Paper read at the Royal Institution of Naval Architects on Tuesday, 11 March 1975

ENGINE PLANTS IN TRIPLE SCREW CONTAINER SHIPS

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Among the several third generation container ships trading in the Far East five vessels are distinguished by being diesel driven triple-screw ships. These vessels are powered by three direct coupled slow speed diesel engines with a total continuous rating of 55 MW (75 000 bhp) and maintain an average service speed of more than 26 kn in the fully loaded condition. This paper deals with the propulsion plants selected for these vessels with special regard to those in the two Danish owned vessels *Selandia* and *Jutlandia*. The first section of the paper deals with the reasons for the choice of diesel machinery and triple-screw propulsion for these vessels. Also, considerations regarding choice of propellers, as well as generating plant, are dealt with. In the second section the engine plant in the two Danish vessels is described. No attempt is made to give a complete description of these engine plants but problems regarding control systems and safety systems for the triple-screw propulsion plants are dealt with in more detail. The third section deals with results obtained during the trials and in the fourth section mention is made of actual service results as well as the special conditions encountered during the energy crisis when the vessels were operated at reduced speed. The last section of the paper describes some initial teething troubles during the first service period.

INTRODUCTION

This paper deals with the engine plant selected for the diesel-powered third generation containerships which were put into service on the Far East route in 1972/73. These four vessels are owned by Wilh. Wilhelmsen, Oslo, The Swedish East Asiatic Co. (Brostrom), Gothenburg, and the East Asiatic Company Ltd., Copenhagen. They are operated by ScanDutch together with two steam turbine powered vessels owned by Nederlandshe Lloyd and one steam turbine driven vessel owned by Messageries Maritime. Apart from these Scandinavian diesel ships, a similar diesel driven triple-screw containership is owned and operated by Mitsui, OSK, Tokyo.

The paper is in five sections:

1) A description of the basic considerations which led to the selection of triple-screw diesel plant for these vessels.
2) A description of some of the more interesting technical construction details of the engine plant.
3) Some interesting trial results.
4) A short description of service results.
5) Description of some initial teething troubles and how they were overcome.

Sections 2–5 are written with special regard to the two vessels owned by EAC.

1. BASIC CONSIDERATIONS REGARDING SELECTION OF PROPULSION PLANT

The task of the preliminary planning of the four Scandinavian vessels was performed by a technical committee with members from the owners' technical staffs. Other working groups formed by the Scandinavian owners had already laid down detailed basic requirements for these newbuildings. These may be summarized as:

a) a sailing schedule with details concerning speed, steaming time at sea, number of port calls, time in port etc;
b) average service speed during ocean passages to be 26·5 kn;
c) bunkers sufficient for a steaming range of 17 000 nautical miles at full service speed;
d) container capacity (20 ft units, slot weight 12·5 t), below deck about 1600, on deck (per layer) about 600;
e) maximum draught limited to 11·25 m (37 ft);
f) hull dimensions to allow for unrestricted passage through the Panama Canal.

Anybody who has been involved in the construction of third generation containerships will be familiar with these requirements.

On basis of preliminary studies of hull dimensions, displacement block coefficient etc, the necessary horsepower for the project was established to be about 55 MW (75 000 bhp). With this power, the speed on loaded trials was estimated to be approximately 28 kn.

Choice of Machinery

An owners' selection of propulsion plant for newbuildings is guided by a number of factors the most important of which may be summarized as:

i) reliability;
ii) overall economy in operation;
iii) off-hire time estimated for maintenance, overhauls and classification surveys;
iv) manning problem;
v) initial investment;
vi) weight of machinery plus necessary bunkers;
vii) space requirement;
viii) vibration and noise;
ix) however, tradition and experience also play important roles.

In the preliminary stages a number of different propulsion plants were studied including gas turbines. Although advanced machinery plant might present interesting possibilities from a technical point of view, it must be borne in mind that containerships represent big investments with a carrying capacity

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equal to about six conventional cargo liners. Due to this the requirement for reliability had especially high priority. For this reason studies were mainly concentrated on conventional propulsion machinery, i.e., steam turbine plant with conservative steam ratings and without reheat and diesel plant (geared medium speed and slow speed direct coupled).

Why Diesel Plant?

In view of the fact that all third generation containerships being planned simultaneously by other owners were twin-screw turbine vessels, the choice of triple-screw, direct coupled, diesel plant meant a departure from established opinion and deserves some comment.

With regard to reliability, no particular difference between steam and diesel was found.

Running cost analysis showed a definite advantage for the diesel powered vessel even with the fuel and lubricating oil prices prevailing in 1969. Admittedly such cost analyses may differ from owner to owner and from country to country, being to a certain extent dependent on one's own repair facilities, availability of one's own repair gangs and past experience with either type of machinery.

Manning problems and time necessary for maintenance, overhauls and classification surveys are major factors, especially with containerships with very strict schedules and extremely short stays in port. This was of course the object of detailed examination based on the owners previous vast experience with diesel driven cargo liners. On the basis of the detailed sailing schedules the manhours necessary for maintenance and overhaul was tabulated in all details. The conclusion was reached that the problem was merely a question of availability of "flying repair squads" and proper planning.

Regarding machinery weight, space requirements and bunkers for the required steaming range it was clear from the early planning stages that the need was for payload rather than cubic capacity. Calculations showed that with steaming ranges exceeding 11 000 nautical miles, the diesel ship had an increasing superiority over the steamship.

Vibration and noise problems were examined carefully and it was found that disturbing vibrations in modern vessels were more often caused by exitation from the propeller(s) than from the propulsion plant. This also applied to slow running diesel engines due to the enginebuilder's thorough knowledge of how to overcome such problems. A lot of experience with noise attenuation had been gained from earlier diesel powered newbuildings. It was concluded that this presented no insurmountable problem.

In view of the above considerations efforts were concentrated on seeking the diesel plant best suited for the purpose.

Twin-screw or Triple-screw Configurations

Because of the limitation in draught it was decided to limit propeller diameters to 6·5 m to obtain adequate submersion in the ballast condition.

A twin-screw vessel called for about 29·5 MW (40 000 bhp) per shaft. This power could only be obtained with geared multi-engine plant or super large-bore diesels. A geared multi-engine plant had of course the merit of optimal choice of propeller speed and lower weight. Such a plant was however found prohibitive because of its complexity and the number of cylinders necessary with available machinery. With regard to super large-bore diesels, the power could be obtained with 2 × 12-cylinder units with c.s.r. at 100 rev/min. At that time only a limited number of these engines had been in service and the low speed resulted in inadequate pitch ratio.

As a result of these considerations a triple-screw configuration was examined. It should be mentioned that at about the same time an article was published suggesting triple-screw plant for VLCC and containerships(1). Tests had been carried out which showed very promising results. It was immediately clear that a three-propeller plant, if acceptable propulsion efficiency could be obtained, offered a number of obvious advantages:

1) the propulsion plant could be chosen from proven large-bore diesel designs of which a great number of units were in service;
2) with direct coupled diesels an acceptably high speed (117 rev/min) could be obtained, thereby reducing propeller diameters; also the load on each propeller could be reduced and a more conventional P/D ratio could be obtained;
3) the reliability of such a plant could be considered the maximum obtainable;
4) with the schedule intended, the wing engines could be regarded as booster engines for obtaining the high speed during ocean passages; between ports in Europe and between ports in the Far East the centre engine was sufficient to obtain the necessary cruising speed of about 19–20 kn.
5) optimal manoeuvrability could be maintained at cruising speed and during manoeuvring with the centre engine and one rudder on the centreline.
6) the problem of maintenance and overhaul was significantly minimized with the possibility of overhauling the side engines at sea whilst cruising between ports in Europe where experienced repair gangs were available;
7) the normal engine room staff could be kept to a minimum;
8) continuous survey by the classification society could be carried out with no difficulty.

Model Tests with Triple-screw Propulsion

During the early investigations the only model available was of a smaller ship (1200 × 20 ft units) and 44 MW (60 000 bhp). A number of propulsion model tests had been carried out with different twin-screw configurations. This model was re-arranged for triple-screw propulsion and a series of tests were performed and compared with the twin-screw test results.

Fig. 1 shows that a five per cent better propulsive efficiency was obtained with the triple-screw arrangement. This result

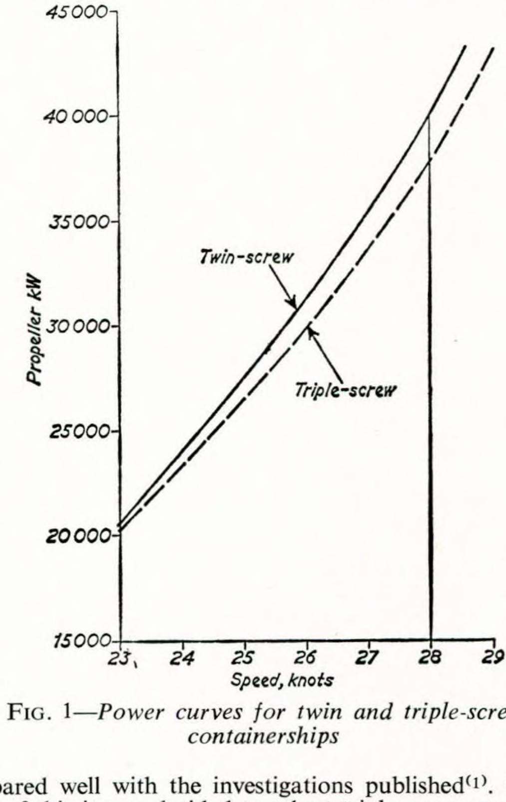


FIG. 1—*Power curves for twin and triple-screw containerships*

compared well with the investigations published(1). On the basis of this it was decided to adopt triple-screw propulsion for the bigger vessel with 55MW (75 000 bhp) and all further tests have been carried out with the bigger model and triple-screw arrangement. The distribution of the power on the three propellers was selected on basis of the stipulated cruising speed of 20 kn with centre engine alone. The power determined for the three engines was:

centre engine c.s.r. 1 × 22 MW (30 000 bhp) at 117 rev/min
wing engines c.s.r. 2 × 16·5 MW (22 500 bhp) at 117 rev/min.

