THE STOWAGE AND HANDLING OF TANKER CARGOES IN THE NINETEEN SIXTIES

A. F. Brereton, C.Eng., A.M.R.I.N.A.,* and

D. J. Gibbons, B.Sc., C.Eng., M.I.Mech.E. (Member)†

Mr. Brereton

Mr. Gibbons

This paper describes the developments that have taken place in the stowage and handling of oil tanker cargoes during the present decade, and the way in which the changing patterns of trade have been accommodated by the authors' company. The marked divergence of tanker design between the general product carrier and crude oil vessel is discussed, together with the particular problems that have arisen in the cargo systems of these different types of vessel. Cargo integrity and ship safety during cargo handling operations are of extreme importance, and the steady progress which has been made in associated systems have all tended towards increased safety, easier operation and at the same time reduced personnel requirements.

INTRODUCTION

The rapid increase in the size of the crude oil tanker in recent years has led to a more marked divergence in the basic requirements of the cargo handling systems of the crude and general product carrying vessels.

The great difference in their respective sizes reduces the possibility of the large crude tanker (ever) carrying refined products and segregation considerations will apply only to differing grades of crude oil, probably no more than two grades in proportions of perhaps 25:75, 35:65 or 50:50. On the other hand the number of refined liquid products has increased considerably and the specifications of many of these is so rigid that contamination by even small quantities of other grades or outside agent would mean rejection by the customer, or acceptance but downgrading of the parcel, both resulting in loss of profit and goodwill.

This has led to the evolution of very definite and dissimilar requirements for each type of vessel and it is the intent of this paper to comment upon fundamental ship requirements and on the more significant developments that have occurred in the equipment and systems associated with the stowage and handling of tanker cargoes.

THE GENERAL PRODUCTS CARRIER

The general products carrier is a vessel which rarely carries the same cargo twice on consecutive voyages and may be called upon to carry simultaneously, without intermixing, a large number of different types of cargo, in specified proportions, to practically any port in the world. Table I shows the major cargo categories which this type of vessel is called upon to handle.

TABLE I—MAIN CATEGORIES OF REFINED PRODUCTS

Products	No. of grades	Specific gravity
Motor gasolines	33	0·713—0·741
Aviation fuels:		
Gasoline	8	0·699—0·712
Kerosene	6	0·774—0·790
Kerosenes	23	0·791—0·808
Vaporizing oils	4	0·774—0·790
White spirits	36	0·63 —0·90
Feedstocks	18	0·64 —0·785
Gas oils	44	0·827—0·845
Diesel oils	11	0·827—0·845
Lubricating oils	18	0·87 —0·93

The following considerations should be taken into account during the design and construction of a vessel of this type.

The cargo deadweight will be the maximum that can be obtained within the limitations of length, beam and draught imposed by the proposed trading routes and port facilities. Current designs are of about 20 000-22 000 dwt on summer draughts in the region of 30 ft 0 in.

The cubic capacity of the cargo space is an important measure of the ability of the vessel to carry the lighter grades of product and also allows greater flexibility in distribution of parcels of differing quantities and specific gravities. Fig. 1 illustrates the advantage that can be taken of an excess of capacity to accommodate parcels of varying size. It is considered that as a standard the cubic capacity of the cargo space should be sufficient to allow the vessel to load to the tropical draught, even keel, with a homogeneous cargo at 50 ft^3/ton occupying 98 per cent of the available cubic capacity. Although full cargoes of 50 ft^3/ton rating are relatively rare they are being encountered and it is of interest to note that a full capacity cargo has recently been taken of Kuwait natural gasoline which has a rating of 57 ft^3/ton.

The required subdivision of the cargo space depends upon the number of parcels which the vessel is to be designed to carry. From a study carried out by the authors' company on the trading of nine clean oil vessels of 20 000 dwt having eleven cargo tanks (33 compartments), it was found that out of 52 cargoes

* Naval Architect, BP Tanker Co. Ltd.

† Propulsion Engineer Superintendent, BP Tanker Co. Ltd.

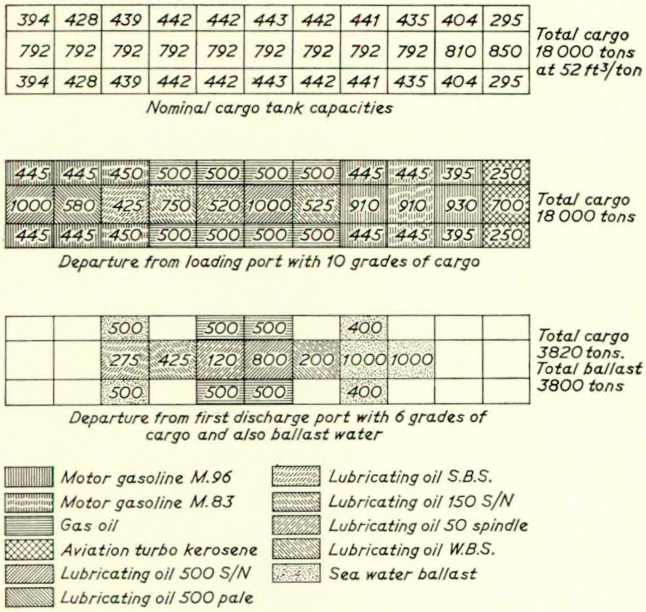


FIG. 1—*Loading pattern for ten grades of product*

carried seven could not have been accepted had there been a lesser number of tanks. Table II summarizes the voyages covered relative to the number of parcels and loading and discharge ports.

Size of individual compartments has also been the subject of considerable study. It is still the policy of some companies to provide a number of special parcel tanks of fixed capacity to meet certain trade requirements. Whereas this can be a very useful facility on a fixed route, it is considered that where the standard tank size is relatively small, say 1000 tons, and the capacity rating is high, it is preferable for world wide trading to have all tanks of equal capacity with one centre tank of approximately equivalent capacity to the two adjacent wing tanks. This arrangement simplifies the problems of contriving, at short notice, cargo loading patterns which may involve as many as twelve parcels with two or three loading and discharge ports for which even keel arrival and departure is necessary, due to limitations in depth of water at the port.

TABLE II—SUMMARY OF CLEAN OIL TRADING

No. of voyages	No. of parcels	Single port load/discharge	Multiple port load/discharge
25	4	25	—
15	5	12	3
7	6	4	3
2	7	1	1
1	8	–	1
2	9	1	1
Total 52		43	9

The cargo piping system design is governed by the parcel requirement, with the lines sized to match the cargo pumping capacity and the designed loading rate. Ideally it should be possible to carry as many parcels as there are tanks and, indeed, this has been the case with smaller special parcel carriers using submerged pumps in each tank, or pump rooms positioned between alternate tanks. For the general product carrier, a reasonable economic compromise can be achieved with four cargo pumps and a line system that will allow any pump to draw from any tank. Fig. 2 illustrates the piping system adopted by the authors' company for vessels of this type. This system is simply a double ring main with interconnected cross-overs at each tank. There are direct loading lines to each of the four main lines to allow the loading of four dissimilar grades, these can be directed to the forward part of the cargo space and by loading through the pump room, a further four grades can be loaded separately into the after part.

Loading of more than eight grades and discharging of more than four grades can be carried out only by accepting a measure of contamination of following parcels or by line washing and draining. This system has now been in operation since 1964 and has been well received by the operating personnel.

Avoidance of contamination is the most important aspect of product carrier cargo systems. It can occur from the following causes:

a) leakage of isolation valves across faces;
b) leakage through valve glands or inspection doors;
c) structural failure of divisional structure or piping system or porosity of welded connexions;
d) leakage at cargo piping couplings, flanges or expansion joints;
e) inadequate cleaning of lines or tanks prior to loading;
f) maloperation of pumps or valves by shipboard personnel.

