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

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The scope of Y·ARD's work in investigating the applications of medium speed Diesel engines to ocean-going merchant ships under contract from the Ministry of Technology is briefly outlined.

The concepts of packaged propulsion machinery installations and auxiliary machinery modules are described and the advantages of both packaging and the use of modules studied.

The designs of propulsion machinery packages, incorporating auxiliary machinery modules, for three specific ship types are then discussed in some detail. The logic leading up to the selection of a twin medium speed geared Diesel installation with controllable pitch propeller and gear-driven alternator(s) as the basic configuration for each of the three ships is described, and each of the auxiliary systems for the propulsion and generating machinery is examined and a standard system for each proposed.

The design of a series of auxiliary machinery modules suitable for medium speed Diesel engines is described. The engines used in preparing these designs are the Ruston and Hornsby AO and the Mirrlees K Major, but the general principles used apply equally well to any of the available medium speed Diesel engines in the 400 to 1000 bhp/cylinder power range. Examples of the resultant module designs and the general arrangement of the machinery installations for each ship are illustrated.

The paper concludes with a consideration of how the concepts of packaging and moduling fit in with current trends in ship construction and outlines the next steps necessary in the evolution of standardized machinery installations to keep pace with these trends.

Mr. Buchanan

INTRODUCTION

Background to Study

This paper is a summary of the findings of a study carried out by Y·ARD into the auxiliary machinery for the main propulsion plant for three ocean-going merchant ships. The study was carried out for and with the guidance of the Ministry of Technology, which has been supporting the design and development of medium speed Diesel engines, and investigations into their applications.

It must be emphasized that this is not a comparison between medium speed and slow speed Diesel engines in the chosen vessels. The case for the medium speed Diesel engine has already been examined[1]. The objectives of the present study were rather to discover and examine the problems of the auxiliary machinery design in vessels for which medium speed geared Diesel engines had already been chosen as the main propulsion machinery.

The increasing proportion of the marine Diesel market claimed by the medium speed engine is illustrated by Table I; almost without exception, the only manufacturers showing an upward trend in volume of production are those predominantly producing medium speed engines. This popularity is likely to be further enhanced by the current development of medium speed engines, which has resulted in a second generation of engines producing approximately 1000 bhp/cylinder as opposed to the previous mean of approximately 500 bhp/cylinder. This development will go a long way towards answering one of the main objections of those opposed to medium speed Diesel engines—the number of cylinders, valves, moving parts etc. required for a given power. Table II illustrates the main characteristics of both first and second generation engines.

Basis for Study

Although the findings of the study are applicable in general to any medium speed Diesel engine in a similar power range, the investigations were specifically based on two British designed and manufactured engines:

i) the Ruston and Hornsby AO;
ii) the Mirrlees K Major.

The design of the propulsion plant for each vessel was also based on two underlying principles:

a) the principle of a packaged machinery installation, by which one main machinery contractor, usually the main engine builder, is responsible, not only for the main engine itself, but also for all the propulsion machinery necessary for driving the ship, including shafting, propellers and electrical generators;
b) the principle of auxiliary machinery modules, in which all auxiliary machinery relevant to a particular system or group of systems is arranged as a single unit and is installed in the vessel in one lift.

Sequence of Work

The following is a brief list of the main tasks completed during the study:

1) in order to ensure that the ships chosen for the study

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TABLE I COMPARISON OF DIESEL ENGINE PRODUCTION 1967/68
(By courtesy of *Shipbuilding and Shipping Record*)

Diesel	1968 No. of ships	1968 No. of engines	1968 BHP	1967 No. of ships	1967 No. of engines	1967 BHP	BHP change	Percentage change
Sulzer	242	261	2 728 350	250	259	2 936 430	−208 080	− 7·09
B and W	196	208	2 176 900	233	245	2 741 970	−565 070	− 20·61
M.A.N.	137	164	1 309 435	170	201	1 515 500	−206 065	− 13·60
Pielstick	53	81	453 260	44	77	428 446	+ 24 814	+ 5·79
Götaverken	26	26	369 800	33	33	490 950	−121 150	− 24·68
Fiat	32	38	314 100	21	24	279 050	+ 35 050	+ 12·56
Mitsubishi	34	34	233 800	41	44	255 070	− 21 900	− 8·56
Deutz	33	41	110 280	29	34	70 715	+ 39 565	+ 55·95
Stork	6	6	93 200	16	16	223 500	−130 300	− 58·30
MaK	37	48	84 000	19	21	43 600	+ 40 400	+ 92·66
Doxford	7	7	79 640	8	8	102 900	− 23 260	− 22·60
Fairbanks Morse	11	31	75 126	13	25	58 182	+ 16 944	+ 29·12
Ruston	13	21	67 300	6	7	14 074	+ 53 226	+378·19
Liebknecht	28	56	64 960	24	28	55 680	+ 9 280	+ 16·67
Total	855	1022	8 160 151	907	1022	9 217 197	−1 057 046	− 11·47

were of a size and type for which there would be an appreciable demand in the foreseeable future, it was necessary to carry out a world market survey of the numbers, sizes and types of ships built over the last ten years and, from this, by statistical methods, to make a prediction of the demand for various ship types over the next ten years; from the findings of this survey it was then necessary to determine which of these ships were most suitable for the immediate application of medium speed geared Diesel propulsion machinery;

2) the ships chosen as a result of the world market survey were then matched as far as possible with existing similar ships;
3) powering calculations were carried out, based on the tank test results for the existing ships, and a choice of the arrangement for the main machinery of these vessels made;
4) from this was established the basic machinery arrangement—propeller size and speed, size of electrical generators etc;
5) having determined the size of the main engines and knowing the requirements for the auxiliary machinery, module specifications were prepared;
6) each auxiliary machinery module was designed in detail;
7) these module designs were incorporated into a detailed machinery arrangement.

The final step, although not completed as part of this study, is to take the three sets of modules developed for the three vessels and expand them into a number of frame sizes of standard modules suitable for the complete range of engine powers available.

It is not proposed to produce here the details of the world market survey, partly because it was simply "a means to an end" in obtaining three typical ship types for which to study the application of medium speed geared Diesel engines, but also because it is always easy to be wise after the event and a number of factors, such as the 1967 war in the Middle East, occurred after the survey had taken place and affected some of the detailed findings. The survey led to the choice of the following ships:

i) a bulk carrier—deadweight 16 000 to 18 000 tons, service speed 15½-16 knots;
ii) a petroleum products carrier—deadweight 22 000 tons, service speed 15 knots;
iii) a refrigerated cargo liner—deadweight 12 500 tons, service speed 18 knots.

It was decided to match these three vessels with three similar ships already in existence in order that the ships' lines and propulsion characteristics of the existing vessels could be used. By doing this:

a) it was ensured that realistic ships' lines were chosen;
b) accurate tank test results could be used in calculating the required powers rather than relying on estimates;
c) capacity plans, ballasting arrangements and some electrical loads could be used from the existing ships to avoid inaccurate estimates.

The adoption of existing vessels designed originally for a different form of propulsion has its disadvantages:

1) to fit the larger slower propeller for the geared Diesel installation, it was necessary to adopt an open-water stern arrangement and to assume some consequent modification to the stern lines of the vessel; the effect of this modification on the propulsion characteristics of the vessel cannot be determined without further tank tests;
2) the three existing vessels were all designed for slow speed direct-coupled Diesel engines and small adjustments to the after lines of the vessels to accommodate more satisfactorily the wider medium speed machinery might well have improved the engine room arrangement;
3) the reduction in main machinery weight by about 50 per cent, resulting from the fitting of the medium speed engines, would necessitate some further modification to the design of the vessel in order to restore the correct trim, although this is in some cases partially offset by a saving in machinery space.

The following vessels were chosen as being the three nearest existing ships to those required for the study:

i) bulk carrier: m.v. *Cape Rodney;*
ii) tanker: m.t. *British Vine;*
iii) refrigerated cargo liner: m.v. *Australia Star.*

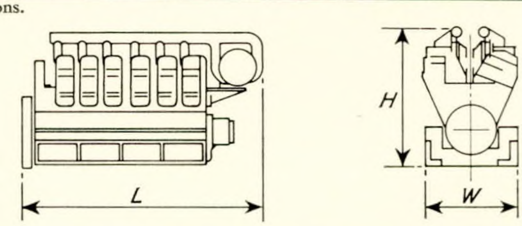
Using the propulsion test data for each of these vessels, the effect of reducing shaft revolutions and increasing propeller size, made possible by the adoption of the medium speed geared configuration, was investigated. Establishment of the most efficient propeller does not indicate the optimum propeller size when the capital cost of the propeller, shafting and gearing have been taken into account. By a process similar to that described by Neumann and Carr[(1)], it was established that, for each of these three ships, the optimum propeller size was the largest that could satisfactorily be accommodated by each vessel within the limitations outlined in Appendix I. The ship's service speed was used as the criterion for the design of the

TABLE II—A COMPARISON OF MEDIUM SPEED DIESELS

		Medium speed range				
		Two-stroke cycle				
		Valve in head			Opposed piston	
Maker		Ruston and Hornsby	Sulzer	Mitsubishi	Mirrlees National	Fairbanks Morse
Designation		AO	Z 40/48	UET 52/65	OP	38A 20
Bore	in/mm	14·25/362	15·748/400	20·47/520	15/381	Top. 10/254 Bot. 20/508
Stroke	in/mm	18·5/470	18·898/480	25·59/650	2 × 15/2 × 381	Top. 10·75/273 Bot. 21·5/546
Swept volume/cylinder	in³	2950	3680	8420	5300	7599
Year first in service		1968		1961	Prototype running trials	1965
Year first in service at sea		1968	1965			
Maximum continuous Output/cylinder*	bhp	500	550	740	1250	1250
Corresponding speed	rev/min	450	430	300	600	450
b.m.e.p.	lb/in²	150	135·5	116	156	145
Mean piston speed	ft/min	1387	1355	1280	1500	upper 806 lower 1612
Specific fuel consumption†	lbs/bhp h	0·342	0·356	0·349		0·334
No. of cylinders available	in line	6,8,9.	5,6,8,9,10,12.	6,7,8,9,12.	8,12,16. (double banked)	6,9.
	vee	12.16.	8,12,16,18. (projected)	—	—	12,18.
Maximum Continuous power Output	bhp	8000	6600	8880	20 000	22 500
Engine weight	tons	67	86	122	150	214·5
Specific weight	lb/bhp	18·75	29·2	30·8	16·8	21·35
Overall length	largest unit	22 ft 0 in	32 ft 0 in	39 ft 2½ in		37 ft 6 in
Overall width		10 ft 0 in	7 ft 9½ in	5 ft 10½ in		14 ft 10 in
Overall height		13 ft 7¼ in	12 ft 9 in	12 ft 10 in		16 ft 0 in

		Medium speed range								
		Four-stroke cycle								
Maker		S·E·M·T· Pielstick	S·E·M·T· Pielstick	Mirrlees National	M·A·N·	M·A·N·	Werkspoor	Fiat	B· and W·	U·D·A·B·
Designation		PC2 400	PC3 480	K Major	V 40/54	V 52/55	TM 410	C 420 SS	45 H	
Bore	in/mm	15·75/400	18·9/480	15/381	15·748/400	20·47/520	16·13/410	16·535/420	17·72/450	20·472/520
Stroke	in/mm	18·11/460	20·45/520	18/457	21·26/540	21·65/550	18·5/470	19·685/500	21·26/540	22·441/570
Swept volume/cylinder	in³	3530	5742	3180	4140	7120	3785	4240	5240	7400
Year first in service		1962	1969	1965	1964		1968		1968	Prototype running trials
Year first in service at sea		1962					1968		1968	
Maximum continuous output/cylinder*	bhp	500	850	464	555	900	500	450	550	1000
Corresponding speed	rev/min	500/520	428—460	525	430	400	500	450	450	425
b.m.e.p.	lb/in²	216	255	220	256	250	206	187	182	247
Mean piston speed	ft/min	1510—1570	1460—1568	1576	1420	1442	1540	1475	1595	1590
Specific fuel consumption†	lbs/bhp h	0·342	0·335	0·335	0·337	0·331	0·35		0·335	
No. of cylinders available	in line	6,8,9.	—	6,7,8,9.	6,7,8,9	6,7,8,9.	6,8,9.	6,7,8,9,10.	5,6,7,8,9.	6,9.
	vee	10,12,14, 16,18.	12,14,16, 18.	12,14,16, 18.	10,12,14, 16,18.	10,12,14, 16,18.	12,16,18, 20.	12,14,16, 18,20.	8,10,12, 14,16,18.	8,10,12, 14,16,18.
Maximum continuous power output	bhp	9000	15 300	8350	10 000	16 200	10 000	9000	9900	18 000
Engine weight	tons	75·8	160	115	105	168	123	121	124	
Specific weight	lb/bhp	18·88	23·4	30·9	23·5	23·2	27·5	30·1	28·05	
Overall length	largest unit	31 ft 7 in	35 ft 7½ in	32 ft 6½ in	32 ft 2½ in	33 ft 10 in	30 ft 9 in	40 ft 4¼ in	34 ft 1 in	
Overall width		11 ft 6¾ in	12 ft 0 in	15 ft 6 in	11 ft 10 in	12 ft 1 in	11 ft 6 in	12 ft 5½ in	13 ft 1½ in	
Overall height		11 ft 2¼ in	14 ft 4½ in	12 ft 1 in	12 ft 11½ in	14 ft 4 in	12 ft 6 in	13 ft 5½ in	13 ft 0 in	

*Makers quoted continuous output for merchant ship propulsion under temperature conditions.
†Specific Fuel consumptions are typical figures based on operation on Diesel fuel.



propeller and, in determining the engine power required, the following allowances were made:

a) a 25 per cent fouling allowance;
b) a ten per cent service de-rating;
c) tropical de-rating as recommended by the engine manufacturers.

The characteristics used in determining the main engine configuration were :

1) Ruston and Hornsby AO engine:
brake mean effective pressure 150 lb/in² (10·55 kg/cm²);
cylinder bore 14·25 in (362 mm);
stroke 18·5 in (470 mm);
hp/cylinder: 500;
rev/min: 450;

2) Mirrlees K Major engine:
Brake mean effective pressure 220 lb/in² (15·466 kg/cm²);
cylinder bore 15 in (381 mm);
stroke 18 in (457·2 mm);
hp/cylinder: 460
rev/min: 525.

The following machinery was therefore chosen for each of the three vessels (see Figs 1, 2 and 3):

Ship	Ruston and Hornsby	Mirrlees National
Cape Rodney	2 x 8 AO in line, cont. hp 4000 each, 450 rev/min	2 x 9K Major in line, cont. hp 4160 each, 525 rev/min
British Vine	2 x 8 AO in line, cont. hp 4000 each, 450 rev/min	2 x 9K Major in line, cont. hp 4160 each, 525 rev/min
Australia Star	2 x 16 AO vee, cont. hp 8000 each, 450 rev/min	2 x 18 K vee Major, cont. hp 8300 each, 525 rev/min.

Ancillary Studies

During the main study, it became apparent that a number of aspects warranted separate investigation and, as a result, some brief ancillary studies were carried out:

i) to determine how best to manoeuvre a vessel fitted with medium speed Diesel engines and a c.p. propeller, and also how this vessel's manoeuvring characteristics compared with other modes of propulsion, a series of computer simulations of various manoeuvring techniques was carried out; the findings of this study form part of the paper by Goodwin, Irving and Forrest[2];

ii) to determine the prejudices of prospective shipowners on various aspects of machinery selection and arrangement, a questionnaire was circulated to as many British shipowners as possible to determine current attitudes in, for instance, the question of motor-driven versus engine-driven pumps;

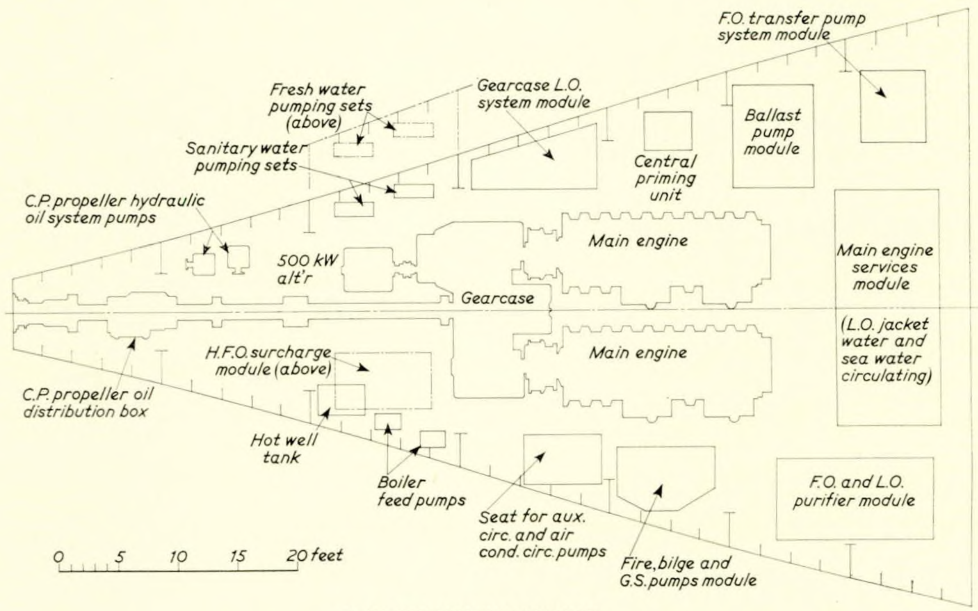


FIG. 1—*Machinery arrangement for 17 000 ton bulk carrier*

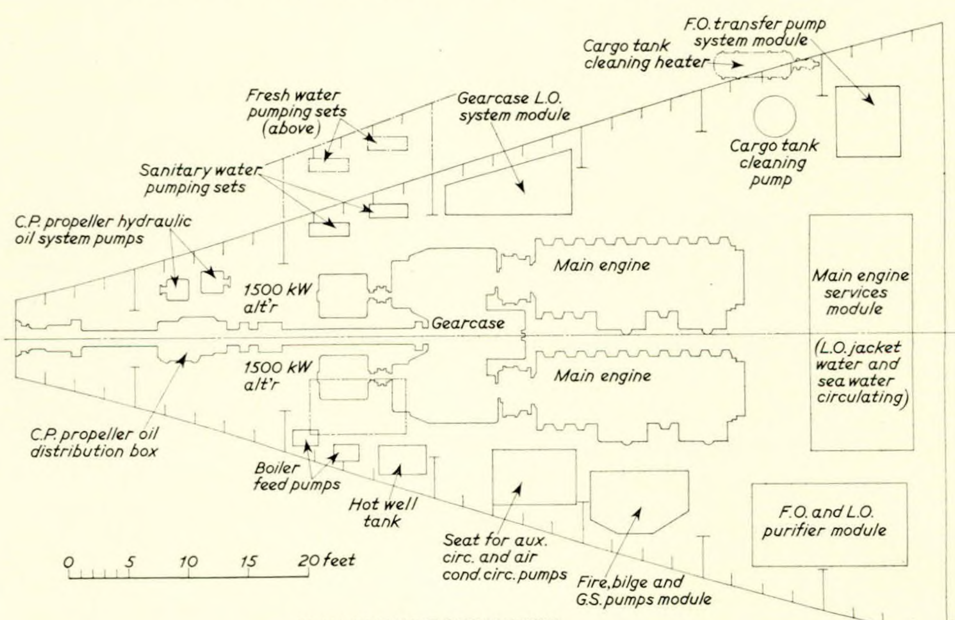


FIG. 2—*Machinery arrangement for 22 000 ton G.P. tanker*

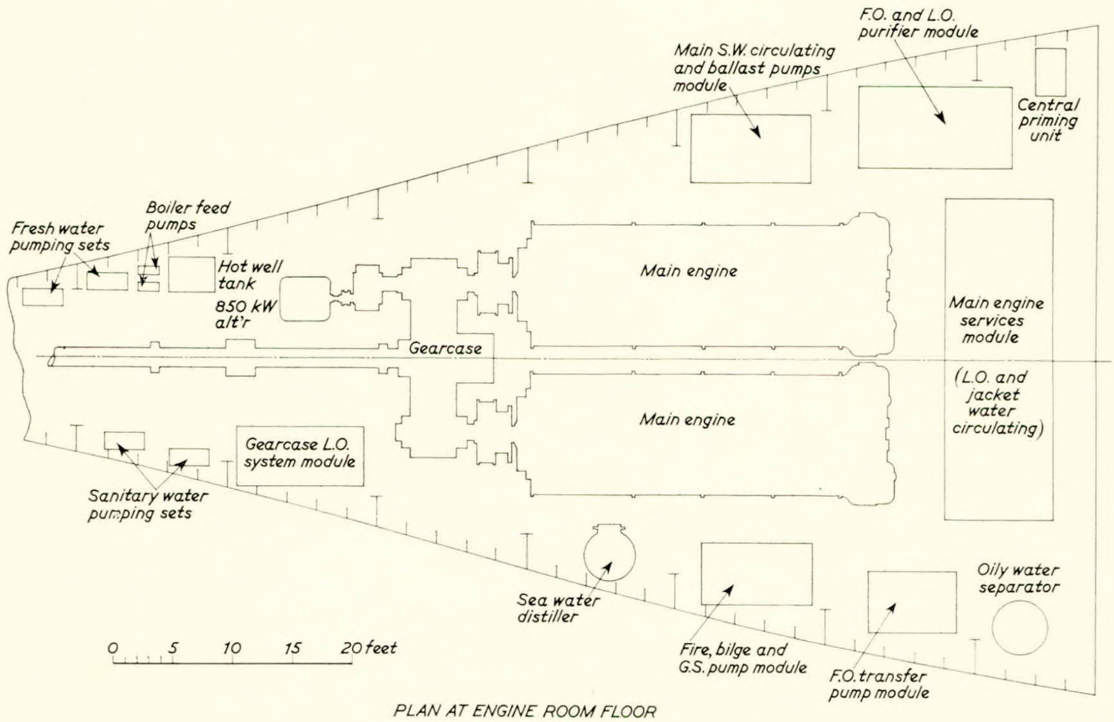


FIG. 3—*Machinery arrangement for 12 500 dwt cargo liner*

iii) as an alternative to the adoption of c.p. propellers, a system using f.p. propellers and gear-driven alternators and an asynchronous generating scheme was investigated;
iv) to examine any contractural problems which might occur in effecting a "package deal" for the supply of the entire propulsion plant for a new vessel, discussions were held with the two main engine manufacturers concerned in the study and these discussions formed the basis of a brief report on these problems and possible solutions.

PACKAGED MACHINERY INSTALLATIONS

The Concept

The concept of the packaged machinery installation can best be illustrated by the recent developments in marine steam turbine installations. The majority of internationally known marine steam turbine manufacturers now advertise, and in many cases have already installed, propulsion machinery packages which include not only the main turbines but also the gearbox, boiler feed pumps, feed heaters etc. The manufacturer guarantees the performance of the whole installation since all the auxiliaries are supplied by him and the plant is installed to his requirements.

In the case of marine Diesel engine installations the choice and arrangement of auxiliaries does not have such a marked effect on overall plant performance as, say, the feed heating arrangements in a sophisticated steam turbine installation. For this reason, progress towards "package deals" in Diesel machinery has not been nearly so marked as in the case of steam turbines, but the move towards larger units in the U.K. shipbuilding industry and the consequent emphasis on rationalization of building in particular yards should lead to an increased requirement for standard machinery installations and therefore towards "package deals".

Aims and Advantages

The aims and advantages of a standardized packaged machinery installation may be summarized as:

a) improved performance and reliability; the matching of the auxiliary equipment exactly to engine requirements enables the main machinery contractor to guarantee the performance of the whole propulsion installation; furthermore, the evolution of standard auxiliary system designs avoids makeshift adaptation of basic slow speed engine auxiliary machinery to the needs of the medium speed engine and results in systems completely acceptable to the main engine supplier for a method of propulsion of which the majority of shipyards have, as yet, only limited experience,
b) reduction in costs: by reducing the time required for installation design and for the accurate matching of auxiliary machinery to the main engines, a reduction in costs to the component suppliers, engine builders and shipbuilders should result, with a consequent beneficial effect on the ultimate cost to the shipowner;
c) promotion of standardization of marine auxiliaries and systems.

There are also two peripheral advantages:

i) the reduction in design time should serve to speed up ship delivery;
ii) one main machinery contractor, responsible for the whole propulsion machinery, simplifies the procedure for installation for the shipbuilder and the establishment of guarantees, and it also simplifies the problems of technical enquiries and spares for the prospective shipowner.

AUXILIARY MACHINERY MODULES

The Concept

As previously stated, the word "module" refers to a group of auxiliary machinery components, such as pumps, coolers and filters, which are assembled together in a suitable structure in a workshop (not on board the vessel under construction) and are tested, flushed and sealed off, and then lifted as one unit into the machinery spaces on board the vessel. The modules designed as part of this study are not arranged for repair-by-replacement of the entire module on the failure of one of its components. The module itself and its position in the machinery spaces are designed to facilitate *in situ* maintenance of the components.

The concept of the modular auxiliary system is not necessarily dependent on the acceptance of the concept of a package deal, although the two logically go together.

The idea of module construction has been current for a number of years and there have also been available on the market certain modules designed by individual machinery manufacturers for machinery components which can with ease be grouped together on one bedplate. A case in point is the number of centrifugal purifier modules currently available. The module principle has been actively supported and developed, notably for application to slow speed direct-coupled Diesel engines, by B.S.R.A. who provided advice and assistance in this study, and also completed the detail design of some of the modules.

The Advantages of Auxiliary Machinery Modules

The advantages of the use of auxiliary machinery modules in the design of machinery installations are:

1) ease of auxiliary machinery erection; because the components of the module are erected in workshop conditions rather than on board the vessel, the assembly of the auxiliary systems is more quickly and efficiently effected;
2) ease of installation; the number of crane lifts from the quayside to the vessel is drastically reduced and once the module has been positioned on its seatings in the vessel, completion of the peripheral connexions provides in the shortest possible time a completed auxiliary system;
3) cleanliness of auxiliary systems; because the testing and flushing out of pipelines and auxiliaries after erection is completed in the workshop, followed by the sealing of end openings against the ingress of dirt, the possibility of machinery breakdown during trials and in the early days of commissioning is reduced, although care must be exercised with off-module piping;
4) improved opportunities for standardization of auxiliary systems and component machinery;
5) ease of preparation of arrangement drawings, pipe layouts and weight and CG calculations; the machinery arrangement draughtsman is dealing with a reduced number of units in that he is concerned with positioning the modules only and not a multiplicity of pumps, coolers and filters. The weight and CG of each module is accurately known and therefore the task of calculating total machinery weights and centres of gravity is simplified; it is perhaps not looking too far ahead to suggest that this is a logical step towards the rationalization of machinery installation design into computer selection of module "building blocks" according to the required duties.

Some Problems of Module Design

In preparing the design of a standard module, the problem of the number of manufacturers of each component has to be considered. There are three alternatives:

i) arrange that the module is capable of accommodating the product of every known manufacturer of that particular item of machinery; this obviously calls for a very large design effort to be devoted to each individual module;
ii) use or create a standard for the external dimensions of every component and design the module to accommodate only these standard items; although in the interests of standardization and rationalization this was obviously a desirable aim, it was necessary to avoid prejudicing the chances of acceptance of the modular principle by forcing the prospective shipowner also to accept items of machinery specially designed to the standard specification which had not previously been proven at sea;
iii) the compromise solution ultimately adopted was to design each module to accommodate between two and four of the more commonly acceptable manufacturers' components; thus, the majority of shipowners' personal preferences can be met by the variation in the components which can be fitted into the range of modules.

Each module design must aim to incorporate as many machinery components and pipes as possible, because:

a) this allows the maximum amount of work to be completed in ideal conditions in the workshop;
b) one crane lift to the vessel installs the maximum number of components;
c) the maximum amount of system pipework can be assembled and flushed under workshop conditions and the number of peripheral connexions which, during the installation procedure on board the vessel, provide an opportunity for the ingress of foreign matter is reduced to a minimum.

There are two primary limitations on module size:

1) the module must be suitable for transportation from the workshop to the vessel; for this reason the module designs in this study are limited to approximately 9 ft 6 in width to allow for transport by road to the shipyard;
2) the larger the module, the earlier it must be delivered to the ship in order that it may be installed without any delay in hull construction; hence it becomes more important that the large module must be delivered to the ship on time and consequently the delivery dates of the component machinery items to the module constructor becomes similarly more critical; the shipbuilder's crane capacity is not likely to prove a limitation on module size, since this crane must in any case be suitable for the installation of the main engines and gearbox.

It had been hoped that by rationalization of the system requirements of the AO and K Major engines for a given power, one set of auxiliary machinery modules could be designed to serve either engine type. This proved in practice impossible, due essentially to basic differences between the two-stroke AO and the four-stroke K Major engine. This has resulted, for the majority of engine dependent modules, in small differences between the module for a given duty for the K Major and AO engines, although the physical size and general arrangement of the module is similar in each case.

In approaching the problem of the sub-division of the auxiliary systems into the various modules, the relative merits of system modules and cooler modules have to be evaluated. A system module contains the majority of machinery items relevant to a particular system, whereas the cooler module contains only the main engine coolers with the relevant pumps, filters etc. grouped elsewhere. Because of the undoubted advantages of both arrangements, an alternative solution was evolved which effectively combines the best of both worlds. Since the relative requirements for jacket water, lubricating oil and salt water will not vary for a given combination of engines, it is logical to consider supplying these auxiliary requirements from a single module. The resulting combined engine services module will supply all the auxiliary requirements for both engines with the exception of fuel oil. The resulting module is obviously large but is acceptable provided it is within the limitations outlined above. In all the machinery installations under consideration, the logical position for this combined services module was across the forward end of the two main engines, a position which also lends itself to minimizing the number and length of interconnecting pipe runs between the module and the engines and which was also suitable for the suction arrangements for the main lubricating oil pumps from the drain tanks below the engines.

BASIC MACHINERY ARRANGEMENT

Single versus Twin Engines

In the case of the two lower powered vessels considered in the study (the tanker and the bulk carrier), it would have been possible to design machinery installations for each vessel with a single geared Diesel engine. It was considered, however, that the operational advantages in favour of twin engines were:

i) the ability to carry out maintenance on one engine in port without immobilizing the vessel; this is of particular importance in the case of the tanker;

ii) the ability to shut down one main engine at sea, either for breakdown maintenance or for reduced speed operation;
iii) the simplification of the provision of standby auxiliaries if it is accepted that repairs can be carried out to a defective main engine or related auxiliary system while operating on the remaining engine;
iv) reduction in size of each main engine, which results in the power of each main engine being more compatible with in-port electrical loads, thereby providing the opportunity of using a main engine-driven generator.

Controllable versus Fixed Pitch Propellers

A comparison was carried out for each vessel of capital and operating costs for machinery installations involving:

a) a controllable pitch propeller and gear-driven alternator(s);
b) a fixed pitch propeller with separate Diesel-driven alternators.

A summary of the figures for one of the vessels is included as Appendix II.

Among the factors which must be taken into consideration in a comparison of the capital costs are:

1) the additional cost of the c.p. propeller itself and its additional installation cost;
2) the need in the f.p. propeller installation for reversing equipment for the main engine and a remote control system (the c.p. propeller cost includes single lever bridge control of engine speed/propeller pitch);
3) increased compressor and air reservoir capacity necessary for the propeller installation because of the increased number of engine starts;
4) the reduction in Diesel-driven generators possible by the adoption of a shaft-driven alternator arrangement.

From the figures it can be seen that the considerable additional cost of the c.p. propeller itself is to a large extent offset by the saving in Diesel generators. The c.p. propeller also has a marginal effect on the operating costs of the vessel. The powering calculations for each of the vessels under consideration show that the ballast speed could be marginally increased (e.g. 0·6 knots for the bulk carrier and 0·5 knots for the tanker) by utilizing the ability of the c.p. propeller to transmit the full engine service power in the ballast condition. If it is assumed that one of these vessels spends about 200 days a year at sea and about 80 of these in ballast, a saving in voyage time per year of approximately three days could be made at an extra fuel cost of £50 to £60 per day.

The f.p. propeller installation used in the comparison does not have a waste heat turbo-alternator. Electrical power will therefore have to be provided by running an auxiliary engine, probably on Diesel fuel, whereas the c.p. propeller installation will derive its power from the main engines burning heavy fuel. Assuming a cost differential of £5 per ton between residual fuel and gas oil, a specific fuel comsunption of 0·35 lb/bhp h and a mean electrical load of 400 kW for 200 days at sea, a cost saving of approximately £2150 per year could be achieved.