Selection of Propellers

In selecting the propellers, a number of requirements had to be fulfilled. Rather than choosing three adequate conventional propellers the company was faced with the problem of choosing a propulsion system suited to cover its special needs:

a) a maximum loaded trials speed of 28 kn;
b) a cruising speed of 20 kn with centre engine at full power and side engines stopped;
c) absolutely safe working conditions during overhaul of the wing engines at sea had to be secured;
d) maximum manoeuvrability to be obtainable at full speed as well as at dead slow.

Requirements under a) and b) called for a centre propeller suitable for transmitting full power from the centre engine at entirely different speeds. The only propeller which fulfills this requirement is a c.p. propeller. Figs 2 and 3 show that with a propeller constructed for cruising speed, maximum power cannot be obtained from the propulsion engine, at full speed, except by overspeeding. With a propeller constructed for full speed the centre engine would be overloaded at cruising speed. Accordingly a c.p. propeller was chosen for the centre shaft.

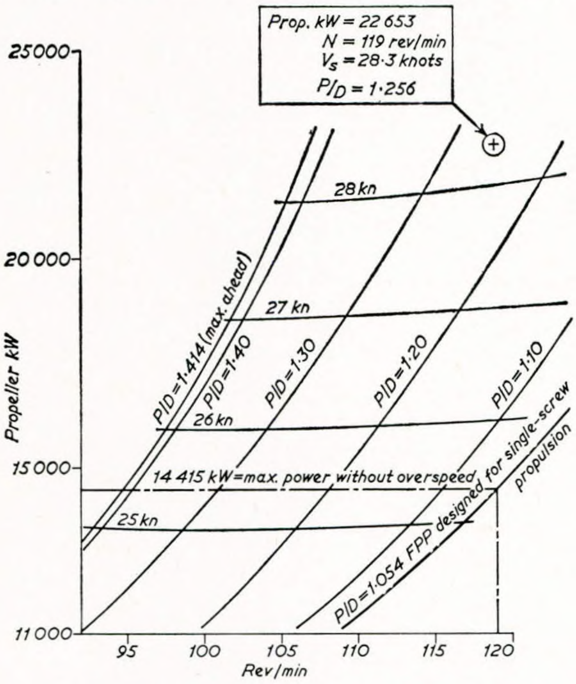


FIG. 2—*F.P. propeller designed for cruising speed—Drawn on c.p. propeller diagram*

To fulfill requirement c), c.p. propellers capable of being feathered were suggested for the wing shafts. This would however lead to slightly lower efficiency and would, furthermore, be a rather complicated and very expensive solution. Six-bladed fixed pitch propellers were chosen and clutches provided on the shaft lines. With the clutches disengaged, the drag of the wing propellers was calculated to be insignificant.

As to requirement d), a c.p. centre propeller and one centrally placed rudder should provide excellent manoeuvrability at all speeds, including dead slow during canal passage and manoeuvring in port.

In this context it should be mentioned that the triple-screw containership owned by Mitsui, OSK, has a fixed pitch centre propeller. It is claimed that a speed of about 16 kn is obtained with wing engines disengaged and without undue overloading of the centre engine. This compares fairly well with the diagram in Fig. 3.

Generating Plant

The electrical power requirements at sea including refrigerated containers was calculated to about 2000 kW and during manoeuvring to about 2500 kW including the bow thruster.

With a total power of 55 MW (75 000 bhp) for propulsion and a considerable part of the total operating time spent at sea under full power there seems to be an obvious case for supplying all electric power at sea from turbo-alternators utilizing the energy in the exhaust gases from the main engines. Such systems had been used by the owners in their motor tanker fleet and the operating economy of such systems was of course well appreciated. In a motor tanker however the large exhaust gas economizers and their auxiliary equipment and piping systems constitute only part of a substantial steam plant, necessary anyway for other purposes. However, from an operation and maintenance point of view there was reluctance, in this case, to introduce an extensive steam plant into a large diesel plant. In addition, diesel driven alternators were considered a more readily available source of power in service, considering the special operating conditions of these vessels i.e. with unmanned engine room.

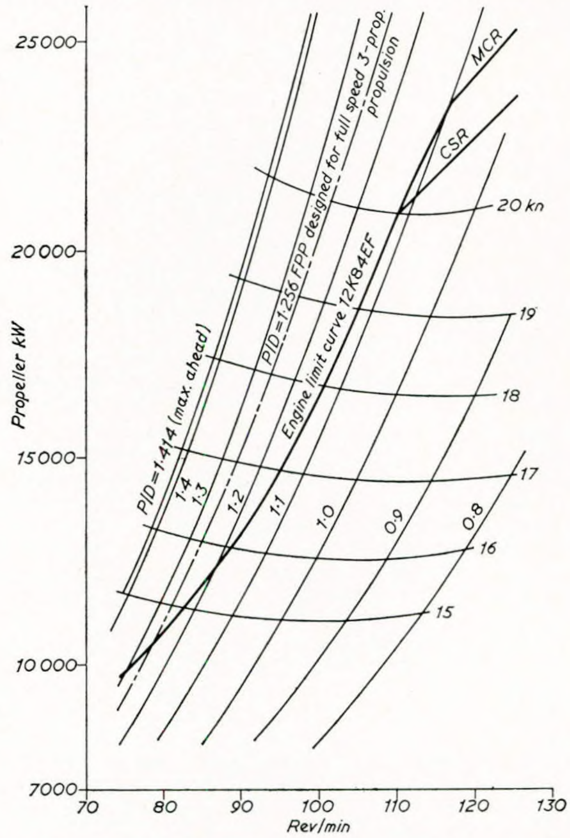
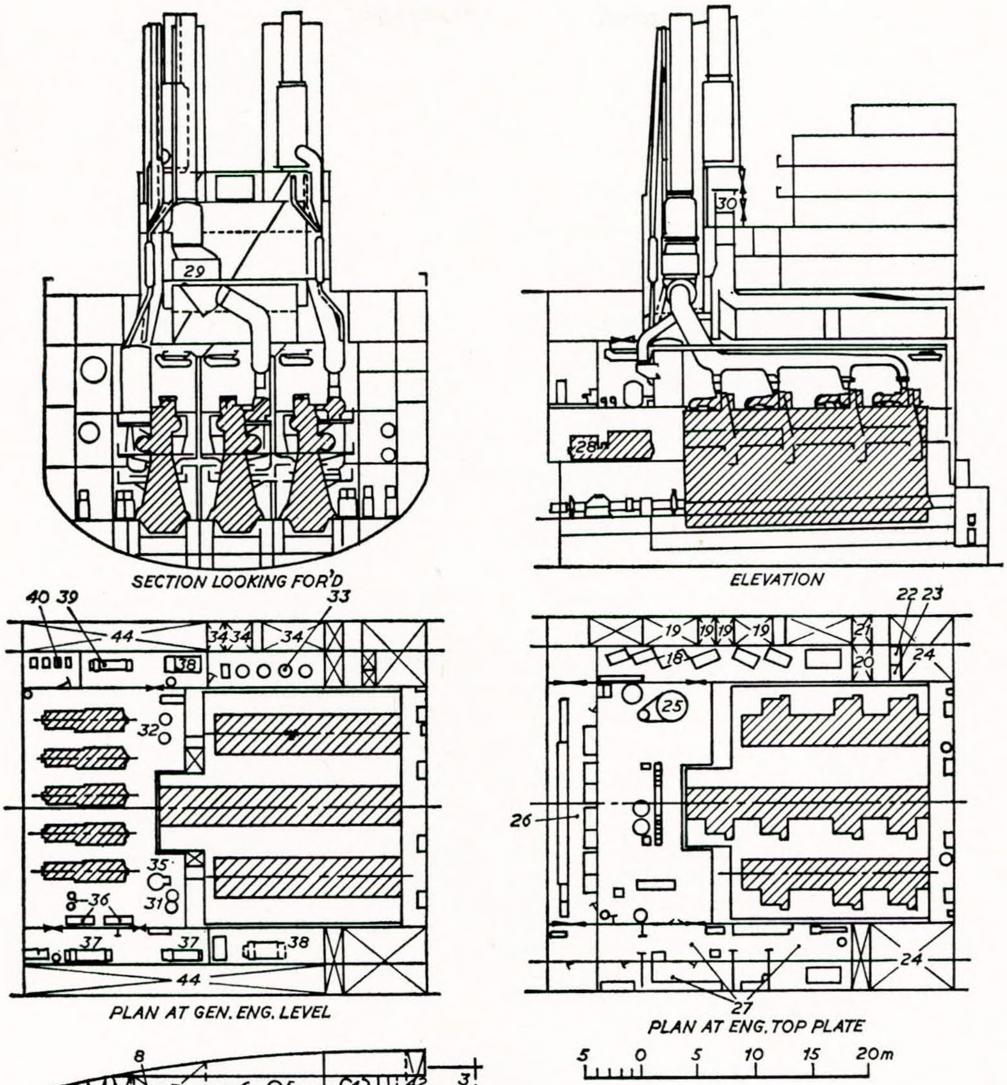