To combat these it is necessary to take the following measures:

i) careful selection of proven equipment designed to meet the conditions encountered during operations in service;
ii) regular routine maintenance procedures in service, including realistic testing procedures with rigorous examination by responsible personnel;
iii) structure—insistence upon full watertesting of at least alternate tanks with careful examination by responsible personnel; piping—supervision to ensure that piping is correctly aligned and assembled during construction with adequate anchors and supports and again careful examination during tests in the shop and of the completed system;
iv) selection of a suitable proven connexion and supervision to ensure that the joints are correctly made, in accordance with good engineering practice;
v) in the line system design valves should be positioned to avoid pockets or dead ends so that through washing can easily be carried out; dips in lines should be avoided so that the liquid will drain freely along the line to the pumps or to the tank served; tank structure to be designed and drain holes provided so that all surfaces drain freely to and across the bottom of the tank, to facilitate mopping operations;

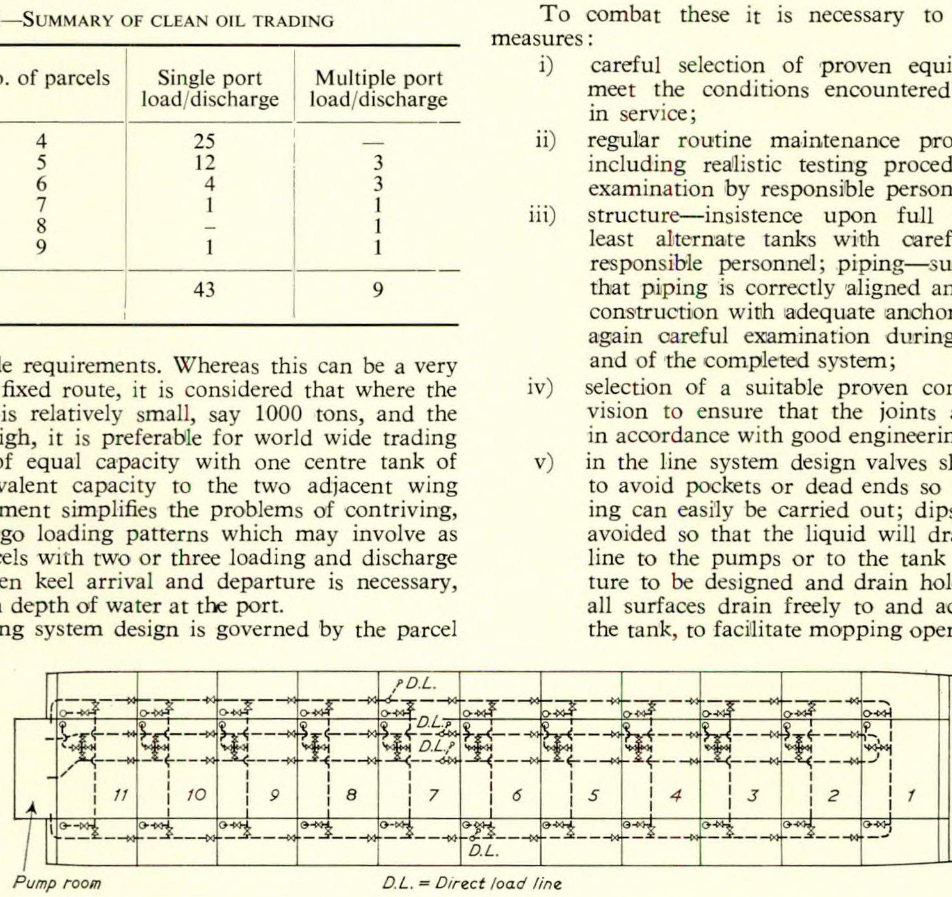


FIG. 2—*19 000 dwt general product carrier cargo piping system*

vi) provide a system in which the controls are easily recognized through a colour coding and standard arrangement for each tank.

It is appropriate to end this section with some remarks on the effects of a suitable paint system within the cargo tanks. Whilst the painting of cargo tanks was introduced primarily as a measure to eliminate corrosion, it is now recognized that other advantages have equal importance. Some of these are: An increase in deadweight results from the reduction in scantlings allowed by classification societies. Lighter forms of construction such as the use of corrugated bulkheads may be used with less risk of premature leakage.

The elimination of scale prevents the significant loss of deadweight from this cause. Furthermore the tanks wash and gas-free more quickly and drain better due to the absence of scale blocking limber holes.

Damage to valves and other fittings by scale particles is minimized and this has been a major factor in eliminating many double valve complexes and their associated undesirable losses on the suction side of the pumps.

Not only is tank washing and venting time considerably reduced, but a very high standard of cleanliness can be achieved in reasonable time. Clearly finer limits may be confidently accepted with regard to product quality.

A most important result of the relative ease with which tanks may be cleaned is that certain completely dissimilar cargoes may be carried successively such as heavy lubricating oil feedstocks followed by aviation spirit, thus increasing the probability of back-loading.

Tanks gas-free quicker and tend to remain gas free and this, in conjunction with the reduced pressure washing time, provides a safety bonus.

The effective days in service are increased by the items mentioned above and by the reduced period in which steelwork renewals take place.

In view of the importance of these advantages it should be stressed that only the highest quality coatings should be used and applied under strict supervision to properly prepared surfaces.

THE LARGE CRUDE CARRIER

Unlike the general product carrier, primarily the crude carrier is designed to carry bulk cargoes between the relatively few terminals capable of receiving ships of these draughts.

For vessels designed for the bulk carriage of crude oils consideration should be given to the following points.

Cubic capacity of the cargo tanks must be sufficient to allow loading to full draught when carrying the lightest grade of crude oils, Table III gives the properties and origins of some of the crude oils which must be considered.

It is suggested that a reasonable standard for capacity rating would be to allow the vessel to load to tropical draught, even keel, with a homogenous cargo at about 45 ft^3/ton occupying 99 per cent of the available cargo space cubic capacity. Capacity in excess of this requirement can be usefully taken for ballast-only tanks positioned within the cargo tank length.

These ballast tanks have a number of advantages. They can be positioned at the point of maximum bending so that when empty in the loaded condition the effect will be to reduce the sagging stresses.

The permanent water ballast spaces which are included in the measurement for gross tonnage are eligible for deduction in arriving at the net tonnage. The consequent decrease in net tonnage reduces the port charges etc., which are based on net tonnage measurement.

In their isolation from the cargo spaces, ballast water can be loaded or discharged without danger of contamination or pollution. The greater the quantity of clean ballast available, the lesser is the number of cargo tanks requiring the high standard of cleaning before receiving the clean ballast which must be discharged before or when loading.

The selection of clean ballast tank position in the cargo tank length gives flexibility in the selection of the desired longitudinal centre of gravity of the cargo spaces. Excess ballast capacity can be artificially provided by increasing the depth of the ship. The excess capacity will allow the provision of sufficient permanent ballast tanks to satisfy a normal ballast condition. The cost of the additional steelwork due to the increased depth is compensated by the greater ease of ballast

TABLE III—CRUDE OIL DATA

Country and name	Terminal	Specific gravity at 60/60°F
Iraq		
IMEG A	Banias	0·845
,, A	Tripoli	0·850
,, B	Tripoli	0·852
Basra	Khor al Amaya	0·846
Iran		
Light	Mashur	0·854
Heavy	Kharg	0·869
Kuwait	Mina al Ahmadi	0·869
Qatar	Umm Said	0·816
Abu Dhabi		
ADMEG	Das	0·839
ADLEG	Jebel Dhanna	0·826
Neutral Zone		
Eocene	Mina Saud	0·956
Ratawi	Mina Saud	0·901
Khafji	Ras al Khafji	0·884
Arabia		
Arabian	Ras Tanura, Sidon	0·854
Safaniya	Ras Tanura	0·891
Algeria		
Hassi Messaoud	Bougie	0·806
Zarzaitine	La Skirrha	0·822
Libya		
Brega	Marsa al Brega	0·829
Oasis	Es Sider	0·815
Sarir	Marsa al Hariga	0·836
Nigeria	Bonny	0·853
	Bonny	0·891
Trinidad	Pointe d'Or	0·873
Venezuela		
Bachaquero	Puerto Miranda	0·975
Tia Juana Medium	Amuay Bay	0·896