A brief investigation was also carried out to determine whether the situation altered radically if the f.p. propeller installation was considered in conjunction with the waste heat recovery system, including an exhaust gas boiler and turbo-alternator. For the cargo liner, the number of days at sea per year proved to be below the break-even point, below which the costs of the steam plant are greater than for the Diesel-driven alternator. For the tanker, the largest auxiliary load is required for cargo discharge in port when no waste heat is available and a large oil-fired steam plant would therefore be required, which would remain idle for a large proportion of the ship's life. In the bulk carrier, the case for and against the waste heat system was most evenly balanced. In the event, it was decided to adopt the gear-driven alternator alternative in this vessel as in the other two, because:

i) there was a strong possibility of the necessity for continuous oil firing of the waste heat boiler under maximum electrical loads at sea;
ii) although no self-discharging gear was assumed for the bulk carrier (unlike the parent or existing ship—*Cape Rodney*), the provision of such gear would certainly tip the balance in favour of the gear-driven alternator alternative, which provides maximum utilization of the main engines in port.

There are also several uncostable advantages of the c.p. propeller installation. Manoeuvring calls for fewer stops and starts of the main engine than with an f.p. propeller. For two similar ships on the same run, the c.p. propeller installation required only 102 starts to the f.p. installation's 2500. The consequent reduction in piston ring and liner wear is difficult to quantify but must result in reduced maintenance costs.

The reduction in number of the Diesel-driven alternators in the c.p. installation will reduce the maintenance and spares for these units, since the maintenance load of a gear-driven alternator is negligible.

The c.p. propeller can be arranged to absorb the full engine power, whatever the hull condition, and therefore avoids the necessity with the f.p. propeller of either designing "too light" when the vessel is new to allow for subsequent fouling, or compromising on the design between the new and fully fouled conditions. The c.p. propeller will also permit one engine to transmit its full power when the other is shut down.

There is, furthermore, the claimed superiority of the c.p. propeller for manoeuvring, which may in certain instances result in reduced towing fees in some ports. C.P. propellers are an almost automatic choice for ferries and other vessels requiring a high degree of manoeuvrability, and a remarkable number of operators of slow speed direct coupled engines are willing to pay the considerable extra cost of a c.p. installation to obtain this and other uncostable advantages.

Based on these considerations, it was therefore decided to progress each of the three vessels' machinery installation designs on the basis of a twin-engined installation with a c.p. propeller and either one or two alternators driven from the gearbox. Detailed electrical load schedules were prepared and a summary of these is shown in Table III.

TABLE III

	Electrical load summary			
	Normal sea load	Stand by	Harbour cargo handling	Harbour basic
Bulk carrier kW	367	484	366	273
Refrigerated cargo liner kW	608*	648*	604*	356*
Tanker kW	373*†	394	1414‡	256§

* Add 80 kW during cargo cooling down period.
† 522 kW when tank cleaning.
‡ 1414 kW represent four cargo pumps in use.
§ 421 kW when tank cleaning.

It should be noted that the loads quoted are not total connected loads but have a diversity factor applied. In preparing the electrical schedules it was assumed that an electrically-driven bow thruster was not provided. Reference to these electrical loads shows that even with a twin engine configuration, the size of the port electrical load represents a relatively small percentage of the total power available from one engine. Due to the well-known troubles attendant on running medium speed Diesel engines for protracted periods on very low loads' it was established with the engine manufacturers that the size of the port electrical load for each vessel was acceptable for protracted running, although it may be advisable to run the main engine on Diesel fuel when operating under these conditions.

Alternator Drive Arrangement

It will be noted that in each of the chosen machinery installations, the alternator is driven from the gearbox, either by quill shaft from the main engine input or directly from the gearing pinion, and that a step-up gearbox is utilized to increase the alternator speed from main engine speed to 1200 rev/min. This increase in speed allows a more compact alternator to be used and it also permits the stepping inboard of the drive to the alternator relative to the centreline of the relevant main engine, thus allowing the envelope of the main propulsion machinery to conform approximately to the lines of the vessel.

The alternative arrangement, in which the alternator is driven directly from the forward end of each main engine, has these advantages:

a) the direct-coupled alternator, although larger than its high speed equivalent, is often cheaper and shorter when the step-up gearbox is taken into account;
b) the complexity of the step-up gearbox and the quill shaft drive arrangement (if required) is avoided.

The outstanding disadvantage of the direct-coupled alternator is that it adds to the overall length of the main machinery block, in way of the maximum width section of the engine room, and would result in an increased machinery space length.

Refrigerated Cargo Liner Arrangement

The machinery installation for this vessel was progressed on the basis of 1 x 850 kW alternator driven from one main engine through step-up gearing by a quill shaft from the input side of the engine clutch. In this way the alternator can be driven by its respective main engine whether or not that engine is clutched to the main transmission. Two 500 kW Diesel-driven alternators are also provided, the gear-driven alternator being used for the supply of power at sea and for cargo discharge in port, while the two Diesel-driven sets act as standby and harbour duty sets. Although the provision of two gear-driven alternators, each capable of providing the full electrical load either at sea or for cargo discharge, would result in an extremely flexible machinery installation, the expense of 100 per cent standby capacity could not be justified when it was considered that separately driven generators had to be provided for standby and harbour duties (see Fig. 3).

Bulk Carrier Arrangement

It was assumed that this vessel had no self-discharging gear on board. The in-port electrical load was, therefore, very limited and represented such a small proportion of the full power capacity of one main engine that it was impracticable to consider running one main engine in port as a generator. The gear-driven alternator drive arrangement is therefore simplified in that the alternator is driven directly from the gearing and not from the quill shaft from one of the main engines. This allows either main engine to be shut down at sea while still retaining the facility of using the gear-driven alternator. The machinery arrangement, therefore, consists of a 500 kW alternator driven directly from the gearing through step-up gears and two auxiliary Diesel generators, each of 350 kW capacity. (see Fig. 1).

One consequence of dispensing with the quill shaft drive arrangement in this vessel is that the electrical load and synchronizing tests would be required to be carried out during sea trials. It might also be necessary, under certain circumstances, to provide an artificial variable load during these tests.

Tanker Arrangement

The machinery arrangement for the products tanker was progressed on the basis of electrically-driven cargo pumps in order to take full advantage of the medium-speed geared installation's facility of supplying the high in-port loads required for cargo discharge (an arrangement with hydraulically-driven cargo pumps, with the hydraulic pumps driven by the main engine, is an alternative). The basic machinery arrangement consists of two 1500 kW alternators, one driven by quill shaft by each engine, and an auxiliary Diesel-driven alternator of 500 kW capacity. The sizing of the main engine-driven alternators must be determined by an evaluation of the likely utilization of the vessel in each specific case. It could be envisaged that this vessel might be used for distributing refined products to a number of ports in limited quantities and that simultaneous use of all four cargo pumps at full power would rarely, if ever, be required. Under these circumstances, reduction of the main engine-driven alternator capacity to 1000 kW each would be justifiable. (see Fig. 2).

MODULE AND SYSTEM DESIGN

Basic Design Philosophy

It was decided that, to take fullest advantage of the capabilities of the medium speed geared installation, the system design should aim at ensuring that failure of any one component on either main engine or auxiliary equipment should incapacitate only 50 per cent of the total power capacity of the machinery. The requirement cannot be met in every case, but in the engine jacket water system, for instance, where one pump supplies both engines, auto-starting of the standby pump is proposed, yet in the main engine lubricating oil system, where one pump supplies each engine from separate systems, manual start-up of the common standby pump is regarded as acceptable (see Fig. 4).

None of the auxiliary systems evolved in the study made use of engine or gear-driven pumps. This is essentially because the ships under study had previously been almost exclusively fitted with slow speed direct coupled Diesel engines, the operators of which, when faced with the medium speed engine alternative, might view with distrust engine-driven auxiliaries. Subsequent indications are, however, that the study during its course has been overtaken by events, since the popularity of engine-driven and, particularly, gear-driven auxiliaries, has grown very swiftly.

All the modules developed were based on tubular heat exchangers, due to the limited popularity of the plate-type alternative at present.

The design of the switchboard, which is located in the machinery control room, is based on a standard series of switchboards on the modular principle, utilizing moulded case circuit breakers evolved as the result of co-operation between Y·ARD and B.S.R.A.

Main Engine Cooling Water System

A number of different systems was investigated, but the choice narrowed down to two:

1) separate systems for each engine with a common standby pump;
   advantages: elimination of contamination of one system by the other; swift identification of which engine is leaking by reference to each header tank;
   disadvantage: necessity of providing three pumps, each capable of supplying one main engine;
2) a combined jacket water system supplied by one pump with a similarly sized standby pump, but separate coolers for each engine;
   advantages: most simple system, also the cheapest;
   disadvantage: contamination of one engine is possible by the other and detection of the source of leakage is more difficult.

There was also the further consideration of the provision of a jacket water evaporator. Since this would be required to operate when either main engine is shut down, some simplification of the change-over arrangements is possible when utilizing a combined system.

The combined system was therefore chosen, with one pump for both engines and a similarly sized standby and a single header tank pressurizing the system. Thermostatic control is provided by means of a bypass valve in the primary circuit at the cooler. Auto-starting of the standby pump is provided in order that failure of the running fresh water pump does not incapacitate the whole propulsion machinery.

Main Engine Lubricating Oil System

As with the engine cooling water system, the relative

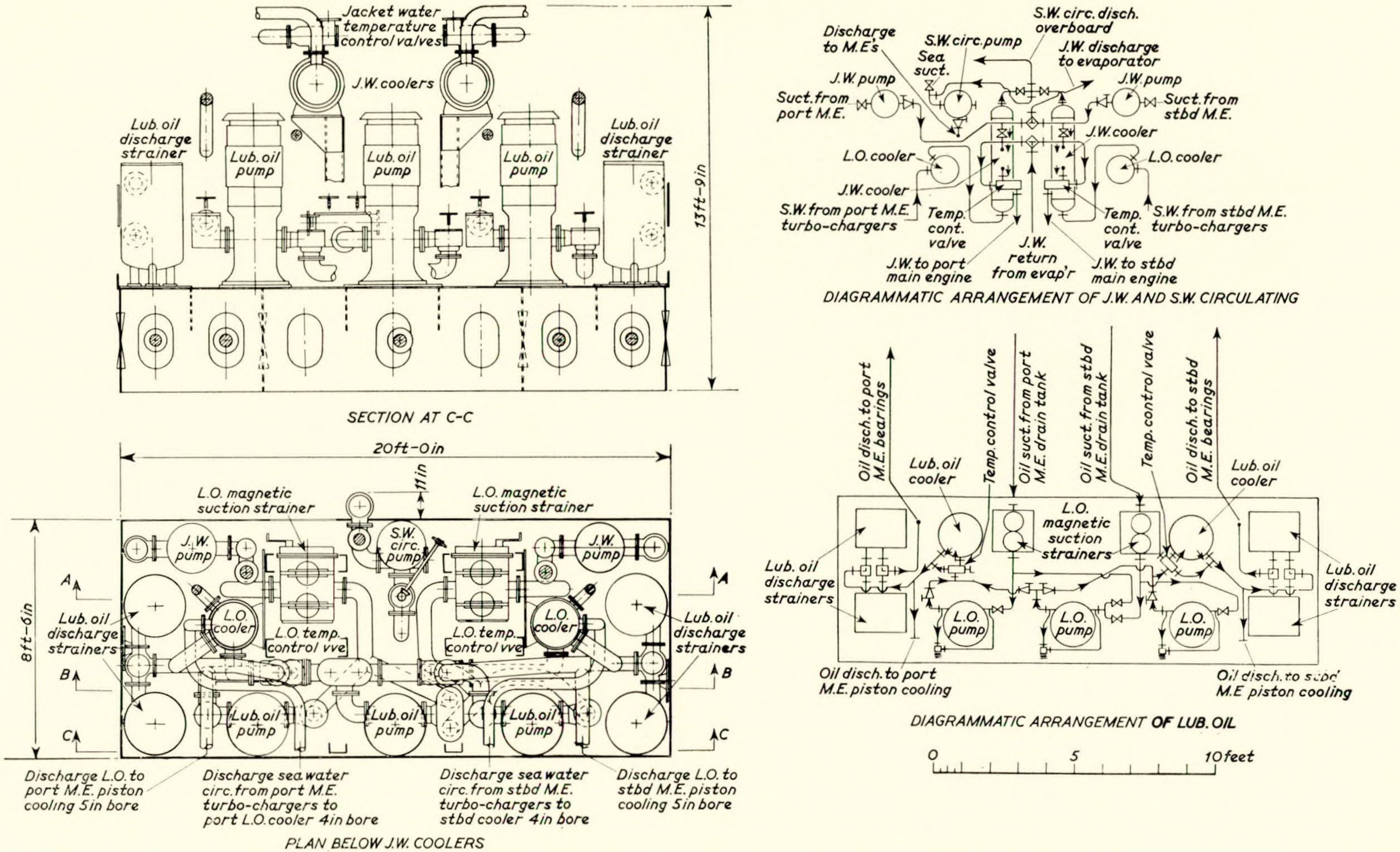


FIG. 4—*Main engine services module—Approximate dry weight of module 19·5 tons*

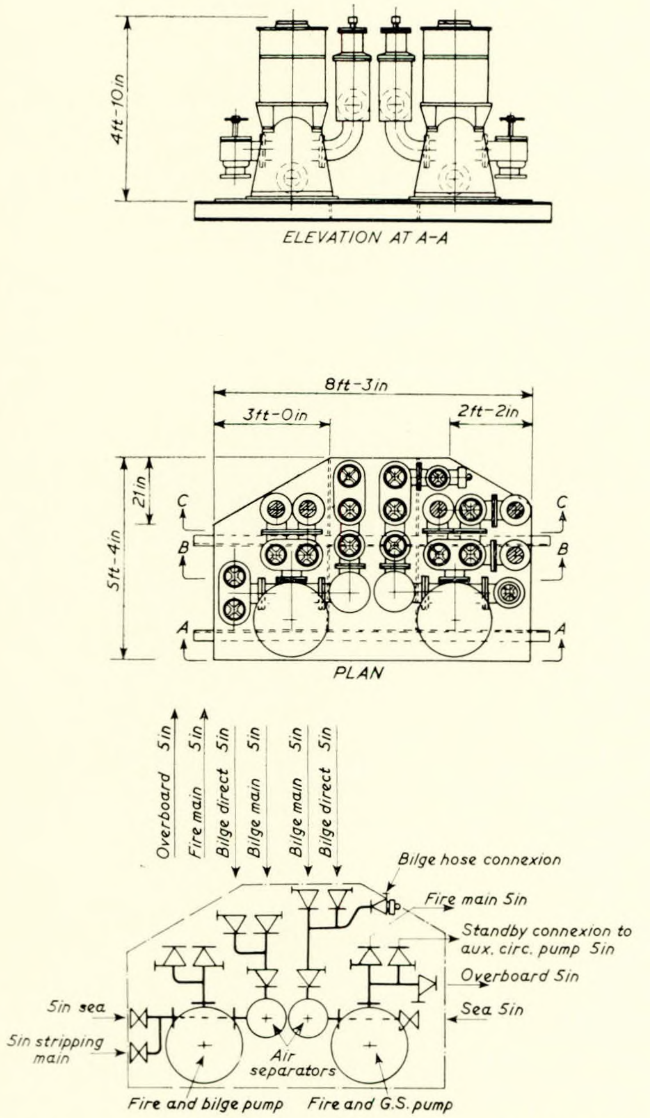


FIG. 5—*Fire, bilge and general service pump module for 17 000 dwt bulk carrier*

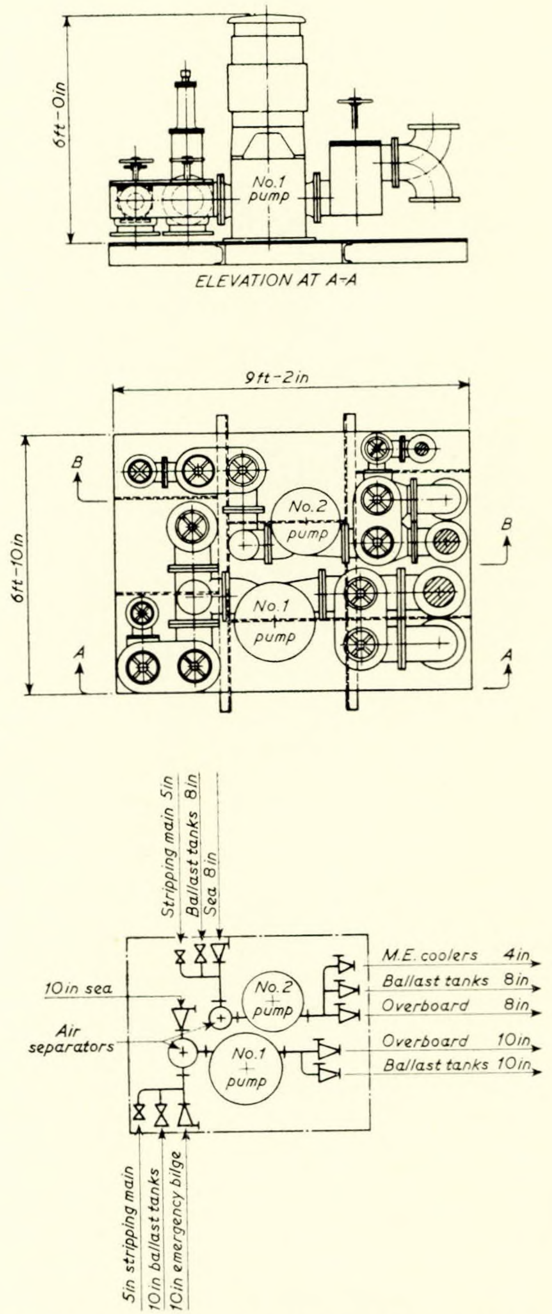


FIG. 6—*Ballast pump module for 17 000 dwt bulk carrier*

merits of combined and separate systems were evaluated. In the case of the lubricating oil system, however, the importance of avoiding contamination of one engine lubricating oil charge by the other was felt to justify the provision of completely separate lubricating oil systems. One pump, therefore, supplies each engine, drawing from its own separate drain tank, with a centre standby pump which is capable of being connected to either system manually in the event of failure of either of the two running pumps. The suction and discharge valves of this standby pump are suitably interlocked to prevent inadvertent suction from one system and discharge to the other. Failure of one lubricating oil pump will result in the relevant engine being de-clutched and stopped automatically and the standby pump being lined up and started manually. Auto-starting of the standby pump is possible, but the motorized suction and discharge valves which would be necessary were felt to be an unjustifiable expense in that 50 per cent full power was still available from the other engine.

It will be noted from Fig. 4 that free standing lubricating oil pumps are shown. The adoption of tank-type lubricating oil pumps is popular. The use of tank-type pumps in conjunction with modular construction is more difficult than with free-standing pumps, since it involves either taking the pump off the module or arranging that positioning the module during installation also positions and effectively seals the tank-type pump in place. A design of combined engine services module is, however, now available for tank-type lubricating oil pumps.

Main Engine Salt Water System

A vessel's salt water system is probably the biggest single source of maintenance load of the various auxiliary systems. It was, therefore, decided, in designing the main engine salt water system, to aim:

i) to reduce the number of salt water pumps;
ii) to minimize the length of salt water piping;
iii) to reduce the number of openings in the ship's side.

To this end, the salt water systems for the main engines include two basic departures from normal Diesel engine practice:

a) a main circulating pump on the same principle as that adopted for steam turbine machinery is utilized, with standby capacity being supplied by the ballast pump (except in the case of the tanker, where an auxiliary circulating pump is provided);

b) a series cooling arrangement is adopted, whereby the sea water from the engine charge air coolers is led into the lubricating oil cooler and thence into the fresh water cooler and then overboard (see Fig. 4).

The series cooling system does fulfil the aims outlined above. It also poses some problems:

1) care must be taken to allow for the increased pump head due to the increased system pressure loss; because the inlet sea water temperature to some coolers is above ambient sea water, the size of the cooler may have to be increased; in practice, the velocity of the primary fluid over the tubes was found to be the critical factor and little increase in cooler size was found necessary;

2) the sea water overboard discharge temperature is higher than would normally be encountered with the conventional parallel system (128°R (53°C) maximum).

A compromise must be struck between optimizing cooler and pump sizes and maintaining a low overboard discharge temperature. This maximum temperature will only be reached in 90°F (32°C) seas when operating at maximum continuous engine rating. with no de-rating for service conditions.

The sea water systems in the standard installations under consideration must be capable of operating with a range of sea water inlet temperature of 28°F to 90°F (−2°C to 32°C). This would obviously require a very large rangeability on the various control valves, with a correspondingly expensive or complex arrangement. This has been avoided by artificially reducing the range of temperatures which the various control valves experience, by recirculating sea water from the overboard discharge line back to the pump suction. By thermostatically controlling this recirculation, an artificial sea water inlet temperature at the pump is created, which will normally never fall below a pre-set value. Hence, for sea water temperatures below, say, 58°F (14°C), a controlled amount of recirculation will take place so that the sea water temperature at inlet to the pump is always 58°F (14°C). Hence the rangeability required at the control valves in the fresh water and lubricating oil systems need only be from 58°F to 90°F (14°C to 32°C) instead of from 28°F to 90°F (−2°C to 32°C).

The following minor differences between the sea water systems for the two engine types should be noted:

i) the Ruston and Hornsby AO engine has a sea water bypass control on the charge air cooling system, which maintains an air temperature of 100°F (38°C);

ii) the Mirrlees K Major engine has a supply of cooling water to a fuel valve and exhaust cage cooling system.

L.P. Steam System

L.P. steam is required on the vessel for heating purposes for accommodation, fuel lines, fuel storage and fuel treatment. The total heat requirement for these duties is very low compared with the heat available in the exhaust gases. Consideration was given to utilizing a steam-operated evaporator in order to simplify the jacket water system, but this was discarded on the grounds that although similar in cost, the maintenance load for the steam evaporator was considerably higher.

An alternative considered was to avoid an L.P. steam system altogether by providing a pressurized hot water system supplied by two compact oil-fired water heaters, the price of which compares favourably with the equivalent boiler. The system has been used with some success in vessels operating on Diesel fuel. The problem occurred, however, in the vessels under study, which burn residual fuels, of providing an observation tank and facilities for removing entrained oil in the return heating lines from bunker tanks in a heating system operating at a pressure of approximately 80 lb/in². It was therefore decided to retain the conventional L.P. steam system.

Boiler Arrangement

The following configurations of boiler arrangement were considered:

a) an exhaust gas boiler with a separate oil-fired boiler;

b) a composite boiler;

c) an oil-fired boiler only.

Capital and running costs were estimated and it became apparent that the composite boiler was the cheapest of the three, but suffered from the disadvantage that a single fault could incapacitate the whole steam system. The single oil-fired boiler was obviously the most expensive to run.

For both the bulk carrier and the refrigerated cargo liner, the basic arrangement adopted is an exhaust gas boiler and a separate oil-fired boiler, the latter being used normally only in harbour. One engine is capable of supplying sufficient heat for the vessel's steam requirements over a reasonable power range and a waste heat boiler is fitted in the uptake of only one engine. The obvious limitation on the flexibility of the installation is felt to be justified in avoiding the provision of a further exhaust boiler for a small steam load. The fitting of an exhaust silencer to that engine which does not have the exhaust boiler will tend to equalize the exhaust pressure drop on the engines.

In the case of the tanker, the normal ship's service requirement is approximately $1{\cdot}25 \times 10^6$ Btu/h. Cargo tank heating requires a further 14×10^6 Btu/h. In order to allow for the contingency of heating the cargo while at anchor, an oil-fired boiler of about 15×10^6 Btu/h is required. The normal harbour load, however, is less than 1×10^6 Btu/h, which represents a higher turn-down ratio than a boiler with a maximum capacity of 15×10^6 Btu/h could cope with satisfactorily. The chosen system is, therefore, a composite boiler, again in only one engine uptake, each part of which is capable of supplying the normal service load, excluding cargo heating, and a separate oil-fired boiler for cargo heating.

The steam capacities of the various boiler arrangements for the three vessels are:

1) bulk carrier: tank type waste heat boiler, 2400 lb/h at 100 lb/in² from feed at 120°F (49°C);
packaged oil-fired boiler—2000 lb/h at 100 lb/in² from feed at 120°F (49°C);

2) tanker: composite boiler, 2400 lb/h and 1500 lb/h at 100 lb/in² from feed at 120°F (49°C) from the waste heat and oil-fired sections respectively;
packaged oil-fired boiler, 13 500 lb/h at 100 lb/in² from feed at 120°F (49°C);

3) refrigerated cargo liner: tank type waste heat boiler, 3500 lb/h at 100 lb/in² from feed at 120°F (49°C); packaged oil-fired boiler, 3000 lb/h at 100 lb/in² from feed at 120°F (49°C).

Various means of controlling steam output from the waste heat unit were considered:

i) Dumping excess steam to a condenser;

ii) bypassing the gas flow around the waste heat unit;

iii) allowing a variation in boiler pressure;

iv) controlling feed water flow rate or boiler water level.

The comparative merits were also evaluated of the fire tube type and the forced circulation economizer type of waste heat boilers. It was found for the small steam outputs required that the forced circulation economizer type of boiler was more expensive when account was taken of the necessary circulating pumps. The design of fire tube waste heat unit chosen has a gas bypass through the centre of the tube nest, controlled by a motorized screwdown shut-off valve. This central valve, when shut, forces the gas flow to pass through the vertical tube nest of the waste heat unit. When open, the major proportion of the gas flows through the central gas passage. This was considered to be the simplest and most effective form of control. The operation of the screwdown gas bypass valve is automatic and can be by air pecker motor or other suitable means. A small water-circulated dump condenser is also fitted, since when the steam demand is very low, the small amount of steam generated, even when on full bypass, may exceed the demand.

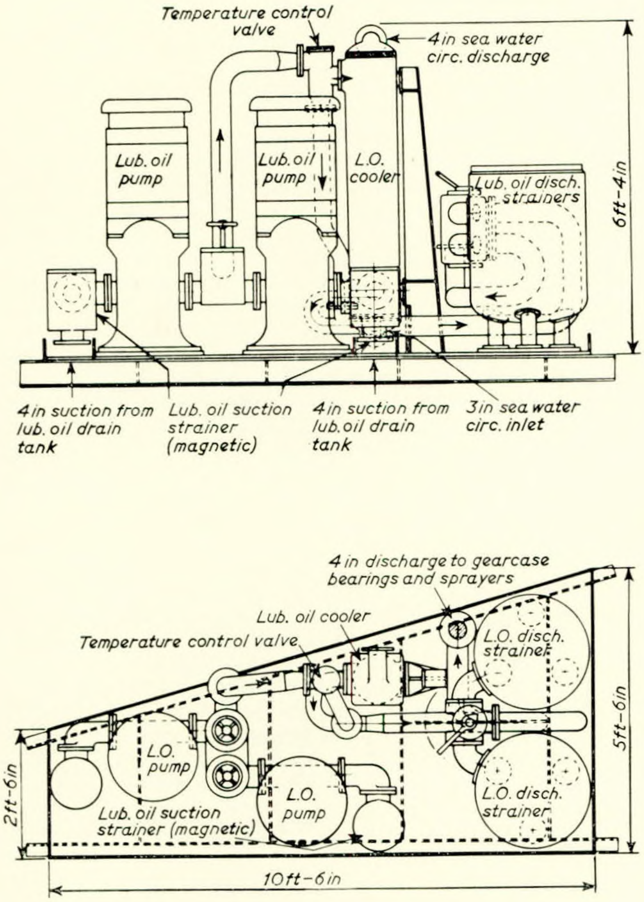
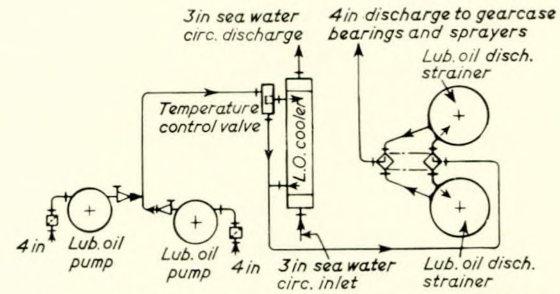


FIG. 7—*Gearcase lubricating oil system module 8000 bhp*



DIAGRAMMATIC ARRANGEMENT OF GEARCASE L.O. MODULE

Gearing Lubricating Oil System

The gearing lubricating oil system may use the same type of lubricating oil as the main engine crankcase. The two systems are, however, maintained completely separate. Although there are several advantages to mounting the gearing lubricating oil pumps, coolers and filters on the gearbox, particularly with respect to space saving, it is almost impossible to achieve a standard arrangement of these components which would satisfactorily fit on all the various gearbox configurations. The gearing lubricating oil module is therefore situated adjacent to the gearbox, with resultant maximum accessibility to gearbox components (see Fig. 7).

Two motor-driven pumps and one 100 per cent duty cooler are fitted.

A gravity tank holding six minutes supply is provided to allow for the occurrence of an electrical "blackout" by providing gearbox lubrication during the rundown of the main propeller shafting (the main engines are automatically de-clutched). The provision of a gear-driven lubricating oil pump in place of one of the motor-driven pumps would obviate the necessity for the gravity tank.

C.P. Propeller Oil System

The variations in design between manufacturers of different types of c.p. propellers have made it necessary to select one particular installation as a basis for the machinery arrangement in the ships under study. In view of the restricted space available in the vicinity of the oil distribution box, a c.p. propeller oil system module has been formed on top of the oil tank, the two motor-driven screw-type positive displacement pumps being vertically mounted on the tank with their associated filters, valves and piping connexions. The module so formed is not of the same type as the other auxiliary machinery modules described in that it is suitable only for this particular type of c.p. propeller and control system.

Oil Fuel System

Fig. 8 illustrates the basic arrangement. The main parameters are:

Maximum fuel viscosity	3500 S.R.1. at 100°F (38°C)
Settling tank temperature	90 to 110°F (32 to 43°C)
Settling tank capacity	24 hours' supply to both main engines at full power.
Temperature at purifier	Approximately 200°F (93°C).
Temperature at HFO ready-use tank	140°F (60°C) or 35°F (19°C) below flash point.
Ready-use tank capacity	12 hours supply to main engines at full power
Surcharge pump discharge pressure	50 lb/in^{2}g, temperature 245°F (118°C) after heater.
Final fuel filter size	5 microns
Temperature at engine supply manifold	245°F (118°C)

The system logically divides itself into three sections:

1) the oil transfer pumps;
2) the purifiers, heaters and sludge tank;
3) the surcharge pumps, heaters and filters.

This presented an ideal opportunity of taking advantage, with very few modifications, of the existing B.S.R.A. designs which have been developed for these modules, after a considerable amount of design work and consultation with shipowners and shipbuilders, for the slow-speed direct-coupled Diesel engine. The modules in the fuel system for the vessels in this study, therefore, are virtually identical to those suitable for the slow-speed engines of similar power. (Fig. 9, 10 and 11).