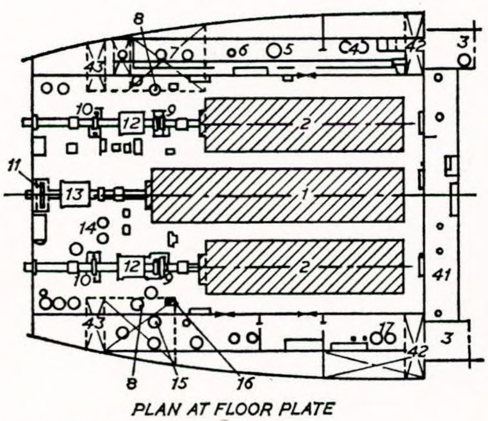
FIG. 3—*F.P. propeller designed for maximum speed—Drawn on c.p. propeller diagram*

As a result of these considerations, diesel driven generating plant was specified. As a consequence of this a conventional steam plant was adopted, with exhaust gas economizer in the centre engine's exhaust only and a single standby oil fired donkey boiler. This steam plant has of course very limited dimensions covering only demands for hotel loads and the heating of bunker fuel. The generating plant selected for the EAC vessels consists of five off, eight-cylinder high pressure charged diesel engines, each coupled to a 1260 KVA 3×440 V aircooled alternator with static exitation. Similar generating plants are used in the other Scandinavian vessels.

2. SOME CONSTRUCTION DETAILS OF THE EAC VESSELS

It is not the author's intention to describe the engine plant in detail. Fairly detailed descriptions have already been published in the technical press. Furthermore, these diesel plants, although exceptional in size, contain a number of conventional features which need no special comment.





1) Centre engine (12K84 EF)
2) Wing engines (9K84 EF)
3) Stabilizer compartments
4) Bilge pumps
5) Bilge water separator
6) Sludge pump
7) S.W. pumps W/E
8) L.O. pumps W/E
9) Hydr. clutch W/E
10) Disc brake W/E
11) Disc brake C/E
12) Thrust blocks W/E
13) Thrust blocks C/E
14) L.O. pumps C/E
15) S.W. pumps C/E
16) L.O. filters
17) Fire and deck wash pumps
18) Starting air comp.
19) L.O. storage tanks
20) Diesel oil serv. tank C/E
21) Diesel oil serv. tank W/E
22) Heavy fuel serv. tank W/E
23) Heavy fuel serv. tank C/E
24) Heavy fuel settling tanks
25) Oil fired boiler
26) Engine control room
27) Workshops and stores
28) Diesel alternators (826MTBH–40)
29) Exh. gas economizer
30) Eng. room ventilators
31) F.W. pumps C/E
32) F.W. pumps W/E
33) Fuel oil purifiers
34) L.O. storage tanks
35) F.W. evaporator
36) F.W. pumps and coolers for generator eng.
37) F.W. and L.O. coolers C/E
38) L.O. coolers W/E
39) F.W. coolers W/E
40) Transformers
41) Pump room for fuel transfer and ballast pumps
42) Forward sea chests
43) Aft sea chests
44) Water ballast

FIG. 4

As an introduction, it seems appropriate to describe some of the interesting features of these engine plants, as well as making some comment regarding the principles on which piping systems, control of propulsion plant and safety systems are laid down. For reference a schematic arrangement of the engine room is shown in Fig. 4.

It will be seen that the main engines are arranged on an extremely stiff tank top. The height of the tank top at the forward bulkhead is about 5 m. Tank top and engine foundations slope aft to obtain adequate submersion of the propellers. The sloping tank top gives the added advantage of an extremely simple engine room bilge system. From the sectional plan it will be seen that four longitudinal bulkheads are carried right through the length of the engine room at the sides for added stiffness. The plan at tank top level shows the three separate thrust blocks, disc brakes on each shaft and hydraulic clutches on the wing shafts. The generating plant is arranged on a 'tween-deck aft, while the control room, work shops and a large working platform are arranged level with the main engines' top platforms. The exhaust pipes are carried to two funnels.

In planning the piping systems, pumps and coolers as well as the control and safety systems, due regard has been paid to the two alternative modes of operation prescribed for this engine plant. With these in mind it seemed natural to divide the engine plant into three sections:

i) the centre engine, which is the main propulsion unit, must be equipped with means for the precise control of output and be capable of reversing the propeller thrust; the auxiliary machinery necessary for its operation must be independent of all other machinery installations;
ii) the wing engines are regarded as a booster propulsion plant for obtaining maximum speed; these engines will always work in unison and may be equipped with common services for their operation;
iii) the diesel generating plant constitutes the third independent part of the machinery plant.

Piping Systems

In accordance with the above considerations, the centre engine has its own piping system, pumps and heat exchangers for salt and fresh water cooling. Also the fuel supply system, including heavy fuel service tank and diesel oil service tank, is entirely independent. The wing engines have cooling water systems and fuel supply systems with service tanks in common. The heavy fuel service tanks for both systems are small and kept at a constant level by the purifiers. There are two fuel valve cooling systems, one for the centre engine and one for the wing engines, served from the diesel oil service tanks. Lubricating oil systems are, of course, independent for each of the three engines. The diesel generating plant has its own cooling water systems. The fuel supply is from either of the two diesel oil service tanks. A separate cooling water system with heat exchanger is arranged for the cooling services of the refrigerated containers.

By dividing the otherwise conventional piping systems, pumps and heat exchangers as described and avoiding the complication of cross-overs, simple and straightforward systems have been obtained.

Control of the Engine Plant

Remote Control from the Bridge

The bridge control system of the main engines in the EAC vessels constitutes a departure from the normally accepted individual control of each engine.

A triple-screw plant arranged for the two alternative modes of operation mentioned lends itself to simple single lever control for the following reasons:

1) all manoeuvring is performed by the centre engine which is uni-directional and the thrust is reversed by means of a c.p. propeller;
2) the wing engines, which are normally only used in open sea, need not be reversed from the wheelhouse, thus control of these engines can be limited to simple control of the engine speed for ahead running; for ease of control this may be linked to the control of the centre engine;
3) as a consequence, remote starting and reversing of the three engines is not installed; this means, of course, a significant simplification of the remote control system;
4) the engines can only be started from the engine control station.

Apart from the above control functions, the remote control system must be equipped with a number of additional functions:

a) means for individual stopping of engines must be provided;
b) engines must be protected against undue acceleration and overload;
c) if the control lever is put into the astern position, the wing engines must be stopped automatically; the same applies if the control lever is put into the stop position (centre engine idling with the c.p. propeller in neutral);
d) at sea it must be possible to reduce speed to dead slow without unintentionally stopping the wing engines.

Stopping the ship during a crash manoeuvre is performed partly by reversing the centre propeller and partly by the braking effect of the wing propellers turning their engines with no fuel admission. This was the object of calculations based on experience from crash stops performed during trials with single-screw vessels. It was concluded that in practice shorter stopping distances would not be obtained by reversing all three propellers.

Control from the Engine Room Control Stand

Normal individual starting and control of the three engines is arranged. In addition, a lever for controlling the pitch of the centre propeller is fitted.

The three engines need not necessarily be reversible. None the less all the engines are fitted for reversing from the engine control station, in case of emergency. The reversing levers which are incorporated in the answer-back engine telegraphs are normally locked in the ahead position.

For the same reason—emergency—the single lever control on the bridge is incorporated in a standard synchro-step telegraph. Bridge telegraphs for the wing engines are of the push-button type. During normal running all push-buttons except "Start" are covered by a perspex screen.

Detailed Description of Remote Control System

Fig. 5 shows a diagrammatic arrangement of the control system for the three engines.

The centre engine and the c.p. propeller is controlled by means of a combinator handle. The speed control of the wing engines is linked by means of a chain drive to the combinator

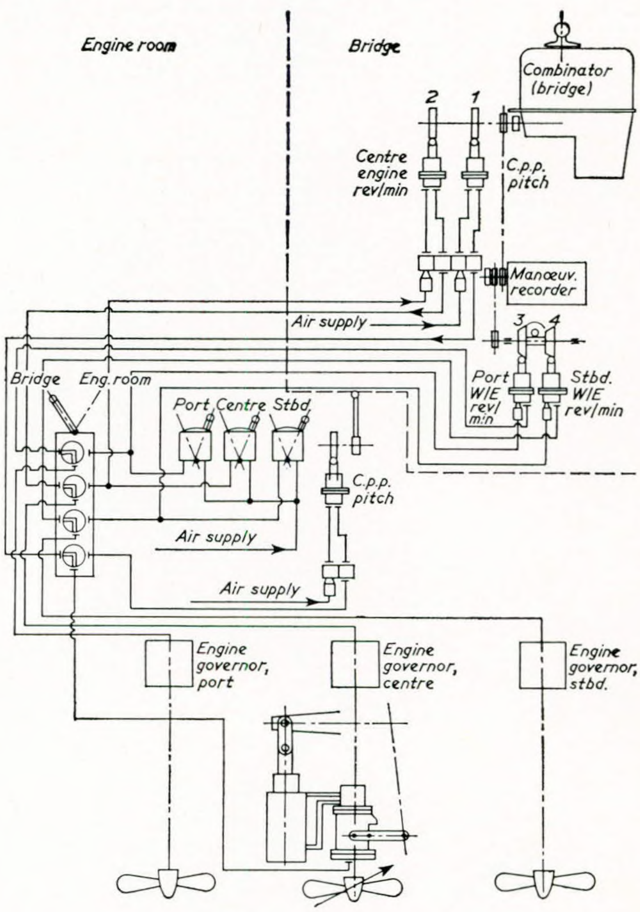


FIG. 5

handle. Moving the handle rotates the four cams:

1) cam for pitch control of the c.p. propeller;
2) cam for speed control of the centre engine;
3) and 4) cam for speed control of the wing engines.