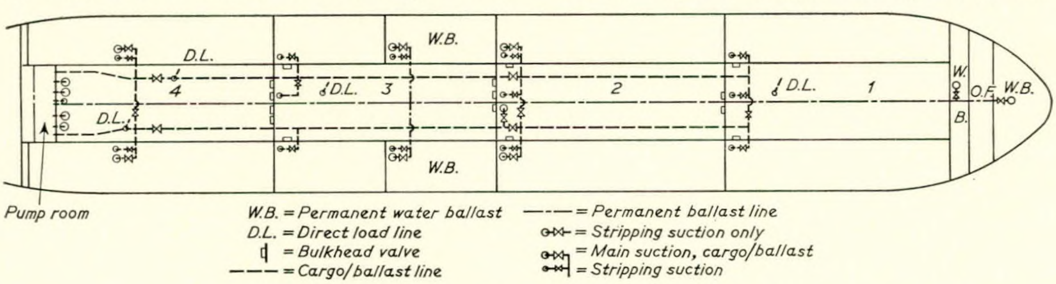


FIG. 3—*213 000 dwt crude oil carrier cargo piping system*

handling, tank cleaning and the simplification of the anti-corrosion measures.

The gross and net registered tonnages will of course be increased and this must be considered in making the overall assessment.

Subdivision—from a cargo handling point of view the least number of tanks is to be preferred provided of course that carriage of two grades of cargo in the proportions previously stated can be achieved.

Fig. 3 shows the tank arrangement of a 213 000 dwt tanker to be built for the authors' company; of the five pairs of wing tanks, one will be exclusively for clean ballast water. In the aftermost of the four centre tanks a smaller tank of 4000 tons capacity has been arranged for the reception and separation of slops and dirty ballast water.

In designing the cargo piping system there is little difficulty in meeting the elementary requirements for cargo segregation. If, however, it is decided to make provision for handling ballast water in the cargo tanks, concurrently with the loading and discharging of cargo, some special provisions must be made to ensure positive segregation.

The advantages of such a system can be summarized as follows.

Loading

When moving into the loading berth, whether offshore or jetty, a minimum of 35 per cent ballast is retained on board to facilitate ship handling. This ballast water, in cargo and permanent ballast tanks, is discharged as loading commences and at no time does the ship have less than 35 per cent deadweight. Not only is time saved by discharging water from the cargo tanks after berthing, but the ship is less sensitive to the development of bad weather which may deter the Master from taking his vessel onto the berth.

Discharging

During the discharging operation it is necessary first to discharge those cargo tanks which are to receive ballast water. The pumps drawing from these tanks can then be switched to general cargo discharge duties and the ballast water allowed to gravitate into the selected tanks.

Towards the end of the discharge period, when the tanks are being drained, one or two of the cargo pumps can be used to bring the dirty ballast tanks to the required ullages for the off jetty condition. At this stage the vessel is, of course, heavily trimmed by the stern to assist the draining operation and more ballast will be required in the forward part of the ship to reach the sea ballast condition.

Other advantages common to both loading and discharging operations lie in the reduction of the very high freeboard if only permanent ballast tanks are being used. The reduction in windage area reduces the range and loading on the moorings, access to and from the vessel is not so difficult and the height and range for which the shore loading/discharging booms must cater is reduced.

The disadvantages of the simultaneous ballast system lie, of course, in the additional piping and valves that may be required and the possibility that mal-operation or faulty equipment may lead to oil being discharged overboard.

With regard to the general considerations of cargo piping systems, the accepted conventional arrangement is to provide one large bore main tank suction line to each cargo pump. The lines are extended through the cargo tanks, with cross-over pipes and suction strums in the tanks, so arranged that each pump will handle an equal proportion of the total cargo. Stripping arrangements can differ quite substantially, but a common solution is to provide a similar, but smaller bore, piping system connected to positive displacement pumps.

In an effort to simplify the cargo handling system for bulk crude oil carriers it was decided by the authors' company to develop a cargo transfer system in which the main cargo suction lines would be omitted. The cargo oil would flow through valves fitted at the bottom of the tank divisional bulkheads to the aftermost cargo centre tank, from which the main cargo pumps would draw. A small ring main would be required with a suction to each tank for ballasting, tank cleaning and any stripping that might be necessary.

A design was prepared for fitting the system to a class of five vessels of 64 000 dwt with six centre and five cargo wing tanks.

The basis of the calculation to establish the required free drainage area at each tank division was taken as follows.

Cargo Discharge

At the full cargo pumping rate (3 × 1980 tons water/hour at 150 lb/in^2) with all valves open, the vessel would develop and maintain a trim of 12 ft 6 in by the stern. The development of the stern trim should, at least, be matched by the bodily lift so that the arrival draught would not be exceeded.

Cargo Loading

From an initial trim of 12 ft 6 in by the stern the vessel would, at full loading rate (6000 tons oil/hour), assume an even keel condition as the loading is completed, thus allowing "topping off" of individual compartments from aft to forward.

Using valve discharge coefficients obtained by experiment, preliminary calculations were made to establish the required number of valves at each bulkhead. Typical loading and discharge operations were then made by an analogue computer to establish the attitude of the ship throughout these operations. The computer results indicated that there would be a small head trim during the first hour of maximum discharge, following which the vessel settled steadily by the stern to meet the designed requirement.

Full scale trials were held on completion of the first vessel from which satisfactory loading and discharge patterns were evolved.

Experience in service has been very satisfactory and it has been found that increasing the stern trim to 15 ft 0 in cuts out the need for stripping in any but the aftermost centre tank and this reduces the overall discharge time. Without shore restriction discharge of cargo has been completed in 15 hours.

The advantages of this system can be summarized briefly as:

a) a saving in weight;
b) a saving in first cost and maintenance costs;
c) improved pump performance due to elimination of suction line losses;
d) reduction in time taken for stripping during cargo discharge — stripping is confined to the aftermost centre tank;
e) a simplified operating technique, particularly during the discharge operation;
f) the flow of oil through the centre tanks has apparently reduced the accumulation of sediment.

The disadvantage of this system lies in a certain loss of flexibility in that only two grades of cargo can be discharged simultaneously, one of these being handled by the relatively small ring main.

For the 213 000 dwt tanker (see Fig. 3) the basic cargo transfer system illustrated has been supplemented by a large bore piping system to those tanks which will be ballasted/deballasted during the cargo discharging/loading operation.

The provision of good drainage from the horizontal surfaces and across the bottoms of the cargo tanks is amply repaid in time saved when stripping cargo and tank cleaning. Generous cut-outs should be arranged where longitudinal stiffeners pass through transverses and large drain holes are provided through the bottom longitudinal stiffening at the after end of the cargo tank in line with suction strums. If bulkhead valves are fitted the sill height should be kept to a minimum if sediment accumulation is to be avoided.

The deck cross-over pipes to connect to the shore hoses or sea lines require special consideration in view of the standard loading booms which are now being generally provided to handle the increased size of hoses.

Recommendations as to pipe line spacing, hose lifting capacities, etc. have been made by the Permanent International Association of Navigational Congresses.

Great as are the advantages of fully painted cargo tanks in the large crude carrier, the cost is very high. In a 200 000 ton vessel the total area to be painted is in the order of 350 000 yd^2. Accommodating this work without extending the constructional programme requires not only careful planning but also having available a large labour force with the requisite skills and equipment to meet the high standards of surface preparation and material application that are essential.

Because of the relatively high cost and possible delays in construction there is a temptation either to discard full tank painting in favour of selected treatment of the areas most vulnerable to corrosion, namely the deckheads and uppermost parts of the tank structure together with perhaps bottoms and tops of other horizontal surfaces, or even to reject coatings altogether in favour of a "short life" philosophy, but this must of course depend upon the economic assessment of the particular case.