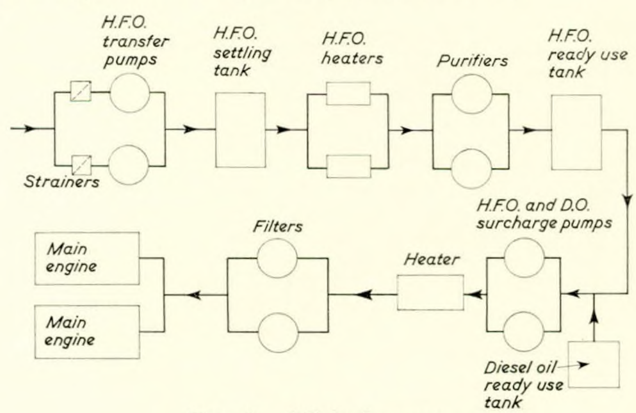


FIG. 8—*Oil fuel system*

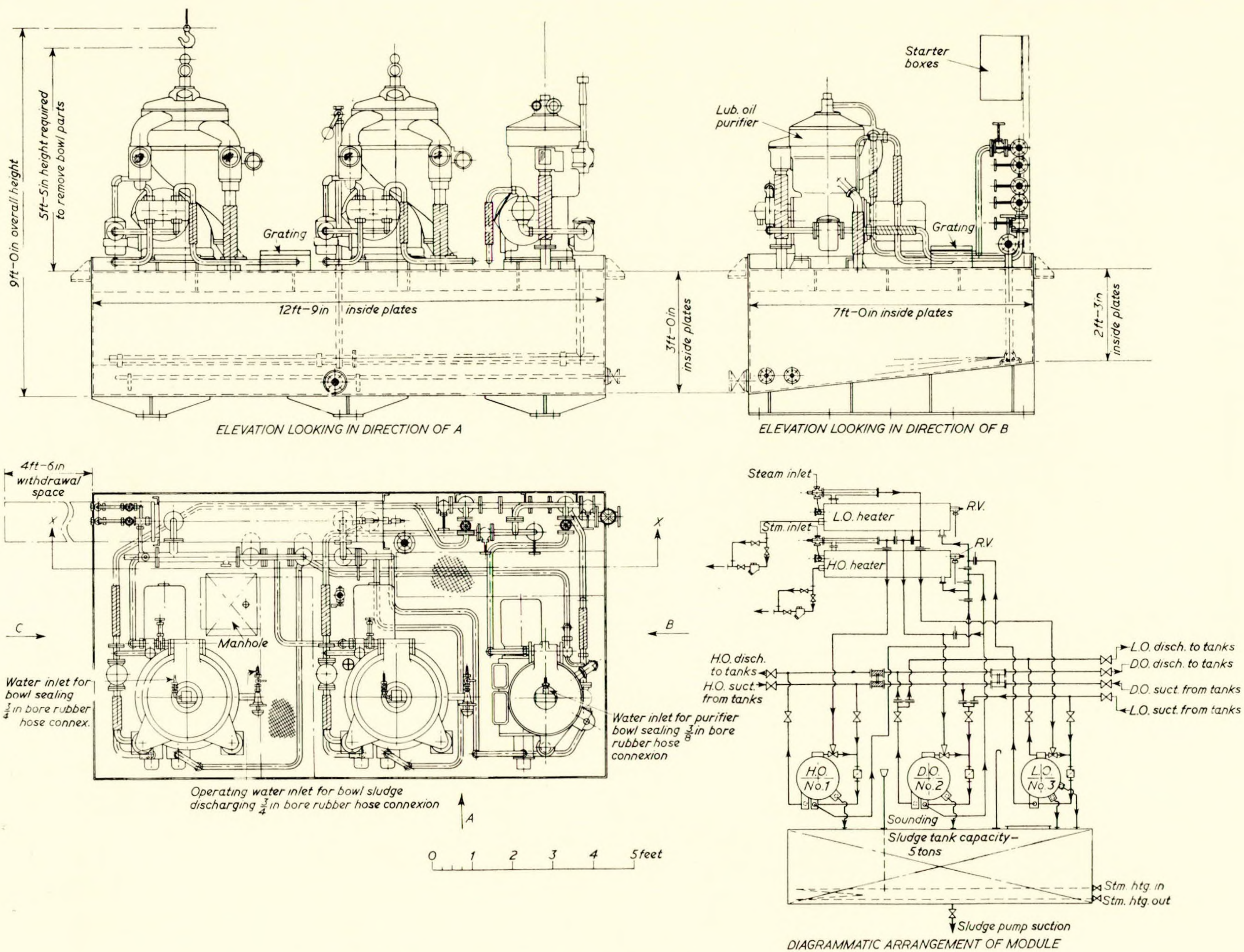


FIG. 9—*General arrangement of H.F.O., D.F.O. and L.O. purification module*

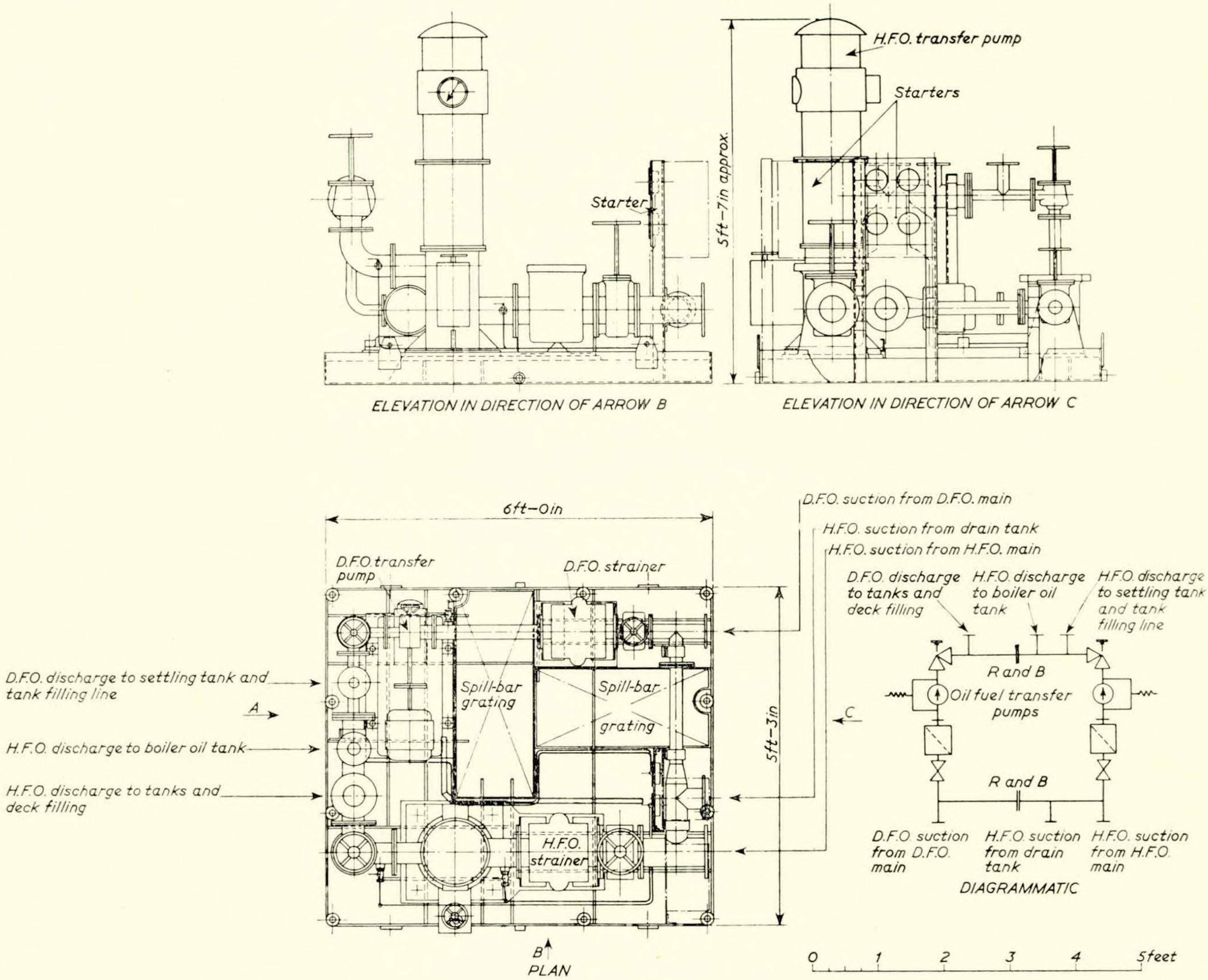


FIG. 10—*Oil fuel transfer module*

MAIN GEARING AND CLUTCHES

Gearing Designs

To ascertain the space required for the gearbox and to include the gearbox lubricating oil requirements in the auxiliary systems, gearing proposals were requested from 14 gearing manufacturers, in the U.K. and on the Continent. For the two vessels in which it was required to operate the engine-driven alternator, whether or not the respective clutch were engaged, there were, among the companies approached, a variety of proposals, including:

i) a separate power take-off box upstream of the main engine clutch;
ii) with the clutch integral with the gearbox: the power take-off is upstream of the clutch at the forward end of the box with an external shaft drive over the top of the box to the alternator;
iii) with the clutch either at the forward end or the after end of the gearbox: the drive to the alternator is by a quill shaft through the pinion and the clutch.

Alternative iii) is probably the neatest and is shown in the general arrangement sketches, but with only a few exceptions the majority of the gearing proposals received could be accommodated with little difficulty in the machinery spaces.

Clutch/Coupling Design

Friction type clutches in combination with flexible couplings of the elastic type were selected to transmit the drive from the main engines to the gearbox input pinions.

In the flexible coupling part of the particular clutch coupling sketched in the arrangement drawings, the input and output halves are connected by rubber elements. These can be tuned to suit particular installations and torsional vibration characteristics, and also allow a small degree of malalignment between engine and gearbox to be accommodated. The clutch part of the unit consists of a series of friction shoes independently mounted on the outer driving element of the clutch. Compressed air admitted to an annular flexible tube between the shoes and the outer casing presses the shoes inward to engage on the outer surface of a drum attached to the driven element of the coupling. The actuating air tube, which automatically compensates for shoe wear, operates at an air supply pressure of approximately 125 lb/in^{2}g.

The reasons for selecting friction clutches in preference to either hydraulic or electro-magnetic slip type couplings may be summarized as:

a) more compact installation;
b) avoidance of two to three per cent slip;

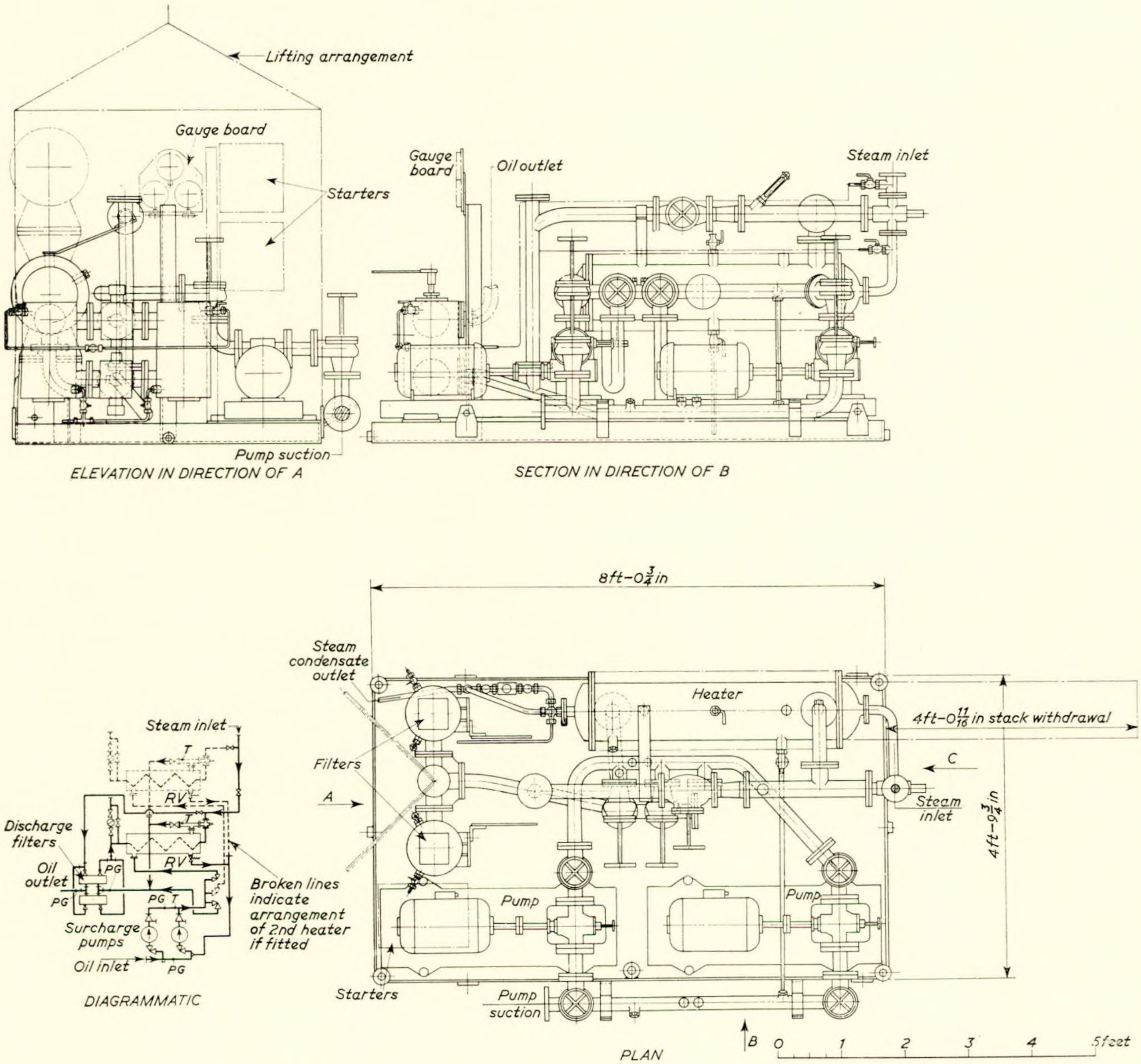


FIG. 11—*Arrangement of surcharge module with thermostatic control*

c) less weight and lower inertia;
d) ability to accommodate the quill shaft drive for the engine-driven alternator;
e) considerably cheaper.

It must be emphasized that in all three of the proposed installations, the friction clutches are required only for disconnexion of the engine from the transmission system and not for manoeuvring the vessel, since the latter is effected by operation of the c.p. propeller. There is, therefore, no question of engaging the clutches at 200 per cent slip with the attendant cooling and friction clutch material capability problems, and, although each case must be reviewed on its merits, it is not envisaged that there would be any difficulty in obtaining classification society approval.

CONTROLS

Control Medium

A detailed investigation was carried out into the design of the control systems for these vessels, which can be mentioned only briefly here. Two parallel designs were produced in conjunction with the engine manufacturers. The first was a pneumatic/hydraulic engine governing and control system of the conventional type. The second was an electronic system. Data logging was also investigated in relation to these installations but was not included as part of the control and surveillance system designs.

The electronic system, despite basic mistrust of the "black box", is gaining in popularity and is in many ways a more versatile governing and control system, the relative cost of which has progressively reduced recently.

Engine Protection

Interlocks are provided to prevent the engine from starting until each of the following conditions has been satisfied:

clutch disengaged;
engine turning gear disengaged;
auxiliary services proved;
gearbox and engine turning gears disengaged;
engine and gearbox lubricating oil pressures satisfactory.

Other protection features include:
- crankcase oil mist detection;
- termination of the starting sequence if the engine fails to fire in set time;
- engine speed reduction and clutch disengagement on low lubricating oil pressure;
- engine shutdown on extra low lubricating oil pressure, high fresh water temperature, low fresh water flow, overspeed.

Control Positions

The machinery can be operated either from:
1) an engine control station in the machinery control room;
2) a bridge control station;
3) a position local to the engines, manually.

From the machinery control room position, the following procedures can be effected:
- start/stop of one or both engines;
- engagement/disengagement of one or both engine clutches;
- selection of propeller pitch;
- selection of engine speed;
- selection of control station.

From the bridge control station, the following can be effected:
- selection of propeller pitch and engine speed;
- emergency stop of both engines.

The console in the bridge control station has single lever control of engine speed, propeller pitch and direction, whereas in the machinery control room there is single lever control for engine speed and a second lever for controlling propeller pitch and direction.

Control Sequence

Fig. 12 shows the sequence of the control functions for the power pitch control arrangements.

Overload of either engine has the effect of fining the propeller pitch automatically, to remove the overload, and load sharing arrangements are provided between the engines. In the event of one engine being shut down under full away conditions, the overload signal from that engine is isolated to prevent it fining the pitch off completely as the engine comes to rest. The other engine will reduce the pitch to an acceptable level in the normal manner.

With the hydraulic governing system, a speed droop type of governor is employed to achieve stability. The governor must, of course, be arranged to operate within the tolerances allowed by the classification societies for the provision of electrical supply frequences. With the electronic control system, two means are available for sharing the load between main engines. The first is droop governing as mentioned. The second is isochronous governing with fuel rack matching between the engines. The electronic system also offers the opportunity of incorporating start sequencing, interlocking, engine protection and monitoring into the electronic system.

Auxiliary System Controls

The system for ensuring that the rangeability required of the control valves in the auxiliary systems is not too great has been described. For the lubricating oil temperature control system and the engine cooling water temperature control system, the conventional wax element thermostat, although perfectly satisfactory for steady operation, may prove to be inadequate for the range of duty required and two term pneumatic control valves can be employed instead.

ELECTRICAL SWITCHBOARD

The switchboard, located in the machinery control room, is of modular construction, designed to enable the units to fit a wide variety of ships. This construction allows sub-switchboards in the vessel to be of the same design and layout, which is advantageous so far as operator familiarity, spares, design requirements etc., are concerned. The aim of the main switchboard is to locate as much generation and utilization features as is conveniently possible in the main switchboard, hence the use of this design is suited to control auxiliary motors also and the intention is that the motors should have thermistor protection controlling the respective moulded case circuit breaker in the main switchboard. The number of conventional motor controllers

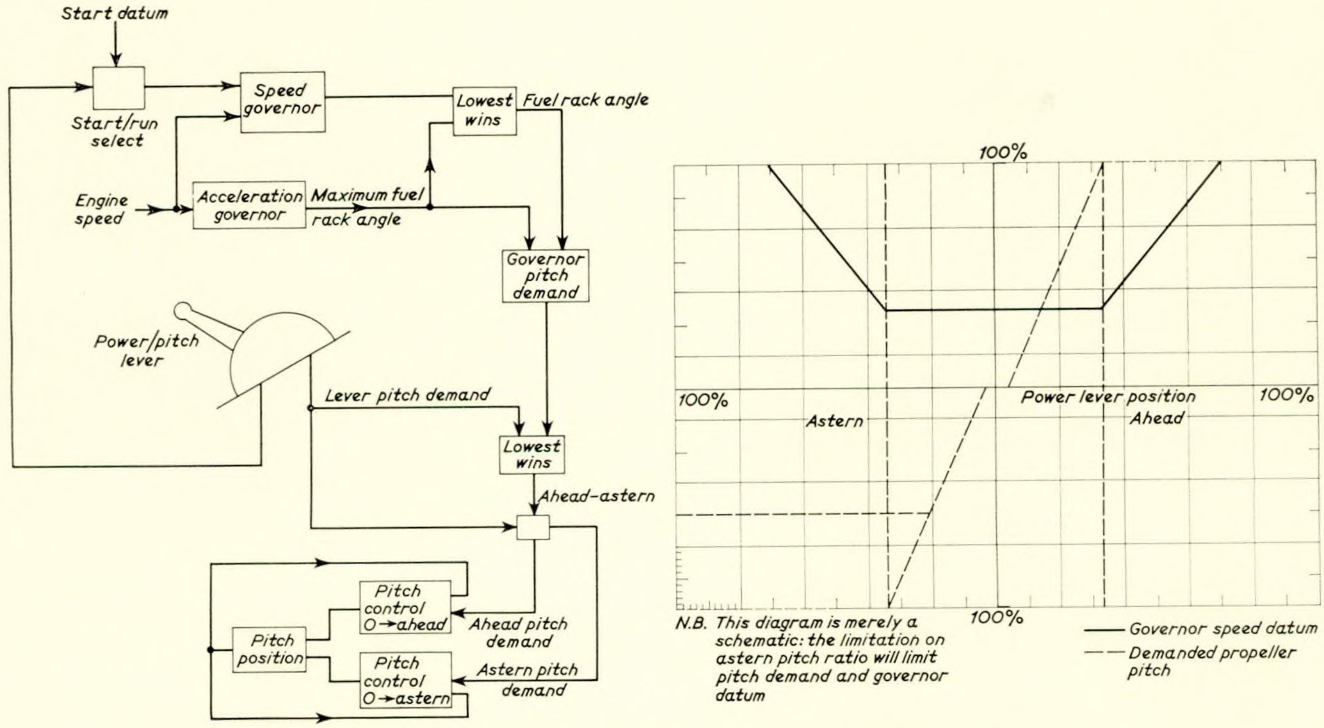


FIG. 12—*Control of propeller speed and pitch*

in a ship will thereby be reduced which, as well as improving reliability, will decrease both the initial and installation costs of the equipment. In view of the advantages of the ring main distribution system, the switchboard design is such as to permit the necessary number of these modules distributed throughout the ship to be fed directly from the main. The low voltage section for lighting on low power uses miniature circuit breakers exclusively for power distribution which, among other advantages, significantly reduces the fire risk.

FUTURE TRENDS

In Machinery Arrangement

The increasing popularity of both engine-driven and gear-driven auxiliaries has already been mentioned. In the case of the main engines, it is to be expected that the main engine lubricating oil and jacket water pumps will be engine-driven, although the case for the engine-driven salt water pump is less well established. The trend towards utilization of the gearbox for as many power off-takes as possible could result in one c.p. propeller oil pump, one gearbox lubricating oil pump and the alternator(s), all driven from the gearing. The transfer of various pumps from the relevant module to either the main engine or the gearbox obviously necessitates a modification to the module design but does not in any way invalidate the principle.

Impetus to the trend towards gear or engine-driven auxiliaries is provided by the current requirement for machinery installations suitable for regular unmanned operation. The ability of the medium-speed engine in general to start from cold without priming or prior circulation of fuel, lubricating oil or jacket water, lends itself, in conjunction with engine-driven pumps, to an extremely simple starting and operating system.

There appears to be a trend towards increased acceptance of filtration of both fuel oil and lubricating oil as an alternative to centrifugal purification. The first cost of the filter installation is considerably lower and the efficiency of the various designs of filter in removing impurities appears to be becoming increasingly accepted. It is on the question of water removal that there appear to be most doubts at present in respect of the filter installations.

In Machinery Installation and Erection Techniques

Recent strides in machinery installation and erection techniques during ship construction, particularly in Japan and Sweden, appear to have been achieved without the intermediate step of the development of auxiliary machinery modules. Although the modules are essentially building blocks which, when connected by closing lengths of piping etc., go to make up a complete machinery installation, modern shipbuilding practice—particularly in Japan—has opted for much larger building blocks, hence an entire tank top or engine room flat or bulkhead is erected in one lift already complete with pipes, machinery components, stiffeners etc., and requiring only short lengths of closing piping to integrate it with the rest of the machinery installation. There is no reason, of course, why each of these large building blocks should not be assembled from a series of modules and some interest is now being shown in this method of assembly, particularly in Sweden. The fact remains also that whether or not it is a necessary adjunct to the method of ship construction, the use of modules is an efficient means of transporting and arranging auxiliary machinery for ship construction. The way is now open, based on the guide lines evolved by this study, for the design by auxiliary machinery manufacturers of complete ranges of modules suitable for the various engine frame sizes.

CONCLUSIONS

The advantages of arranging a "package deal" for the whole machinery plant in terms of efficiency and the establishment of clear-cut lines of responsibility are evident, and a number of shipowners and shipbuilders have already capitalized on them. Even in the complex and more costly realms of warship machinery, it appears that a similar system is evolving. With the advent of the "package deal" also comes the opportunity for the main machinery contractor to develop a carefully designed and standard propulsion machinery plant which, with a minimum amount of modification, is applicable to as wide a range of ship types and power as possible. It was to lay down the ground rules for such a standard installation that the study was carried out.

The study has shown that it is possible to develop a logical basic arrangement, on which can be built an effective machinery installation, by choosing from standard series of auxiliary machinery modules.

The use of modules for certain auxiliary systems has been, accepted for some time; fuel and lubricating oil purifier modules, evaporator modules and packaged boilers are commonplace, each to individual manufacturers' designs; a complete engine services module for a number of slow speed Diesel engine types is also available, built on a one-off basis. The acceptance of a series of standard modules capable of accommodating a variety of different types of each component has however, been more limited. With the slow speed Diesel engine, this is partly due to the size of some of the auxiliaries occasionally necessitating modifications to each module to allow it to fit a particular vessel's machinery spaces, but it is also largely due to the problems of designing a module which is capable of accommodating not only a reasonable range of different manufacturers' components, but also a range of variations in operators' preferences. These preferences, built up over many years of operating this type of machinery, may vary from the number of stop valves in a sea water line to a choice between "one-cooler-or-two". Hence, although the basic module may be very simple, it has to be sized and arranged to take a multiplicity of optional extras.

Because the medium speed engine is a relatively new entry as a contender for main propulsion for ocean-going merchant ships, such preferencees may not be so well established and the consequent compactness and simplicity of the modules will, it is to be hoped, lead to their acceptance.

There can be little doubt that the medium speed geared Diesel installation will be favoured increasingly for a considerable range of ship sizes and types. Some of the initial medium speed installations suffered basically because they were unsatisfactory adaptions of established slow speed engine practice and the study described here has set out to establish the basic requirements for a rationalized installation to act as a basis for the standard propulsion package.

ACKNOWLEDGEMENTS

The author wishes to thank the Ministry of Technology and the Directors of Y·ARD Ltd. for permission to publish this information.

In addition to the many manufacturers of individual items of equipment, such as gearing, coolers, pumps, valves etc., who willingly supplied a great deal of information but who are too numerous to list here, the author also wishes to express his appreciation to:

a) the members of the Advisory Committee set up by the Ministry of Technology during the study, for their guidance;
b) the engine manufacturers, Ruston and Hornsby Ltd. and Mirrlees National Ltd., for the detailed information which they supplied;
c) B.S.R.A. for their help and co-operation in module design;
d) the shipbuilders and owners for allowing their vessels to be used as models. Lyle Shipping Company, Blue Star Line, BP Tanker Company, Lithgows Ltd., Austin and Pickersgill Ltd., and Harland and Wolff Ltd;
e) N.P.L. for their help in matching existing ships with those required for the study and for the use of their tank test results;
f) the Secretary and members of the British Marine Equipment Council for the information concerning auxiliary machinery.

The author also owes a personal debt of gratitude to a large number of Y·ARD Ltd. staff who worked on the original study, and helped in the preparation and correction of this paper, the figures and the slides.

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Appendix I

SPECIMEN POWERING CALCULATION
FOR THE TANKER FOR ONE ENGINE TYPE

Calculations were conducted to determine the required machinery power and optimnm rev/min at the given service speed, given the ship resistance data, for a range of propeller diameters for the ship. The propellers used are all Troost B series four-bladed propellers. The range of permissible propeller diameters has been chosen to lie between ⅔*D* and ¾*D* (*D*=loaded draught) and the use of a clearwater stern has been assumed with a 3 in bottom clearance between propeller tip and the moulded base line. The ship resistance data have been taken from the N.P.L. report for the ship. 25 per cent has been added to the trial ehp to allow for service conditions. The blade area ratio has been arrived at by consideration of commonly accepted cavitation criteria. Total transmission losses have been taken as three per cent, which includes a gearing loss of two per cent, the other one per cent being made up of losses at shaft bearings, stern gland, etc., i.e. dhp=0·97 bhp.

22 000 dwt (general products) tanker—
British Vine
Loaded draught 30 ft. Service speed 14·5 knots.
Thrust=66·2 tons

The electrical load for the converted *British Vine* was estimated. The steady steaming full away load is 373 kW. Therefore, assuming a generator efficiency of 0·9, the power input required by the shaft alternator will be 556 bhp.

The maximum continuous rating of the engine under consideration in accordance with BS. 649 is:
2 × Mirrlees 9K Major 8320 bhp at 525 rev/min.
Applying the shipowners' normal ten per cent de-rating, the continuous service power available is:
7488 bhp at 525 rev/min.
therefore the power available for propulsion is:
7488—556 = 6932 bhp.
Taking into account the transmission losses:
dhp = 0·97 bhp.
Using the Q.P.C. (η_D) obtained for each size of propeller:
ehp = dhp × η_D
From the N.P.L. model propulsion tests, a curve of trial ehp against ship's speed is available. The service ehp required for a given speed is 1·25 × trials ehp.
From this curve it can be shown that the ship speed for a range of propeller sizes is:

14·41 knots with a 20-ft diameter propeller;
14·51 knots with a 21-ft diameter propeller;
14·6 knots with a 22-ft diameter propeller.

Hence for the general products tanker, 2 × 9 Mirrlees K Major engines will give a service speed of 14·5 knots with a 21-ft diameter propeller (turning at 90 rev/min; this is obtained from the curve of bhp and rev/min for optimum propeller efficiency at the given ship speed).

Using fuel conforming to BSS 2869/1957 Class A of net calorific value 18 400 Btu/lb; specific fuel consumption of Mirrlees KV Major range of engines at 220 lb/in² b.m.e.p. and 525 rev/min is approximately 0·336 lb/bhp h; hence daily fuel consumption will be 27·3 tons total (25·3 for propulsion, 2·00 for electrical power); typical daily lubricating oil consumption is 46 gallons.

As stated, the fuel consumptions are based on a Class A fuel of net calorific value 18 400 Btu/lb. However, it has been assumed throughout that both engines would burn heavy fuel under full away conditions. When burning fuels other than Class A fuels, the fuel consumptions would be increased on a *pro rata* basis corresponding to the reduction in net calorific value. Thus, if a heavy fuel of, say 3500 S.R.1. (net calorific value 17 200 Btu/lb) is used, the fuel consumptions as shown should be increased by approximately seven per cent.

These calculations must only be regarded as approximate as a number of assumptions have had to be made.

Propellers commercially available, both c.p. and f.p. are manufactured in proprietary designs, which are said to give two to three per cent greater efficiencies than the Troost B series. The only series results, available in a suitable form for this study, are the Troost B series and the results therefore represent Troost B fixed pitch propellers. It has been assumed that the loss of efficiency due to the larger boss of a c.p. propeller (two to three per cent) is compensated by the use of a proprietary design of c.p. propeller.

The stern has been assumed to be of "clearwater" design rather than the traditional stern frame design on the existing ship. This should give an improved hull efficiency, but this is offset by the fact that the thrust deduction factors, hull efficiencies, wake fractions etc. used are for the existing ship design taken from the N.P.L. report, and some of these will be adversely affected, although only slightly, by the use of an increased propeller diameter.

Throughout the calculations, where assumptions, estimations or "rounding off" have been necessary, it has been the practice to be slightly pessimistic.

Propeller diameter ft	rev/min	Minimum B.A.R.	Pitch, ft	η_O	η_D	DHP	BHP	EHP
20	99·5	0·476	15·32	0·576	0·777	6830	7030	
21	90·5	0·455	16·48	0·592	0·798	6630	6840	5300
22	81·6	0·43	17·90	0·609	0·818	6470	6670	

GLOSSARY

η_O	=	propeller open water efficiency
η_D	=	propulsive efficiency (QPC)
dhp	=	horse power delivered to screw
bhp	=	horse power at engine output
ehp	=	effective horse power
Wo	=	effective work done (ehp × ship speed)
t	=	thrust deduction fraction
R	=	relative rotative efficiency
H	=	hull efficiency

Appendix II

COST COMPARISON FOR BULK CARRIER WITH C.P. OR F.P. PROPELLERS

The following is a comparison of the capital and installation costs for controllable and fixed pitch propellers for the bulk carrier used in the study:

	C.P.P.	F.P.P.
1—c.p. propeller (19 ft 6 in) complete with shafting and combinator controls:	£53 100	
Additional installation costs for c.p. propeller	£ 3 100	
1—f.p. propeller :		£17 500
Shafting for f.p. proeller:		£ 3 000
Reversing equipment for main engines, 2 sets:		£ 4 400
Remote control system for main engines:		£ 8 300
Increased compressor and air reservoir capacity:		£ 1 000
3—350 kW Diesel alternators, 1200 rev/min at £11 370 each, complete with all ancillary equipment:		£34 110
Fitting on board for above:		£ 1 000
1—shaft-driven 500 kW × 1200 rev/min alternator:	£ 3 000	
1—set step-up gearing for above:	£ 3 000	
2—Diesel alternators 350 kW at 1200 rev/min complete with all ancillary equipment:	£22 740	
Fitting on board for above:	£ 900	
Total:	£85 840	£69 410

Differential=£16 430

Discussion

COMMANDER E. TYRRELL, R.N. (Member) said it gave him very great pleasure to open the discussion on the paper and he felt sure that the clarity with which it had been presented would contribute greatly to the resolution of many of the problems which were and would be associated with the installation of medium speed Diesel engines in ocean-going ships.

The Ministry of Technology was particularly pleased to be able to give permission for the publication of the material which was the result of two contracts placed with Y.ARD, as the prime, and B.S.R.A. as secondary, contractor; the results of the detailed study had already been made available over the last year to the British shipowners and shipbuilders on a confidential basis.

The author had drawn attention to and had added to what was probably the most important piece of information to be found in Table I, which clearly showed the growth in popularity of the medium speed engine and the decline in installed horsepower of the slow speed, direct coupled engine. It must be expected that the trend would continue as the author had shown, it was doing, with further information. Commander Tyrrell questioned, however, the information given on the percentage of ships fitted with medium speed engines, showing that in the United Kingdom it had been constant at about one-third of the ships for some time; that percentage, he thought, would, in fact, be found to be higher and still growing in a number of Continental countries who competed with the U.K. in shipbuilding.

When the trend of the growth in the popularity of the medium speed engine was less apparent than now, the Ministry of Technology decided, against the opposition of various interests, to investigate problems associated with medium speed engines, and the results of two of those studies had been made known. A third study was now under way and, in due course, would be made available at the Institute, if the Council would permit it.

The paper was not a comparison between medium speed and slow speed engines in the chosen vessels. This case had already been examined by Neumann and Carr*. The author in particular emphasized the advantages of modular construction in the auxiliary systems. The paper must be of great interest to all those engaged in designing the most suitable and economic ship for a particular purpose.

The Ministry of Technology had commissioned another study on the use of medium speed engines as opposed to slow speed engines for a 24 000 ton product carrier tanker. The basis of comparison on which the study was carried out was a B and W 6–74 VTF 2 BF engine producing 990 bhp at 119 rev/min, already fitted in a number of ships of that type. With a propeller diameter of 21 ft and a running speed of 83 rev/min as opposed to the 18 ft 6 in diameter propeller running at 115 rev/min fitted with the slow speed engine, the propulsion horsepower required was reduced to 6600 hp. An allowance was made of $2\frac{1}{2}$ per cent for the stern tube and bearing losses and an allowance for gear

* Neumann, J., and Carr, J. 1967. "The Use of Medium-speed Geared Diesel Engines for Ocean-going Merchant Ship Propulsion". *Trans. I.Mar.E.*, Vol. 79, p. 89.

efficiency of 97·5 per cent. The engine power required was 6950 hp.