Compressed air at 6·9 bar is fed to the four cam-actuated transmitters which transmit pneumatic signals to the hydraulic actuating unit for the c.p. propeller and the speed setting devices in the hydraulic governors of the propulsion engines. Thus for each control lever position, a preselected combination of centre engine speed/c.p. propeller pitch and wing engines speed are ordered.

Automatic Adjustment of Centre Engine Output

The c.p. propeller pitch is further adjusted by means of an electronic load control device which is continuously fed with signals for actual speed and fuel pump index. These signals are correlated and the pitch is adjusted automatically so as to obtain a given output from the centre engine. In other words, for each position of the bridge control lever a given engine output/speed is selected irrespective of variations in the propulsive resistance due to variations in draught, weather conditions or fouling of the hull.

Adjustment of Wing Engine Output

When at full speed in open sea, equal distribution of the load between the three main engines must be aimed at. While adjustment of the centre engine output is performed by adjusting the propeller pitch, similar adjustment of the wing engine output can only be obtained by variation in the engine speed. This adjustment is made by axial displacement of the camshaft for the speed control of these engines. As shown, cams 3 and 4 have a conical shape and adjustment is carried out manually by turning a small handwheel mounted in the bridge control console.

Indication of the power on each engine can be read from instruments on the control desk. These show speed and fuel pump setting for each engine. No torsionmeters are fitted. A layout of the bridge control console is shown in Fig. 6.

Preset Program for Load Distribution Between Main Engines

The use of a single combinator lever means that the load distribution between the three engines, in all control lever positions, can be controlled through a preselected program which must be determined with a view to optimal manoeuvrability at all speeds.

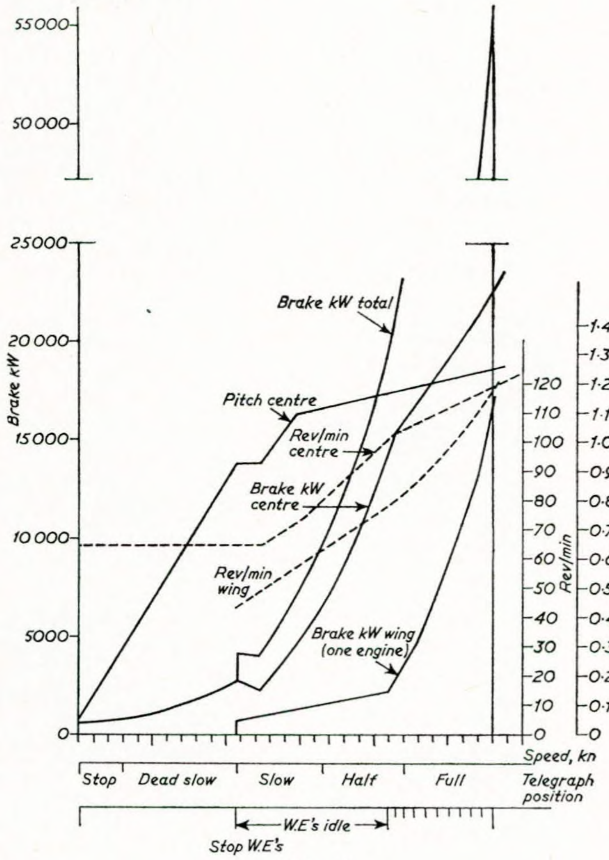


FIG. 7

Fig. 7 shows power and speed of the centre engine and side engines, and basic pitch of the centre propeller as a function of the control lever position from "Stop" to "Full Ahead". It will be seen that good manoeuvrability at slow speeds is obtained by placing the loading primarily on the centre engine. When under bridge control the wing engines cannot be started with the control lever below position

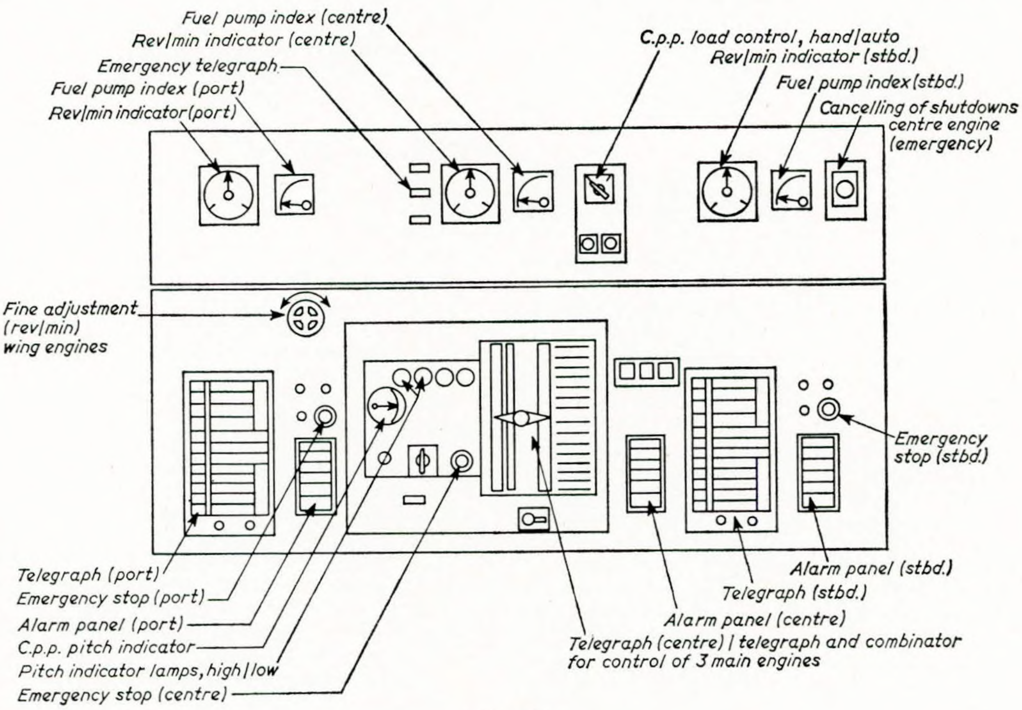


FIG. 6

"Slow". From "Slow", centre engine output about 2·5 MW (4000 bhp), and up to between "Half" and "Full", centre engine about 15 MW (20 000 bhp), the wing engines, when started, are idling at constant fuel pump index with their speed increasing with the increased ship's speed from about 45 to 80 rev/min. Further load increase means a steep rise in power on the wing engines. When moving the control lever from "Full Ahead" to "Stop" the wing engines are stopped when passing "Dead Slow". At "Stop" the centre engine idles at about 65 rev/min with propeller pitch at zero. Moving the control lever to "Astern" means increasing revolutions with the centre propeller in reverse pitch (not shown in diagram).

Overload Protection

A comprehensive system of overload protection devices is arranged (see Fig. 8).

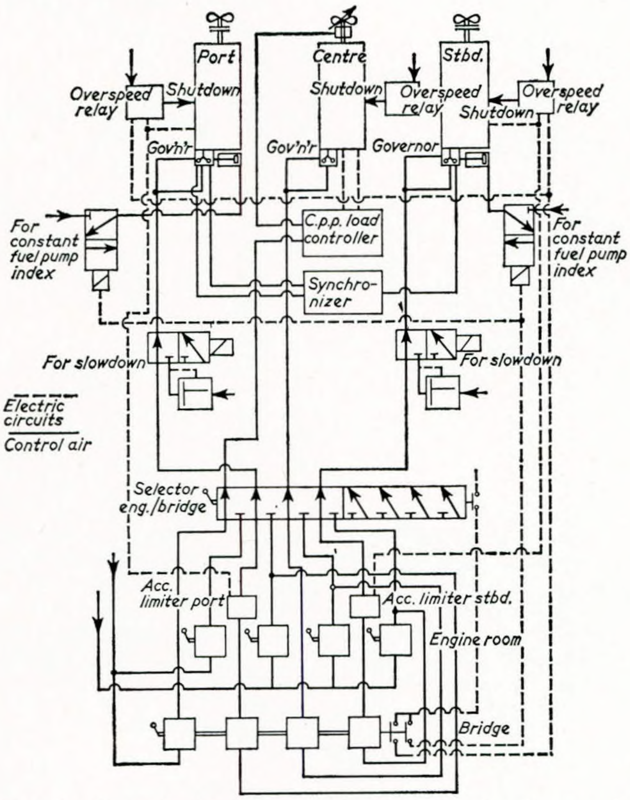


FIG. 8—*Control, slow-down and shut-down for three main engines*

As mentioned the c.p. propeller is equipped with an electronic load-limiting device. Additionally, the main engine governors are provided with a speed dependent load limiter, i.e., for each speed setting of the governor, the fuel pump setting cannot exceed a preset value.

During warming up, the engines are controlled by the engine room staff even when under bridge control. As will be seen in Figs 5 and 8 the pneumatic revolution transmitters on the bridge control desk are piped in series with similar revolution transmitters controlled by the fuel admission handles mounted in the engine room control desk. Thus a speed setting signal from the bridge control is always over-ruled by the speed setting ordered from the engine room control station. On termination of the warming up period, the fuel admission handles are put to "Full Ahead" and further control of the engine output is left to the bridge.