CARGO VALVES

The selection of cargo valve type must be made in the full recognition of the fluids to be handled. The maximum system design pressure and the range of line pressures, over which the valve must be guaranteed to be tight, must be defined.

Of the various types of valve available, the most commonly used for oil tankers has been the wedge gate valve. With the increases in pumping rate of the larger crude carriers and consequent increase in valve size, the butterfly valve is now achieving greater popularity from both economic considerations and the relative ease of operation.

For the cargo transfer system it was necessary to develop a valve which would fit directly to the lowest part of the cargo tank divisional bulkhead. A vertically operating rectangular gate valve was selected to avoid interference with the tank structure and to achieve the best possible drainage over a minimum height of sill. Two hand operated prototypes were fitted to a vessel then under construction and tests were carried out in service to establish discharge coefficients, tightness and drainage performance. Some difficulties were encountered with the first complete power operated installations in getting the valve to be tight without jamming. This was due to a number of reasons, including bulkhead and valve frame distortion under load, method of face adjustment and wedge taper. After the adoption of a standard procedure for the setting up of the valve, using temporary closing plates with a hose connexion to obtain the service liquid head, the difficulties were finally overcome. Adjustments made to the design of the latest type of valve take account of this experience and further difficulties are not expected.

Tightness of the valve must be the prime consideration, though ease of operation, maintenance and minimum losses through the valves are also important.

A valve may leak through the body, across the face or through the gland. The following test procedure has been adopted by the authors' company to ensure initial tightness of the valve.

Shop Test

All valves are to be hydraulically tested with the gate in the open and closed positions to twice system working pressure.

Leakage Test

All valves to be hydraulically tested using water to a pressure of 30 lb/in^{2}g. Tests are to be made on both faces with the valve in the closed position and hand hole door open.

Ship Tests

All valves to be hydraulically tested *in situ* using water to a pressure of 20 lb/in^{2}g. Tests are to be made on both faces with the valve in the closed position and hand hole door open.

The most common causes of valve leakage are:

i) elastic distortion of the valve body;
ii) leakage past the valve spindle packing;
iii) leakage past the faces damaged by scale or sand;
iv) leakage past the faces when valve will not close due to scale or foreign bodies fouling the wedge.

These will now be considered.

Valves and adjacent piping should be so supported that loading from ship and pipeline movement is not transmitted to the valve. Before installation the pipelines should be checked to ensure correct alignment and the correct sequence of bolting should be followed.

The problem with packing is to ensure a tight gland whilst retaining ease of operation. Trials carried out with nitrile sealing rings have been reasonably successful though swelling of the ring in certain products has occurred making the valves more difficult to turn.

Damage to valve faces can be reduced by using a harder face material than the standard gunmetal ring. There are also available valves with flexible seats and others with "viton" or "nitrile" sealing rings. Valves incorporating these features have been used extensively by the authors' company for single valve separation in product carriers. Difficulties have in some cases been experienced due both to swelling of the nitrile ring and to incorrect clearances, but it is hoped that these have now been overcome.

Scale is difficult to combat and its absence is an advantage which comes from painting of the cargo tanks. Foreign bodies in the pipelines can be countered only by thorough inspection during the installation or at maintenance periods.

POWER OPERATION OF CARGO VALVES

Many of the vessels entering service during the past few years have been fitted with power operation to the cargo valves. The extent and types of system that have been used varied considerably according to the physical requirements of the cargo systems and the preferences and philosophies of the various owners.

Before deciding upon the provision of power operation it is necessary first to establish the limits of necessity and desirability. Necessity is determined by the physical ability of the operating personnel to operate the valves manually within the time available.

The operating efforts for conventional types of gate valve vary with the lead of the thread, length of time in service, length of spindle and types of spindle support bearing. For a thread lead of 2/in it is considered that gate valves of 16-in diameter and above should be provided with power operation.

The case for the power operation of valves of below the strictly manual limit of 16-in diameter requires a much wider study embracing operational techniques, manning policies and utilization factors.

Product carriers engaged on coastal trade or short hauls are faced with valve setting almost every day as well as during the loading and discharging operation when time is of utmost value and fatigue of operating personnel most likely. In this case the improvement in operational efficiency can outweigh the penalty of the additional first cost.

Another factor which cannot be ignored is safety within the pump room. Power operation and remote control of pump room valves and cargo pumps with suitable monitoring devices reduces the number of occasions when the pump room must be entered during pumping operations.

Once it has been decided that a proportion of the valves require to be power operated, it becomes a more attractive proposition to make fuller use of the centralized power source and transmission lines by extending the application to cover all of the valves commonly used in the cargo system.

Having accepted the need for power operation, the type and capacity of the system to be employed must be decided and what system of control should be adopted.

Capacity of the system is determined by the following requirements: the speed of valve operation and the number of valves to be operated within a given period.

The standards adopted by the authors' company allow for two valves to be operated simultaneously with a closing or opening speed of 2 ft/min.

With regard to selection of the type of system, there are the alternative power sources of electric, hydraulic and compressed air. Electric motors for oil tankers can be rejected for obvious reasons of safety.

Each of the other systems has its advantages and disadvantages, but one consideration which may well be overlooked is whether the installation contractor understands, or is capable of achieving, the quality of engineering that is required. This is particularly important in the case of those hydraulic systems with low tolerance to particle inclusion. For systems of this type a very high standard of engineering hygiene must be maintained under what can be very difficult shipyard conditions.

Hydraulic Operation

The most popular choice so far appears to have been for an hydraulic oil system with piston-type actuators mounted directly onto the cargo valve. Alternative systems are available in which rotary actuators are fitted directly onto the pump room and deck valves and to the deck spindles of the tank valves.

The advantages of the rotary system lie in the ability to operate the valve mechanically by hand from the deck in the event of failure of the hydraulic system, and the ease with which positive valve position indication can be arranged.

The system using piston-type actuators has the important advantage over the rotary type of a lower first cost, less fittings on deck liable to damage from heavy weather and a smaller line size.

The major problem with any hydraulic oil system has been the leakage of the hydraulic fluid. If the leakage is not traced and isolated the result will be eventual failure of the whole system and in the case of product carriers there is possibility of contamination of the cargo if the leakage is into the cargo tank. There is also the danger of low flash point cargoes entering the hydraulic system. To counter these possibilities piping should, as far as is possible, be installed in continuous lengths and together with the associated equipment, be of a material resistant to the corrosive atmospheres on deck and in the cargo tanks. Couplings and connexions must be carefully selected and positioned so that they may be easily inspected and maintained.

Another problem that has been encountered principally with piston-type actuators has been the jamming of wedge gate valves in the closed position. To avoid this it is necessary to select the closing effort with care, to avoid over driving the valve wedge into the seat, and to provide a suitable margin on the designed system effort to unseat the valve.

An alternative hydraulic system, which is available but not yet in use, utilizes water from the tank cleaning line to drive a rotary water turbine actuator fitted to the deck spindle of the valve. The use of an available power supply makes this an attractive proposition for conversions of ships in service, provided suitable arrangements are made to control the disposal of the exhausting water.

Pneumatic Operation

Compressed air has been used successfully by many tanker operators. Rotary actuators can be fitted directly to the pump room and deck valves and to the deck spindles of the cargo tank valves. Compressed air is supplied from a main line on deck with an exhaust from the motor to atmosphere.

A plentiful supply of dry air is required and the air delivery lines must be arranged so that they may easily be drained.

Systems of this type give little trouble and the compressed air which is available when the valves are not being operated can usefully be tapped for other power tools used about the deck.

Disadvantages of this system lie, as in any deck unit, in the maintenance of the exposed equipment, possible heavy weather damage, and in the relatively greater power requirement (five or six to one) as compared with the hydraulic ram system.