A very extensive discussion took place with a major tanker owner on the derating which he would require for continuous service and it was agreed that the engine should be derated by ten per cent to give the continuous service rating. This was generally in line with common practice and gave an allowance to cover fouling of the turbocharger and unequal power distribution in the cylinders. The engines were also derated to BS 649 for sea water temperature of 93°F (34°C), and this amounted to a further three per cent derating for tropical conditions.

The horsepower required for medium speed Diesel engines to give the same speed, depending on whether an engine driven alternator was used or not, was in general a little over 8000 hp, as opposed to 9900 hp for slow speed engines. A number of installations ranging from a single engine of something over 8000 hp to two engines of between 4000 and 4500 hp each were investigated technically and economically in detail. The cost differential varied from a saving of £121 000 for a single medium speed engine installation to £23 000 for a controllable pitch propeller installation (see Table IV).

TABLE I

Item	Basis Machinery Installation	Single engine geared machinery installation		Twin engine geared machinery installations							
				1		2		3		4	
		16AO Ruston	18K Mirrlees	8AO Ruston	9K Mirrlees	8AO Ruston	9K Mirrlees	8AO Ruston	9K Mirrlees	8AO Ruston	9K Mirrlees
Cargo oil pumps	Steam	Steam		Electric		Electric		Steam		Steam	
Couplings engine/gear-box	Solid	Holset/Fawick		Holset/Fawick		Holset/Fawick		Holset/Fawick		Holset/Fawick	
Boiler, watertube	1 at 55 000 lb/h	1 at 55 000 lb/h		—		—		1 at 55 000 lb/h		1 at 55 000 lb/h	
do. Package	1 at 3000 lb/h	1 at 3000 lb/h		1 at 20 000 lb/h 1 at 10 000 lb/h		2 at 10 000 lb/h ea.		1 at 3000 lb/h		1 at 3000 lb/h	
do. waste heat	1 at 5850 lb/h	4400 lb/h	5550 lb/h	4400 lb/h	5550 lb/h	2000 lb/h		4400 lb/h	5550 lb/h	4400 lb/h	5550 lb/h
Generators Diesel	1 at 500 kW	1 at 500 kW		1 at 500 kW		2 at 650 kW ea.		1 at 500 kW		1 at 500 kW	
do. turbo	1 at 500 kW	1 at 500 kW		1 at 1250 kW		—		1 at 500 kW		1 at 500 kW	
do. engine driven	—	—	—	2 at 1250 kW ea.		2 at 1250 kW		—		—	
do. gear driven	—	—	—	—		—		—		1 at 500 kW	
Propeller type	Fixed pitch	Fixed pitch		Fixed pitch		Controllable pitch		Fixed pitch		Controllable pitch	
Capital cost differential £ A		121 000	91 500	62 300	56 200	47 000	43 500	87 500	82 700	25 700	23 400
Lubricating oil cost/year £ B	2000	3600	3800	4400	4300	4400	4300	4200	4100	4200	4100
Fuel cost/year £ C	68 600	67 050	64 000	63 000	61 250	62 300	61 900	67 040	64 000	65 610	62 600
Maintenance cost/year £ D	2740	4940	5440	4940	5440	4940	5440	4940	5440	4940	5440
10% Interest on capital, £	Basis	−12 100	−9150	−6230	−5620	−4700	−4350	−8750	−8270	−2570	−2340
E 5% Depreciation on capital, £	Basis	−6050	−4080	−3120	−2810	−2350	−2180	−4380	−4140	−1290	−1170
A+B+C+D+E, £	73 340	57 440	60 010	62 990	62 560	64 590	65 110	63 050	61 130	70 890	68 630
Annual operating cost differential, £		−15 900	−13 330	−10 350	−10 780	−8750	−8230	−10 290	−12 210	−2450	−4710

Data:

Gearing:	Renk co-axial	
Engines: basis—	B. and W. 6-74-VT2BF	9900 bhp mcr at 119 rev/min
Single geared—	Ruston 16AO	8000 do. do. at 450 do.
	Mirrlees 18 KVDMR	8320 do. do. at 525 do.
Twin geared—	Ruston 8AO	2×4000 do. do. at 450 do.
	or Ruston 9AO	2×4500 do. do. at 450 do.
	Mirrlees 9KDMR	2×4160 do. do. at 525 do.

Fuels costs: Based on Diesel oil at 255/- a ton; heavy oil at 178/- a ton (1500SRI)

Propeller: Basis— 18 ft 6 in Diameter running at 115 rev/min;

geared Diesel— 21 ft 0 in Diameter running at 115 rev/min.

Maintenance Costs based on a rate of 30/- per man hour.

Cargo capacity increased by 170 tons with geared Diesel installations.

Diesel alternators:

Paxman 12 YLCZ 1250 kW at 900 rev/min.

Paxman 12 YJXZ 650 kW at 1200 rev/min.

Paxman 12 RPHCZ 500 kW at 1200 rev/min.

Paxman 6 RPHCZ 240 kW at 1200 rev/min.

It was hoped that British shipowners would note the lower horsepower possible with the medium speed Diesel installation due to the improved propulsion efficiency gained by using a larger propeller operating at lower revolutions. Some shipbuilders were asked recently to tender for a medium speed engine as an alternative to the slow speed engine—they proposed to fit medium speed engines of considerably higher horsepowers than calculations indicated were necessary. Unless realistic assessments of the type he had shown were made, British shipbuilders would stand a very grave chance of losing contracts to their foreign competitors.

An important problem associated with the use of the medium speed engine was to be found in how the engine could best be reserved. Investigations showed that although the fixed pitch propeller and direct reversing engine driven through an air operated friction clutch provided the cheapest arrangement, the basic disadvantage of that arrangement was in manoeuvring. The most convenient and technically satisfactory reversing arrangement was the controllable pitch propeller, but it was more expensive. If there were a breakdown in the controllable pitch propeller mechanism, the blades were usually designed to go into full ahead pitch, when once again the difficulties of starting and reversing engines were obtained. However, it had been shown that if the electrical load on the ship was high and arrangements were made to drive a high speed alternator from the gear-box, the saving in cost and installation of auxiliary Diesel generators, and the saving gained from the generation of electric power from heavy fuel, could well pay for the extra cost of a controllable pitch propeller.

In view of the difficulties attendant on manoeuvring with the medium speed engine installation, the Ministry had found it prudent to make yet another study of other ways of dealing with those problems. Technically a very satisfactory method of obtaining reverse power was to use the reversing engine driving through a fluid coupling. The conventional fluid coupling, however, had the disadvantage that it was large and expensive, if the slip losses were to be cut down, but even so these were, in general, not much less than three per cent which increased fuel consumption by that amount. There was at present under development a fluid coupling which operated with a slip of about seven per cent. This had the effect of reducing price and size; but it could be locked solid and operated as a friction clutch when the propeller speed was 90 per cent or more of the normal operating speed. The slip losses occurred only when the ship was manoeuvring or operating at very low speed. The capital cost of such an installation seemed to be about mid-way between that of a controllable pitch propeller and the comparable friction clutch arrangement, and about the same or possibly just slightly more than that of the fluid coupling. Operating costs showed a considerable saving over the normal fluid coupling arrangement and it was comparable with, and only very slightly more (due to using more fuel when manoeuvring) than if the controllable pitch or friction clutch installation were used. There was little doubt that shipowners would ponder whether ease of manoeuvring with that slightly more expensive arrangement than the straight friction clutch was justified. However, the position was becoming clear that for a modest sum one could achieve much more satisfactory reversing arrangements than with the air operated friction clutch. The Ministry would make the results of the further studies available to British shipowners and shipbuilders and, it was hoped, to the Institute. He apologized to the author for providing extra information relative to the paper rather than commenting on it in detail, but expressed the hope that in so doing he had forestalled criticism in that a number of points which had been thrown up as the result of study and experience and not foreseen when the original contract was placed had now been covered.

Lieutenant A. M. Scrimgeour, R.N. (Associate Member) said that he had studied the paper and had listened to the presentation, in the role not only of a marine engineer but also an interested taxpayer. In the latter capacity, it was one's duty to assess just what the new super Ministry of Technology, using taxpayers' money, was in fact gaining from the Yarrow-Admiralty Research Department. He was at a loss to decide precisely what the Ministry was seeking from its Consultants, hence appreciation of the value of the paper was difficult.

The author defined his objectives as "to discover and examine the problems of the auxiliary machinery design in vessels for which medium speed geared Diesel engines had already been chosen", but concluded that his paper "has set out to establish the basic requirements for a rationalized installation to act as a basis for the standard propulsion package". Although complementary it must be admitted that they were different propositions. Mr. Scrimgeour said he also believed that many of the problems highlighted stemmed from the author's own proposals. Having thus, it was hoped, not over rudely and belligerently nailed his colours to the mast, he wished to take issue with the author over what were felt to be serious misconceptions and omissions from the paper. He had restricted himself to ten items, roughly in order of their appearance in the paper and not of relative importance.

The few module designs illustrated, in spite of the assurance of their being designed to facilitate *in situ* maintenance, appeared from accessibility considerations extremely cramped. He referred particularly to Fig. 4.

The overall advantage of well designed modules was agreed, but although the reasons given might not necessarily be listed in order of importance, to stress and indeed re-stress the attraction of reducing the number of crane lifts—after all, operations taking only minutes compared with possibly hours, accumulating to days, of frustration working on inaccessible machinery items during the ship's subsequent operational life—seemed a bit far fetched.

The sequence of work mentioned incorporation of module designs into detailed machinery arrangements. One was shown only one sketch plan at one level for each ship. To appreciate, to any useful degree, any proposed installation, one would expect at least plans at the various levels, elevations to port and starboard, and typical sections looking forward and aft.

Reference to the sketch plans revealed little groups of items including fresh water pumps, sanitary pumps, boiler feed pumps, auxiliary circulating and air conditioning pumps, and a Butterworth heater and associated pump. Surely they were all auxiliaries ideally suited for "modulation"?

Again probably dealt with in arrangement drawings not included, Diesel generator sets, steam raising plant, refrigerating machinery, starting compressors and their associated air reservoirs and, for these up-to-date vessels, even sewage systems, all of them potentially in the single lift module class—the siting of which was of prime importance in the production of a well laid out installation—were absent.

In the alternator drive proposals examined, the main disadvantage of siting engine driven units ahead of the main machinery was given as the consequent lengthening of the machinery space. That was precisely what would appear to have resulted from the proposed siting of the main engine services module.

Sea-going engineers and superintendents, one felt certain, would be rather taken aback to say the least at the specified design aim for a bulk carrier to "provide maximum utilization of the main engines in port".

There were inconsistencies in the attitude to finance where costly hydraulically driven cargo pumps were mentioned, admittedly only as a possible alternative, but two motorized suction and discharge lubricating oil pump valves were ruled out on price and yet again a fairly involved thermostatically controlled sea water service re-circulatory system was advocated. The necessity to reckon on running generator engines on Diesel fuel, although advantageous to the economic reasoning, was questionable.

The argument for c.p. propellers lost its force when pneumatically controlled clutches were proposed. A relatively minor additional complication, but logical extension of the case for the latter, was to include reverse reduction pneumatically controlled gear-boxes. That was not advocated from any conservative anti-c.p. propeller standpoint, but from recent personal experience of just such an installation which proved straightforward to maintain and simple to operate. That also eliminated the rather specious cause of number of engine starts influencing starting compressor sizing.

Control system proposals were confusing. Assuming the use of pneumatically controlled clutches and noting also the probable necessity for pneumatic control valves on the auxiliary systems, an electronic control system was proposed for the main engines. The resultant differing media facing the operators and maintainers of those ships was quite a mixture.

As a parting shot following that final point, the complete absence of any reference to the requirements and conditions affecting the engine room staff of these ships was noted. That was an all too prevalent omission of would-be designers and critics of installations. It was well known that with air conditioned machinery control rooms and watchkeeper-less installations, much of the drudgery had gone, but the paramount necessity for ships' engineers to have simple, robust, accessible and hence reliable main propulsion and auxiliary machinery, became of ever increasing importance as the shipowners' "units of investment" decreased in number, but increased in size and/or value at such an unprecedented rate.

Mr. A. Norris (Member) agreeing with the general principles laid down in the paper, said that it was logical that smaller, lighter and more highly rated machinery should be used for ship propulsion, and it was rational that the potential of the propulsion plant should be exploited to supply ancillary and ships' services loads. The auxilliary machinery module concept was desirable since it should enable installation costs to be reduced and the quality of work improved. Development along the lines indicated was inevitable, but the human resistance to acceptance of change might make it rather later than some would like.

The case for controllable pitch propellers was well presented and their use was technically desirable in cases where an engine or gear driven alternator was employed. That was, of course, due to the constant shaft speed which could be maintained allowing the electrical supply frequency to be kept steady. However, the economic case could not always be proved for general purpose ships similar to the tanker dealt with in the paper. Even if the auxiliary powering arrangements permitted the differential cost to be reduced to the £16 430 given, in Appendix II, for the bulk carrier, it might still be necessary to demonstrate that the operational savings justified the increased initial cost and the higher propeller maintenance cost which would probably follow use of the more complex arrangement.

With regard to gearing lubrication oil systems, the author mentioned that the main engines were to be automatically declutched in the event of an electrical blackout and lubricating oil might be supplied during the run-down period from a gravity tank or from a gear driven pump. Machinery was at hazard under such conditions, and there had been cases where a steam turbine plant had been allowed to run down in protracted blackout conditions and bearing damage had followed exhaustion of the gravity tank supply. In an investigation carried out some months ago, his company arranged for steam to be shut off from the turbines in a large ship to check on the run-down time. It was found that from an initial 70 rev/min the shaft continued to rotate for 45 minutes. Even if a gear driven pump were provided, it would have a difficult duty during the protracted stage of run-down when shaft speed was low. Had the merit of declutching Diesels under protracted blackout conditions been demonstrated in practice in cases where shaft brakes had not been fitted?

It was mentioned that the main machinery weight was reduced by about 50 per cent by the use of medium speed engines. It would probably follow from that, that the total weight of the machinery installation would be reduced by about one-third of that of the slow speed Diesel installation. As that overall machinery weight was an important consideration in many cases, it would be of interest if the weight saving in tons could be given.

The author had commented on the use of waste heat recovery plant in conjunction with a turbo-alternator for the fixed pitch propeller installations and had given reasons why the study was not prosecuted along those lines. The low main engine power required for two of the ships considered and the small number of days at sea probable for the cargo liner could, in any case, be expected to make the auxiliary powering decision marginal. The decision taken for that study should not be accepted as being gospel, however, for all ships in those power ranges. The determining factors for decisions were always the differential capital cost, the main engine power and, hence, the auxiliary loading, the number of days at sea per annum, and the maintenance work load and cost of other forms of power generation. The steam turbine attributes of low maintenance following long periods of continuous operation at sea and in port, and the ability to carry the maximum rated load without fuss or bother should not be overlooked. In the many slow speed Diesel engined ships in service in Mr. Norris's company, in which turbo-alternators were used, the machines were expected to operate from dry dock to dry dock apart from any short stops which might be necessary for unscheduled maintenance work.

One would have thought that the electric generating plant in the cargo liner would have justified some detailed scrutiny in view of the considerable maintenance load which would be imposed by use of Diesel engines to drive the alternators in port. The suggestion was made in the paper that one of the main engines should supply the power during cargo discharge. That would involve building up operating hours on a large engine instead of on units which could be overhauled on sea passages if necessary; the author might care to comment on the strength of the case behind the proposal.

The paper referred to power take-off connexions from the gearing for generator and pump drives, and it followed that power from prime movers other than the Diesel engines could be fed back through the gearing to the propeller shaft. Fig. 13 illustrated a simple scheme which would enable the all-purpose fuel consumption of a 20 000 hp medium speed engined installation to be reduced by over eight per cent in addition to supplying normal electrical and heating loads. The scheme involved use of near maximum heat recovery from the exhaust gases and a steam turbine which could supply power to both an alternator and the main engine gear-box. The system was straightforward and only established engineering practices were followed.

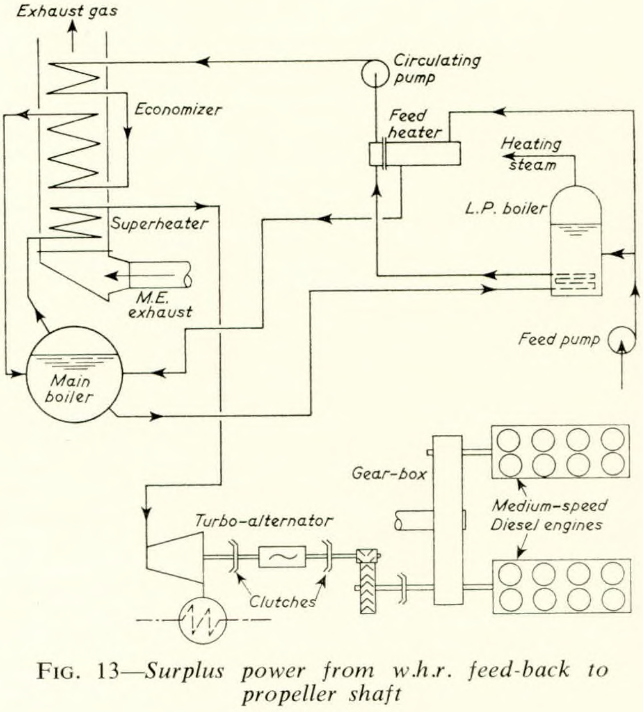

Fig. 13—*Surplus power from w.h.r. feed-back to propeller shaft*

He had made a first order estimate of applying the principle to the liner installation described in the paper, assuming that about 15 000 bhp was required for propulsion and electrical loads. If the Diesels were operated at 14 000 bhp it would be possible to provide an additional 1250 bhp from a waste heat recovery installation and, if the ship were at sea for 200 days per annum, the annual fuel bill reduction could exceed £6000. A saving of that magnitude over the ship's life could justify an increase of initial investment of over £50 000 if investment grant were taken into account. In addition it could allow the number of Diesel engine cylinders to be reduced with some engine configurations, which would also reduce the cost of the initial medium speed installation. The lower cost fixed pitch propeller could also be used.

While he would not expect that such considerations were included in the design study covered by the paper, he would suggest that the potential of medium speed engines for the higher power ranges could be further exploited by integrated schemes along those lines.

Mr. E. C. Brook said that the manufacturers of medium speed engines had been supplying machinery packages, in one form or another, to both the industrial and the marine market for a considerable number of years. They were thus generally well equipped to cope with the demand for those packages, which had, to date, come especially from the smaller shipbuilders, where technical manpower was limited. There was, however, little standardization, either in the extent of the package supplied or in the specification of the equipments making up that package.

The extent of the package was usually dictated by the shipbuilder, who presumably weighed the advantages of limiting his personal technical responsibility against the financial savings and more direct control which he might achieve by ordering direct from the manufacturer. Thus, the extent of the engine builder's supply might range from little more than the bare engine to a complete package including engines, gear-box, propellers, generators, complete control systems, purification equipment, air receivers, Diesel generating sets, waste heat boilers and silencers, and of course all the ancillary heating, cooling and filtering equipment associated with those items. There was little hope of any standardization, of the sort envisaged in the paper, while that variation existed and, in order that both the shipbuilder and engine builder could plan their future technical manpower requirements, it was highly desirable that there should be some general agreement on the normal extent of the engine builder's supply. Once such an agreement was reached, it would become possible to consider the question of standardization of the equipments making up the package.

The author had suggested the role of the engine builder as main machinery contractor, supplying all the machinery necessary to propel the ship. That did, however, raise a number of problems. How far was the shipbuilder willing or even able to permit the amount of work carried out by the engine builder's staff, actually on board the ship, to be increased? At the moment the engine builder's role was usually limited to supervision of the installation of his machinery with the shipyard personnel carrying out the work. Clearly, the responsibility of the main machinery contractor must be limited by the control which he possessed over the whole of the installation.

It had been suggested that the engine builder be responsible for the supply of the main shafting and propeller. Although those items had sometimes been supplied by the engine builder, there were a number of drawbacks to such an arrangement. Firstly, the engine builder could not be responsible for the hydrodynamic performance of the propeller. That was the joint responsibility of the naval architect and the propeller maker. Secondly, the type of propeller attachment, stern gland and shaft bearings employed were matters for the shipbuilder, in consultation with the shipowner, to decide. Thirdly, the number and position of the shaft bearings must be decided from considerations of shaft deflexion and that was more within the province of the shipbuilder than the engine builder.

It was true that the engine builder, through his responsibility for torsional vibration calculations, might influence the diameters of the shafting and that, in the case of controllable pitch propellers, he became involved in the control aspects, but it was not thought that those factors outweighed the disadvantages given.

The normal extent of the engine builder's package should be based on those items which directly influenced the performance of the main engines or whose specification was mainly dependent on that performance. Items which were specified mainly by the type and performance of the ship as a whole would be excluded. Thus the package would include, besides the main engines themselves, the couplings, clutches and gear-box, the main engine driven generators, the control, instrumentation and alarm systems, the lubricating oil and cooling systems and the final fuel boost and filtration system. In addition, since many of the main engine builders also produced smaller engines, it would be appropriate to include the Diesel generating sets, when required, thus providing a complete power production package. The shipbuilder might also find it advantageous to order the switchboard and control equipment associated with the generating sets in the same package. The medium speed engine builder had close associations with electrical equipment suppliers, who were often within the same group, and was used to handling such items in the contracts he carried out for large industrial power stations.

If the philosophy outlined were accepted, the question of who made which module would be automatically resolved. Taking the module designs detailed in the paper, for example, the engine builder would be responsible for the main engine service module, the gearcase lubricating oil system module and the fuel surcharge module, while the shipbuilder would supply the fire, bilge and general service pump module, the ballast pump module, the purification module and the oil fuel transfer module. The acceptance of the module principle depended largely of course on the shipbuilders, who would be influenced by the views of the shipowners. If full benefits of modular construction were to be experienced, however, it was imperative that there should be quantity production. It was no use adopting the module principle for one ship and then reverting to traditional methods for the next.

Of the module designs illustrated in the paper, clearly the main engine services module (Fig. 4) was the most important, in that it represented a logical development of the module principle to maximize the advantages listed. Unfortunately, in order to obtain those advantages it was necessary to lose, almost completely, the flexibility which was inherent in the established layouts of ancillary equipment. That loss of flexibility applied whenever the modular principle was accepted, but it increased greatly with the size and complexity of the particular module. In the case of that module, the shipbuilder must accept a unit of size comparable to that of the main engine, resulting in virtually the same limitations on its location within the engine room.

When considering the basic machinery arrangements chosen for the study and the reasons for the selection of the controllable pitch propellers, it might be of interest to note that, of the 19 ships in service or on order, propelled by Ruston AO engines, no less than 16 had c.p. propellers and, of those, only one was a ferry. Twelve of the vessels would have main engine driven generators, thus showing the acceptance of that principle by shipowners.

The question of whether to drive the generator from the gearing or from the forward end of the engine was important. In cases where the generator was not required in harbour, the drive from the gearbox (Fig. 3) was an automatic choice. Where that was not the case, however, the drive from the gear-box led to a fair degree of additional complication. A wide variety of different arrangements were possible, depending on the type and location of the clutches employed, and the arrangement selected would probably be mostly dependent on the views of the particular gear designer.

A recent study carried out by his company showed that an arrangement with two 1100 kW generators driven from the forward end of the engines cost £4000 more and was four feet longer than an arrangement similar to Fig. 2.

As engine builders, his company would not normally expect the main engine driven generators to be used to produce the basic harbour electrical supply, as might be implied by the final paragraph under *Controllable versus Fixed Pitch Propellers*. It was more economical to use the Diesel generating sets for that duty. The main engine driven generators did, however, provide a standby facility which justified the installation of comparatively few conventional generating sets.

MR. B. E. WELBOURN (Associate Member) said that there was no doubt that the technique in application of clutch couplings was increasing, but it certainly would not be correct to assume that it was widely accepted or that there was a full understanding of the improvement in performance which could be offered by application of a friction clutch. This is clearly demonstrated when shipowners persisted in specifying slip type couplings for the transmission, regardless of overall cost or technical limitations. This practice made it increasingly difficult for the engine builders to adopt the package deal, since interchangeability, between slip couplings and friction clutches, was extremely difficult.

On the basis of past experience, one had to accept that the slip coupling was capable of providing maximum torsional isolation between the prime mover and propulsion system, in addition to the facility of control during manoeuvring sequences. However, the natural movement towards highly rated engines of relatively light construction had introduced new problems, in many cases exaggerated by application of such couplings.

Appreciation of these problems by the designers of flexible transmission couplings, had resulted in introduction of new product ranges, the designs of which were able to meet the technical requirements of both transmission contractors and classification inspection authorities.

This had quite naturally led to design integration of transmission couplings with friction clutches.

The author had made specific mention of the concept of clutch couplings being adopted for a transmission system, incorporating two engines through reduction gearing to a c.p. propeller.

In fact, this was an extremely easy duty for the clutch, since it was required for disconnexion duties only, and limited slip only occurred at the clutch as a by-product of the engagement cycle.

However, with most types of air operated clutches, the engagement cycle could be effectively extended to suit the specific installation, with corresponding increase in the efficiency of the manoeuvring operation. This facility had already been used to advantage in that engagement cycles of the order of eight to ten seconds were able to provide:

i) minimal shock torque to the gearing system on pick-up of the propulsion train;
ii) a torque load consistent with the specific stall characteristics of the prime mover.

Clutch engagement was usually programmed with minimum pitch setting at the propeller and the engine speed set at idling; interlock could be provided between the fuel racks of the Diesel engine and the clutch control circuit, so that propeller speed and pitch could not be increased until the engagement cycle was completed.

Further advantage could be derived by utilizing the clutch as an overload protection device for the reduction gearing, this being a particularly useful facility for vessels with unmanned engine rooms or for vessels subjected to navigation in ice conditions. This facility could be provided in several ways, but in all cases the end result would be disconnexion of the clutch on receipt of an appropriate signal to the air control system:

a) signal dependent on torsional deflexion at the flexible coupling;
b) signal dependent on "break away" or slip torque of the clutch;
c) signal dependent on fuel rack setting at the Diesel engine.

In the majority of cases, the latter was considered to be the most reliable method.

Finally, with the advent of Diesel engines developing powers in excess of 500 hp/cylinder, there was evidence that there might be a significant change in the approach to clutch technique for propulsion systems. Certainly, where twin engine installations were to be considered, there was an increasing problem regarding the magnitude of misalignment which could be experienced between prime mover and reduction gearing.

The concept of clutch couplings might not provide the best solution to this problem and, in the event of the package deal being extended to this size of transmission system, considerable improvement might be added by consideration of quill shaft mounted clutches at the aft end of the reduction gearing.

This form of application would still require a flexible coupling between engine and reduction gearing, with part weight of the quill shaft supported by the coupling. An aft end bearing between quill and pinion shaft supported the remaining mass of the quill shaft, in addition to providing a fulcrum point adjacent to the clutch position. It was then designed, that affects of subsequent misalignment were compensated by bending in the quill shaft and that torsional vibration characteristics were satisfied by combined use of the elastic properties of couplings and quill shaft.

This technique was relatively new, particularly when considering it in relation to transmission of this size. However, present navigation indicated that resultant stress in the quill shaft would be acceptable to classification inspection authorities and that angular misalignment at the clutch would be sufficiently small as to give a life at the friction linings of the clutch, consistent with normal marine requirements.

ENGINEER LIEUTENANT P. A. KNOWLES, R.N. (Associate Member) said that the author had mentioned the trend towards increased acceptance of filtration of both fuel and lubricating oil as an alternative to centrifugal purification.

The module technique applied to filtration had also been raised by Budd and Wilkinson* and in discussion on that paper Shilp advised that it had been aptly carried out at least as far as lubricating oil was concerned.

The units referred to were the high efficiency streamline filters using stacked disc edge filtration and vacuum dehydration. The module was complete with vacuum, pumping and heating equipment and was capable of continuous conditioning of the lubricating oil on a bypass basis (immediate protection of bearings and gears was maintained by in-line full flow filters). However that method of purification, although very successful with straight lubricating oils, had not yet been adapted for the new high dispersent oils as used in Diesel engines where a large amount of carbon was held in suspension. Nevertheless, the example illustrated at that least one manufacturer had been aware of the need for filter modules capable of running unattended for long periods and had satisfied that need on certain applications.

With regard to fuel purification, a filter module for heavy fuel (1000 gal/h) had been developed which also operated continuously without attention (see Fig. 14). It consisted of a pump, with provision for a standby pump, heater, primary filter/dehydrator, Duplex filter and control panel.

The overall measurements were 7 ft 9 in by 5 ft with a height of only 4 ft 6 in. The primary filter carried out a double function of removing large contaminant and stripping out—not coalescing—the water. It automatically cleared itself, without interruption of flow, as the pressure drop across the elements built up to a predetermined figure which could be adjusted to suit the application. The reduction in contaminant level achieved by that primary filter served to prolong life of the secondary Duplex filter which controlled the ultimate level of cleanliness of the fuel being pumped to the engines.

The Duplex filter elements which were made from Vaflon synthetic fibre, had already been proved over millions of hours use all over the world on heavy fuel applications. It should be noted that the elements were always positioned after the purifiers or separators, leaving no doubt that the present module would give a satisfactory performance in terms of removal of particular contaminant. The new module had in fact completed six months

* Budd, E. B. and Wilkinson, H. C. 1968. "Modular Design of Marine Auxiliaries" *Trans. I.Mar.E.*, Vol. 80, pp. 301-313.

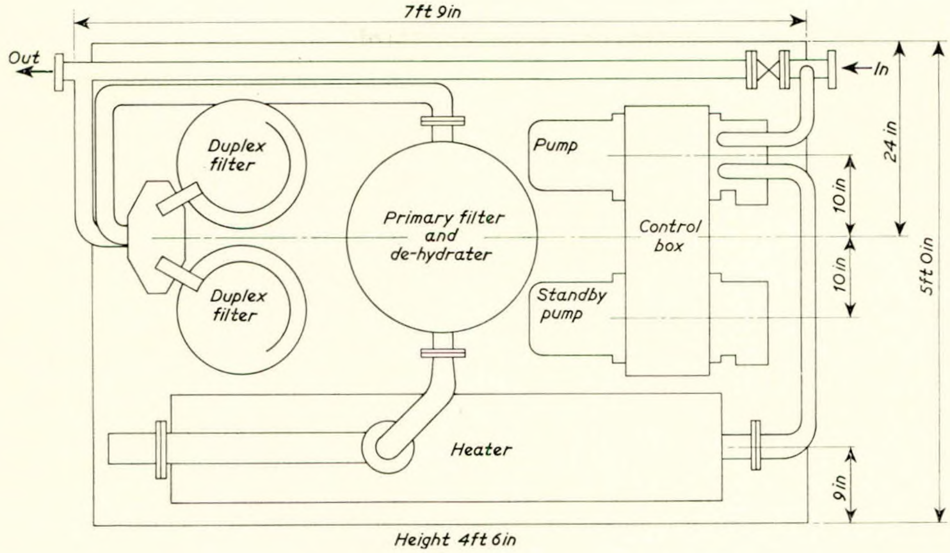

FIG. 14—*Vokes 1000 gal/h, heavy fuel oil filter/dehydrater module—Weight two tons approximately*

continuous running on a Mirrlees engined power station and was now being fitted in a ship for sea trials.

Referring to the parameters given under *Oil Fuel System*, existing oil fuel systems were divided into three sections. They could be reduced to two sections by using the new filter module which combined the functions of sections (2) and (3), purification and filtration respectively. That represented a considerable saving in space and initial cost. Maintenance consisted of infrequent change of the element in the Duplex filter—not less than six months. It was believed therefore that such a module would now establish filtration as an alternative to centrifugal purification with considerable installation and operating advantages.

MR. G. VICTORY (Member) said that he did not intend to be drawn into a discussion on medium speed versus slow speed engines, except perhaps to reiterate what the author had hidden in a vast number of facts i.e. that many of the improvements he had proposed in respect of modules were equally applicable to slow speed installations.

However, the question had been reopened in discussion and Mr. Victory was not convinced that the medium speed engine with very high mean effective pressures had yet been shown to have the degree of reliability and availability required for marine installations; certainly not very long voyages. The use of the c.p. propeller in conjunction with clutch and gearing required these engines to run at idling speeds and no load, or very low load, for considerable periods of time whilst manoeuvring in and out of port, or in fog, and this usage could apparently have deleterious effects on pistons, valves, or ports, and could be accompanied by an unacceptable level of lubricating oil consumption; he would like to hear that these difficulties had been overcome. The disadvantages consequent on running these engines at very low load rating, and also the doubt whether under supercharge they would follow varying loads which could be obtained in port, quickly enough, made one wonder whether, unless they could be run at high rating, their use in port as generating engines was worthy of consideration.