From the load distribution diagram (Fig. 7) it appears that with a control lever position near "Full Ahead", a further increase in the power means a steep rise in the load on the wing engines. In order to prevent unintended fast acceleration of the engines the pneumatic control signal to the governors passes an acceleration limiter (see Fig. 9). By introducing a non-return valve in the bypass, the orifice has a one-way action only.

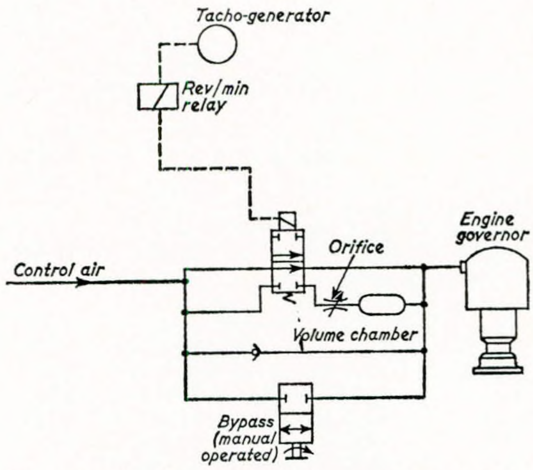


FIG. 9—*Acceleration limiter for wing engines*

Shut-down and Slow-down

The propulsion engines are provided with automatic shut-down and slow-down, which is normal practice, and the classification society rules for UMS notation.

Shut-down for failing lubricating oil pressure is arranged to follow the normal practice in EAC vessels since 1964 (see Fig. 10). Header tanks with level switches are used in lieu of

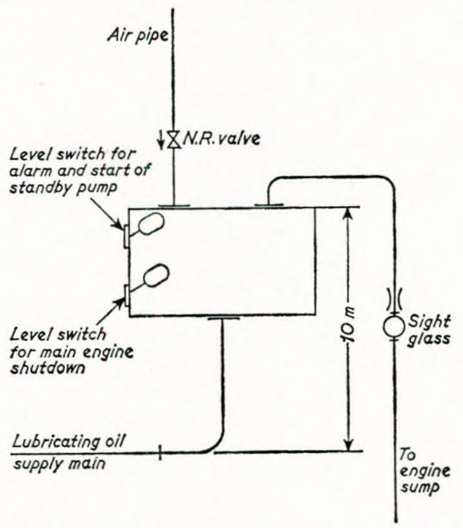


FIG. 10—*200 litre level tank for shut-down on falling L.O. pressure*

pressure switches. Due to the interaction of the engines, one with another, the three engines act in unison on shut-down and slow-down (see Table I).

Shut-down of the centre engine can be cancelled in the engine control room and also on the bridge in case of emergency.

Protection Against Windmilling of Centre Propeller During Crash Stop Procedure

From trial trips with earlier conventional single-screw cargo-liners with c.p. propellers, the experience was that the crash stop might result in overspeeding due to the propeller windmilling. This was dependent on the speed of the vessel and the rate of pitch change. It was obvious that in the case of the present triple-screw plant, the centre engine might reach dangerous overspeed levels, apart from the fact that a shut-down would be initiated. The problem was overcome by fitting an anti-windmilling relay in the pneumatic control circuit to the c.p. propeller.

The function of the relay is as follows (see Fig. 11). The independent overspeed relay (1) is adjusted for closing the contactor (2) at a 4 rev/min lower set point than the ordinary

TABLE I

Failure	Resultant Action Port wing engine	Centre engine	Stb. wing engine
Blackout	shut-down	shut-down	shut-down
Lub. oil failure on port or stb. W/E	shut-down	slow-down	shut-down
Lub. oil failure C/E	slow-down	shut-down	slow-down
Lub. oil failure turbocharger port or stb. W/E	shut-down	slow-down	shut-down
Lub. oil failure turbocharger C/E	slow-down	shut-down	slow-down
Overspeed port or stb. W/E	shut-down	full power maintained	shut-down
Overspeed C/E	slow-down	shut-down	slow-down
Turbocharger vibration port or stb. W/E	slow-down	full power maintained	slow-down
Turbocharger vibration C/E	slow-down	slow-down	slow-down
Scavenging belt fire port or stb. W/E	slow-down	full-power maintained	slow-down
Scavenging belt fire C/E	slow-down	slow-down	slow-down
Thrust block port or stb. W/E temp. high	slow-down	full power maintained	slow-down
Thrust block C/E temp. high	slow-down	slow-down	slow-down
Cooling f.w. port or stb. W/E/ low	slow-down	full power maintained	slow-down
Cooling f.w. C/E low	slow-down	slow-down	slow-down
C.P. propeller oil dress. low	slow-down	slow-down	slow-down

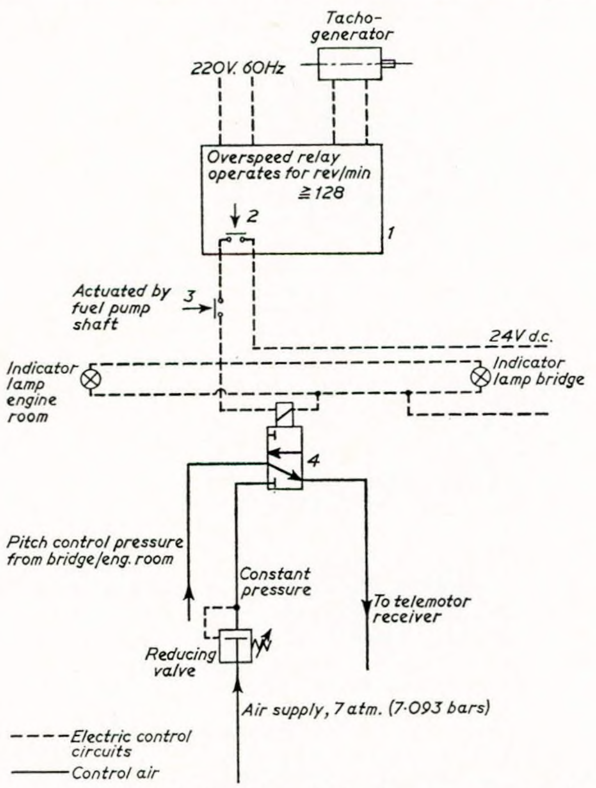


FIG. 11—*Anti-windmilling device for centre propeller*

overspeed shut-down relay. The contact (3) is actuated by the fuel pump regulating shaft and is closed at low fuel pump setting.

During normal operation the pneumatic control signal from the bridge transmitter to the c.p.p. telemotor receiver passes through the solenoid valve (4) while the constant air pressure is shut off. In case of overspeed occurring simultaneously with low fuel pump setting (windmilling) the control circuit to the solenoid valve is closed. The pneumatic control signal is thereby changed to the constant pressure signal which corresponds to a preset safe propeller pitch. This pitch is maintained until ship's speed falls below the critical where upon normal pitch control is restored and the pitch is quickly reversed to astern.

Automation and Alarm Systems

Considering the size of this engine plant it was an obvious requirement with a view to safe operation and adequate supervision that the engine installation had to be equipped with automation and alarm systems for obtaining a UMS notation. The installation is arranged along the lines used in previous EAC vessels operated with unmanned engine room i.e., a straightforward system with no data loggers or computers.

Automatic equipment is installed for maintaining correct cooling water and lubricating oil temperatures, as well as the viscosity of fuel supplied to the engines. Automatic starting of important standby pumps has been arranged. The steam plant is equipped for maintaining constant steam pressure and purifiers have programmed desludging.

The alarm plant consists of a total of 223 relays grouped in four alarm panels, one for each main engine and one for auxiliary machinery. A rotating flashing beacon on top of each alarm panel indicates which panel is giving an alarm.

In earlier automated EAC vessels, conventional pressure and temperature sensors and level switches with on/off contacts were used in closed loops for non-alarm conditions. This type of equipment has generally given good performance. However in the case of the temperature sensors, this type has certain drawbacks, notably it does not fail safe in the case of leakage from the temperature bulb or capillary tube. Furthermore, the checking of such equipment is a very time consuming matter. For this reason it was decided in this present case, that temperature sensors on the main engines and shaft bearings should be of the analogue type. Should a future need arise, these may be connected to data loggers or computers.

Certain other important alarms which do not fail safe are duplicated and connected to independent loops, e.g., the level switches in bilge wells. On the main engine top platform, a number of photocells are arranged to give an alarm in the event of a fuel oil spray from the high pressure injection system. In addition the whole engine plant is covered by a separate fire detection system.

While the engine room is unattended, the alarm is switched via a selector to the duty engineer. Additional alarm buzzers are located in officers' salocns. If an alarm is not acknowledged, it is automatically (o\er a time relay) switched to the chief engineer's cabin and, if still not acknowledged, to the wheelhouse. Engine alarm par els in the wheelhouse are otherwise limited to alarms for cor.ditions directly affecting propulsion and steering (see Fig. 6).

Instrumentation in the engine control room is simplified as far as possible. All instruments are of the electric type with rotatable dial so that all pointers are normally pointing in the same direction. Operating lights are fitted only for machinery which is not constantly running, for instance air compressors, transfer pumps, boiler feed water pumps, etc. These lights are grouped for easy survey. No mimic diagrams are fitted.

3. TRIAL RESULTS

Speed Trials

During the speed trials a speed of 28·02 kn at 55 MW (75 000 bhp) was attained at full draught 10·92 m. At a draught of 8·40 m the maximum speed recorded was 30·52 kn.