Cargo Valve Control Systems

Control by the operator may be arranged locally to the valve, conveniently grouped in stations or completely centralized in a control room remote from the cargo tank area.

At the control position it is necessary for the operator to know the valve position and in the case of cargo tank valves the liquid level in the associated tank.

Control in the case of local or grouped systems can be arranged by fitting a multi-port two-way direct control valve in the power supply line, giving open, closed and locking positions. For remote operation the direct control may also be used, or alternatively a secondary system, electric or pneumatic, may be introduced in series to reduce the complexity of piping leads to the control room. Although this can be an important consideration with regard to ease of installation and future pipeline maintenance, the introduction of another control element must reflect against the reliability of the system.

Selection of valve position indicating equipment depends very much upon the type of actuator being used and the position at which the indication is required. Obviously for local control of rotary actuators standard mechanical indicating gear is all that is required and for ram-type actuators mounted to pump room and deck valves with local control, simple mechanical systems can easily be arranged.

For position indication, remote from the valve, there are a number of possible solutions, using either flow meters, slave cylinders or pressure differential secondary systems. All of these are designed to indicate the actual position of the valve, which has been considered to be of some importance. It has been found in service that the accuracy of indication can be seriously impaired if there is air in the lines; it is necessary for the system to be bled and the pointers reset whenever this is evident. A simpler and more reliable form of indication, but giving only fully open or fully closed positions, can be provided by a pressure-type gauge connected to ports on the actuator cylinder. Intermediate positions are then obtained by proportioning the time taken for the full stroke of the valve movement and setting the control to the lock position.

A further simplification can be achieved by using a simple flow meter in the hydraulic powerline which indicates that the valve is moving. When the flow meter stops rotating it is assumed that the valve has completed its travel and is either fully open or closed depending on the movement of the control lever.

Tank Level Indication

Accurate tank level indication at the deck, whether by hand dipping or mechanical gauge, has never presented any problem. Remote control systems demand reliable tank level indication, as without it all the advantages of the remote control are lost and the system is dependent upon information from the deck. The degree of accuracy required will depend upon the type of ship, for a crude carrier probably ¼-in accuracy over the top and bottom 10 ft 0 in, with 6-in range at intermediate levels will be all that is necessary, but for product carriers where ullages vary considerably throughout the ship due to parcel sizing, the gauge should be accurate to at least ¼-in over the full depth of tank.

CARGO PIPELINE MATERIALS

The materials most commonly used for cargo piping systems have been cast iron, steel and spun or spheroidal graphite cast iron.

Cast iron is a relatively cheap material with very good resistance to corrosion, but its proneness to brittle fracture and greater weight for equivalent duty has led to its virtual disuse in cargo systems with large bore piping.

Steel piping would be a very suitable material were it not for the rapid internal corrosion that occurs at the bottom of horizontal piping, which does not drain easily. It is not uncommon for piping of about ½-in thickness to fail after four to five years in service.

Spun cast iron and spheroidal graphite cast iron piping has the corrosion resistant qualities of cast iron but is much less susceptible to fracture due to its superior mechanical properties. Fractures do occur, primarily because of mechanical damage during installation, but these are usually found during the testing of the system or in the first year of service.

New materials and applications are currently on test or under review, the most promising of these would seem to be steel piping coated internally, after shot blasting, with epoxy resin based paints. Piping of fibreglass resin laminate has also performed well under test. It is flexible, easily handled due to its low weight and is apparently unaffected by crude oils or sea water. The cost level for this material has yet to be properly established, there seems no doubt that as its use becomes more widespread there will be a sharp reduction in its present high cost and it may then be possible to justify a complete installation taking into account the saving in weight that would be achieved.

Other materials such as aluminium brass or stainless steel have been considered, but the high material cost, which is unlikely to reduce, would not seem to justify their use.

ASSOCIATED SYSTEMS

Tank Cleaning

For many years tank cleaning has been carried out by portable water jet washing machines of about 40 tons/h capacity and 50 ft 0 in throw. The machines are lowered into the cargo tanks through small openings in the deck and operated at various drop levels. The oil and sediment, washed from the structure, drains to the tank bottom and is discharged to the slop tank by drain pumps or eductors.

During this operation the vessel is trimmed heavily by the stern to promote good drainage to the suction strum. Good drainage is most important when washing the tank bottom, where the sedimentation is greatest, as accumulated water protects the sludge from the jets of the washing machines and the oil floating on the surface of the water is above the level of the strum.

The washing of cargo tanks on the product carrier does not generally present any problem. The tanks are relatively small and though the required state of cleanliness is high the residues of the various products carried are generally easily dispersed. The most arduous task lies probably in the mopping up of the small pools of liquid lying on the tank bottom prior to the carriage of a high specification parcel.

Tank washing of the crude carrier with its heavy residues and waxes has become progressively more difficult as the size of ship has increased. With a standard deck crew it is difficult to operate continuously more than 10 portable washing machines at one time, so that many companies have now adopted in the larger vessels the practice of washing thoroughly only those cargo tanks into which clean ballast water will be loaded and then discharged at the loading port. The remaining cargo tanks are either lightly washed each voyage to avoid sediment accumulation or thorough washing is carried out in one third of the remaining tanks on each voyage.

The obvious solution to the problem posed by the larger ships is to install a fixed system with the machines fitted in the cargo tanks. One proposal recently developed makes use of high capacity washing units mounted at the deckhead. Each unit has a capacity of about 150 tons water/h and at a pressure of about 150 lb/in² will produce a jet with a throw of about 150 ft. The machine is rotated by an air motor and the variable angular vertical movement of the nozzle is adjusted to give the required cleaning pattern. The number of units required per tank depends upon the tank size and structural arrangement. In trials carried out recently in the centre tanks of a 64 000 tonner, tanks size 98 × 54 × 58 ft it was found that four high capacity deck mounted machines produced the required standard of cleanliness in two hours of washing time as compared to eight conventional machines taking a washing time of six hours. The water temperature in both cases was 58°F.

A separate tank cleaning pump has been provided for many years, but for complete fixed washing systems the high water throughput can best be provided by the cargo pumps. A design presently under consideration utilizes cargo pumps drawing from the secondary slop tank to provide both the washing water and the motive power for the eductors which are drawing from the tanks which are being cleaned.

The use of chemicals to assist in tank cleaning has developed rapidly over the past three years. The chemicals used are emulsifying agents which are injected into the washing water at a rate of about one gallon to three or four tons of water. The chemicals can be selected to give either an unstable emulsion, which will break down rapidly in the slop tank, or to give a stable emulsion, if the slops are to be pumped overboard.

Following a series of trials carried out by the authors' company the following broad conclusions were reached:

a) The use of emulsifying detergents can halve the time required to clean tanks to a standard of cleanliness at least equal to that obtained by conventional means.
b) Whilst most effective in emulsifying recent deposits of sediments in which the light ends of crude oil are still present, chemicals are less effective in dealing with sediments of long standing. In these cases the highest water temperature possible should be used.
c) Although good results can be obtained with water pressures of 120 lb/in², higher pressures give very much better and quicker results.

Cargo Tank Venting Systems

During the cargo loading operation, or when ballast water is being taken into the cargo tanks, the air or gas/air mixture in the tanks is vented to the atmosphere. When discharging from the cargo tanks, air or gas must be drawn freely into the tanks to replace the evacuated liquid. At other times, during the loaded or ballast passage, variations in temperature cause the gas or air to be expelled from or drawn into the cargo tanks. Under this random breathing condition the state of the tank atmosphere is controlled by pressure/vacuum valves fitted in the vent lines.

In the design of cargo tank ventilation systems there are four important requirements which must be fulfilled.

a) Undue pressure or vacuum conditions must not occur in the cargo tanks. The usual criteria taken are of 2 lb/in² pressure and ½ lb/in² vacuum.
b) The gas/air mixture, venting from the cargo tanks, should be dispersed to the atmosphere in such a manner that dangerous concentrations about the ship are avoided.
c) Where dissimilar cargoes are being carried, cross contamination through the venting system must not occur.
d) During the breathing condition gas evolution from the cargo oil must be restrained to avoid excessive loss of the lighter ends of the cargo.