His next point concerned the inability, for some reason, of marine engineers to get away from mixing steam and Diesel machinery. After all, the Scott-still engine was a good idea on paper, but there was a great deal of blood, toil and sweat taken out of marine engineers before it finally came to its demise. One admitted the importance of economics, but economics were not everything. Marine engineers were difficult to get and, having got them, they were difficult to retain, and one thing most liable to make them pack up and leave was to have trouble with boilers and steam pumps, or to have numerous steam joints leaking while they were trying to do repairs to generators or to the main engines. It would appear therefore, that, while tanker operations probably called for a steam installation, it was doubtful whether the simplification and reduction in maintenance obtained by eliminating steam circuits in the case of the bulk carrier and cargo liner would not justify having a fully electric system or, as a poor second, the hot water system suggested.

In looking at some of the modules he was surprised at the rather archaic philosophy that had been used. For example, Fig. 5 showed a fire, bilge and general service module, where dual duty pumps were still being used for fire duty. Surely in this day and age the advantages of having a pressurized fire main system, which required the pump to start automatically by a reduction of pressure in the circuit, or to have the fire pump directly on line and started at will by remote control, were such as to make the old idea of fire and general service pumps entirely untenable.

It was also surprising to find that the oil/water separator had been fitted only in the case of the cargo liner. Were there double bottomed tanks for the carriage of fuel oil in the bulk carrier and, if so, how were the requirements in respect of oil pollution met? Mr. Victory also questioned the concept of putting oil pumps on the same module as fresh and salt water pumps. He would prefer to see all the oil section modules, both fuel and lubricating, and comprising pumps, coolers, purifiers and everything else, put into one section and isolated from the remainder of the engine room by large coamings. In that way in the event of leakage or joint failure a reasonable amount of oil could be isolated in the section without overflowing the coaming or floating across the tank tops. That was, of course, a very great safety factor and could be a preliminary to having the oil installation isolated from the rest of the engine room by bulkheads which would be a better idea still from the safety point of view.

The author had shown the requirements of a well designed module and had related these to fuel oil modules. In Mr. Victory's view one had not covered the requirements of a well designed module until a "safety assessment" had been made. For example, the fire safety of a module could be improved by trying to eliminate the possibility of oil leakage on to hot surfaces or of spraying on to electrical equipment. Joints and unions, particularly of some of the small pipes leading to pressure gauges, could be improved. If safety could be designed into machinery layout, one would have gone a long way towards making the task of providing adequate fire extinguishing coverage after the installation of the machinery much simpler and more effective.

MR. K. E. LEA (Associate Member) said that it was interesting to note the methods and suggestions given by the author with regard to the development of the concept of medium speed machinery packages for marine use. The organization for which Mr. Lea worked had for some considerable time been concerned with the application of modular principles to its four-stroke Diesel products, both marine and industrial.

The author's statement that the supply of the entire package should be the responsibility of the engine builder could become commercial suicide, unless the engine builder was prepared to carry a full marine consultancy staff, with specialization in those aspects of the design which were not directly concerned with their product. The far sighted ideal of the author would be an intolerable burden for the engine builder to assume overnight, when no such structure existed within the company. Obviously the setting up of such a team, with its consequent financial increases, would detract from that which had been to date part of the shipbuilders' established supply and financial gain. There would seem to be two ways in which to tackle this problem:

a) that the package be split into smaller packages and, as the engine builder's experience broadened, to take more of these package units, e.g. cooling package, boiler package etc;
b) to standardize the product ranges; this of course suffers from the same difficulties as modular constructions in that it restricts the choice available.

The large "engine services" module imposed rigid limitations in location. It would be interesting to know whether the author felt it wise to restrict the forward bulkhead in such a manner. One of the concepts of the medium speed engine was that major repair of the unit could be designed into the vessel without being forced to remove all the upper deck structure etc. This could be accomplished by using the forward engine bulkhead and vessel cargo space. Further such a concept as shown in the paper, although producing a shorter engine room than its slow speed counterpart, was not the best from that standpoint. Mr. Lea referred to Fig. 9 of a paper by Henshall,* illustrating the concept of a twin engine, single screw cargo vessel, where use had been made of the idea of one engine being standby to the other. Separate systems for each engine were little, if any, more expensive than combined ones, and were more attractive not only for the reasons given, but because they contained more readily "proved in service" components and standardized equipment packs. They were clearly illustrated in the isometric views shown in Fig. 1 of Henshall's paper. They were applicable in principle to one, two, three and four-engine arrangements.

It was useful to bear in mind that medium speed engines were built in quantities to serve the marine, as well as industrial power generation and other markets and that kept the price down in terms of £/hp. The other applications could and did accept the modular forms of ancillary equipment. If modules were designed to serve a particular engine of a design for any environment, marine or otherwise, then they might well be cheaper than those designed only for marine services. That was another argument for the small, separate cooling module system.

The question of clutch slip appeared to have been side stepped by assuming c.p. propellers, but would every owner accept them? If a fixed pitch propeller were specified, would the author have retained his choice of mechanical drive and chosen clutches capable of withstanding the slipping condition under the crash manoeuvre condition?

The paper given by Adley and himself† to Branches of the Institute set out to analyse that problem and it had now been borne out in practice, as was also shown to be the case in the paper by Goodwin *et al*‡. Therefore, the case of technical ability was not in question, but the financial costings given by the author did not give a large enough difference between c.p. and f.p. propellers.

The c.i. spare propeller usually fitted to fixed pitch vessels, against the spares required for c.p. installations, such as spare blade, controls servo system etc., seemed to have been ignored.

Mr. P. J. G. Mack (Member) said that for a number of years during discussions at the Institute, those present had listened to countless views from, on the one hand, exponents of the direct coupled, slow speed engines, and on the other hand, from their medium speed competitors. Those views were generally biased one way or the other and, no doubt, the majority of members of the Institute would really appreciate follow up papers or composite papers giving the operators service experience of the respective machinery. Computer simulations, however complex, were by no means conclusive. The proof of the pudding really was in the eating.

The opening remarks of the paper related to medium speed geared Diesel engines developing up to and in excess of 1000 bhp/cylinder, and included the term "first and second generation engines". That description would seem to imply that second generation engines, developing in excess of 500 bhp/cylinder were a natural development of the first generation engines based, one would hope, on service experience. Table II however appeared to belie that interpretation and it might, in fact, be that a large percentage of the first generation engines in the various cylinder combinations had not yet been built let alone seen satisfactory service. Would the author care to say whether the alternative main machinery arrangements selected for each of the three ships had a counterpart or near counterpart in service, operating and manoeuvring satisfactorily on 3500 sec fuel?

Having decided on a c.p. propeller installation and either one or two alternators driven from the gearbox, the author claimed to have been advised by the engine manufacturers that, for protracted port use at low engine loads, it might be advisable to run on Diesel fuel. There were few present who would commend that practice on the basis of having operated medium speed generators, albeit of lower engine outputs, on Diesel fuel for many years. Two-stroke cycle trunk piston engines especially do not take kindly to light load operation. Perhaps there had been a fundamental break through in technology which permitted the running of two-cycle and four-cycle trunk piston engines with impunity at eight to ten per cent load on 3500 sec fuel. Apart from consideration of limited loading of these engines in port, they would also be required to operate at idling speed at zero propeller pitch for extended periods during standby and in fog. Presumably it might be even more desirable to resort to Diesel fuel in such circumstances. The design of conventional governing and fuel injection equipment did not readily lend itself to protracted operation in the idling condition, and one would expect to find that more sophisticated controls, perhaps embodying dual injection systems, would be necessary as cylinder power increased. That might also be coupled with the need for auxiliary blowers particularly in the case of two-cycle engines so as to ensure combustion of the fuel charge and smooth and safe response to load demand from idling or light port load during the full service life of the machinery.

The electronic pitch/speed control system shown in Fig. 12 provided an alternative arrangement to the conventional pneumatic/hydraulic system. Experience to date with the latter type of control system indicated that when failures took place they did so rapidly and sometimes without apparent warning; it might be pertinent to quote just two incidents.

The first concerned a Class VII cargo ship. The main engine was designed to operate with an unmanned engine room although at the time of the casualty, the fourth engineer was on watch in the engine room control room, but the vessel was being manoeuvred into a lock under bridge control. In accordance with the owners' usual practice, power was being provided by one engine only whilst manoeuvring the ship slowly into the lock, to a position where it would be possible to close the entrance lock gates. When the vessel was approximately 150 feet from the exit gate, which was closed, the order was given to stop the engine and

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* Henshall, S. H., 1969. "The Development of a Range of Standardized Power Packs". I.Mar.E. International Marine and Shipping Conference, Section 4—Main Propulsion Machinery (Medium Speed Diesels).

† Adley, A. A., and Lea, K. E., 1968. "The Selection and Application of Medium-speed Machinery Systems". Paper read to I.Mar.E., South East England Branch, 15th February.

‡ Goodwin, A. J. H., Irvine, J. H., and Forrest, J., 1968. "The Practical Application of Computers in Marine Engineering". *Trans. I.Mar.E.*, Vol. 80, p. 209.

the propeller control moved into the neutral position by the first mate. The master noticed, shortly after, that the vessel was still moving forward instead of quickly losing headway, gave the order "slow astern", and then became aware that the vessel was not responding to the engine controls. The first mate had altered the engine control to slow astern, he glanced at the lock wall and could see that the vessel was proceeding faster than normal for her position in the lock. He glanced at the pitch indicator, saw that it indicated an excessive pitch in the ahead direction, and immediately moved the engine control to full astern. The mate, realizing that bridge control had failed, declutched the engine from the propeller shaft and reported what was happening to the master, who ran into the wheelhouse from the wing of the bridge, where he had been with the pilot, and rang full astern on the emergency telegraph as an order from the engineer on watch to put the engines full astern. The master then ordered the mate to stand by the astern anchor. Due to the engine increase in speed and with the propeller pitch in the normal fail position of full ahead, the vessel's speed was also increased. The vessel struck the lock gate which immediately burst open, the water level dropped, the bow portion of the ship charged through the opening and was eventually stopped with one-third of the vessel protruding beyond the lock gate.

In the meantime, the fourth engineer had observed a change in noise of the engines, looked at the pitch indicator and noticed that the pointer was indicating ten degrees ahead and rapidly increasing to maximum. As the increase in pitch was much faster than he had normally seen, he concluded something was wrong and immediately decided to declutch the engine, but by that time the pitch had reached 20° and the bridge emergency telegraph rang full astern. He realized there was an emergency and ran down to the manual control above the variable pitch propeller mechanism, which connected up, by which time the chief engineer had arrived in the engine room, clutched in the engine and the propeller was made to operate full astern. It was estimated by the engineers that it took 30 seconds to get the engine under manual control from first having observed the emergency telegraph signal.

It would appear that the casualty was caused by a blockage of the pencil filter gauze, or a restriction of the oil orifice incorporated within the pencil filter, causing a failure of the oil supply to one side of the differential piston of the telemotor control gear, which directly controlled the supply or discharge of oil to or from the variable pitch propeller hydraulically operated unit.

No alarm or indicators were provided either on the bridge or in the engine room control room which would give any indications to the officers on watch that a filter and orifice was becoming partially blocked. Subsequent modifications included the fitting of pressure switch operated alarms at the oil filters and of a pitch control system electrically operated from the bridge as an emergency back-up to the normal combinator controls.

The second case involved a Class II passenger vehicular ferry which, manoeuvring from bridge control within the confines of a harbour, sustained a total electrical black out. The Diesel engines were provided with gear driven lubricating oil pumps. The subsequent failure of the independent electrically driven c.p. propeller servo-pumps and of the electro-hydraulic steering gear resulted in bridge control being lost. The ship, completely out of control, steamed at the maximum revolution of the engines, miraculously on a safe heading, six cables length out of harbour before control was regained.

As a result of that experience, amongst others, the owners had now specified shaft driven c.p. propeller servo-pumps in addition to stand-by independent pumps.

Having regard to the foregoing and referring back to Fig. 12, it would be desirable to embody a feed-back arrangement of actual propeller pitch and speed. These signals should be taken positively from the propeller mechanism and the propeller shafting, and utilized in the same manner as the "old hat" wrong way alarm. Irrespective of the design of the system, all critical signals should be constantly monitored and, where dial type meters or gauges were specified, they should be banded over the permissible working range. It should be standard procedure to test these systems efficiently before departure, and to check their response in clear water before arrival at each port.

A safety assessment of each module design should be carried out, observing, for instance, that the spade blanked cross-connexions between fuel and lubricating oil systems shown in Fig. 9 should not be allowed. The provision of daily service tanks having a capacity equivalent to 12 hours full load requirements was to be commended where, during unmanned operation, automatic fuel transfer and purification should not be permitted.

The foregoing comments represented Mr. Mack's own personal opinions and do not necessarily reflect the official views of the Board of Trade.

MR. L. F. MOORE, B.SC., (Associate Member) asked, firstly, whether consideration had been given to mounting the modules on anti-vibration mountings with a consequent reduction in structure-borne vibration and noise to accommodation. For instance, with a single pump, anti-vibration mountings were not a good proposition because of the high ratio of height of C.G. to base area and consequent instability; but with the large base area of the modules that problem was overcome and the selection of suitable mountings should be much easier.

Secondly, in the paper it was stated that a pneumatic/hydraulic governing and control system as well as an electronic system had been considered. Mr. Moore asked whether a system incorporating fluidic controls had been investigated and, if so, what were the relative merits of that type of system.

MR. J. J. STENNETT said that the author had sought the opinion of shipowners with regard to the relative merits of engine versus motor driven engine auxiliary pumps and had suggested that there was a feeling in favour of engine driven pumps. Perhaps he could indicate the reasons for that.

Under The Advantages of Auxiliary Machinery Modules was listed the cleanliness of the auxiliary system. The physical size of the lubricating oil filters meant they must be mounted away from the engine and there were almost bound to be cleanliness problems with the interconnecting pipe, as acknowledged by the author. Did the author recommend the introduction of individual filtration for the purpose of protection during commissioning?

In Fig. 4, did the sea water enter the module from the charger cooler and not from the turbocharger?

With regard to future trends, the suggestion that medium speed engines could be started without priming the lubricating oil system was a little surprising. Would the author confirm that that was so?

Author's Reply

Replying to the discussion, the author said that he was indebted to Commander Tyrrell for his description of the investigation sponsored by the Ministry of Technology in the field of medium speed Diesel engine propulsion and for allowing the results of some of these investigations to be published. The growth in popularity of the medium speed engine referred to by Commander Tyrrell was illustrated by a further analysis of the figures shown in Table I. Table V showed the outputs for the top eight manufacturers in the original table split up to illustrate the relative differences in growth between the medium and slow speed engines. The consistent upward trend in the medium speed field and downward (except for Fiat) among the slow speed engines was readily apparent. There was admittedly an overall 11½ per cent fall in the total Diesel brake horsepower installed in 1968 compared with 1967, accompanied by a sudden rise in steam turbine brake horsepower installed of 83½ per cent and this would obviously have adversely affected the slow speed engine figures, but the medium speed engine was undoubtedly increasing in popularity as a contender in the lower half of the power range. The trend was continued in figures for the first half of 1969 compared with the same period in 1968 shown in the two end columns. Once again with the exception of Fiat, the only two overall increases were two exclusively medium speed designs, Pielstick and Deutz.

One must, of course, keep a sense of proportion. A study of percentage changes in individual manufacturer's outputs was meaningless unless associated with an assessment of the proportion of the marine Diesel market claimed by the slow and medium speed types of engine. Fig. 15 indicated, for yards in the United Kingdom only, the percentage of the total number of Diesel engine ships which had medium speed Diesels and this would perhaps help to put in perspective the popularity of the medium speed engine, although as Commander Tyrrell stated, one would expect that had the graph been prepared for a European country, the proportion might well have been higher. The emergence of the second generation of medium speed engines in the 1000 hp/cylinder class should serve to increase this popularity and Fig. 16 illustrated what had been achieved by the designers

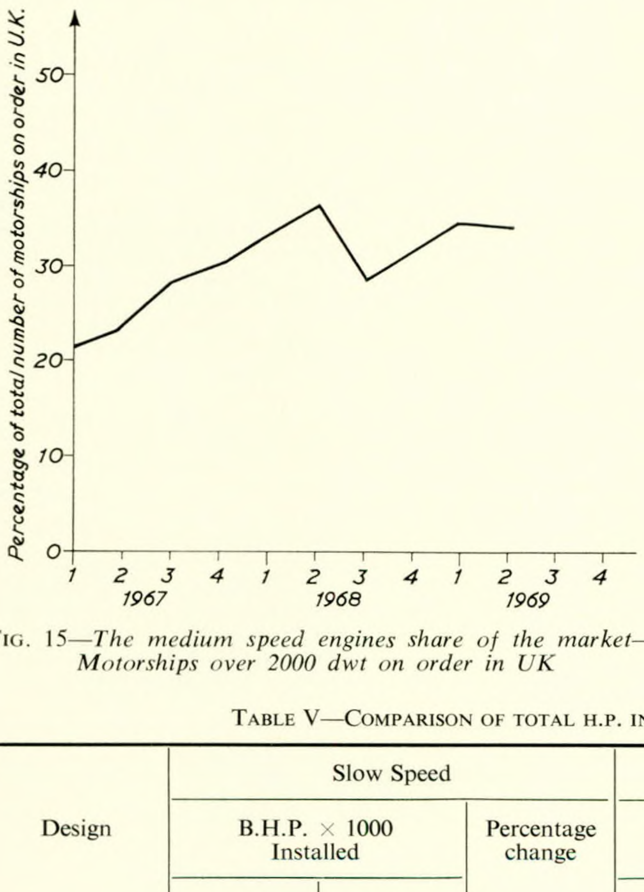


FIG. 15—*The medium speed engines share of the market—Motorships over 2000 dwt on order in UK*

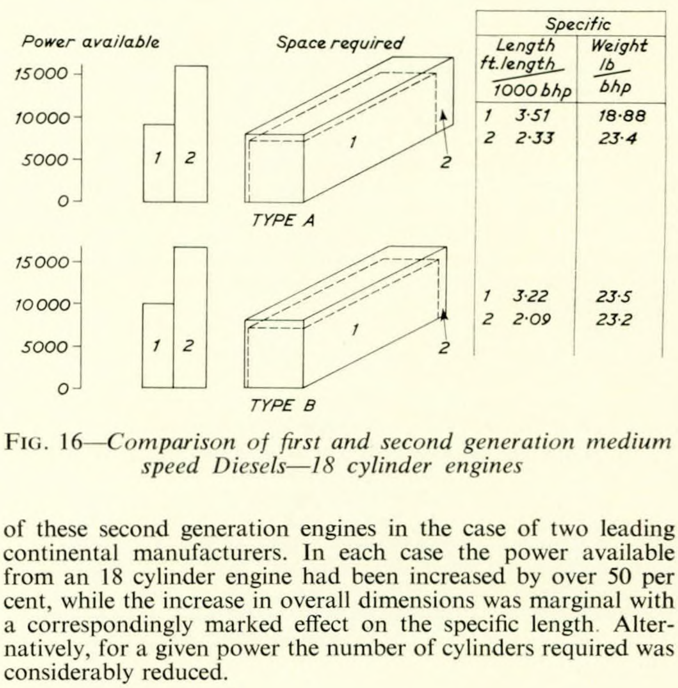


FIG. 16—*Comparison of first and second generation medium speed Diesels—18 cylinder engines*

of these second generation engines in the case of two leading continental manufacturers. In each case the power available from an 18 cylinder engine had been increased by over 50 per cent, while the increase in overall dimensions was marginal with a correspondingly marked effect on the specific length. Alternatively, for a given power the number of cylinders required was considerably reduced.

TABLE V—COMPARISON OF TOTAL H.P. INSTALLED OF LEADING MARINE DIESEL DESIGNS

Design	Slow Speed B.H.P. × 1000 Installed 1968	Slow Speed B.H.P. × 1000 Installed 1967	Slow Speed Percentage change	Medium Speed B.H.P. × 1000 Installed 1968	Medium Speed B.H.P. × 1000 Installed 1967	Medium Speed Percentage change	Overall percentage change 1968/1967	Overall percentage change 1st half '69/1st half '68
Sulzer	2635	2859	−7·8	93	77	+20·8	−7·1	−7·1
B. and W.	2118	2728	−22·4	59	14	+321	−20·6	−20·0
M.A.N.	1046	1366	−23·5	263	149	+76·5	−13·6	−10·1
Pielstick	—	—	—	453	428	+5·8	+5·8	+63·2
Götaverken	370	491	−24·7	—	—	—	−24·7	−37·3
Fiat	291	278	+4·7	24	1·3	+1745	+12·6	+38·5
Mitsubishi	127	149	−14·8	107	106	+0·9	−8·5	−1·0
Deutz	—	—	—	110	71	+55·9	+55·9	+39·8

Turning to Mr. Scrimgeour's spirited contribution, the aims and scope of the Ministry's investigations had already been outlined and the issue of whether the Ministry was obtaining value for money was surely one which must be taken up with the Ministry itself. Referring to the itemized points raised:

i) *in situ* maintenance of modules: the diagrams in the paper of relatively large items of equipment had, of necessity, to be reduced to quite small proportions and this might give the impression of inaccessibility in some instances; in every module design, however, great care had been taken to maintain adequate accessibility and the advice and experience of B.S.R.A. in module design has been utilized throughout; B.S.R.A. were, in fact, responsible for a number of the module designs themselves; in each case, furthermore, the module designs were submitted to the Advisory Committee set up by the Ministry of Technology on which were represented a number of leading British ship operators and the designs were amended in the light of their comments;

ii) the number of crane lifts: the crane lift itself is, as Mr. Scrimgeour suggests, itself a relatively unimportant item; the total number of lifts required for a complete installation, however, is indicative of the extent of pre-fabrication and of the size of units being transported for installation into the vessel and thereby gives a guide to the extent to which the present day trend in machinery installation techniques of pre-assembling as large a proportion of the machinery as possible in ideal workshop conditions has progressed;

iii) plans, elevations and sections: the necessary limitations placed by the Institute on the number and capacity of drawings which can be accommodated in the published technical papers naturally limited the extent to which detailed machinery arrangements could be shown; an attempt had been made to create a general impression of the nature of the machinery installations; the more detailed drawings which Mr. Scrimgeour required could be found in the full report published by the Ministry of Technology and widely distributed within the industry in the U.K.;

iv) other items suitable for modules: there were certainly a wide variety of items of machinery which were suitable for grouping into various modules; the study, however, had concentrated on those which logically fell within the propulsion machinery package; a dividing line was therefore drawn between those auxiliaries which were ship dependent and those which were engine dependent, with the emphasis in the design of standard modules on those which were dependent on the propulsion machinery; by doing this, the basis was established for a range of modules, any one of which for a given power could be fitted in a wide variety of vessels;

v) absence of Diesel generators, boilers, etc.: this was again a question of lack of space for the inclusion of machinery arrangement drawings and all these items are shown in the arrangements in the full report; perhaps Fig. 17, however, will serve to give some indication of the position of some of the more important items mentioned;

vi) siting of the main engine services module: the position of the main engine services module certainly occupied valuable machinery space immediately forward of the two main engines; the point, of course, was that, had it been decided to drive the engine-driven alternators from the forward end of the main engines as well, the machinery space bulkhead would have to be removed further forward; the present position of this module minimizes the length of pipe runs and forms almost as compact an arrangement as could be achieved were the module dispensed with and the items of equipment dispersed; the provision of the engine or gear-driven alternator aft of the gear-box enabled the envelope of the main propulsion machinery to conform to the pre-dominantly triangular shape of the aft end machinery space;

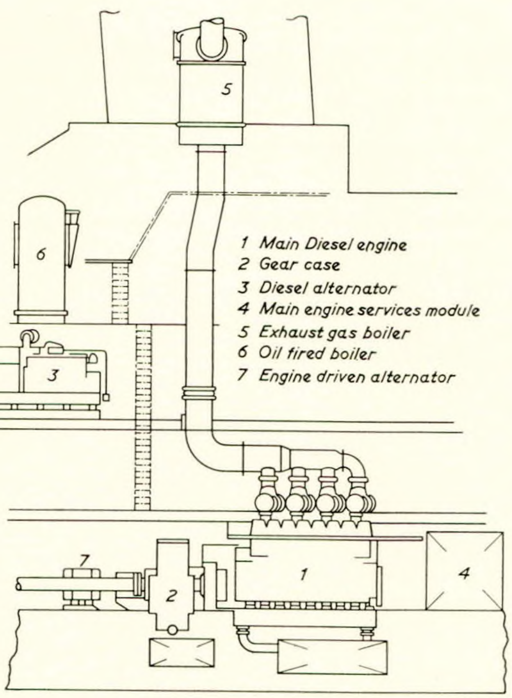


FIG. 17—*Elevation showing typical machinery layout*

vii) utilization of the main engines in port: the bulk carrier is the only one of the three vessels in the study in which the main engines are not utilized in port; in the other two vessels, one main engine was utilized in each case to provide the relatively high cargo handling loads, so that there was at all times at least one main engine available for maintenance;

viii) financial inconsistencies: the hydraulically driven cargo pumps referred to by Mr. Scrimgeour as a costly alternative could prove economically very attractive; in another part of the tanker study already mentioned by Commander Tyrrell, a comparative evaluation was made of the operational capabilities and costs of various methods of cargo pump drive, including steam, electric and hydraulic and the capital and running cost differentials were roughly as indicated in Table VI for otherwise similar installations.:

TABLE VI

Cargo pump drive	Steam	Electric	Hydraulic
Capital cost differential, £	Basis	+20 200	+9600
Annual operating cost differential, £	Basis	+2050	−3660

The recirculatory sea water system is a relatively simple one involving only one thermostatic valve and a cross-over pipe from the sea water discharge to the inlet. It is used in the Freedom Ship Liberty replacements and considerably reduces control valve cost;

ix) reversing gear-box: the reversing gear-box was certainly a possible means of achieving astern power as an alternative to the c.p. propeller although Mr. Scrimgeour's contention that the argument for c.p. propellers lost its force when pneumatically controlled clutches were proposed is not supported by the number of installations with just this combination currently at sea; reverse reduction gear-boxes with a few notable special-purpose exceptions have not proved economically or operationally attractive to merchant ship operators at powers much above 3000 shp;

x) the control system: the control systems proposals were explained in more detail in the full Ministry of Technology report; the electronic control was offered as an optional alternative to the more conventional pneumatic system; any control system covering the entire propulsion machinery must, of necessity, be a mixture of differing media and even the conventional pneumatic system which Mr. Scrimgeour presumably prefers, is, in effect, a mixture of the hydraulic and pneumatic.

Mr. Scrimgeour's parting shot was noted and if he cared to wait until the outcome of the third in the series of studies placed by the Ministry of Technology which covered the design of similar installations for periodically unmanned operation, he would no doubt find the reference for which he sought to the requirements and conditions affecting the engine room staff.

With reference to Mr. Norris's query regarding the merits of de-clutching the main engines in the event of electrical black-out, this was provided essentially to prevent the motoring of an unlubricated engine by the propeller with consequent damage to the engine. It should also serve, incidentally, to prevent the type of catastrophe described by Mr. Mack in one of the electrical black-out situations which he described. The author did not know of any specific demonstration of the effect of de-clutching the engines under protracted black-out conditions.

A rough estimate of the weight saving in tons on the overall machinery weight is 170 tons for the medium speed version of the basis ship in the tanker study mentioned by Commander Tyrrell compared with the direct drive version representing a saving of approximately 20 per cent for this particular case.

The author was fully in agreement with Mr. Norris's suggestion that each individual vessel must be investigated in determining the case for the extent of waste heat recovery to be utilized. Relatively minor differences in the operating profile of the vessel could make a marked difference in the most economically feasible installation. Mr. Norris had questioned the advisability of building up operating hours on the main engine. In the case of the cargo liner, this vessel was, of course, the one of the three under study in which the accumulation of running hours for propulsion duties at sea would be lowest and there would, therefore, appear to be a good case for the utilization of one engine for cargo discharge in port, thereby reducing the total capacity required of the separately driven generators.

Mr. Norris's concept of power feed-in to the gear-box was attractive and might well, as he suggested, help to exploit the potential of higher power medium speed engines. The only problem which the author had encountered with the application of the principle to a specific project was one of machinery arrangement in the relatively confined spaces adjacent to the gear-box, but this was, no doubt, a problem which could be overcome.

In replying to Mr. Brook, it was necessary to outline briefly the sequence of events which the author envisaged in the construction of a hypothetical vessel. In Fig. 18 it could be seen how the engine builder, presuming that he was the main machinery contractor, supplied and guaranteed the main propulsion machinery package from the engines through to the propeller in much the same way as other sub-contractors would provide the air-conditioning plant, "fridge" plant, etc as the result of the specification by the shipbuilder of the basic performance requirements. The question of who actually installed the components of the package in the vessel was a thorny one. The Geddes' Report, however, indicated that this was the shipbuilder's job and the figure had therefore been prepared accordingly, although as might well be the case with some of the other sub-contractors, the arrangement might well be for the machinery contractor also to carry out the installation if he wished.

The detailed demarcation between what was in and what it was out of the package would, in general, depend on the preferences of the specific shipbuilder, engine builder, and ship-

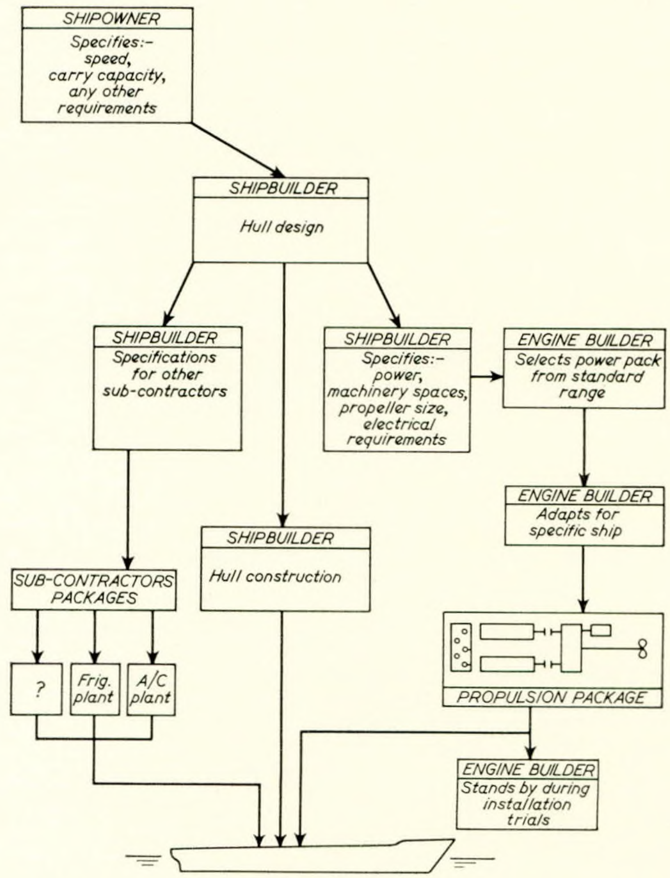


FIG. 18—*The propulsion machinery packaging concept*

owner, but the following was an indication of the major item which had in this study, been considered as being within the package:

1) main engines;
2) gear-box, clutches and couplings;
3) shafting and sterntube;
4) propeller;
5) electric generators (gear and Diesel driven);
6) exhaust gas boiler;
7) oil-fired boiler;
8) auxiliaries necessary for propulsion (engine and gear-box lubricating oil, engine and gear-box salt water, compressed air starting and control, c.p. hydraulics);
9) main machinery controls including bridge console;
10) switchboard.

The following were some of the components which, although normally found in the engine room, were in this case excluded from the package:

a) air-conditioning plant;
b) machinery space ventilation plant;
c) cargo or domestic refrigerating machinery;
d) ballasting system;
e) funnel fittings and auxiliary uptakes.

In the case of the propeller, the ship designer and therefore the shipbuilder was responsible for the design of the propeller and for its propulsive performance, whereas the main machinery contractor was responsible for providing the propeller and for its mechanical performance. Perhaps the main machinery controls should also be mentioned as one of the items which most benefitted from having both the design and the responsibility of the manufacture clearly established at the outset, since other-

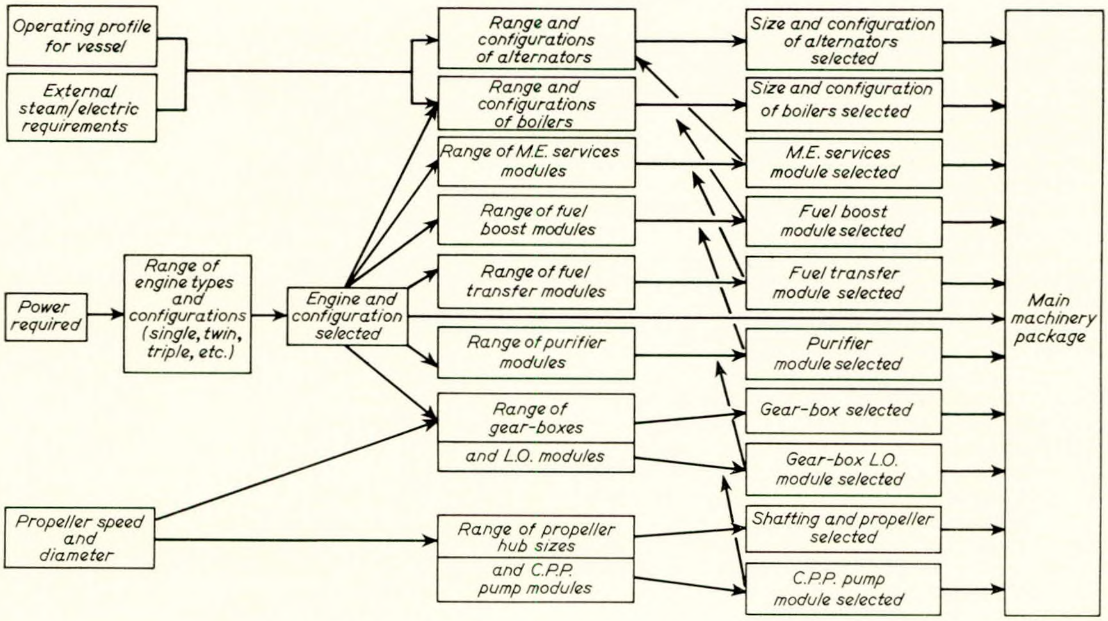


FIG. 19—*Standard machinery selection sequence*

wise the control may prove to be a last minute blot on an otherwise well thought out installation. It was perhaps also worth noting that the Ministry of Defence was likely to embrace the concept of the main propulsion machinery package in some future warship construction.