With the centre engine developing 22 MW (30 000 bhp) and the wing engines stopped the following speeds were attained (draught 9·0 m):

i) wing engines disengaged from their shafts, 20·20 kn.
ii) wing engines at maximum obtainable power (not recorded) centre engine turned by propeller at maximum pitch } 22·5 kn

The steering abilities of the vessel were excellent during these tests. The manoeuvrability at slow speeds with the centre engine alone especially proved very satisfactory. The single lever remote control also came up to all expectations.

During the speed test with wing engines operating and centre engine stopped, the wing engines were under control from the engine room control stand. The bridge control lever was in stop position, i.e., with the c.p. propeller in zero pitch. It is interesting to note that in this condition the ship would not answer the helm, the rudder being screened by the propeller. After moving the pitch only a little to "Ahead" the steering was perfect.

Crash Stop Test and Turning Circle

The results from the crash stop test are shown in Fig. 12.

The tactical diameter for starboard and port turns at 28 kn was 1250 m.

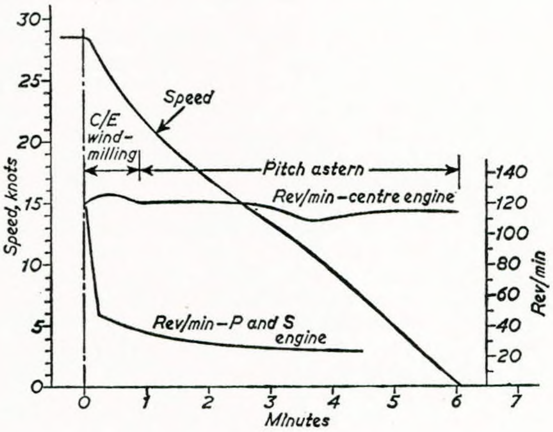


FIG. 12

Starting the Wing Engines with Centre Engine at Full Power

With the centre engine at full power and wing engine clutches engaged, the wing engines are turned by their propellers at about 35–40 rev/min. When a further increase in power is required, the wing engines must be started from the engine control stand. Off-hand it might be expected that the engines would start readily, needing little or no starting air. This is, however, not the case. Due to the comparatively high rotational speed of the engines, the starting air admitted per stroke proved insufficient to start the turbochargers, as was evident from the turbocharger tachometers. Now that the problem has been recognized, more air is admitted during the starting procedure and no further difficulties have been met.

Measurement of Overspeed due to Centre Propeller Windmilling

Windmill action of the centre propeller and engine was the object of a series of tests in order to establish with accuracy the correlation between ship's speed, propeller pitch and tendency to overspeed (>122 rev/min). Fig. 13 shows the maximum speed reached during crash stop tests at different speeds. It will be seen that centre engine overspeeding occurs at about 23·1 kn. The overspeed shut-down trips at 132 rev/min, corresponding to a speed of 24 kn. Accordingly the windmilling relay was adjusted to 128 rev/min.

It will be realized that much time was spent during the trials in testing and adjusting the comprehensive safety equipment in these vessels. A number of conventional trial tests were of course carried out and need no comment.

Vibration and Noise

Measurements taken during the trials showed that the vessels were remarkably free from vibration. However some vibration, presumably excited by the centre propeller, was felt on the after deck near the mooring winches. Extra stiffening by means of pillars had to be installed.

In order to reduce possible vibrations excited by the nine-cylinder wing engines' 1st and 2nd order moments, these engines are provided with a synchronizing device attached to the hydraulic governors (see Fig. 8). Due to the very stiff

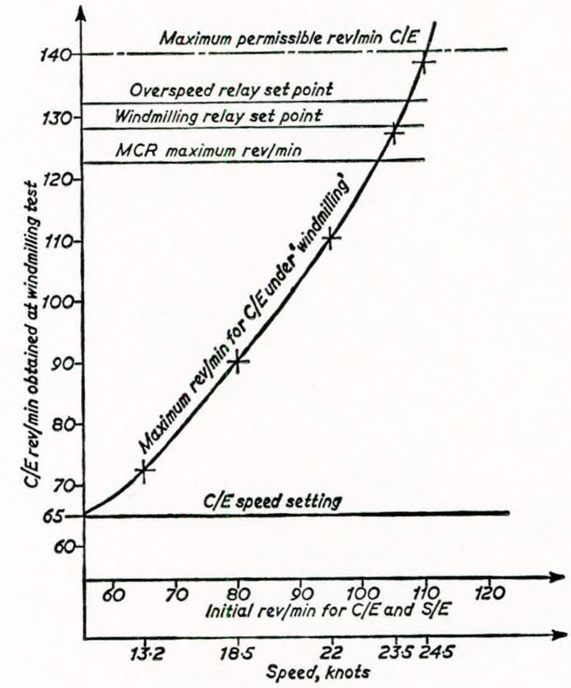


FIG. 13

engine foundations, this installation turned out to be superfluous, having no traceable effect.

The noise in the engine room with all engines operating at full power was of a normal level. From the noise measurements taken the following are typical:

engine top near turbochargers	105 Db(A)
working platform	100 Db(A)
floor level aft	101 Db(A)
auxiliary diesel platform	102 Db(A)

In the control room a comfortable level of 70 Db(A) was measured. In workshops, which are entirely closed by steel bulkheads, a maximum level of 90 Db(A) was measured. In the engine room, the engineers wear ear protection. For this reason rotating flashing beacons are installed for calling attention to telephone calls. With regard to alarms the sirens are so penetrating that they are easily heard. Between the engine room and the accommodation, one deck height is used solely for store rooms. The noise level on this deck is, in certain locations, at a level of 80 Db(A). The accommodation on the upper deck contains cabins for the repair gang and is normally only used when the wing engines are disengaged. On this deck the crew's cafeteria and day room are also located, and it has been necessary to fit further sound insulation to the floor in the day room, where noise levels of 70 Db(A) were measured with the propulsion plant at full power. In the remaining accommodation noise levels between 56 Db(A) and 64 Db(A) were measured.

4. SERVICE RESULTS

On her first round the world voyage, the first of the two EAC vessels, *Selandia*, attained an average sea speed of 27·4 kn. Normally these vessels are operated at an average speed of 26 kn. This means that a round trip to the Far East takes a little less than two months or, on average, six round voyages will be completed per year.

Fuel consumption at sea for the main engines at full power (c.s.r.) is about 290 t/day. Heavy fuel at 1500 seconds Redwood I is used. The consumption of the diesel generating plant (operating on marine diesel fuel) is about 9–10 t/day.

The vessels normally call at Gothenburg, Hamburg, Bremerhaven, Rotterdam, Singapore, Hong Kong, Tokyo, Kobe.

The round trip in European waters normally takes about a week. This time has proved sufficient to carry out necessary overhauls and surveys. A repair gang of six to eight engineers boards the vessels in Rotterdam (homebound) and stays on board during the round trip. The normal engine room staff consist of one chief engineer, four engineers, one electrician, four junior engineers and four engine room ratings. The maintenance overhauls and surveys are carried out on a planned maintenance scheme. All exhaust and fuel valves for main and auxiliary engines are normally overhauled and checked in the EAC workshop in Copenhagen. The main engine pistons and cylinders are checked through a close follow-up of running conditions by means of indicator cards, as well as visual inspection of piston rings and running surfaces through the scavenging ports. Thus no superfluous piston drawing is carried out. Average cylinder wear has steadied on 0·08 mm/1000 h. Cylinder oil feed rate is about 25 kg/cylinder a day corresponding to 0·57 g/kW h. Similar wear rates are obtained in the sister ship.

The generating plants have run troublefree. The normal power load including air conditioning and about 80 × 40 ft refrigerated containers averages 1800 kW. This means that two diesel generators are operating at sea. With the bow thruster in operation, three diesel alternators are normally sufficient due to manoeuvring being carried out with the centre engine only.

Hydraulic Couplings

The hydraulically operated couplings on the wing shafts (a new construction by the engine builders) have been entirely satisfactory in operation. The time needed for disengagement or engagement is less than five minutes. The couplings were recently surveyed by Lloyd's Register of Shipping and proved to be in excellent condition, carrying no trace of wear in the conical mating surfaces.

Stern Tubes

Stern tubes have performed well. Operating temperatures in tropical waters are below 50°C.

In one of the vessels, the forward port-side stern tube gland had to be changed afloat because of oil leakage. The wing stern tubes have three bearings each and each tube holds 20 t of lubricating oil which had to be drained to a double bottom tank constructed for this purpose.

Propellers

Propellers have shown no signs of cavitation erosion. Some problems were however encouutered with the c.p. propeller. A crack detection survey of the bolt holes in the four blades revealed that small circumferential cracks had developed in the fillet of the nut recess. The blade material is stainless steel. Bigger fillet radii were ground and the nuts were fitted with O-rings to prevent penetration of sea water to the recesses. Later however, the bolt hole recesses had to be reinforced by welded on material in order to reduce local stress concentrations.

Operating During Energy Crisis

During this period the average speed was lowered to about 21 kn. This speed called for a total horsepower of about 48 per cent c.s.r. corresponding to a saving in fuel consumption of about 150 t/day. This presented the problem that the power was too high for centre engine propulsion and rather low for prolonged operation on three engines. The engine builder, however, had no particular concern as long as the load was evenly distributed between the engines (it must be borne in mind that under bridge control the preselected load distribution would mean about 18 MW (24 000 bhp) on the centre engine and about 5 MW (7000 bhp) on each wing engine). By adjusting the fuel admission handles in the engine control station it was, however, possible directly to achieve an even distribution of the load on the three engines. The bridge control lever could, thereby, be used in the usual manner.