Practice varies considerably in respect of the types of venting system employed, and considerations of safety in respect of the best dispersal of gas during loading and tank washing operations are of primary importance. In the authors' company, it has been the practice for some years to fit an inert flue gas system in all new crude oil carriers: in this system, the main vapour manifold is used to distribute washed boiler flue gas to the tanks, in which is maintained an atmosphere containing insufficient oxygen to support combustion. An alternative to the inert gas system favoured by some companies provides a very high throughput of ventilating air during tank washing to maintain the tank atmospheres below the lower explosive limit.

TABLE IV—INSTALLED CAPACITIES

	Year	Nominal deadweight, tons	Number of pumps	Capacity, tons/h		Discharge head			Rev/min	Approximate horsepower	Prime mover
				water	oil	ft	lb/in^2 water	lb/in^2 oil			
Crude oil carriers	1952	28 000	3—H	1080*	920	290*	125*	106	1500	400	Steam turbine
	1962	50 000	3—H	1750*	1530	290*	125*	106	1500	800	Steam turbine
	1965	100 000	2—V	1550	1860*	408*	205	150*	1775	1250	Electric motor,
			2—V	2200*	1250	357*	150*	150	1775	1250	3·3 kV
	1970	200 000	4—H	4700	4000*	460*	205	170*	960	2900	Steam turbine
Product carriers	1964	19 000	4—V	370	450*	370*	165	125	1750	290	Electric motor, 440V
	1968	22 500	4—V	990	750*	460*	205	155	1750	510	Steam turbine

H=Horizontal axis V=Vertical axis *Design condition

It is not within the scope of this paper to describe either of these systems in detail, or to discuss their relative merits. Suffice to say that, although the incidence of serious tank explosions is, mercifully, low, tanker owners today are vividly aware of the importance of designing for maximum protection from these dangers, and are working closely together to ensure that the highest possible standards shall be set and maintained by international agreement.

CARGO PUMPS—TYPES INSTALLED AND SYSTEM CAPACITY

Many types of pump have been used for handling tanker cargoes in recent years, including reciprocating, screw displacement, centrifugal and deep well designs, but that which is now most generally installed is the horizontal, or vertical, single stage centrifugal design. The reciprocating pump which was common in product tankers in the 1950s, has now been superseded by the centrifugal design, which has also retained its place as a satisfactory design in bulk oil tankers, even allowing for the greatly increased capacity that is now required for ships in the 200 000-500 000 dwt range. The largest pumps so far considered by the authors' company have a capacity on crude oil of 4700 m^3/h at a head of 140 m. Four such pumps are to be installed in one vessel; this is believed to be the largest capacity cargo pumping installation in any vessel to date, requiring no less than 12 000 hp to develop the rated output. It is believed that a reduced number of larger pumps has been installed in some cases.

The increase in installed pump capacity in product and crude oil tankers which has taken place in recent years is shown in Table IV. All these pumps are of single stage centrifugal design, with either single or double entry impellers. In the majority of cases horizontal pump designs have been used, associated with steam turbine power units and steam turbine vessels. The recent development of mechanically, satisfactory vertical turbines of the required power has enabled vertical units to be installed in a number of recent motorships (not shown in the table), and there are no sound reasons why vertical pumps should not also be used in steamships, with a potential saving of space, particularly in the length of the engine room. At the same time an operational advantage is obtained by siting the turbines in improved environmental conditions, and in improved access to the units for maintenance. The large vertical units were adopted by the authors' company some years ago when it was decided to install some ships of a class of 64 000 dwt tankers with steam, and some ships with Diesel, main engines. The way in which this was achieved is shown in Fig. 4, utilizing the same engine room dimensions by constructing a flat within the pump room beneath the settling tanks on which the cargo pump turbines were installed. The drive between the turbine and pump is transmitted by a vertical universally jointed and splined shaft, connected to a stub shaft, passing through a gas and watertight deck head seal. It is considered important for the pump and motor axes to be offset so as to ensure satisfactory functioning of the flexible drive.

Steam turbine power units have been used for many

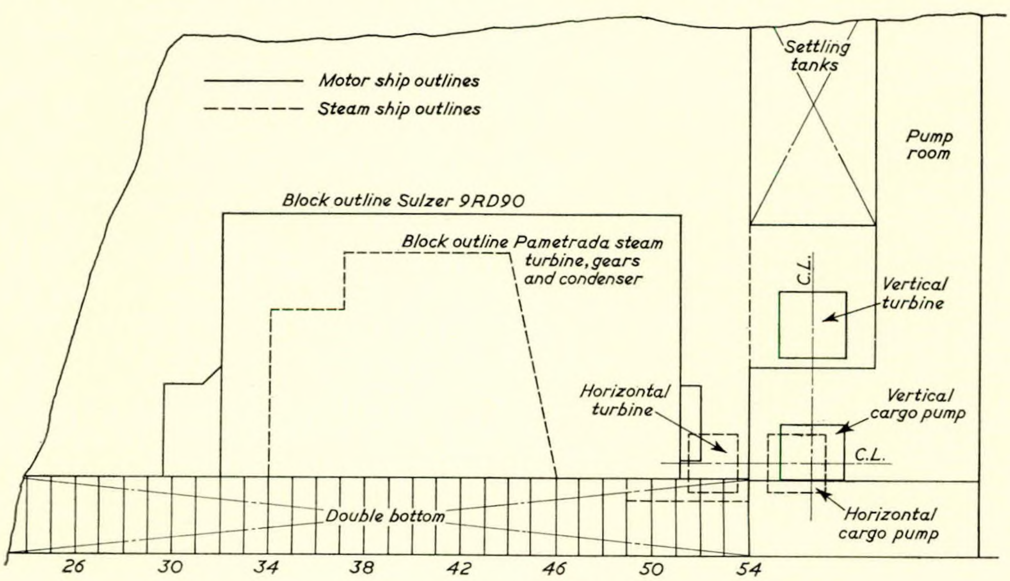


FIG. 4—*Comparison of steam and Diesel installations*

years almost without exception and the design of two classes of vessel with widely differing capabilities using electric cargo pumps owed something to the satisfactory experience obtained some years previously with the turbo-electric T2 tankers. In those ships the cargo pumps were supplied with power from the main propulsion generators, and some measure of speed control was possible. In the vessels now considered, the cargo pumps are driven at constant speed, and at the same time the cargo pumps are also used for cargo tank stripping duties. A steam reciprocating stripping pump is however also installed, and is used principally for line cleaning and draining.

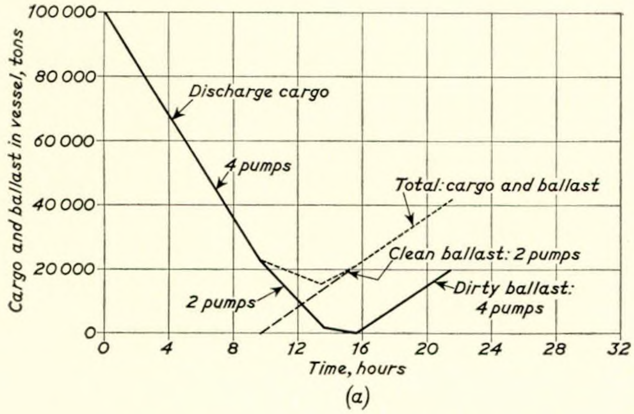


FIG. 5(a)—*100 000 dwt vessels—Proposed discharge procedure before modification*

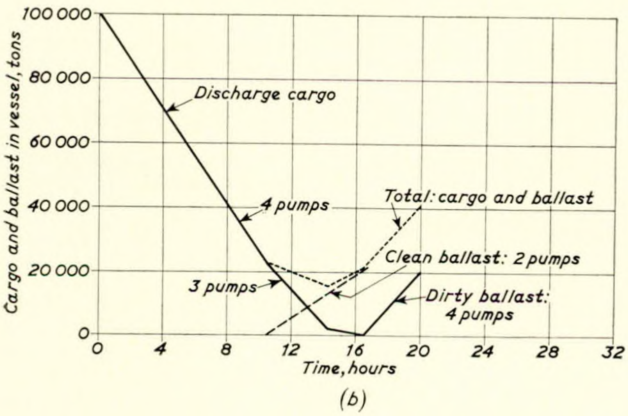


FIG. 5(b)—*100 000 dwt vessels—Proposed discharge procedure after modification to pumps*