The manner in which the concepts of the use of auxiliary machinery modules and the standard package could be combined was important and Fig. 19 illustrated how, from the provision by the shipbuilder of the basic requirements regarding power, propeller diameter etc, the main machinery contractor could, by a virtually automatic process evolve the main propulsion machinery package. Hence having selected the engine configuration and the specific size of the engine required, the module for each of the auxiliary duties could be selected from each of the module ranges. These, in turn would determine the electrical and steam requirements for the main machinery indicated by series of diagonal link-ups from the chosen modules. Together with the requirements of those components outside the package, these would determine the electrical and steam generating capacities required. The division of the electrical generating capacity between gear driven and Diesel driven generators would, of course, depend on the operating profile of the vessel in question, one of the basic requirements supplied by the shipowner. Each item selected from the various ranges then all went together to make up the main propulsion machinery package. The opportunities were obvious for carrying the selection procedure one stage further and removing the human element altogether by allowing a computer to select the required item from each range and thereby assemble the package. The problem inevitably was the amount of preparation which must be carried out beforehand for each of the various permutations and combinations of engines and it would readily be appreciated how a relatively small alteration in, say, maximum rating on a given engine would radically alter subsequent ranges in the chain of events.

As indicated in the reply to Mr. Scrimgeour, the author would prefer to extend Mr. Brook's suggestion as to the limits of the package to cover those items which directly influenced the performance of the propulsion machinery as a whole, rather than solely the main engine, and he still felt that the necessity of integrating the propeller controls into the machinery control system demanded the inclusion of the propeller in the package.

Mr. Brook was right to doubt the implication that the main engine driven generators were to be used to produce basic harbour electrical supply. In the case of the G.P. tanker and the refrigerated cargo liner, the main engine driven generator was used solely in port when handling cargo.

The author was grateful for Mr. Welbourn's further comments on the friction clutches.

Lieutenant Knowles' contribution illustrated the possibilities of dispensing with centrifugal type purifiers. The aspect of performance of this type of unit likely to be examined most closely by ship operators was the ability to remove water since this had been the area of greatest doubt in some previous "no centrifuge" installations. Once this capability was proven, the probable reduction in first cost and undoubted reduction in maintenance effort and costs compared with the centrifuge made this an attractive alternative.

With reference to Mr. Victory's comments, there was no suggestion that the advantages claimed from the use of auxiliary machinery modules did not apply equally well to slow speed Diesel engines. Standard modules had, in general, not found great acceptance in the slow speed field. B.S.R.A. had, for some years, been developing standard ranges of auxiliary machinery modules suitable for a wide variety of slow speed engines. What had been attempted here was to lay the foundation for the same sort of standard series of modules for medium speed engines, in the belief that some of the factors which had resulted in the only limited acceptance of the principle for slow speed engines did not apply in the case of the medium speed alternative. This primarily referred to the fact that because the slow speed Diesel engine had been the almost exclusive choice for main propulsion duties for a wide variety of vessels for many years, each individual ship operator had evolved his own preferences for the arrangement and operation of the auxiliary systems for those engines. The development of an auxiliary machinery module for a particular slow speed engine type had therefore to make allowance for a wide variety of owners' preferences for the manner in which the system should be arranged or operated. The resultant module although it might be basically very simple must have so many optional alternatives as to defeat, to some extent, the basic concept of a simple standard arrangement. In the case of the medium speed engine, the history of whose application to ocean-going merchant ship started only relatively recently, it should be possible, by establishing standard configurations and methods of operations as early as possible, to produce ranges of module designs which would be acceptable to the growing number of owners and operators of medium speed engines.

The compatibility of the combination of the medium speed engine and the c.p. propeller could surely now be in little doubt for routine manoeuvring as far as the effect on the engine components was concerned. The successful performance of this combination had been amply demonstrated in such vessels as the container ships operating on the Manchester/St. Lawrence service. With reference to the specific question of low load power generation, the dangers were noted by the author and the specific engine builders early in the study and care was taken to ensure that the engine loading did not fall below acceptable limits. In the medium speed engine field in general, the engine manufacturers recognized the need for further study and development of this problem and a number of means of ameliorating the effects of prolonged low load running were being investigated. These included pre-heating of the induction air and automatically cutting out certain cylinders for low load running.

The ultimate desirability of dispensing with L.P. steam systems was agreed but the contention that economics were not everything was almost akin to a blasphemy in an investigation of this type. As yet, no acceptable alternative had been found. Perhaps if the maintenance load of the L.P. steam system could have been accurately assessed, the extra expense of the electric installation for the all-electric ship could have been justified.

The apparent fitting of an oily water separator only in the cargo liner was due to the fact that in the other two ships the separator was on the flat above and therefore appeared in drawings for which there was insufficient space in the paper.

Isolation of the fuel oil system from other systems had been adhered to. Isolation of the lubricating oil system was not so easy with water-cooled lubricating oil coolers, nor one might argue, quite as important from the safety aspect. In both cases, the baseplate of the module formed the coaming to limit spillage. Safety was, of course, a cardinal issue in every aspect of any machinery design and, in addition to their own investigations, the author's organization had submitted the entire report, including the module designs, to both the Board of Trade and Lloyd's Register of Shipping for their advice and comments.

The argument by Mr. Lea for separate modules for separate engines raised some involved points. At the outset of the study, the author's organization felt that the twin-engined installation was destined to be initially the most logical and popular configuration of the medium speed Diesel engine and this had proved to be the case. The module design was therefore progressed along the lines of one module per installation. i.e. per two engines. To break this up into two modules would be a reversal of the trend to prefabricate more and larger pieces of equipment before installation.

Acceptability of the concept of the one engine module depended on the acceptability to both the prospective owner and the classification society of having one engine incapacitated by the failure of part of its auxiliary system with the remaining engine or engines continuing to operate. If this was not acceptable, then the resultant standby pumps and cross-overs made the auxiliary systems more complex and expensive. The one-engine module was also limited in its application. It could not be used alone on a single-engined installation since some standby must be provided and connected into the systems and the same applied to the one engine per shaft installation, assuming that shaft sets of machinery were to be independent of each other.

Turning to the question of cross-fertilization from land applications, there were a number of land installations involving pairs of medium speed engines and a large variety of proven marine pumps, coolers, etc, adequate for the combined requirements of the two engines, were available.

The question of whether or not one must leave the forward engine room bulkhead clear for machinery removal had already been argued elsewhere (see discussion to Henshall, Imas '69 paper). It was sufficient to say that because in the case of a tanker such an arrangement might mean breaching bunker tanks or in a refrigerated cargo liner, insulated cargo spaces, the design of the vessel could be arranged to allow removal of large pieces of machinery upwards and aft rather than forwards and up as suggested.

Naturally, as Mr. Lea had stated, c.p. propellers might not be acceptable to every owner, although the proportion men-

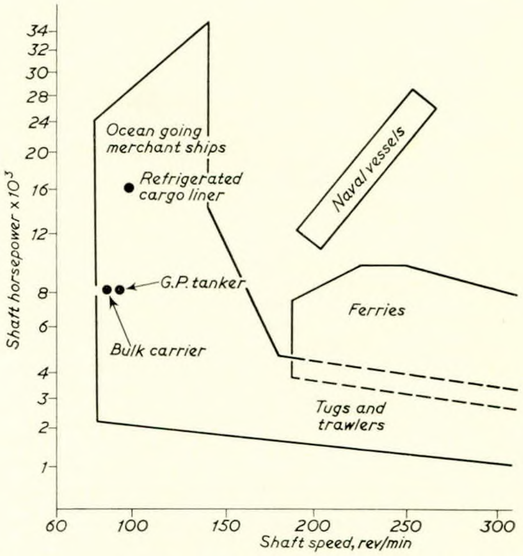


FIG. 20—*Power/shaft speed distributions of controllable pitch propellers in service or on order*

tioned by Mr. Brook was impressive and Fig. 20 illustrated that the c.p. propellers chosen for these installations were well within the limits of currently accepted practice. It was debatable what type of clutches would have been chosen for the vessels had fixed propellers been selected. It would be necessary to analyse the installations to compare the relative merits of each clutch type on the grounds of cost, reliability and the ability to withstand crash manoeuvres. Of the available types, the hydraulic coupling was the most proven but was large, expensive and suffered from the continuous slip loss at full power. The magnetic coupling was fairly expensive and relatively unproven in marine installations. The friction clutch was the least expensive, the most compact and potentially one of the most attractive, provided it was selected so that the duty was well within its capabilities. The lockable hydraulic coupling referred to by Commander Tyrell might well prove to be an attractive contender also.

The spare propeller had indeed been omitted from the economic comparison of c.p. and f.p. installations. There were some ship operators to-day who preferred to keep a single bronze or stainless steel spare propeller at a base for a number of identical ships rather than have a cast iron spare on each vessel. To avoid being drawn into arguments on the various philosophies of propeller spares, it was simpler to omit them from the comparison. If the price of a cast iron spare fixed pitch propeller were to be included and the c.p. propeller spare blade and other spares, the result would probably help to reduce the extra expense of the c.p. propeller. With reference to Mr. Mack and the pedigree of the second generation medium speed Diesel engines, some of these 1000 bhp/cylinder engines were natural developments of their 500 bhp/cylinder forerunners, such as the Pielstick PC3 (developed from the PC2) and the M.A.N. V52/55 (developed from the V40/54). Others were basically new engines, such as the Mirrlees OP and the UDAB engines. Both the engines used for the installations in the paper had satisfactory operating experience at sea on high viscosity residual fuels. There was no one characteristic of any of the installations proposed which had already not been successfully utilized at sea and there were a number of vessels with similar, but not identical, machinery arrangements already in service.

The author was indebted to Mr. Mack for his interesting and daunting account of the two controls catastrophies and for the conclusions he had drawn from them. In the machinery controls proposals embodied in the study for the Ministry of Technology, actual shaft speed and actual propeller pitch were used as feedback signals in the controls operating sequence.

The question of low power operation had already been covered in the reply to Mr. Victory, as has the reference to a safety assessment. Ring and blank figures or other suitable safety devices would, of course, be fitted to the fuel/lubricating oil cross-connexions.

In answer to Mr. Moore, although the question of anti-vibration mounts had not been considered in detail, the module arrangements fortuitously lent itself, if required, to this form of mounting. The alternative of fluidics as a control medium had been investigated by the author's organization mainly in connexion with warship applications. In the merchant field, the present state of development of fluidic system design, together with its current cost, ruled it out as a possible basis for standardizing the controls arrangements.

Regarding Mr. Stennet's query on engine driven pumps, as a subsidiary part of the work for the Ministry of Technology, the author's organization had distributed a questionnaire to a large number of U.K. ship operators to try and discover their preferences and preconceptions. The detailed results were contained in an appendix to the report to the Ministry but, broadly speaking, the majority did not favour engine driven pumps due, no doubt, to previous experience on predominantly slow speed engines. In the intervening two years, however, there had been a growing number of owners who, from the aspects of simplicity of control, safety of operation and capital cost, were considering engine driven pumps. It was a regrettable fact that the drives for auxiliary pumps had often been an aspect of the engine design which had received least attention on a newly evolved engine and initial teething troubles with pump drives were not uncommon.

With reference to the question of the cleanliness of "off module" piping, every practicable measure must be taken to ensure that the preparation procedures resulted in a standard of cleanliness at least as good as the "on module" pipework, and careful flushing with the use of temporary filters during the first few hours of operation would be necessary.

The use of the phrase "from cold" in the paper, when describing the ability of the engine to start without priming, was misleading. The engines in question could be started from rest on the engine driven lubricating oil pump alone, provided the engine had not stood at rest for so long that all the oil had drained from each bearing, etc and bearings were dry. Before starting after several days at rest, therefore, priming would be necessary. Needless to say, the priming of even a long inactive medium speed engine was considerably shorter and simpler than its slow speed counterpart since there was no need for pre-heating.

The author would like to thank all those who contributed to the discussion for their interest and comments, including those in the Branches around the country. He would also like to take the unconventional step of thanking Mr. J. Hansen of Y-ARD for his help in the preparation of this reply in the last few days before the author's departure for Australia and for manfully accepting the task of delivering the paper to four other local Branches.

Marine Engineering and Shipbuilding Abstracts

No. 1, January 1970

* Patent Specification

Swedish Laboratory for Testing Propulsion Devices

Construction has recently begun on a large laboratory for Karlstads Mekaniska Werkstad (KMW).

Advanced methods of propulsion, for example jet propulsion for hovercraft, will be tested here for use in future projects. The first test programme in the new laboratory is scheduled to start during December 1970.

The finished KMV building will have a volume of about 13 500 m^3 (477 000 ft^3), housing two cavitation tanks for model tests on propellers and other propulsion equipment for ships. One of the test units will replace the present KMW cavitation test tank. It will be of the closed type with an overall length of 46 ft, height of 26 ft and a volume of 4130 ft^3. Maximum water velocity will be approximately 40 ft/sec, which will permit tests under vacuum corresponding to the maximum practicable ship speeds.

The second unit, which will have a free surface, will be 82 ft long, 39 ft high and have a volume of approximately 14 120 ft^3. A maximum water velocity of approximately 40 ft/sec will be attained with a pump driven by a 1300 hp motor. The vacuum setting can be adjusted as required.

The conventional closed tank is of moderate size, as a very large tank of this type at the National Shipbuilding Experimental Tank at Gothenburg is already available to KMW for tests requiring a bigger tank.—*Shipping World and Shipbuilder, September 1969, Vol. 162, p. 1333.*

East German Built Vessel for Africa Trade

The first of six ships of the same type on order from VEB Mathias-Thesen Werft, Wismar, *Wismar* is a cargo vessel for the Africa trade. Designated *Afrika* type, these vessels are being built for VEB Deutsche Seereederi, the State-owned shipping line of the German Democratic Republic. At the end of last year, this line owned 162 cargo ships totalling 1 049 000 deadweight tons. These are operated over 15 regular services, six of which are run in conjunction with other countries.

Wismar has been designed for the transport of many kinds of general cargo, including especially heavy items of industrial equipment, and bulk cargoes, particularly grain, ore and semi-finished metal products.

Principal particulars are:

Length, o.a.	129·5 m
Length, b.p.	119·0 m
Breadth, moulded	17·3 m
Depth from second deck ...	7·2 m
Depth from main deck	10·0 m
Draught (full scantling vessel) ...	7·58m
Draught (shelter deck)	6·72 m
Dwt (full scantling vessel) ...	10 770
Dwt (shelter deck)	9335
Displacement	10 635/9335
Operating range	9000/13 000 miles
Gross tonnage (full scantling vessel)	5968·59
Net tonnage	3340·33
Gross tonnage (shelter deck) ...	3917·77
Net tonnage	2163·27
Machinery output ...	7000 hp at 165 rev/min
Trial speed	17·55 knots

A combined refrigerating plant is fitted, which comprises one unit for the refrigerated hold and two units for air conditioning, plus another for the provision store room. The whole of this plant has been designed, manufactured, supplied and installed by VEB Kühlautomat, Berlin. Each of the systems works with single-stage compression and expansion.

The main engine installed in *Wismar* is a single-acting seven-cylinder two-stroke crosshead Diesel engine with an exhaust-gas turbo-charger. This engine is a M.A.N. type K7Z60/105E, built under licence in the German Democratic Republic by VEB Maschinenbau, Halberstadt, having an output of 7000 hp at 165 rev/min. It is designed to run on heavy oil and is coupled directly to the propeller shaft, which carries a five bladed propeller having a diameter of 4200 mm and a mean pitch of 3520 mm.—*Shipping World and Shipbuilder, September 1969, Vol. 162, pp. 1276–1278.*

Advanced Levant Trader

Aktiebolaget Transmarin of Halsingborg, Sweden, has taken delivery of the 6300 dwt cargo motorship *Marianne* from Jos. L. Meyer of Papenburg-Ems, West Germany.

The hull form is of the yard's own design with an accentuated bulbous bow, clear-water afterbody and transom stern. Machinery and accommodation are all aft. There is an extended fo'c'sle, raised quarterdeck and five holds, of which Nos 3 and 4 are twinned.

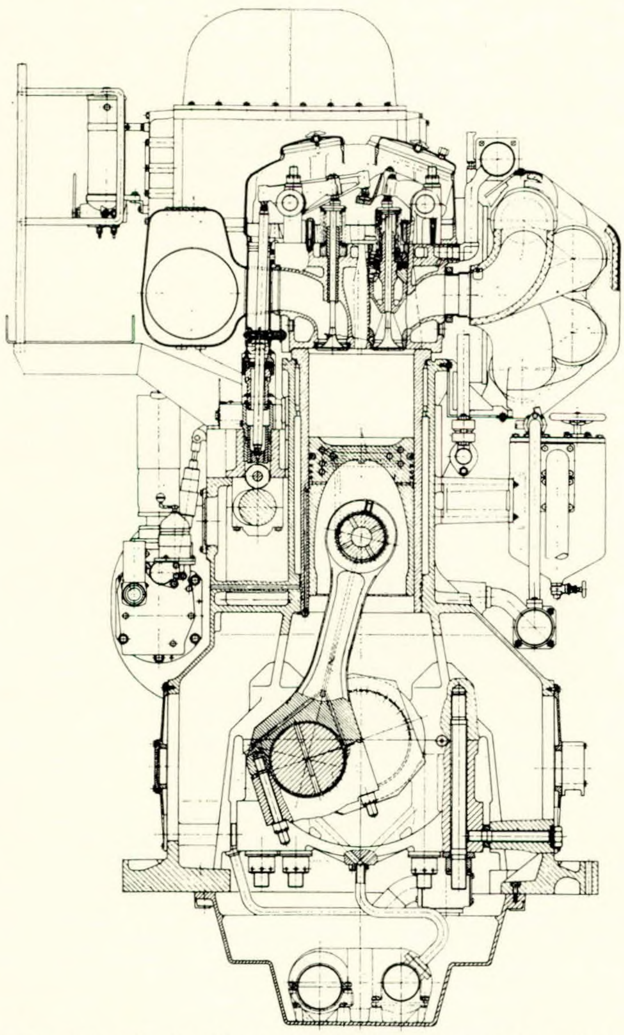
All hatch covers are of Von Tell hydraulic-type, opening fore and aft on the weatherdeck, fore and aft for each end for two panels of the 'tween deck hatches and laterally for Nos 1 and 5 holds and in the centre of Nos 2, 3 and 4 holds.

Principal particulars are :

Length, o.a.	393 ft 8 in
Moulded breadth	56 ft 5 in
Draught	24 ft 0 in
Cargo space, including refrigeration	347 000 ft^3
Refrigeration	30 217 ft^3

Forward there is a 400 hp LMG Tornado bow thruster and a large water ballast tank beneath No. 1 hold. Amidships there is a stabilizer tank and a small pump room with a 200-ton ballast pump for rapid trimming. The hatches have been designed to accommodate standard 20 ft containers.

Propelling machinery consists of two eight-cylinder Deutz engines of the latest RBV8M540-type, each rated 3500 bhp at 600 rev/min and geared to the 174 rev/min KaMeWa cp propeller through a Renk plain reduction gear. These engines run on Class B fuel of 200 to 400 sec viscosity. One engine is fitted with direct-reversing gear to provide for the unlikely event of propeller damage. Auxiliary power at sea is provided by a 600 kVA alternator driven from a power take-off aft of the gear-box and in harbour by three 330 kVA high-speed alternators driven by 12-cylinder Vee-type Deutz BF12M716 engines. These are installed in a pocket at the forward end of the engine room between two wing bunker tanks.—*Marine Engineer and Naval Architect, June 1969, Vol. 92, pp. 248–249.*



Section through the Deutz RBV540 in-line engine—One of the current generation of medium-speed heavy fuel-burning four-strokes

Electrical Repairs Carried Out on Tanker

Rebuilding of a 960-kVA alternator was accomplished in only six days by The London Graving Dock Co. Ltd., when the 103 785 dwt tanker *Bedford* was in the Thames recently after discharging a cargo of crude. The alternator, weighing some two tons was seriously damaged as the result of a mechanical failure on the d.c. rotor, and after consultation with the original manufacturer, the stator coils were sent to London by air, together with a new rotor assembly.

Following rewinding, London Graving Dock applied its Ayrodev Process III to ensure that the electrical, physical and mechanical characteristics were compatible with those of the company's patented impregnating varnishes. Installation of the crew rotor was carried out on board the ship with only $\frac{1}{32}$ in working clearance, and this was followed by load tests, synchronizing and all necessary safety checks of protective equipment before the ship sailed.—*The Motor Ship, August 1969, Vol. 50, pp. 240.*

Chemical Tanker with Stainless Steel Tanks

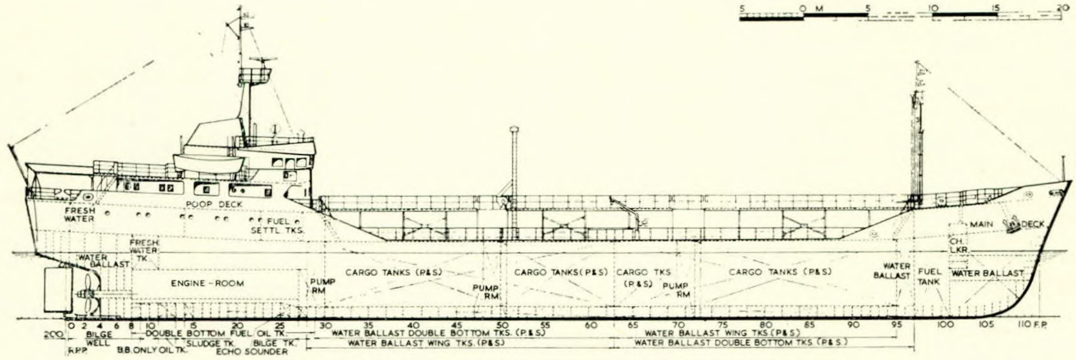
Main particulars of *La Hacienda* and her sistership *La Quinta* are as follows:

Length, o.a.	264 ft $1\frac{1}{4}$ in
Length, b.p.	246 ft $0\frac{3}{4}$ in
Breadth, moulded	41 ft 4 in
Depth, moulded	19 ft $8\frac{1}{4}$ in
Draught, design	17 ft 0 in
Corresponding deadweight ...	2300 tons
Service speed	12 knots
Capacities:	
Cargo capacity abt.	77 163 ft^3
Water ballast	27 722 ft^3
Fuel	4944 ft^3
Potable water	1236 ft^3
Tank washing water	2119 ft^3
Slop tank	1589 ft^3

The vessel is being constructed to Lloyd's Register of Shipping Class ✠ 100 A1 "Oil and Chemical Tanker", Ice Class 2 and U.K. B.o.T. requirements. The initial list of possible chemicals which can be carried totals 108 including phosphoric acid.

The cargo spaces are isolated from the ship's hull by a double bottom and water ballast side tanks. All the tank plating, which is solid stainless steel of about 6 mm in thickness, was supplied by Samuel Fox and Co. Ltd., Sheffield. For each ship, the company supplied 65·5 tons of A.I.S.I. 316-L low carbon content S.S. with a yield stress of 13·5 ton/in^2, and 70·5 tons of Hi-proof 316 S.S. having a yield stress of 20·5 ton/in^2.

The cargo spaces are arranged as eight tanks, each with its independent centrifugal pump and line, and its own vapour line. Each of the eight pumps, of 100 ton/h at 8 kg/cm^2



General arrangement of La Hacienda

capacity, are of Hamworthy manufacture. The pumps are located in three pump-rooms, to avoid passing cargo pipe lines through the double bottom spaces.

Cargo heating is effected by circulating hot water at temperatures of up to 95°C through ducts on the outside of the tanks. This enables cargo to be maintained at 58°C with a sea temperature of 0°C and an air temperature of −10°C. Arrangements will be made for circulating cargoes to prevent heat layers forming (some chemicals are bad heat conductors). A cargo temperature indicator will be installed in the bridge.

The main propulsion machinery comprises two Ruston AP3M engines, each having a maximum continuous rating of 1000 bhp at 1000 rev/min. Daily fuel consumption is about 6·75 tons of Diesel oil. The engines are connected to a twin-input, single-output MWD single-reduction gear-box to give 180 rev/min at the propeller shaft. The propeller is a KaMeWa controllable-pitch unit arranged for bridge control. A 350 kW alternator is coupled to the free end of each main engine.—*The Motor Ship, May 1969, Vol. 50, pp. 66–67.*

Italian-built Car and Passenger Ferry

Designed for their domestic and inter-Scandinavian routes, *Aalborghus* has been built for the United Steamship Co. of Copenhagen (DFDS) by Cantieri Navali del Tirreno e Riuniti, Italy. This twin-screw drive-on/drive-off, car and passenger ferry, which also has provision for carrying commercial vehicles and containers, is powered by two B & W Diesel engines which give the vessel a speed of 21 knots.

Principal particulars are :

Length, o.a.	406 ft 3 in
Length, b.p.	353 ft 7 in
Breadth, moulded	62 ft 6 in
Depth to saloon deck	56 ft 5 in
Depth to car deck	23 ft 4 in
Draught	16 ft 10 in
Deadweight	900 tons
Gross tonnage	7698
Passenger capacity	950
Crew	87
Car capacity, approximately ...	135
Hold capacity	22 073 ft^3
Machinery output	12 000 ihp at about 220 rev/min
Service speed	21 knots

Propulsion machinery consists of two turbocharged B & W engines of the 42-VT2BF-90 type having a combined output of 12 000 ihp at about 220 rev/min. A 1000 hp bow thruster is fitted, and this can be controlled from the bridge. Auxiliaries comprise four turbo-charged six-cylinder B & W engines, type 23-MTBH-30, each having an output of 660 hp and coupled to a 580 kVA generator. The electrical supply is at 440 V, three-phase, 60-cycles a.c. For lighting, navigational instruments, etc. the power supply is 110 V a.c. Sperry roll stabilizers are fitted.—*Shipping World and Shipbuilder, September 1969, Vol. 162, p. 1272.*

First of Seven Aker Group Built Tankers for Hilmar Reksten

Kong Haakon VII is a vessel of 220 000 deadweight tons. Propulsion of this single-screw ship is by an MST-14, non-reheat, steam turbine fed from a single boiler. On trials she attained a speed of 16·25 knots.

This vessel is of all-welded construction with longitudinal frames connected to web frames. There is a single continuous deck, poop and forecastle. The 'soft nose' stem has a bulbous bow and the stern is of transom form.

Over the length of the cargo tanks the deck is without sheer, and longitudinal and transverse bulkheads divide the cargo-carrying space into 16 tanks.

Principal particulars are:

Length, o.a.	1068 ft $8\frac{1}{2}$ in
Length between p.p.	1023 ft 8 in
Breadth, moulded	152 ft 0 in
Depth, moulded to No. 1 deck ...	85 ft $4\frac{1}{2}$ in
Draught, moulded to summer load line	67 ft 0 in
Corresponding dwt	220 050
gt	109 422
Net tonnage	89 461
Lightship weight	33 736 tons
Machinery output ...	30 417 bhp at 80 rev/min
Speed on trials 16·25 knots at an output of 29 400 shp	
Service speed	16·0 knots
Cargo oil tank capacity ...	283 650 m^3, 10 017 071 ft^3

Propelling machinery in the *Kong Haakon VII* comprises a steam turbine set of the General Electric MST-14 (non-reheat) type. This has a maximum continuous rating of 30 417 bhp at 80 rev/min and is arranged with a single pass condenser and scoop circulation. Steam is provided by a single Kvaerner/Foster Wheeler ESD III type boiler having a maximum steam output of 115 000 kg/hr and a normal steam output of 9000 kg/hr. Steam pressure and temperature conditions are 61·3 kp/cm^2 and 513°C respectively at the inlet to the main turbine. No provision has been made for "get-you-home" purposes in the event that the boiler breaks down. For cargo and fuel oil heating and domestic services, a low pressure steam/steam generator having a maximum output of 40 000 kg/hr and normal output of 2000 kg/hr at a pressure of 12 kp/cm^2 has been fitted.

The steam turbine drives a single six-bladed Cunial bronze Lips propeller having a diameter of 8800 mm.—*Shipping World and Shipbuilder, September 1969, Vol. 162, pp. 1323–1326.*

First Bulgarian Built 25 000 dwt Cargo Vessel

Bulgaria is about to start building her first ship of 25 000 dwt and is already contemplating building one of 50 000 dwt for which the plans have been completed by Bulgarian naval architects at NIPKIK, the Institute of Scientific Research and Designing for Shipbuilding, at Varna. Concurrent with the launching of this major undertaking, the shipyard responsible, the Georgi Dimitrov shipyard at Varna, is at work on a series of coal/ore carriers of about 10 000 dwt for the home market and on a series of 5000 dwt tankers for the U.S.S.R. A contract for the supply to Poland of 11 cargo motorships of 3050 dwt has just been completed.

Details of this new large ship show her to be a bulk carrier, intended in the first place for the Bulgarian merchant fleet, to cope with Bulgaria's great and expanding imports and exports of such commodities as apatatite, phosphates, light and heavy ores, coal, grain, etc. She is a single-screw, single-deck ship, with raised forecastle and poop, having the engine room and superstructure abaft the seven holds. The specific volume of the holds is 1·40 m^3/ton, that of holds and tanks combined being 1·54 m^3/ton.

The hull, the form of which has been tank-tested at the model basin in Vienna, is of all-welded construction and has a bulbous bow 5·0 m (16 ft $4\frac{7}{8}$ in) in diameter. Standard ship plate is used excepting for the deck, sheerstrake and tanks under the deck, which are of increased tensile strength. The stem and stern pieces are of welded construction. The frames are V-shaped forward, while those aft are of a modified V shape. The curvature of the beams is normal, the camber of the deck, however, being rather below average.

The main engine is a two-stroke Sulzer Diesel type 7RD76, made under licence in Poland, by the Ciegelski factory. It develops 11 200 bhp at 122 rev/min.—*Shipping World and Shipbuilder, September 1969, Vol. 162, pp. 1273–1274.*

Unusual Combination

It would be difficult to think of two less compatible cargoes than refrigerated containers and manganese ore. Nevertheless a ship is now under construction for the Australian National Line at the State Dockyard, Newcastle, NSW which has been designed to carry both. The widely differing stowage rates of the two cargoes, as high as 17 ft^3/ton for the ore, and an average of around 80 ft^3/ton for the containers has been put to advantage. The excess buoyancy resulting from inevitably high broken stowage of container cargo is needed to carry the heavy deadweight cargo. The ship will carry containerized general cargo from Melbourne to Sydney, Newcastle, Queensland ports and Darwin, Northern Territory, and return with manganese from Groote Eylandt, Northern Territory, to Bell Bay, Tasmania.

There are to be eight cargo spaces, Nos. 1, 3, 4 and 6 for containers and Nos. 2, 5 and 7 for ore. Full width deep tanks amidships will carry fuel. The ship will have capacity for 190 containers each of 20-ton capacity, and about 50 railroader type "flats". Principal particulars are as follows:

Length, o.a.	458 ft 0 in
Length, b.p.	425 ft 0 in
Moulded breadth	70 ft 0 in
Moulded depth	47 ft 6 in
Container draught	23 ft 6 in
Container dwt	8000
Ore draught	30 ft 0 in
Ore dwt	12 000
Speed at 30 ft draught	$15\frac{1}{2}$ knots

Machinery consists of twin-geared 16PC2V Pielstick engines developing 6500 bhp each. The ship will be automated and will have accommodation for a crew of 38. Cargo will be handled by a 25-ton gantry crane and a 15-ton travelling jib crane. Because of the high tidal range at Darwin, special attention has been given to the design of the cranes.—*Marine Engineer and Naval Architect, August 1969, Vol. 92, p. 335.*

General Purpose Bulk Carrier for Scottish Owners

A bulk carrier of approximately 26 000 dwt has been built by the Scotstoun Division of Upper Clyde Shipbuilders Limited, Glasgow, for the Scotspark Shipping Company.