Several round trips were performed at reduced speed by the two vessels with no signs of effects of adverse running conditions on the three engines. On the contrary the diesel plant showed its superiority in maintaining its low specific fuel consumption over wide power ranges.

With regard to the diesel generating plant this had of course to be operated at normal sea load (about 1800 kW). It is worth noticing that turbo-generating plant with steam supply from waste heat economizers would only be able to supply about 25 per cent of the electric load under these operating conditions.

Operation with Unmanned Engine Room

After a running-in period, these plants were operated with unmanned engine room for 16 hours a day, as well as over the weekends.

The average number of alarms during the unmanned periods could be expected to be about three times the average for single-propeller ships of similar lay-out. The average number of alarms in single-screw cargo liners has been about 0·25 alarms/day. In these triple-screw containerships, the average number of alarms recorded is about 0·65 alarms/day.

5. SOME TEETHING TROUBLES

During the construction of these engine plants, close co-operation was maintained between the shipbuilder, the engine builder and the owner. During the planning stages it was always the aim, as far as possible, to use conventional reliable components and equipment, and to seek simple and straightforward solutions to technical problems. None the less, some of the troubles encountered during the first service period were experienced with entirely conventional construction details and had their origins in the high service speed of these vessels.

Cracks in Forward Sea Chests

Salt cooling water services were supplied by four sea chests of conventional design, two of which were placed at the forward engine room bulkhead and two aft in the engine room (see Fig. 4).

As shown in Fig. 14, the forward sea chests are very deep due to the exceptionally high tank top. The ballast pumps and some smaller sea water pumps take their suction from these chests.

During the second round voyage both vessels suffered cracks in the floors and girders surrounding the sea water chests resulting in sea water leakage to the adjacent fuel tanks.

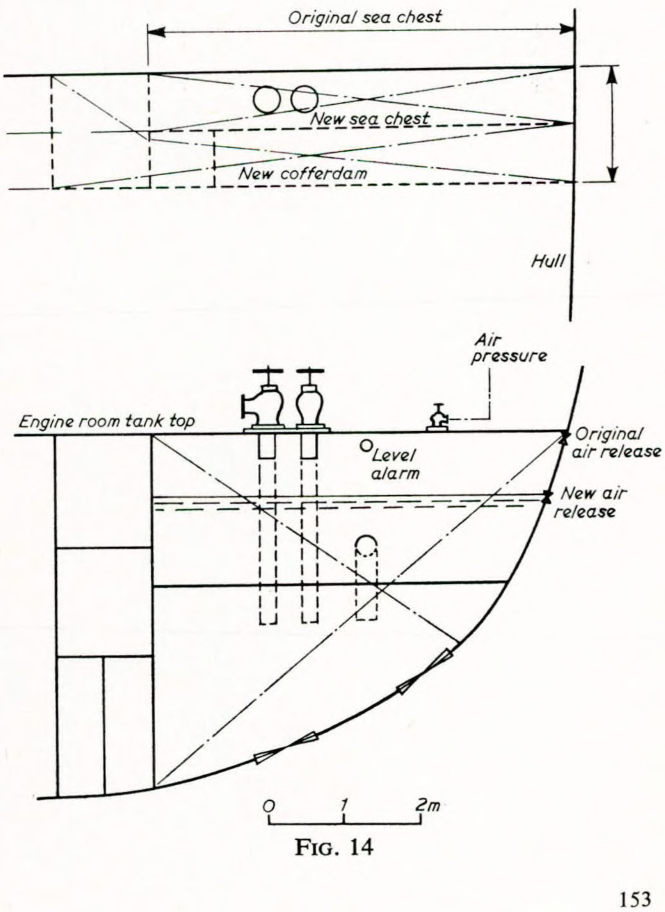


FIG. 14

Temporary repairs were carried out afloat followed by substantial reinforcing during a later drydocking. For some time everything went well, but new cracks developed with resultant leakage. The chief engineer in one of the vessels reported that on some occasions fairly heavy vibration could be felt in certain areas of the steel construction surrounding the sea chests. Vibration specialists were sent out in one of the vessels and a thorough investigation was carried out, the results of which were:

1) vibration existed in the forward sea chests;
2) at full speed the maximum pressure amplitudes inside the chests were 490 mb at a frequency of 9 Hz;
3) vibration completely disappeared with only little compressed air admitted to the sea chests through the ice cocks, thus forming an air cushion on top of the sea chests; this also accounted for the fact that continuous measurements over long periods showed that vibration occurred and faded away at irregular intervals;
4) measurements performed at different speeds showed that the vibration amplitude decreased with decreasing speed and disappeared at speeds below 24 kn; astonishingly the frequency increased at lower speeds to 12 Hz;
5) similar vibration but with much lower amplitudes was measured in the aft sea chests.

It was clear that the vibration was not propeller excited but probably due to vortex formation in the water layer close to the hull openings. During a short drydocking the cubic capacity of the forward sea chests was halved and a dry tank was formed around the chest for extra safety. Air at low pressure could be supplied to the sea chest and a float switch was fitted to give an alarm in case the air cushion disappeared. In addition, the pump suctions were carried deeper into the sea chest (see Fig. 14). New measurements were carried out on the resulting smaller sea chests during service. With no air cushion, the pressure amplitudes and frequencies were unaltered. A theory that the sea chests formed a sort of resonator in resonance with the unknown excitation force could, therefore, be abandoned. The conclusion was drawn that the phenomenon had something to do with the configuration and size of the hull openings, the height of the water column in the sea chest and obviously the high speed of the vessel. Air collection in the top of a sea chest is normally something one tries to avoid by introducing air release pipes on top of the sea chest or air holes in the hull plating. In this instance air had to be introduced deliberately to cure the problem. No further trouble has arisen at this point, but the alarm float switch fitted to the top of the sea chest revealed that, in these ships, the air cushion can only be maintained continuously by a steady slow admission of compressed air.

Salt-laden Atmosphere in the Engine Room

At full speed in strong head winds, these vessels, of course, take a lot of salt water spray over the bows and the whole ship is shrouded in an atmosphere of salt water droplets. To a lesser extent this is also the case in comparatively fair weather. Some of the water droplets find their way to the engine room ventilation grids which are of conventional design. This phenomenon is of course well known in conventional ships, but in this case the introduction of finely dispersed salt water particles into the ventilation air was so massive that it proved difficult to keep the engine top clean. Layers of salt even built up on hot surfaces in some locations. Apparently coincident with this phenomenon the time between overhauls of the main engine exhaust valves was unusually short. In single-screw plants of exactly the same type and operated continuously at the same cylinder output, with fuel oils bunkered at the same places, the period between overhauls averaged 5000 h. In the triple-screw plants, some exhaust valves had to be overhauled after less than 1000 h running. It must be emphasized that it still has to be proved that sodium chloride was the principal cause of this. Extensive research was carried out to solve the problem which was not experienced in the other third generation diesel driven containerships with same type of propulsion plant. This research has been temporarily interrupted by the decrease in normal power which followed the energy crisis. From the engine arrangement plans (Fig. 4) it will be seen that the engine room ventilators are located in a casing house separated from the accommodation house. The inlet grids are facing forward and sidewards. The forward facing grids may seem shielded by the accommodation house but experience shows that high and very turbulent wind effects are found between the two structures. Eight axial ventilators each of 105 000 m^3/h capacity and two high pressure centrifugal fans are in operation when the engine plant is operating at full power. In order to solve the annoying problem of salt apparently in the intake air, GRP grids with vertical ribs have been fitted in the air intakes. The grids are formed in such a way that water droplets are captured and drained away. The effect of this remedy has not yet been finally ascertained.

ACKNOWLEDGEMENTS

The author wishes to thank The East Asiatic Company Ltd. for their permission to publish this paper and also to thank Burmeister and Wain Engineering Company Ltd. and Burmeister and Wain Shipyard as well as his colleagues in EAC newbuilding department for their assistance in the preparation of the paper. The opinions expressed are however solely those of the author.

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Discussion

MR. L. SINCLAIR, F.I.Mar.E., said that, in the quest for increasing speed and, thus, for increasing power, triple-screw propulsion had, in his view, long been worthy of careful consideration for a number of very valid reasons. It was gratifying, therefore, that Mr. Kongsted's company had had the courage to build such ships and the data now provided was a welcome addition to our knowledge of the subject.

If, as a result of an increased power requirement, a departure from the conventional single centre-line shaft arrangement was forced upon us, it would seem reasonable to expect best performance from the triple rather than the twin-screw alternative. The gain from the wake would be retained if most of the power was on the centre-line and the relatively lightly loaded wings screws should also give high performance. By this means maximum efficiency should be achieved with good manoeuvrability, even when operating solely on the single, centre-line screw, as this was situated in front of a conventional rudder. Add to this the introduction of a controllable pitch propeller to the centre shaft and it would appear that an extremely flexible and highly efficient installation had been achieved on *Selandia* and *Jutlandia*.

It was no secret that his own company favoured triple-screw propulsion, but it must be added that this was not conditioned by commercial considerations. The containership posed special propeller problems, mainly in relation to its relatively limited draught, as compared with the VLCC, and consequently its high propeller speed. As a result cavitation was an increasing problem as power increased and a change from single-screw propulsion was necessary around the 29·5 MW limit. It was, thus, surprising that the triple-screw configuration had not found a more ready acceptance on the container ship and even on the large tanker where draught limitations did not seem apparent. When the new IMCO regulations began to bite, ballast draught limitations on the VLCC might restrict propeller diameters on this class of ship, thus increasing propeller speed and precipitating the point at which single-screw propulsion must give way to multi-screws.