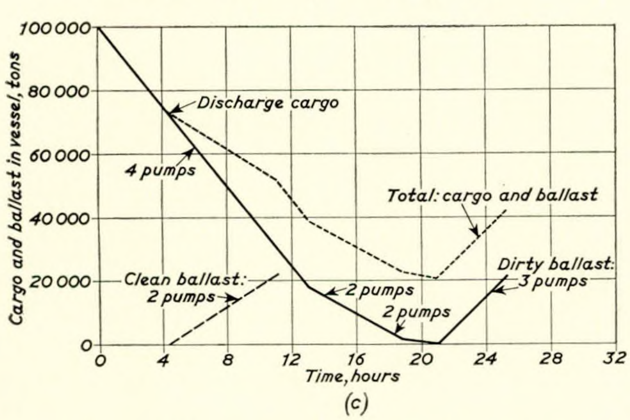


FIG. 5(c)—*100 000 dwt vessels—Actual results of incorrect procedure after modification to pumps*

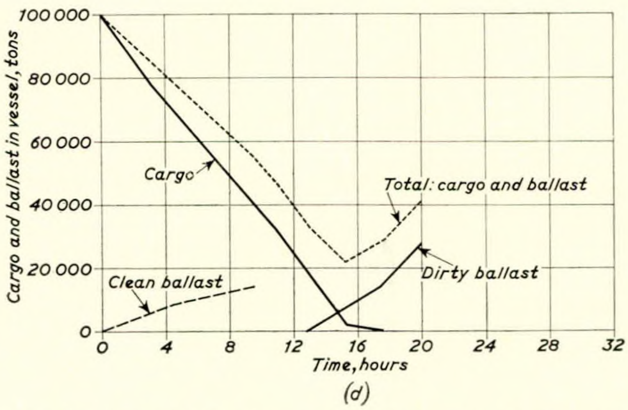


FIG. 5(d)—*100 000 dwt vessels—Proposed discharge procedure after pump and tank modifications*

These ships of 19 000 and 100 000 dwt capacity have Diesel and steam turbine main engines respectively, and in the latter class, electric power is generated at 3300 volts. Initially, in both cases, the pumps and cargo handling installation were designed around a water capacity. At a later date the rating basis for the pumps was changed to the cargo that they were expected to carry most frequently, and primary consideration was given to obtaining a good cargo discharge rate at the expense of ballasting and deballasting rates, since it became necessary when handling sea water to throttle the pump discharge to prevent overloading of the driving motors. This criterion was later reviewed in the light of operating experience, and modifications were made to the installations or pumping procedures adopted to increase the rate at which sea water could be handled. In the case of the 100 000 dwt vessels, two out of the four cargo pumps installed in each vessel were altered to improve their performance as ballast pumps, which, although reducing the rate at which cargo could be handled, enabled the overall time that the vessel was alongside the discharge jetty to be decreased by several hours. This effect is shown in Figs 5a and 5b. The effects of maloperation of the system are clearly shown in Fig. 5c, based on actual discharge records, when it is seen that the overall discharge time has been extended several hours beyond the design figure.

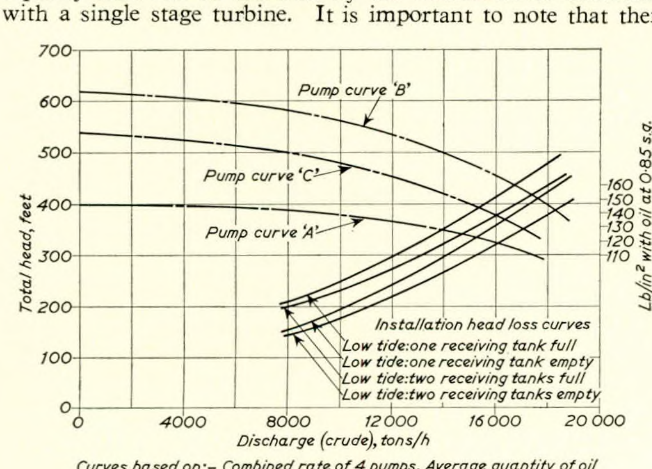
In this case the clean ballast was admitted too early and the heavy stern trim which should have developed was much reduced, a factor which contributed to a lower average rate of bulk discharge, and substantial discharge using two or three pumps only, much of this being taken out through the ring main.

A further modification was later made to the vessels following the 1966 load line revision, when certain clean ballast tanks were converted to cargo duty. By adopting a procedure (see Fig. 5d), in which cargo is discharged and dirty ballast loaded simultaneously, the minimum deadweight of the vessel whilst alongside the jetty was increased, and at the same time the overall discharge time was not significantly extended. The principal advantage of this system, however, arises from the loading operation where significant time saving can be achieved whilst at the same time maintaining sufficient displacement for security at the berth. The special provisions required in the cargo system to prevent accidental discharge of oil to the sea has been discussed in an earlier section of the paper.

In the 19 000 dwt vessels, the ballast performance has been improved by throttling the flow at the pump inlet, using the pump isolating valve. Throttling in this way creates a condition of cavitation within the pump, and it has been found that the ballast rate has been doubled without overloading the driving motors. Subsequent inspections of the pump and discharge piping have not revealed any significant damage to the pumps as a result of this unorthodox procedure. As a means of increasing the rate at which ballast can be handled both into and out of the vessel, inlet valve control has proved to be most successful, although sensitive to suction conditions limited by a basically unstable power characteristic in which any increase in back pressure causes a rapid increase in the power input to the pump. As a result, the system is considered suitable for manual operation only, and when pumping ashore to a ballast cleaning plant it is sometimes necessary to revert to the normal discharge valve control to avoid overloading the motors.

Screw displacement pumps have been studied with interest, with a view to installation in product tankers, where they have a considerable attraction based on their inherent self priming and stripping ability. However, the progressive increases in pressure and capacity required in pumps for this class of vessel are such that, if continued, it is likely that they will preclude the use of this pump type.

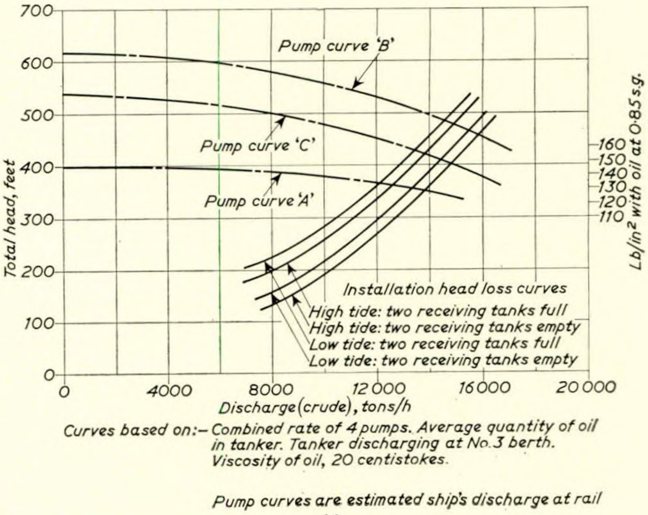
There is no single universal formula which can be used to establish the installed cargo pumping capacity appropriate to any particular size or type of vessel. From an analysis of tankers in service, it will be seen that the installed capacity is frequently about 10 per cent of the nominal deadweight of the vessel, which can be expected to give approximately the same discharging time for each vessel, regardless of size. New developments in shore installations, coupled with ship designs that are frequently to the maximum dimensions that can be admitted to the berth, now make it necessary to design the pumping installation to match the terminal characteristics closely, besides giving detailed consideration to the economics of ship operation. Close co-operation between shipowner and terminal is now essential in determining the technical and economic factors in the installation design. In some installations it may be that ship movements can only be undertaken within a few hours of high water, and in this case the normal tidal cycle will play a large part in the selection of the system capacity and number of units installed. At other terminals the state of tide is only important at loaded draughts, and the installation can be designed independently of tidal considerations. The curves shown in Figs 6a and 6b show the expected performance of 200 000 dwt vessels now building when discharging at two installations, and with alternative pump capacities. Pump A is the original proposal submitted by the shipbuilder. Pump B is based upon the maximum steam availability from the installed boiler plant, and necessitates the adoption of a two stage turbine design. The incremental cost of this design is substantial and cannot be justified. Pump C has the largest capacity that can be achieved by the manufacturer concerned with a single stage turbine. It is important to note that there