Scotspark has been built to the latest regulations of Lloyd's Register for a vessel of this type and is classified ✠ 100 A.1 (ore in alternate holds) Ice Class II. She is a single deck ship of the self-trimming type with all the accommodation and propelling machinery located aft.

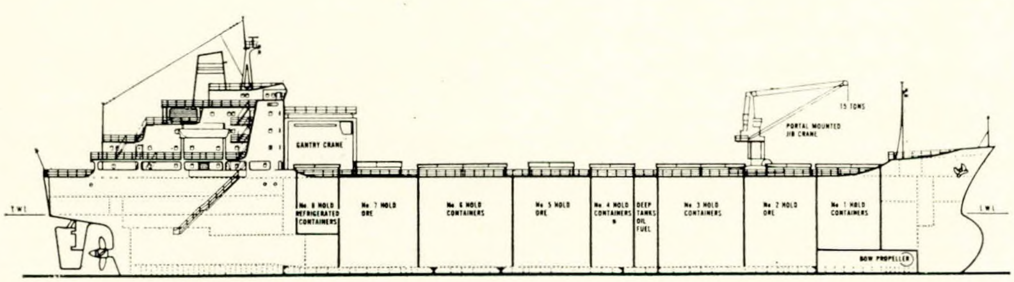
Principal particulars are:

Length, o.a.	580 ft
Length, b.p.	560 ft
Breadth, moulded	75 ft
Depth, moulded	47 ft
Draught summer	34 ft $8\frac{5}{8}$ in
Deadweight, approximately ...	26 000 tons
Machinery output	11 600 bhp
Speed	$15\frac{1}{2}$ knots

The main propelling machinery consists of a B & W Diesel engine having a maximum continuous output of about 11 600 bhp to give a service speed of about $15\frac{1}{4}$ knots. There is an air-conditioned, soundproof control room in the machinery space which accommodates the engine room control console, the main switchboard, group starter board, etc. The main and auxiliary machinery is arranged for automatic control to U.M.S. standard. A CO_2 fire extinguishing system serves the engine room, and a Pyrene fire detection system for an unmanned engine room has been installed.—*Shipping World and Shipbuilder, October 1969, Vol. 162, pp. 1418–1419.*

Salvage Vessel for the P.L.A.

The first of two vessels which when delivered will complete the Port of London Authority's modernization of their salvage fleet, *Crossness* is capable of lifting 120 tons over her bow horns. An innovation is the fitting of a self-contained crane, with telescopic jib, of a type not previously fitted as



ANL container/ore carrier

ships' deck equipment. The vessel was designed by the PLA's Engineering Department and built by James W. Cook and Co. (Wivenhoe) Ltd.

The 60 ton salvage winch is fitted with two independent main barrels, arranged in tandem, each barrel capable of exerting a maximum pull of 30 tons at full load at a speed of 20 ft/min. Controls for this winch are situated on a console forward of the deckhouse. Also fitted, at the stern, is a seven ton triple drum winch capable of exerting a maximum pull, from each drum but not simultaneously, of seven tons at full load at a speed of 50 ft/min.

Principal particulars are:

Length, o.a. (including bow horns)	127 ft 6 in
Length, b.p.	110 ft 6 in
Breadth, moulded	28 ft 0 in
Depth, moulded	11 ft 0 in
Gross tonnage	268·24

Crossness is propelled by two Rolls Royce type C6TFLM Diesel engines, each developing 262 bhp at 1800 rev/min, driving twin screws through Capitol gearboxes having a ratio of 3·5:1. At the forward end of each engine is a Fluid-drive coupling and a universally jointed shaft connected to a hydraulic power pack unit. This unit supplies the hydraulic power to the deck machinery, winches and capstans.—*Shipping World and Shipbuilder, September 1969, Vol. 162, p. 1293.*

Cathodic Protection

Reliable and complete control of the corrosion of ships' hulls in sea water can only at present be practically achieved in one way—that is by suppressing the activity of the galvanic couples which cause corrosion. All underwater corrosion that matters to the shipowner is of electrochemical origin and it is caused by the flow of current through sea water between parts of the structure which are at differing electrical potential. Causes of potential difference are many—dissimilar metals, differential stresses arising from working, differences in molecular structure, variable surface conditions of the metal and differences of environment such as those caused by variations in water speed, local turbulence and aeration. The anodic areas corrode whilst the cathodic areas are protected.

This action can be stopped by reversing the flow of current at the anodic areas. Practically this is achieved by making the whole underwater hull of the vessel into one large cathode by introducing, external to the structure, an anode which is of lower potential than the original complex. The anode is in electrical contact with the hull and current flows through the water from anode to hull and returns thence to anode *via* the metallic path of the hull.

In the practical field, it has been found that low-potential zinc anodes, manufactured from the correct alloy, provide the ideal anode material for general use. They give the cathodic-protection engineer considerable flexibility in scheme design and the vessel owner the most satisfactory level of protection combined with the best economic advantages, as well as achieving the most suitable length of scheme life to fit in with drydocking programmes.

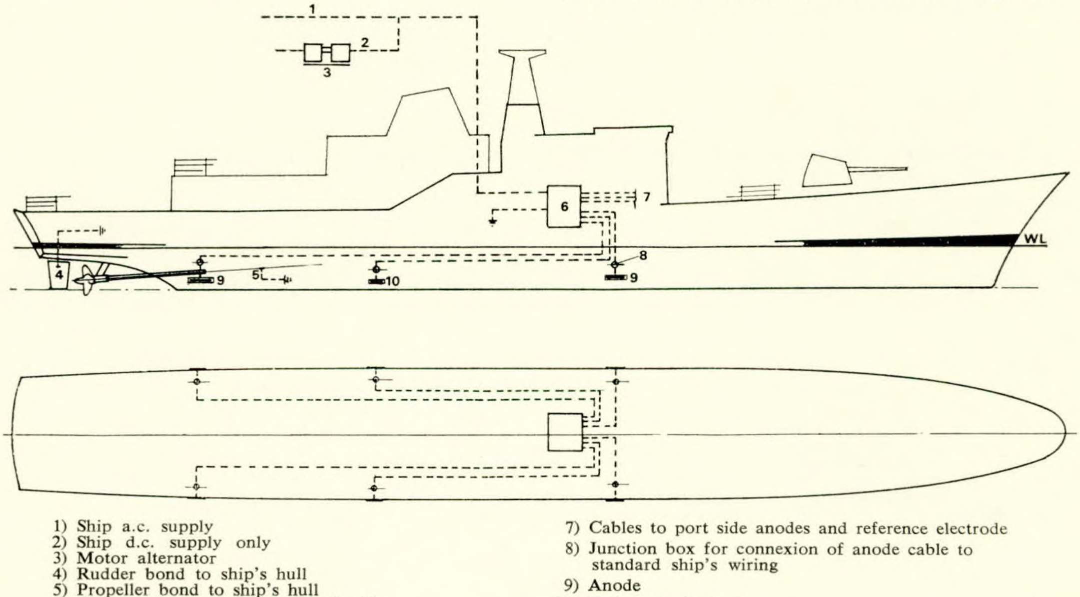
Magnesium-alloy anodes, which are manufactured from a high-potential anode material, also have their place in providing protection for certain craft. They are of particular interest when Kort nozzles are used and especially when cast-iron propellers are being considered.

The "impressed-current" method does not use sacrificial anodes in the way that a galvanic scheme does. Instead, inert platinized-titanium anodes are employed, capable of passing current into the water without deterioration. The electrical energy required is provided by the ship's normal electrical system after conversion into low voltage d.c. by suitable equipment within the ship. Because the current is impressed upon the anodes (instead of being generated by consumption of the anodes) it is referred to as the impressed-current system.

It is not sufficient just to povide a random value of current flow. Too little and the natural corrosion cells upon the ship's plating may not be fully suppressed. Too much and there is a waste of electrical energy, plus the possibility of disturbing the paint coating on the hull.

The correct amount of current varies constantly, depending upon the ship's speed, draught, water salinity and temperature, type and condition of paint coating.

The impressed-current system incorporates automatic control of current output. Sensing electrodes are fitted to the hull and are connected to the controller. Every variation in current need is detected by the sensing electrodes and the controller automatically varies the output of the protective current to suit the circumstances.—*Twynam, S. G., Ship and Boat International, June 1969, Vol. 22, pp. 14–16.*



1) Ship a.c. supply
2) Ship d.c. supply only
3) Motor alternator
4) Rudder bond to ship's hull
5) Propeller bond to ship's hull
6) Guardion/Cematic controller (2 off)
7) Cables to port side anodes and reference electrode
8) Junction box for connexion of anode cable to standard ship's wiring
9) Anode
10) Reference electrode

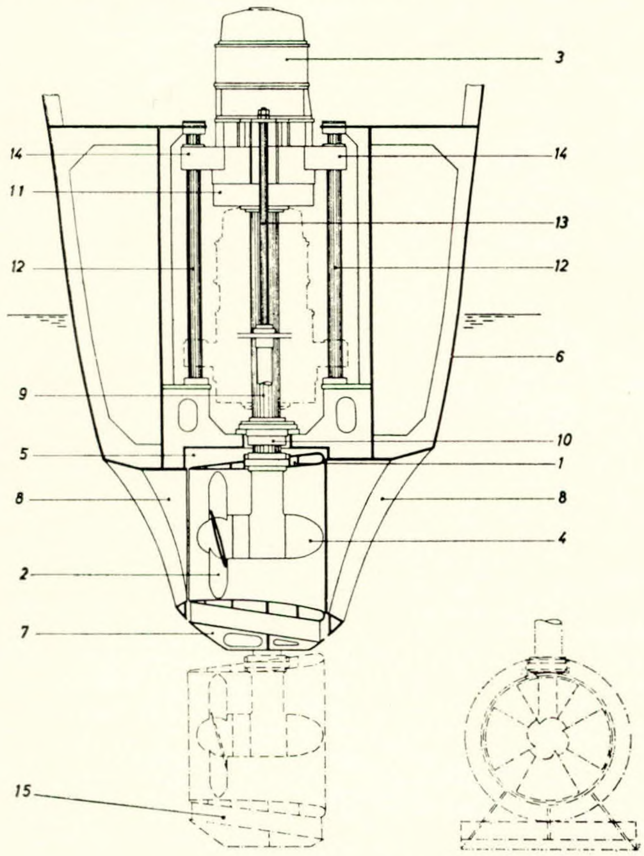
Arrangement for impressed-current system

Bow-thrust unit

Research vessels, cable layers and supply vessels require a high degree of manoeuvrability when operating in the open sea at either very low speeds or stopped.

To assist in this some vessels have been fitted with a Pleuger active rudder or a bow thruster and in some cases with both.

Vessels so equipped can carry on with their work in sea and wind conditions that would force other vessels without such manoeuvring aids to wait for the sea conditions to moderate. Ships fitted with only the traditional form of bow thruster have also found themselves in difficulties manoeuvring in bad weather due to the rise and fall of the bow. If this is severe, the unit may have little or no effect.



1) Centre tunnel section
2) Propeller
3) Prime mover
4) Gear housing
5) Chamber
6) Ship's hull
7) Bottom section
8) Tunnel end sections
9) Rotatable column
10) Lower bearing
11) Crosshead
12) Guide column
13) Lifting cylinder
14) Guide
15) Spaces between ribs

Bow thrust unit

In order to ensure that it is always submerged, a retractable and rotatable type can be used. It is housed inside the vessel when not required for use and lowered through the hull to project beneath the keel when required for operation. In the working position it can be revolved through 360° to direct its thrust in any required direction.

An oceanographic survey ship recently built was fitted with one of these units, which incorporated a controllable-pitch propeller and shell doors.

In the retracted position, this unit can be used as a conventional bow thruster with controllable-pitch propeller after the shell doors have been opened. In the extended position, the shell doors remain closed and the unit is used as a 360° rotatable propulsion unit with controllable-pitch propeller. With a 450 bhp prime mover a thrust of 5·5 tons is developed in the tunnel and 6·25 tons in the extended position.

The propeller pitch is adjusted to the prime-mover output under full load conditions by means of a contact feeler-type ammeter. This ensures that the propeller blades, due to lower hydraulic loss to the propeller in the extended position of the unit, maintain a coarser pitch thus achieving a better thrust than in the retracted position.—*Ship and Boat International, June 1969, Vol. 22, p. 25.*

Improved Reverse-osmosis Process for Desalinating Seawater

Although reverse-osmosis processes have been used successfully to convert brackish water containing 2400 ppm of natural salt to potable water having a salt content of 250 ppm, they have hitherto proved impracticable for the desalination of seawater, which contains about 35 000 ppm of salt. It is therefore interesting to note that an improved process, developed in the U.S.A., is reported to have overcome the limitations of other reverse-osmosis systems and, in fact, a pilot plant embodying it is currently producing 250 gal of drinkable water per day. Even more important, however, is the fact that desalination is accomplished in a single pass through the plant. In this connexion, the general rule, according to the accepted state of the reverse-osmosis art, is that one pass of saline water will reduce its salt content by a factor of 10, so that two passes have been considered necessary to reduce the salt in seawater to less than the maximum of 500 ppm postulated for drinking water. The elimination of this second pass in the new process therefore constitutes a major advance.

Like other reverse-osmosis systems, the pilot plant is based on the application of pressure and the use of a suitable diaphragm or membrane between the salt water and fresh water. It differs, however, in that the membrane is a special design of precisely the correct composition to keep back the salt and allow the fresh water to filter through at an acceptably high rate. More specifically, the membrane is in the form of a tube having a wall thickness of 0·004 in and is cast at room temperature from a mixture of cellulose acetate, formamide, and acetone.

The membrane is wrapped in two layers of a nylon-net material, and the assembly is then inserted into a copper tube of 1 in in diameter and 10 ft in length, with small holes, from which fresh water flows and is collected, at 3 in intervals along its length. In the pilot plant, the daily output of 250 gal of fresh water is produced by an assembly of 27 tubes through which the seawater is supplied under pressure by means of a reciprocating pump.—*Engineers' Digest, September 1969, Vol. 30, pp. 57; 59.*

Open-type Bulk Carrier

Built by Cammell Laird and Company (Shipbuilders and Engineers) for Wm. France, Fenwick and Company, who have fixed the vessel on a long term charter with the Star Bulk Group, *Star Pinewood* is the first of a new series of specialized bulk carriers being constructed by Bergens Mek. Verksted and Verolme's Cork Shipyard, as well as Cammell Laird's, for operation by Star Bulk.

Of all-welded steel construction, the vessel is built to the requirements of Lloyd's Register of Shipping class ✠ 100 A1, ✠ LMC, UMS, "strengthened for ore cargoes", to comply with B.o.T. regulations and, in some respects, with Det Norsek Veritas "F" class. It has a curved raked stem incorporating a bulbous bow, a transom-carrier stern and the superstructure and machinery arranged aft. Transverse bulkheads subdivide the hull into fore-peak and chain locker, five cargo holds with double bottom fuel tanks below and water ballast in the upper wings, machinery space surmounted by the superstructure, and the aft peak, housing fresh water tanks.

Principal particulars are:

Length, o.a.	171·907 m
Length, b.p.	161·544 m
Breadth moulded	25·908 m
Depth, moulded to upper deck ...	14·884 m
Draught, summer	10·770 m
Deadweight, about	29 200 tons
Gross register, about	18 500 tons
Propulsive power	2 × 6520 hp
Speed, trials	15·75 knots

The Kvaerner Piggy-Back hatch covers the first of this design to be installed, comprise two transverse airtight box girder sections to each hatch. In operation two synchronized pairs of hydraulic pillar jacks lift the foremost of the two sections sufficiently to enable the after section, which is mortorized, to roll under it; the raised section is then lowered to ride "piggy-back" on its partner. In this condition the two sections may be operated as a single self-propelled package capable of being driven to any position over the hatch or to any available location over adjacent hatchways.

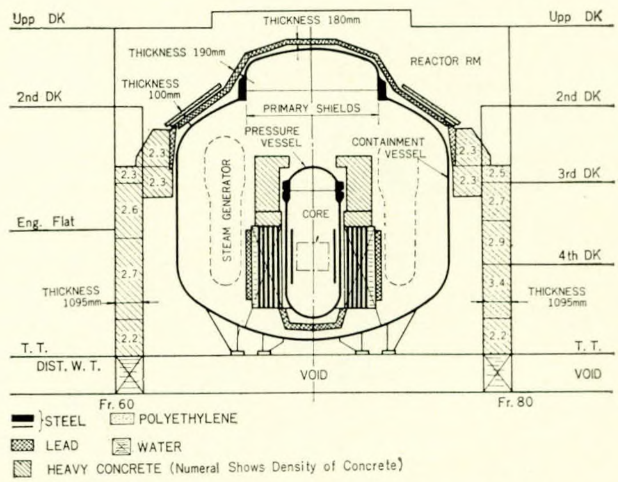
The two Munckloaders, installed on *Star Pinewood,* are each designed to have a safe working load of 25 tons, including the weight of special handling devices such as grab, vacuum or pulp bale clamps, etc. Designed to travel the length of the vessel's hatched area, with a rack and pinion drive system, the cranes incorporate all the established Munckloader principles with all movements being wholly controlled from the operator's wide-vision cab slung under the trolley. The main controls comprise a left-hand joystick for simultaneous gantry and trolley travel and a right-hand lever for the hoist motor. The lower lifting yoke is designed to carry a variety of fittings including magnets for handling scrap, electro-hydraulic grabs for bulk cargo, and pulp bale clamps for the handling of timber products.—*Shipbuilding International, September 1969, Vol. 12, pp. 15–19.*

Japanese Nuclear Merchant Vessel

Launched from the Tokyo yard of Ishikawajima-Harima Heavy Industries (IHI), *Mutsu* will never lay claim to being an attractive ship: she is a stubby three-hold cargo liner type of vessel, with which Japan emphasises her interest in the future of nuclear propulsion.

Principal particulars are:

Classification ...	Nippon Kaiji Kyokai NS (Nuclear ship) and MNS
Length, o.a.	130·00 m
Length, b.p.	116·00 m
Breadth, moulded	19·00 m
Depth	13·20 m
Designed, full load draught ...	6·90 m
Designed, full-load displacement tons	10 400
Gross tonnage	8350
Trials speed	17·00 knots
Speed with auxiliary power ...	10·00 knots
Reactor:	
Type	1 × indirect-cycle light water reactor
Thermal output	36 MW
Average temperature of primary coolant (at 100% output) ...	278°C
Flow rate of primary coolant (at 100% output) ...	approximately 1800 ton/h
Pressure of primary coolant (at 100% output)	110 kg/cm²g
Fuel charge ...	About 2·77 ton of uranium oxide of low enrichment
Main steam generators:	
Type	2 × vertical U-tube
Evaporation (at 100% output) ...	30·6 ton/h each
Steam pressure (at 100% output)	40·0 kg/cm²g
Steam temperature (at 100% output)	Saturated
Main coolant pumps:	
Type	2 × vertical, enclosed-motor type
Capacity, each	900 ton/h
Head	3·5 kg/cm²g
Reactor containment vessel:	
Type	1 × vertical cylinder type
Internal diameter	10·00 m
Internal height, approximately ...	10·6 m
Designed internal pressure ...	12 kg/cm²g
Designed external pressure ...	3 kg/cm²g
Main engine:	
Type	1 set of IHI cross-compound double-cylinder saturated steam turbine with 2-stage gearing
Max. cont. output	10 000 shp × 200 rev/min
Normal	9000 shp × 193 rev/min
Propeller	1 × constant-pitch 5-bladed solid type



Radiation shields fitted to reactor—Mutsu

By the end of January, 1972, Japan's major step towards a leading part in nuclear propulsion will enter service.—*The Motor Ship, July 1969, Vol. 50, pp. 183–185.*

Automated Cutter Dredger for French Owners

The cutter dredger *Maria-Carolina,* built by I.H.C. Hollandyard Smit Kinderdijk for a Panamanian contractor, is a modified version of the I.H.C. Stancutter 8000. The Stancutter series of dredgers consists of three basic non-demountable models of higher capacity than those of the I.H.C. *Beaver* and *Giant* series, which are standardized craft required to be demountable for transport by road, rail or ship.

Maria-Carolina has been built to Bureau Veritas class 3/3D1.1. and the equivalent R.I.Na classification. She is powered by a Diesel-electric installation. Her pontoon measures (54·80 m (179 ft 9 in) by 13·00 m (42 ft 8 in). Her cutter is driven by two electric motors with a total m.c.r. of 1200 hp. These are mounted on the ladder and are fed by two generators coupled to an M.A.N. Diesel engine type V6V22/30 ATL. This engine is a 12-cylinder four-stroke supercharged Diesel engine with charge-air cooling and direct injection. It has a continuous output of 1560 ehp at 900 rev/min. Twin dredgepumps of an I.H.C. pattern are installed and these operate in tandem. Each is driven by a nine-cylinder Smit-M.A.N. Diesel engine developing 2950 hp at 280 rev/min. A fourth Diesel engine drives the generator supplying current for lighting and general purposes. This is also an M.A.N. Diesel engine. It is on the type R6V22/30 ATL, a six-cylinder, four-stroke in-line engine, supercharged and having charging-air cooling and direct injection. Its continuous output is 860 ehp. A standby generating set brings the total output of the machinery installed to more than 8000 hp. There is an auxiliary set, consisting of a Lister Diesel engine type JK4MA, of 62 hp at 1500 rev/min, which drives a 40 kW Smit-Slikkerveer generator and a Hatlapa compressor, type LHD 18 N, with a capacity of 30 m^3/h at an endpressure of 30 kg/cm^2. All of these are built together on a common bedplate.

Maria-Carolina is equipped with anchor booms having a radius of 83 ft 8 in, and 109 ft long, hydraulically operated spuds. No spud carriage is installed. All dredging operations are controlled from a console situated in the elevated cabin.

Maria-Carolina incorporates a very high degree of automation. In addition to loading rate, mixture concentration and production indicators, she is fitted with an automatic cutter controller developed by I.H.C. Holland. This is of the digital type and its function is to regulate the forward swing winches and the vacuum relief valve in such a way that maximum production is achieved without

a) overloading the cutter motors;
b) overloading the swing winches;
c) sedimentation occurring in the delivery pipeline;
d) choking of the dredge pump as a result of excess vacuum;
e) exceeding the limits of the planned channel;
f) interruption of the dredging process following, for example, the collapse of a vertical face.

A digital computer was chosen in preference to the analog type in view of the greater reliability which it affords, bearing in mind that many of the data processed in this equipment are digital.—*Holland Shipbuilding, September 1969, Vol. 18, pp. 62–64.*

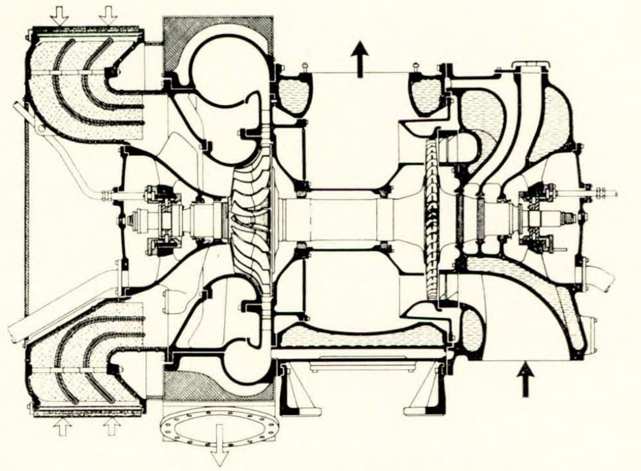
Turbocharger for Super Bore Engines

Brown Boveri has recently made available a further turbocharger in its VTR range of units which, it is claimed, will be suitable for all expected increases in engine output in the foreseeable future.

The VTR 900 turbocharger comprizes a single-stage axial gas turbine driving a single-stage centrifugal compressor and, as previously mentioned, is a direct development of the smaller VTR-type turbochargers. Designed for a maximum pressure ratio of about 3:1 with a maximum speed of 7600 rev/min, the blower is able to operate with exhaust gas temperatures of up to 580°C and can be used for charging two-stroke engines using either the constant pressure or the pulse system to produce up to 4200 hp per cylinder. Six units to be supplied for the Eriksberg-B. and W. 8K98FF engines will each have an air throughput of 30 m^3/sec at a pressure ratio of about 2:1 with a speed of 6200 rev/min on maximum continuous rating, corresponding to an output of about 3750 hp per cylinder. Overload speed is 6600 rev/min which corresponds to 4200 hp/cyl with a pressure ratio of 2·35.

The compressor volute casing is of cast iron, instead of light alloy, in order to reduce the noise level of the blower.

The gas turbine wheel and shaft are forged in one piece, and the solid turbine blades, of austenitic chrome-nickel steel, are of the fir-tree root type, mounted in longitudinal slots in the periphery of the wheel. The blades, about 8 in long, are supported by a bracing wire of about $\frac{3}{8}$ in diameter and are reduced at the tips to a knife-edge section to minimize damage should a "rub" occur. The compressor



Brown Boveri VTR 900 turbocharger

rotor is constructed of a light alloy known as peronal and is in two parts, the inducer and the impeller being manufactured separately then shrunk and keyed to the shaft, the complete shaft weighing 1·2 tons. A heat-resistant partition separates the turbine rotor from the compressor and is arranged as part of the rotor assembly so that the rotor can be withdrawn easily, when required, without disturbing the exhaust or scavenge connexions: special overhauling tools are provided.—*The Motor Ship, July 1969, Vol. 50, pp. 170.*

Some Problems in Crankshaft Design

Although the problem of axial vibrations of crankshafts is not of comparable importance to that of torsional vibrations, it may be useful to calculate their natural frequencies as accurately as possible. This article is intended to show how it is possible to check the accuracy of the axial vibrating system on the basis of recorded resonance amplitudes at both ends of the crankshaft.

The same dynamic equations of the torsional vibrations are also used for the axial vibrations, the moments of inertia being replaced by the masses, and the torsional elasticities by the longitudinal ones. An additional relationship, which does not appear in the torsional vibrations, is given by the response forces of the thrust bearing. For a crankshaft coupled directly with a propeller, the elasticity of this connexion is that of the thrust bearing girder added to that of the foundation, as long as the constant thrust is larger than the periodic thrust variation. When no constant thrust is provided, the axial clearance of the thrust bearing acts in accordance with the natural frequencies giving additional elasticity to the girder.

Axial vibrations can be excited by the engines as well as by the propeller. The sources of the engine excitations reside in the radial harmonic components of the gas forces and the peaks of those forces at the moment of firing act like a succession of shocks. These "shocks" do not affect the torsional vibrations, since their tangential components are negligible.

There is, indeed a coupling effect between the torsional and axial vibrations, due to the axial contraction in a twisted shaft portion. However, only the natural frequencies of very high modes are affected by this effect and, therefore, calculations can be made separately for both kind of vibrations. Nevertheless, recorded axial vibrations often show traces of the main torsional resonances, which are considered as forced vibrations due to the coupling effect in the axial direction of the periodic torque variations. For instance, the angular amplitudes of the propeller at critical speeds of torsional vibrations produce axial force variations due to the inclination of the blades which must be considered. The axial excitations generated by the propeller have a frequency corresponding to the number of blades multiplied by the rev/min, and there is a coupling effect between the bending vibrations of the blades and the axial vibrations of the crankshaft. The mass of the propeller should, therefore, be introduced in two parts, separated by an equivalent, or adequate, axial elasticity.—*Beguin, F., Special Survey, The Motor Ship, October 1969, Vol. 50, pp. 91–93.*

Automation

A recommended Code of Procedure for Marine Instrumentation and Control Equipment has been produced by the British Ship Research Association to provide a lead in ships' automation work. It has been distributed to British shipbuilders and shipowners.

Preparation of the Code was undertaken by BSRA at the request of its members who felt that there was an urgent need for guidance in the design and installation of merchant ship instrumentation and control systems. While preparing the Code, BSRA received great assistance from a panel comprizing representatives of automation firms together with specialists from shipbuilding and ship-owning companies in the U.K.

A guide to good practice and an aid to designers in the field of marine automation, the Code identifies the many aspects of instrumentation and control engineering in the marine environment which require understanding among shipbuilders, shipowners and sub-contractors. There is no other code available elsewhere giving a similar degree of detailed practical guidance. While it has been prepared primarily for use in the U.K., it is expected that the Code will also be in demand abroad for use in negotiations between shipbuilders and shipowners.

Control design procedure is outlined through all stages from the basic requirements of a system to its commissioning: for example, one chapter is entirely devoted to documentation and is intended to ensure that proper information is set out during the tendering and contracting stages of supply. Chapters on installation and commissioning give guidance in the conduct of these further stages of instrumentation. An important appendix is devoted to recommended environmental tests for marine equipment.

Future BSRA policy will be to extend and up-date the Code, which is provided in loose-leaf format, to preserve its value as leading documentation in a rapidly developing field.—*Shipbuilding International, September 1969, Vol. 12, pp. 11.*

Research Vessel

Built to Norske Veritas Class ✠ 1A1, IceC, EO, *Karen Bravo* is a shelterdeck vessel with soft roundnosed stem and transom stern. The cargo hold is located amidships and has a clear hatch opening on the shelter deck of 63 ft × 26 ft 3 in whilst the main deck clear hatch opening is 66 ft 11 in × 26 ft 3 in. Steel folding hatch covers are employed on the shelterdeck whilst flush fitting pontoon-type covers are used for the main deck hatch.

Deck machinery comprizes a Bodewes electric windlass, with two warping drums forward and a double-drum, 1½ ton electric winch by the same manufacturer immediately aft of the accommodation superstructure. This latter serves a three-ton derrick. In addition there is a two-ton capstan on the starboard side aft. A class C polyester lifeboat is carried under a single-arm davit on the starboard side of the bridge deck, the arm being equipped with a boom to support the boat at the forward and aft ends. An inflatable liferaft is carried on the port side of the deck. The steering gear is of Tennfjord manufacture.

Main propulsion machinery comprizes a June Munktell type V10-260M Diesel engine of 1200 hp at 425 rev/min together with a type 340 marine box for the Berg controllable pitch propeller which is driven directly. This high propeller speed was selected after taking into consideration the reduced draught when *Karen Bravo* is operating as a research vessel. The June Munktell V10-260M is a four-stroke pressure charged and intercooled 10-cylinder 45°V-form engine having a bore of 260 mm and a 400 mm stroke. A low V-form propulsion engine was chosen for this vessel as it permitted a flush deck over the engine room. The engine is located at the aft end of the engine room resulting in a very short shaft requirement.

Karen Bravo should only be regarded as the first of a series as the owners wish to try out this unconventional vessel in a variety of trades before a final decision is made regarding the design.

The design was based on experience gained from a number of ships involved in geophysical research at sea. In addition to being able to operate as a research vessel, the owners required that *Karen Bravo* should be able to compete with conventional coasters in conventional trades, including carrying containers. After considerations with regard to tonnage, complement, cubic capacity, etc., it was determined that the optimum size of the vessel would be 399 gt, a dwt of about 1000 and that it should have an easily accessible cargo space which did not have to be abnormally large.—*Shipbuilding International, September 1969, Vol. 12, pp. 42–43.*

New Electronic Analyser for Diesel Engines

A new device which promises to become a major tool for rapid diagnosis of Diesel engine performance has been developed jointly, by the Austrian research company of Anstalt für Verbrennungsmotoren (AVL) and Mobil Oil. Using electronic equipment, the unit operates by detection and measurement of instantaneous pressures in the injection system lines of an engine. With this system, pressure signals from the engine's No. 1 cylinder are used to produce synchronization with crankshaft rotation. Signals are transmitted by calibrated quartz pressure transducers which are easily inserted in the lines. No other connexions to the engine are required and hook-up can be made quickly in the laboratory or repair shop.

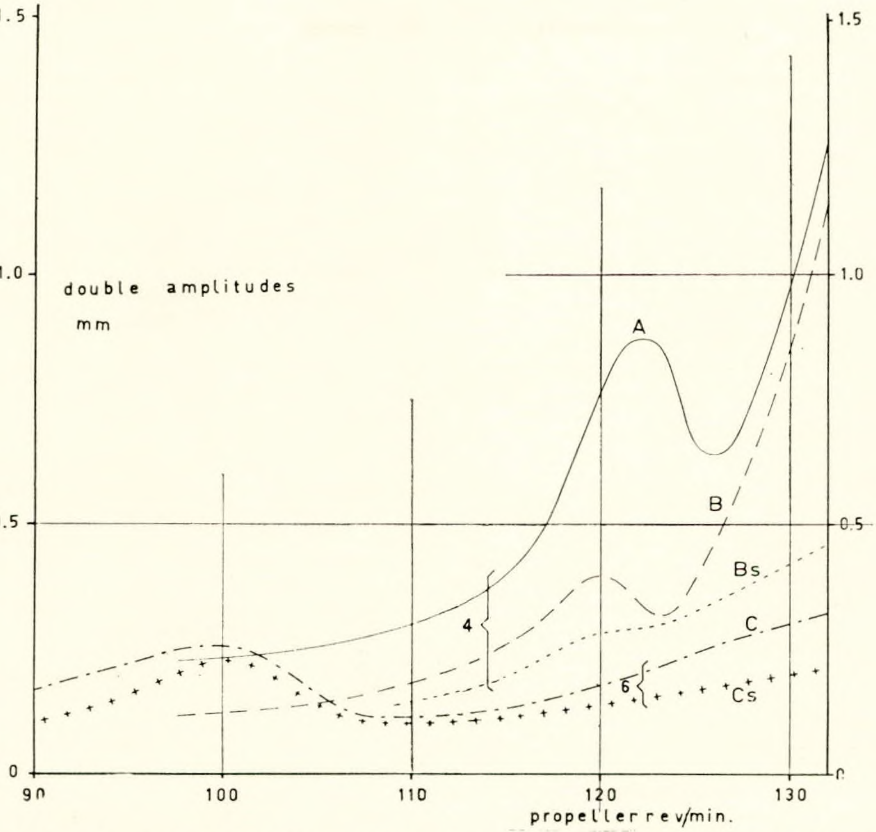
Traces for each injection line are displayed on a conventional cathode ray engine analyser, and shown in a vertical arrangement to permit immediate comparisons. The instrument also includes circuitry which gives precise measurement of each nozzle static opening pressure as well as timing advance at various speeds.