The East Asiatic Company had decided to have a controllable pitch propeller on the centre-line, with fixed pitch screws on the wings. The Japanese ships apparently had only fixed pitch propellers and it would be of interest to know how the ship performances compared. From the manoeuvring point of view c.p. propellers on the wings might have been preferable with fixed pitch on the centre-line. Mr. Kongsted's comments on this decision would be appreciated.

It was noted that when operating solely on the centre-line shaft, the wing propellers were declutched and allowed to idle, or windmill. This presumably gave lowest drag, but it would be of interest to know if model tests had been used in coming to this decision. Presumably, if controllable pitch wing screws had been fitted. they could with advantage have been locked in a feathering position and it would be of interest to know if this had been considered in the design stages. It would also be useful to know if the considerable amount of cavitation tunnel testing that took place prior to the production of these vessels led to satisfactory propeller performance. In the design stages, concern was expressed regarding the performance of the centre screws as this could perhaps be influenced by the slipstream of the wing screws.

During his presentation, the author made reference to the *Queen Elizabeth* which of course had quadruple screws. In this case the centre screws were to some extent in the shadow of the wings and consequently suffered from erosion and gave a considerably shorter propeller life.

MR. B. E. WELBOURN, M.I.Mar.E., said that, at the beginning of the presentation the author had justified the solution of using triple-screw design on the basis of rising costs of fuel—for steam against diesel—and also the machinery in existence at the time. It would be very interesting, in the light of subsequent experience, to know if his views would change with the introduction of the newer medium speed diesel machinery. Since the ships had been designed, there had been a great change in the machinery available, and it would be very helpful to have the author's comments.

MR. J. CLIFFE, said that he had been intrigued by the comments about operating problems with salt laden atmosphere and would like further information on this point. This seemed to be a relatively new trouble with diesel vessels. Was there any sign of salt build-up in the turbochargers, which was something one might expect when a significant amount of salt was drawn in? Further comments about the exhaust valve problem would also be of interest.

Correspondence

MR. R. G. BODDIE, F. I. Mar.E., wrote that the author had mentioned in his talk the numerous design differences in the five ships of the same class, and in these days of economy and standard ships it was surprising to hear that such basic features as the direction of rotation and design of the propellers, the design of the stern of the ships, the main shafting and the layout of the main engine had not been standardized.

What was the reason for the lack of standardization and the apparent duplication of design and development work, not to mention the logistic problems where not even the propellers were interchangeable between ships.

MR. A. NORRIS, F.I.Mar.E., wrote that Section I of the paper admirably listed the basic considerations regarding propulson plant, making it appear that the final selection was inevitable and arrived at in a truly rational manner. An enormous amount of high quality technical effort must have been put into the design study and the technical committee responsible must be sincerely congratulated on the courage of their conviction in departing from the easy answer of accepting twin-screw steam turbine plant.

The Mitsui container ship, mentioned in this paper and described in the July 1972 issue of *Shipbuilding and Marine Engineering International*, had a generally similar power plant. In this ship (*Elbe Maru*), the designers used turbo-alternators, supplied with steam from waste heat recovery plant, despite the considerations discussed by Mr. Kongsted in the *Generating Plant* section of his paper, which led to his selection of diesel generators which required 9 to 10 tonnes/day of marine diesel fuel.

In an article* in the press, Mr. Norris had pointed out that the total potential power output from waste heat recovery plant, in a ship of *Elbe Maru* power, could have been as high as 5500 kW and that the number of main engine cylinders could then be reduced by two, if surplus electrical power was fed back to the propulsion shafting. Could Mr. Kongsted comment on his view of the feasibility of such procedure for any future generation of large powered diesel ships; also, on whether any operating cost differences had shown up as yet for the electric generating plants of the two types, steam turbo-alternators and diesel alternators in the separate vessels?

When discussing *Operating During Energy Crisis*, the author had mentioned that in the reduced power condition only about 25 per cent of the electrical load could have been supplied from a waste heat recovery plant with steam turbo-generators. It was obvious that the balance of the 1800 kW

* NORRIS, A., 1972. "Why Waste It". *Shipbuilding and Marine Engineering International*, September, p. 445, Figs. 1 and 2.

load could be made up by operating a diesel generator in parallel (2 × 1700 kW + 1 × 980 kW in *Elbe Maru*) which would still save 2·5 tonnes of expensive diesel fuel per day. Alternatively the steam production could be supplemented from an oil fired boiler to allow the turbo-alternator to meet the full electrical load. When using waste heat recovery plant under this kind of reduced power condition, it should be remembered that the heat input from the boiler fuel might be required for the latent heat phase of steam production only, as the sensible heat input requirement could be met from relatively low temperature exhaust gases. Hence the specific fuel consumption of the steam turbine under this part load firing condition was reduced and the total fuel cost might well be comparable to that of a diesel generator as the cheaper residual fuel was burnt under the boiler.

Author's Reply

Replying to the discussion, the author thanked all the contributors for their interesting remarks.

Mr. Sinclair was, of course, well known to the author as one of the leading specialists on propeller problems and it was true to say that Mr. Sinclair's paper on triple-screw propulsion of large tankers and fast containerships had had an influence on the choice of the triple-screw concept for the diesel powered containerships.

He agreed that it was astonishing that the triple-screw configuration had not been adopted in later designs of third generation containerships with comparable total power. In these twin-screw ships, however, engines of sufficiently high horsepower at sufficiently high shaft speed were available. There were strong indications that first cost had been in favour of twin-screw installations. On the other hand an up to date, triple-screw plant would require a much shorter engine room than the ships constructed in 1969. Also the flexibility of the triple-screw plant from an operational point of view was of such great advantage that the extra cost would be worth while considering.

With regard to the question of the drag from the declutched wing propellers, no model tests were carried out in order to get exact figures. The drag was a function of the propeller speed and the friction moment of the shafting and was calculated to be insignificant (corresponding to 220 kW/shaft). In the design stages the possibility of adopting c.p. propellers for the wing shafts was considered, but the cost and complication of c.p. propellers which could be locked in a feathered position was considered prohibitive.

With regard to performance of the Japanese owned vessel *Elbe Maru*, with three fixed pitch propellers, the author had no knowledge of any exact figures. According to the builders, who were also responsible for the construction of the Norwegian owned *Toyama*, the performance of the former has been entirely satisfactory.

Regarding the problem of the influence the slipstream from the wing propellers might have on the centre propeller, a number of tests were carried out in the Gothenburg cavitation tunnel with a full size model. Also open water propulsion tests, including strain gauge measurements, were carried out at an early stage to ascertain this influence.

According to these tests, the wing engine shafting was diverging slightly and no cavitation erosion was found on the centre propeller.

In reply to Mr. Welbourn, the author said that, in the design stage, medium speed engines were considered, but the extremely large number of cylinders needed with available engine makes was then considered prohibitive. Mr. Welbourn correctly pointed out that the situation was different today, when medium speed engines with outputs of about 1100 kW/cylinder were available. There was no doubt that such engines would have to be considered for a new project. On the other hand, the slow speed engine also had seen considerable development in power and engine speed and would, in the author's opinion, still be a strong competitor to the medium speed engine from an overall operational point of view. There might, of course, be an advantage in first cost in favour of a plant with medium speed engines.

With power requirements above about 36 775 kW/shaft (up to say 44 130 kW) medium speed engines might be the only choice when diesel propulsion was preferred.

The author, replying to Mr. Cliffe, said that of naval vessels and merchant ships equipped with gas turbine propulsion plant he had heard of problems with compressor blade fouling by salt particles. Whilst a salt laden atmosphere was certainly a problem in the EAC containerships, this caused little trouble to the turbocharger compressor sides. Most of the salt was captured by the filters and silencers in the air intakes, which had to be cleaned rather often. Also the compressor rotors had to be cleaned from time to time, but this could be done while the engines were in operation and had little influence on the performance of the vessels.

In reply to Mr. Norris, the author recalled some very interesting discussions, years back, with him regarding waste heat recovery from diesel engines.

The decision to choose a diesel driven generating plant for the containerships was not based solely on economical considerations and the decision was certainly not an easy one. These engine plants, with their enormous quantity of waste heat and long periods at sea at full power, represented a strong case for a waste heat recovery system feeding turbo-alternators, and with back-up power from oil fuel heated boilers and diesel generating plants for manoeuvring and port use. Such a plant, even though it was quite feasible, would, however, mean a complication of the whole installation, and the final choice of a diesel generating plant for the Scandinavian owned vessels was in fact a choice for simplicity against economy in operation.

The difficulty in this choice was demonstrated by the fact that other owners of diesel propelled containerships had (as pointed out by Mr. Norris) arrived at a different conclusion.

With the prevailing fuel prices, a project for similar vessels would today most probably include a waste heat recovery plant. One might even accept the added complication of a plant with maximum waste heat recovery and feed-back surplus electrical power to the propulsion shafting as proposed by Mr. Norris. On the other hand one would also have to take into consideration that the economic advantage gained would be completely jeopardised by a reduction in propulsion power, as experienced during the energy crisis.

The author, in reply to Mr. Boddie, said that it might seem surprising that the construction of the four Scandinavian owned containerships was not standardized and that such basic features as direction of rotation and design of propellers, machinery layout, etc, were not the same for all the vessels.

The reason for this was that these vessels were constructed by three different shipbuilders, each having the full responsibility for their own design. To avoid any delays in the design, it was accepted by the owners that the four vessels would not be identical, like sisterships. It was, however, a strict requirement that they should be compatible from an operational point of view.