The pressure patterns evolved provide information on functioning of the injection pump, delivery valve and nozzle. Extensive testing in engine laboratories and repair shops, with many engines of different manufacture, has indicated that numerous types of operational problems can be detected. Among these are worn pump plungers, cams, and camshaft bearings, broken plunger springs, incorrect pump volume settings, worn or incorrect delivery valves, stuck or intermittantly sticking nozzle needles, leaking or choked nozzles, and high or low nozzle opening pressure.

Two principal applications are envisaged for this advanced electronic analyser. One is in research laboratories conducting Diesel development work where it is expected to save much time in establishing the performance of research engines. The other is a relatively compact, easily used diagnostic unit aboard ships and at stationary installations, also in maintenance shops for smaller engines.—*The Motor Ship, September 1969, Vol. 50, pp. 290.*

Vibration Cured by Increased Number of Propeller Blades

Vibration troubles are one of the problems for marine engineers and naval architects alike. Many cures are possible, from additional localized strengthening to dampening either



Double amplitudes of longitudinal vibrations on top of gear-box—Curves A, B and C recorded before six-bladed propeller was fitted—Curves C and C's show final results

in the vicinity of where movement is greatest or in the prime mover if it is reciprocating machinery which is required to operate at a set speed. However, in the case of vibration on a vessel having a rotary motor prime mover, i.e. steam turbine, the cause is more likely to be faulty propeller design.

The effect of the number of blades of a propeller under certain conditions is well known, and a recent cure for severe vibration on a 5120 gross ton training ship *Pedro de Alvarado* clearly showed the influence of this fact.

The turbine-driven ship was initially equipped with a four-bladed propeller transmitting 7000 hp at 140 rev/min. Excessive longitudinal vibrations were observed at the top of the gearbox and vertical vibrations at the bridge superstructure.

On the advice of Bureau Veritas, extra stiffening was added. This gave an improvement which is shown by curves B and Bs in the accompanying diagram. The bridge construction was stiffened by adding a pillar located in way of the largest amplitudes.

Frequency of vibration under various conditions:

Curve	*Frequency*	*No. of blades*	*Condition*
A	4 N	4	With four-bladed propeller, before measures were taken.
B	4 N	4	Bearings and double bottom structure reinforced.
Bs	8 N	4	As B and with stretches between bearings and tunnel.
C	6 N	6	With six-bladed propeller; reinforcements as at B.
Cs	6 N	6	As C and with stretchers having rubber connexions between them.

Although the foregoing improvement was achieved, the vibration level was still not yet satisfactory. Therefore, on the advice of Lips propeller works, the four-bladed propeller was replaced by a six-bladed one, produced by the factory Navalips in Cadiz. This provided an adequate solution to the problem as can clearly be seen from the other plots in the diagram. Only a sixth order vibration of minor amplitude remains; curves C and Cs show the results. The increase in the number of propeller blades had no effect on the ship's speed.—*The Motor Ship, May 1969, Vol. 50, p. 73.*

Tanker for Great Lakes Service

Marine Industries Ltd. has recently completed the 8500 dwt products tanker *Lakeshell* and delivered her to Shell Canada Limited. In service the tanker will transport petroleum products on the Great Lakes and the East Coast from Fort William to St. John, NB.

Of all-welded construction, the hull has been strengthened for navigation in ice to Lloyd's Class 3, and the ship is one of the few Canadian-built tankers that complies almost in all respects with the proposed requirements of the International Convention on Load Lines, 1966.

Six transverse oil-tight bulkheads and two longitudinal bulkheads divide the cargo section into 15 tanks. The transverse bulkheads are corrugated as are the wash bulkheads which are arranged midway along each tank. A slop tank is incorporated into No. 5 port wing tank.

In order to prevent corrosion the cargo tanks, together with fuel oil and ballast tanks, have been coated with epoxy paint and given a final coat of International Paints' Intergard. Provision has been made for the installation of a complete system of helical type cargo heating coils in the cargo tanks. The coils, although actually fitted in only two tanks, are standard and completely interchangeable, enabling re-location or removal to suit the cargoes carried. A unique tank venting system coupled with a closed cycle tank cleaning system allows automatic tank cleaning and gas freeing with a minimum of effort on the part of the crew.

The pump room, located just forward of the engine room, contains three electrically-driven Stothert and Pitt cargo pumps. These horizontally screw displacement-type pumps each having a pumping capacity of 380 tons of cargo per hour. The pumping system is so designed that three separate grades of products can be loaded or discharged, completely segregated at all stages of the operation, by using local or remote control push buttons for starting and stopping of the cargo pumps.

For main propulsion, a Ruston & Hornsby two-stroke turbocharged eight-cylinder 8AO medium speed Diesel engine capable of developing 4000 bhp at maximum continuous power, and with a service output of 3800 bhp, is installed. The engine is connected to a Modern Wheel Drive, vertically stepped reduction gearbox and turns the controllable pitch propeller at 150 rev/min. Control of the engine can be effected from three places in the wheelhouse, from the main control room and, for emergency use, from the floor plate level. The two generators, also Ruston & Hornsby supply, are equipped with an automatic synchronizing system for paralleling operations. The total bunker capacity is sufficient for not less than 6250 miles steaming and *Lakeshell* will operate at a service speed of almost 13½ knots.—*Shipbuilding International, September 1969, Vol. 12, pp. 20–21.*

Japanese Paper Roll and General Purpose Cargo Carrier

A small cargo vessel containing a number of interesting features on deck, in the holds and in the engine room, has been built by the Miho Shipbuilding Co. Ltd., Shimizu, for the Kuribayashi Steamship Co. Ltd. of Tokyo. This vessel, *Shinju Maru* 3084 dwt, has been specially designed for the carriage of paper rolls for newsprint from Tomakomai, Hokkaido, to Tokyo; and automobiles, containerized and palletized general cargo on the return trip.

The cargo handling arrangements on board *Shinju Maru* are extensive and include two sets of K-7 overhead cranes, each of three tons s.w.l., which run through the 'tweendeck space; one K-7 five-ton s.w.l. derrick forward and a K-7 reversible derrick of ten tons s.w.l. amidships. There is also an elevator on the starboard side of the cargo hold, which can be used to handle trucks, etc. between the two decks.

Principal particulars are:

Length, o.a.	312 ft 11 in
Length, b.p.	284 ft 8½ in
Breadth	47 ft 10⅝ in
Height, keel to main deck	21 ft 7¾ in
Height, keel to shelter deck	38 ft 4½ in
Draught, maximum	19 ft 8½ in
Deadweight	3084·26 tons
Gross tonnage	2175·910
Cargo capacity	211 800 ft³
Ballast	37 165 ft³
Newsprint rolls	3256
Trial speed (mcr)	15·80 knots
Service speed (85 per cent mcr)	14·50 knots
Cruising range	4500 sea miles

The overhead crane for the 'tweendeck was developed while the vessel was under construction. It has no built-in prime mover but is driven by wires from hydraulic winches installed in a separate compartment at the forward end of the main deck and remote controlled electrically.

Because there is no motor in the crane it is so compact that the overall height is less than 0·8 m. The rail length is 60·5 m and its range covers the entire cargo hold. Each of the two cranes on board has a rate of travel of 60 m/min, the traverse is 30 m/min and the hoisting rate rate 30 m/min. It is claimed that the efficiency of this system is higher, and the cost of operation less, than that of the conventional fork-lift truck.

The propelling machinery in the *Shinju Maru* consists of a nine-cylinder in-line N.K.K.-SEMT-Pielstick type 9PC2L trunk-piston Diesel engine developing 4185 shp at 500 rev/min, reduced to 260 rev/min through gearing. The main engine, as well as the auxiliaries, have been designed to be operated on low grade fuel oil, and to ensure that this is satisfactory a special system of fuel oil treatment has been installed

The conventional centrifugal treatment of fuel oil is not employed. In place of this a K-7 strainer and K-7 ultrasonic fuel oil reformer (U.S.R.) has been installed. The K-7 strainer is a unit of very fine mesh about 10μ, which is cleaned automatically by the oil flowing down and counterflowing through the admision of compressed air. This strainer is installed between the settling tank and the daily service tank. A sensor detects the presence of any water and rings an alarm bell if there is the slightest quantity present.

Fuel oil to the engine from the service tank is treated by the K-7 U.S.R. unit before entering the fuel injectors.

The K-7 U.S.R. unit homogenizes the fuel oil by means of an ultrasonic generator, thus breaking up the heavy molecular material so that the light fractions are not rapidly burned while the heavy fractions burn more slowly and form carbon: the water particles that may be contained in the oil are emulsified and the combustibility of the fuel oil in the cylinder increased.—*Shipping World and Shipbuilder, November 1969, Vol. 162, pp. 1560–1564.*

Dutch-built Product Tankers

The first of two 2300 dwt product tankers, *La Hacienda*, ordered by Buries Markes Ltd. from Wilton-Fijernoord N.V. is now in service.

The principal feature of the Dutch-built vessel are, of course, the cargo tanks, which are all stainless steel construction and enable the *La Hacienda* to carry about 108 different types of chemical cargoes. The possibility of having coated tanks or tanks constructed of steel and clad with stainless steel was considered, but was eventually rejected because of the difficulties envisaged during construction and of maintaining the very thin stainless steel skin contact.

Principal particulars are:

Length, o.a.	264 ft 1¼ in
Length, b.p., ...	246 ft 0¾ in
Breadth, moulded	41 ft 4 in
Depth, moulded	19 ft 8¼ in
Draught, design	17 ft 0 in
Corresponding dwt	2300
Service speed	12 knots

The ship is built to Lloyd's Register of Shipping, Class ✠ 100 A1 "Oil and Chemical Tanker," Ice Class 2, and to U.K. B.o.T. requirements. The hull is of double-skin construction, with water ballast double-bottom and side tanks. There are two fore and aft gangways, on the main deck, as well as a more conventional flying bridge.

The main propulsion machinery, supplied by the Ruston engine division English Electric Diesels Ltd., comprises two Ruston six-cylinder compact engines of the AP3M type. Both engines can be controlled from the bridge, and each has a maximum continuous output of 1000 bhp at 1000 rev/min; they are coupled to a twin-input single-output MWD single-reduction gear-box, and thence to a KaMeWa c.p. propeller.—*The Motor Ship, September 1969, Vol. 50, pp. 259–260.*

On the Reduction of Slamming Pressures

This paper presents a revised and improved theory of flat water impact which takes account of the air trapped between the falling body and the water surface. This is shown to provide a good general description of the physical situation. Two sets of experiments are described: in the first set, pressure measurements on a flat plate were made which showed that end flanges would sharply reduce the peak pressures because they entrapped a greater volume of air; the second set of experiments was on a small scale Marine model in head seas. They showed that slamming pressures under the forefoot could be sharply reduced if the air cushion was artificially reinforced.—*Paper by Lewison, G. R. G., presented at a joint meeting of the Royal Institution of Naval Architects and the North East Coast Institution of Engineers and Shipbuilders, 3rd November 1969.*

On Modelling Cavitation Damage

Similar to the definition of the intensity of cavitation damage, the intensity of cavitation bubble collapse is defined as the power transmitted per unit surface area of the bubble when the collapse pressure is a maximum. The efficiency of damage given by the intensity of damage divided by the intensity of collapse is shown to depend principally on the dissolved gas content of the liquid (using the data obtained from the vibratory cavitation damage apparatus).—*Thiruvengadam A., Journal of Ship Research, September 1969, Vol. 13, pp. 220–233.*

Investigation of Shaft Relief Grooves in Tailshaft Assemblies

Various relief grooves, located in the shafts between the liners and the hubs, were investigated with regard to their effects on the fatigue strengths of conventional tailshaft assemblies. It was found that tailshaft assemblies could be designed which would fail in the groove, regardless of whether single or increment loads were used, and that fatigue strength of these grooved assemblies was higher than the fatigue strength of comparable ungrooved tailshaft assemblies.—*Calarmari, P. L., Marine Technology, July 1969, Vol. 6, pp. 274–280.*

System for the Computer-aided Structural Detailing of Ships

The U.S. Naval Ship Engineering Centre is sponsoring the development of a computer system whose output will be information necessary to fabricate and assemble the structural components of naval surface ships. This output will be in the form of graphics, printed output and numerical control tapes. The system, given the acronym CASDOS for Computer-Aided Structural Detailing of Ships, is now at a point where implementation trials can begin.—*Romberg, B. W. and Roth, C. E., Marine Technology, July 1969, Vol. 6, pp. 303–317.*

Analysis of Impeller and Volute Losses in Centrifugal Fans

The measured losses from tests on five centrifugal fan impellers of different blade angle, running at constant speed in a given volute, are analyzed. Impeller and volute losses, expressed as a fraction of the dynamic exit pressure relative to the impeller and volute respectively, are correlated with a diffusion factor over a wide volume flow range. The results are applied to other impellers and volutes.—*Paper by Myles, D. J., presented at a meeting of The Institution of Mechanical Engineers, 12th November 1969; paper No. P14/70.*

Contrarotating Propeller Propulsion—A State of the Art Report

This report presents a synthesis of existing knowledge on C.R.P. as applicable to merchant ship propulsion. It presents propulsive performance data for two types of ships, tankers and containerships, based on model tests, and discusses briefly the stopping, cavitation and propeller-induced vibration problems of C.R.P. From the analysis it appears that the greatest potential for employment of such a propulsive device, offering significant reductions in installed s.h.p. is on high-powered, fine-formed ships for which the only current alternative is a twin-screw installation.—*Hadler, J. B., Marine Technology, July 1969, Vol. 6, pp. 281–290.*

New Design Measures to Reduce Siren Tones Caused by Centrifugal Fans in Rotating Machines

The interaction between the fan in a compact rotating electrical machine and axial ribs of the surrounding casing can often give rise to troublesome siren tones. Experimental studies on a model fan show that one way to overcome these tones is to modify the ribs: ribs of trapezoidal cross-section, for example, with the short parallel side facing the impeller, lower the intensity of the tone by up to 7 dB, compared with the conventional rectangular cross-section.—*Ploner, B. and Herz, F., Brown Boveri Review, 1969, Vol. 56, No. 6, pp. 280–287.*

Influence of Flue Gas Recirculation on Combustion of Fuel Oil

A review of the influence of recirculation, particularly of cooled combustion products, on such flame properties as temperature, combustion intensity, carbon formation, heat transfer, and SO_3 formation has been carried out. Whilst the influence of recirculation on the combustion of gases and distillate fuels is now reasonably well understood, no extensive comparative data are available for residual oils.

A rig consisting of a vertical, refractory-lined water-cooled surface, 10 in inside diameter by about 12 ft total length was used.—*Monaghan, M. T. and McGrath, I., Journal of the Institute of Petroleum, September 1969, Vol. 55, pp. 303–321.*

Combined Marine Power Plants

In combined ship power plants two types of main drive engines connected with the propeller shaft are installed. The drive unit preferably constitutes a combination of Diesel engines, gas turbines and steam turbines integrated in different systems. The drawback of such power plants is their complexity and the necessity of using specially designed transmission gears, clutches and some other elements.—*Nocon, P., Budownictwo Okretowe, 1968, No. 6, pp. 202–205; Polish Technical and Economic Abstracts, 1969, No. 1, Abstract No. 12131.*

Fatigue Tests on a Diesel Crankshaft

This paper describes fatigue-test rigs developed at the Admiralty Engineering Laboratory, West Drayton, to test single-throw crank test pieces cut from multi-throw Diesel engine crankshafts under bending and under torsion. The pieces tested under torsion indicated a fatigue limit for failure at the oil hole under a stress range equal to the tensile strength of the material; the effects of imperfect finish to the oil hole surface and of non-metallic inclusions in the material were also shown.—*Paper by Barton, J. W., submitted for written discussion to The Institution of Mechanical Engineers, 1969; paper No. P13/70.*

Low Cycle Fatigue at Elevated Temperatures

In order to clarify the dependence of the number of strain cycles to failure upon temperature and frequency of straining in low cycle fatigue a number of fatigue failure tests were carried out by the authors. In order to gain further insight into the process of fatigue failure, a comparison of the cracks appearing on the failed specimens was undertaken. The fatigue tests were carried out in a push-pull testing machine on AISI 321 stainless steel specimens at various temperatures and frequencies of straining conditions.—*Kanazawa, K., Iwanaga, S., Kunio, T., Iwamoto, K. and Ueda, T., Bulletin of the Japan Society of Mechanical Engineers, 1969, Vol. 12, No. 50, pp. 188–199.*

Automated Cracked Nut Sorting with Eddy Current Test

An automated system has been developed for the inspection and sorting of nuts at the rate of 100 parts per minute or greater. The system has been designed to accommodate a variety of nut shapes and sizes with minor tooling changeover and set-up adjustment. The multistation system utilizes eddy current NDT instrumentation to detect forging cracks, seams, bursts, quench cracks, etc. on both the bearing surfaces and crown surfaces of nuts.—*Niskala, J. H. and Carson, R. D., Materials Evaluation, July 1969, Vol. 27, pp. 153–158.*

On the Strength of Container Ships

The first container ship in Japan, *Hakone Maru*, was recently completed. Many important problems are to be solved in the design of container ships. Distortion of hatch openings of a container ship is six times as large as that of an ordinary cargo ship. The major problem in the design of a ship with a large hatch opening is to investigate torsional behaviour. Theoretical and experimental investigations were carried out in this respect.—*Nakagawa, M., Umezaki, K., Mori, M., Hagihara, K. and Terada, K., Mitsubishi Technical Bulletin, May 1969, No. 57.*

Solid State Control for Shipboard Power Plants

Dramatic changes in specialized ship construction and in power plant design involving combined cycles and new machinery approaches are on the horizon. Control systems will employ a greater use of micro-miniature electronics, stored programs, and time-shared techniques. The extent of these systems will overcome the engine room boundaries and extend to the limits of the ship. Navigation, cargo handling, inventories and other ship functions will be included. Modular construction of complete shipboard control systems will be required.—*Nitsch, A., Naval Engineers Journal, June 1969, Vol. 81, pp. 113–121.*

The Consumable Nozzle Electroslag Welding Process with Movable Current Terminal

The consumable nozzle electroslag welding process with movable current terminal, is an automatic singlepath vertical welding method as is the ordinary type of consumable nozzle welding. The wire is fed through a stationary, consumable nozzle of non-flux coating to the weld area, and weld metal is continuously cast under a shielding medium which is a blanket of molten flux. In the new process, both a current terminal and copper shoes are designed movable upwardly with weld progression.—*Nakajima, M., Ishikawa, Y. and Shimamoto, T., Mitsubishi Heavy Industries Technical Review, 1969, Vol. 6, No. 1, pp. 1–9.*

Methods of Studying the Effects of Blast Waves on Ships' Superstructures

Techniques are described which have been used to predict the possible effects of blast waves on ships' structures. The basic physical properties of a blast wave, the factors which affect these properties, and the techniques for measuring them are discussed. The interaction of shock waves with scaled rigid models is studied in the laboratory and the results are used to predict the blast loading on a full-scale structure.—*Dewey, J. M., Marine Technology, July 1969, Vol. 6, pp. 268–273.*

General Review of Marine Fuel from its Origin in Crude Oil to its Ultimate Use

A subject of interest and concern to all marine engineers is the fuel oil which supplies the energy required to operate various marine craft. This paper describes the chemical and physical properties of the hydrocarbons which make up fuel oil, the refining processes used in its manufacture, the various types of fuel oil marketed and some of their properties as well as reviewing the basic fundamentals of fuel oil from its origin in crude oil to its ultimate use.—*Allen, J. B., Transactions of the Institute of Marine Engineers, Canadian Division Supplement, September 1969, No. 37, pp. 15–21.*

Experiments on Supercavitating Plane Hydrofoil with Jet-flap

An experimental investigation of the supercavitating flow around a plane hydrofoil equipped with a jet-flap at the trailing edge has been carried out in a water tunnel in which the working section has rigid walls. Results are presented for a range of cavitation numbers between 0·3 and 1·2, and for jet momentum coefficients up to 0·5. The pressure forces exerted on the model have been calculated from the measured distributions of static pressure on the surface of the model.—*Dinh, N. N., Journal of Ship Research, September 1969, Vol. 13, pp. 207–219.*

Trends in the Offshore Oil Industry in Relation to Marine Environment

This paper describes the wide range of specialized marine structures and craft used by the offshore oil industry. An attempt is made to assess the future trends in the development of equipment and structures used by the offshore oil industry, and is of potential interest to the naval architect and shipbuilder.—*Paper by Bell, A. O., Brereton, A. F., Marriott, G. B., Stras, J. C. and Wilson, R. O., presented at a joint meeting of the Royal Institution of Naval Architects and The Liverpool Engineering Society, 15th October 1969.*

Marine Environment—Some Features of Concern to Naval Architecture

Those physical features of the marine environment which are of concern in the design and operation of ships, offshore platforms and other structures are described. These include the effects of waves, tides and surges, currents and changes in temperatures and density. The propagation of light and sound and the properties of the sea bed are other features of importance in certain applications. Emphasis is given to those aspects which are of increasing significance in view of new developments in underwater technology, and techniques for measuring the relevant parameters are discussed.—*Paper by Bowden, K. F., presented at a meeting of The Royal Institution of Naval Architects, 9th October 1969.*

Massive Recirculation as a Method of Minimizing Corrosion in the Combustion of Residual Fuels

Metal surfaces exposed to combustion gases from residual fuels suffer from fouling and severe corrosion. Several methods currently in use to combat these effects include chemical treatment of the fuel, combustion at near stoichiometric conditions, and the use of protective coatings. This study introduces another method based primarily on improved burner technology, specifically, combustion with massive, external recirculation.—*Reba, I., Transactions of the ASME, Journal of Engineering for Power, July 1969, Vol. 91, pp. 198–206.*

Cavitation in Centrifugal Pumps

A theoretical model is proposed to explain the development of cavitation in a centrifugal pump. A basic assumption is that the pump always behaves as a head generating-volume flow device, hence the density of the cavitating fluid becomes important, and this is estimated assuming that cavitation is a bubble growth problem. Essentially the model applies to best efficiency flowrate and complete breakdown, and is semi-empirical since test data at ambient conditions are required. From these data, predictions are made of the variation of breakdown and inception points with temperature and speed.—*Paper by Chivers, T. C., submitted to I.Mech.E.; paper P2/70, 1969.*

Application of Microwaves to Non-Destructive Testing

The properties which make microwaves a suitable tool for certain non-destructive testing applications are outlined and some aspects of microwave engineering are explained. Examples of the application of microwave techniques to the detection of flaws, study of composition, measurement of dimensions and high frequency eddy-current testing are discussed and possible future applications are outlined.—*Owston, C. N., British Journal of Non-destructive Testing, June 1969, Vol. 11, pp. 26–30.*

Ultrasonic Crack Growth Monitor

An ultrasonic nondestructive test procedure has been developed to measure and record the extent of crack growth encountered in fatigue and stress corrosion tests involving the Wedge-Opening-Loading fracture toughness specimen. The essence of the technique is to relate the position of an ultrasonic transducer on the specimen surface to the tip of the propagating crack such that crack length can be interpreted in terms of transducer location.—*Clark, W. G. and Ceschini, L. J., Materials Evaluation, August 1969, Vol. 27, pp. 180–184.*

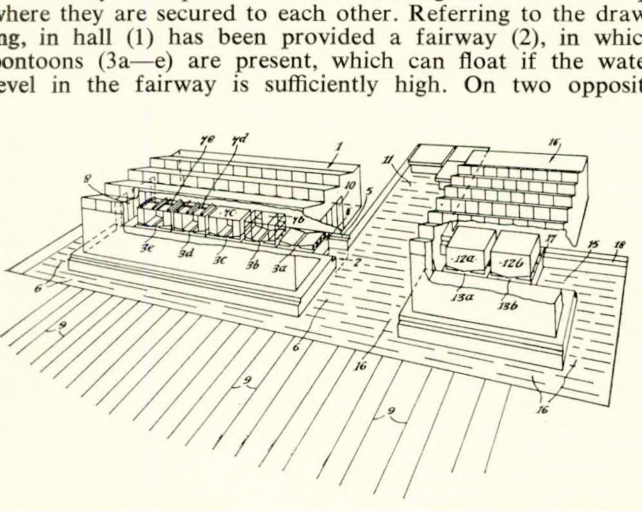
Pump Driven Lateral Thrust Unit with Ejector Augmentation

The idea of using cargo pumps to improve the low speed manoeuvrability of tankers is evaluated. The main drawback of this system is the high kinetic energy loss in the jets, which results in an unfavourable thrust-to-installed-pump-power ratio. To improve the system, the exit velocity can be lowered, and the mass flow rate can be increased, by mounting the nozzles in contoured tunnels with open ends at both sides of the ship (ejector principle).—*Witte, J. H., Marine Technology, July 1969, Vol. 6, pp. 291–302.*

Patent Specifications

Process for Building a Ship

This invention relates in the first instance to a process for building a ship, according to which process, in at least one assembly hall ship sections are built, which sections are successively transported to the building site of the ship, where they are secured to each other. Referring to the drawing, in hall (1) has been provided a fairway (2), in which pontoons (3a—e) are present, which can float if the water level in the fairway is sufficiently high. On two opposite



sides the pontoons have protruding parts (4), each of which is adapted, when the water level in the fairway is relatively low, to rest on a horizontal supporting face of stepped projecting parts (5) of the sides of the fairway (2). Outside the hall the fairway (2) is extended to a channel (6). On the pontoons (3a—e) inclusive, which form one continuous line, complete ship sections (7a—e) are built. When all building operations have been finished, the water level is raised until the pontoons will float, after which they are simultaneously moved to a next station. Pontoon (3e) is sailed from the hall via an opening (8) and is transported in channel (6) to a place where the completely finished ship section (7e) on pontoon (3e) can be easily transferred to the building site of the ship. —*British Patent No. 1 157 917 issued to N. V. Koninlijke Maatschappij "De Schelde." Complete specification published 9th July 1969.*

Moving Weight Stabilizer

Referring to Fig. 1, across the hull of a ship (1) extends a glass fibre reinforced plastic tube (2) lined with an epoxy resin to provide a good clearance for a weight (3) which may be of lead filling a steel casing. Centrally at each end of the weight is a steel tube (5) in the form of a mandrel. The respective ends of the tube (2) are closed by end plates (6) from the centre of the inner side of each projects a tube (7) into which the mandrel (5) is a tight fit. From adjacent ends of tube (2) there are pipes (8) each of which extends down the side of hull (1) through a one-way valve (9) to a shut-down valve (10), normally open. The stabilizer control of Fig. 6 is controlled from the roll of the vessel, or a suitable combination of its derivatives. A signal is given to the hydraulic control (23) which transmits hydraulic pressure to the hydraulic cylinder of control valve (16). During operation, the energy dissipated in decelerating the weight would cause the water to heat up to an excessive level, and to reject this heat, water is transferred from the sea by the small water pump (17) and ejected through a non-return valve (18). Thus, during the stabilizing cycle, as the weight

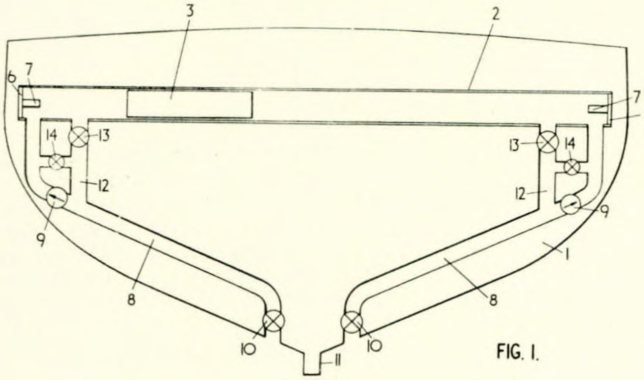


FIG. I.

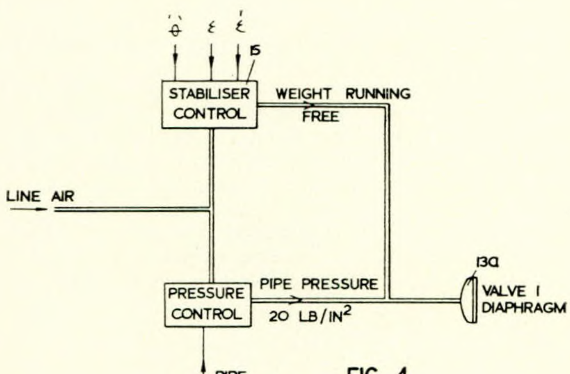


FIG. 4.

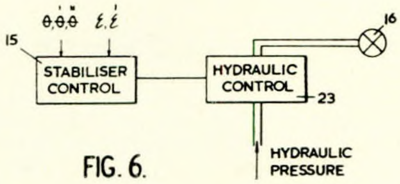


FIG. 6.

is travelling towards valve (19), the water pressure in the pipework will be high at valve (19) and low at valve (18). Therefore the two non-return valves will open and some water will enter pipe (8) through the valve (18) and some will leave through valve (19). By having the pipework placed in a vertical plane close to the inner hull surface, an extra stabilizing effect is produced by the inertia of the water moving round the pipework.—*British Patent No. 1 160 153 issued to Vosper Ltd. Complete specification published 30th July 1969.*

Bulk Cargo Carrier

This invention relates to a bulk carrier. Referring to Figs 1 and 2, the carrier (1) at a loading port of call is loaded with bulk dry cargo through the cargo hatchways (18) on upper deck (17) and is, after its voyage, moored at dolphins (2) and, at the same time, alongside berth (3). In order to connect the carrier with the reservoir tank (6) on the shoreside, the supply pipe (19), the supply pump (22), the cargo unloading pipe (20), the booster pump (35), etc. are properly arranged on the carrier (1), berth (3) and the pipe and walk way (4). When the supply pump (22) is driven, the conveying liquid medium beforehand stored in the reservior tank (6) is fed to the collecting tank (25) through a series of feeding means including the supply pipe (19), the flexible coupling (23), the feed pipe (24) and the level-adjusting valve (26). On the other hand both the valve (37) on the feed pipe (24) and the valve (48) of the charging pump (28) are closed. When the collecting tank (25) is filled with the conveying liquid medium, the valve (48) and the valve (49) of the charging pump (28) are opened. In addition, the valve (50)

FIG. 1

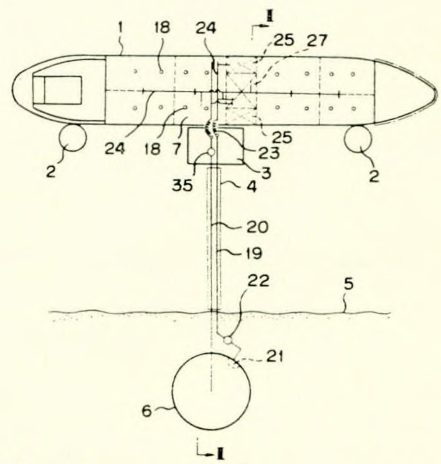
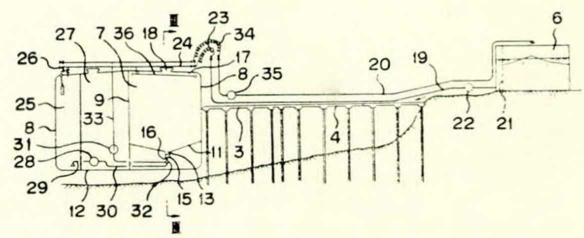


FIG. 2



and the valve (51) of the cargo discharge pump (31) are opened. Thus, the conveying liquid medium in the collecting tank (25) is returned to the reservoir tank (6) on the shore through a series of discharge means including the discharge pump (3) the discharge pipe (33), the booster pump (35) and the cargo unloading pipe (20), provided that the charging pump (28), the cargo discharge pump (31) and the booster pump (35) are driven. If valve (52) is opened, the conveying liquid medium in tank (15) and also in the cargo receiving groove (13) flow down in together with the cargo therein and through the valve (52) to the eductor (32).—*British Patent No. 1 161 103 issued to Mitsubishi Jukogyo Kabushiki Kaisha. Complete specification published 13th August 1969.*

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