(a)

Installation A: Estimated crude offloading rates with 213 000 dwt vessels—Pump curves taken from builder's specifications

FIG. 6(a)—*Ship and shore discharging characteristics*



(b)

Installation B: Estimated crude offloading rates with 213 000 dwt vessels—Pump curves taken from builder's specifications

FIG. 6(b)—*Ship and shore discharging characteristics*

is no significant "step" in the pump cost figures, which only increase slightly within the same pump frame size.

AUTOMATIC STRIPPING SYSTEMS

Commercial considerations dictate the need for high ship utilization, rapid discharge of cargo and turnround of the vessel, features which, when coupled with a continuing shortage of labour, have led to the development and installation of semi-automatic discharge systems. In these the pump output, pump priming, air and gas removal, and change from bulk discharge to stripping duties is carried out automatically in accordance with the system design. Several different systems, out of the many that have been commercially proposed, have been evaluated by the authors' company.

It may generally be stated that in each case the output of the cargo pump running at constant speed is controlled by a discharge valve, while air and gas is removed from a vessel adjacent to the pump inlet by a water ring vacuum pump. In product carriers, completely independent priming systems have been adopted to prevent any possibility of cargo contamination due to equipment mal-function, whereas in bulk tankers, one priming pump has been used to serve two or more cargo pumps.

One such stripping system that has been satisfactorily employed is illustrated in Fig. 7. In this system the cargo pump draws the cargo through a separator vessel fitted adjacent to the pump inlet which is so designed that any air or gas entrained with the liquid cargo separates out and collects at the top of the vessel. The liquid/gas interface is continuously measured by a level controller, which is used to control the pump discharge control valve and also to start and stop the vacuum pump acting as a gas exhauster. The same air signal from the level controller operates the gas valve at the moment of starting the vacuum pump. The interceptor vessel between the separator and vacuum pump serves to separate any liquid droplets that may be carried up with the gas, and these are returned to the separator vessel through a non return valve. The system illustrated is for a pump with electric motor drive and load control operated from the phase current of the driving motor. The air signals from both the level controller, and the motor load controller are preferentially selected by the selector relays, Relay 1 and Relay 2, which also receive an input from the manual loader at the control station. In this way, any system tending to close the discharge control valve will be preferentially selected, and the control valve will respond to the selected signal.

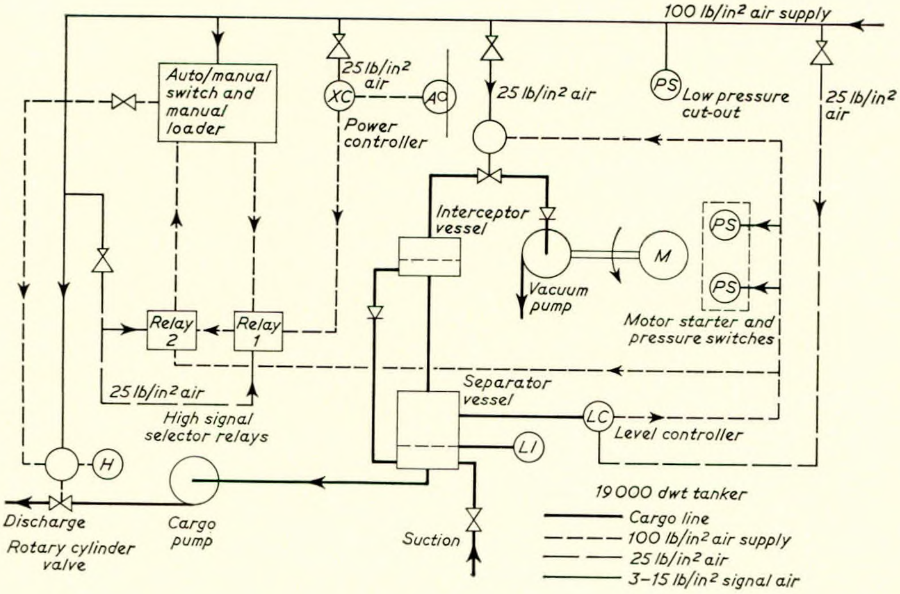


FIG. 7—*Cargo discharge control system*

The other systems evaluated have included a rather similar system in which two fixed position level switches replaced the level controller, and another in which an attempt was made to use the pressure at the pump inlet as a controlled variable, and to set the control point to a value which prevented the pump from losing suction. In this latter case however, although the equipment could be made reasonably satisfactory, difficulty was encountered in selecting the correct set point for the controller, too high a setting resulting in a large amount of cargo discharged at the reduced stripping rate, and too low a setting causing a loss of suction which would not be regained without manual intervention.

These automatic control systems have been used with both constant speed and variable speed pumps with some degree of success in both product and bulk oil vessels, and in installations ranging from 290 to 1250 hp. In the modern bulk oil tanker, with few tanks, short suction piping, good drainage and little stripping, it is now considered that this type of system is an unnecessary complication. For a product carrier, however, with a large number of relatively small tanks, and with the simultaneous discharge of multi-parcel cargoes, the ability to complete the discharge semi-automatically is attractive, and reduces the detailed attention to tank level otherwise necessary when discharging cargo.

CONTROL VALVE DESIGN

In all the semi-automatic systems described, the most important elements are the control valve and control means. Several different types of valve have been installed, not all of which have been satisfactory. The perfect valve for this purpose awaits discovery, but top quality butterfly valves can be expected to give satisfactory service. The conditions under which these valves are required to operate are very severe and, apart from having a satisfactory control characteristic, the valves require to be extremely robust to withstand the severe mechanical forces imposed. In the case of the largest pumps to which this type of semi-automatic system has been fitted, there is a power dissipation of over 600 hp across and down stream of the valve when stripping the cargo tanks and handling water ballast. The pressure difference across the valve may be in excess of 400 ft, with theoretical liquid velocities of about 160 ft/s. These conditions are illustrated in Fig. 8. The high liquid velocity and large energy dissipation can set up severe vibration in the control valve and discharge piping, with resultant mechanical damage to components of the piping system, and in particular the rather delicate pneumatic instrumentation, particularly valve positioners. In some cases it has been necessary to adopt a remote mounting for the valve positioner to overcome this problem. Even so, the connexion between valve and positioner remains a source of weakness.

The large pump sizes now under consideration combined with the high discharge pressures required together with the vibration experienced in the higher powered installations under throttled conditions lead the authors to consider that speed control of the cargo pumps is essential if they are to be operated satisfactorily over a wide range of output and against a wide range of back pressures. The pump speed should be automatically adjusted to maintain a constant discharge pressure, combined with a control related to the flow to the pump so that a smooth change from bulk cargo to stripping conditions can take place.

CARGO PUMP MOTORS

Previous reference to both steam turbine and electric motors as the power units for cargo pumps illustrates that

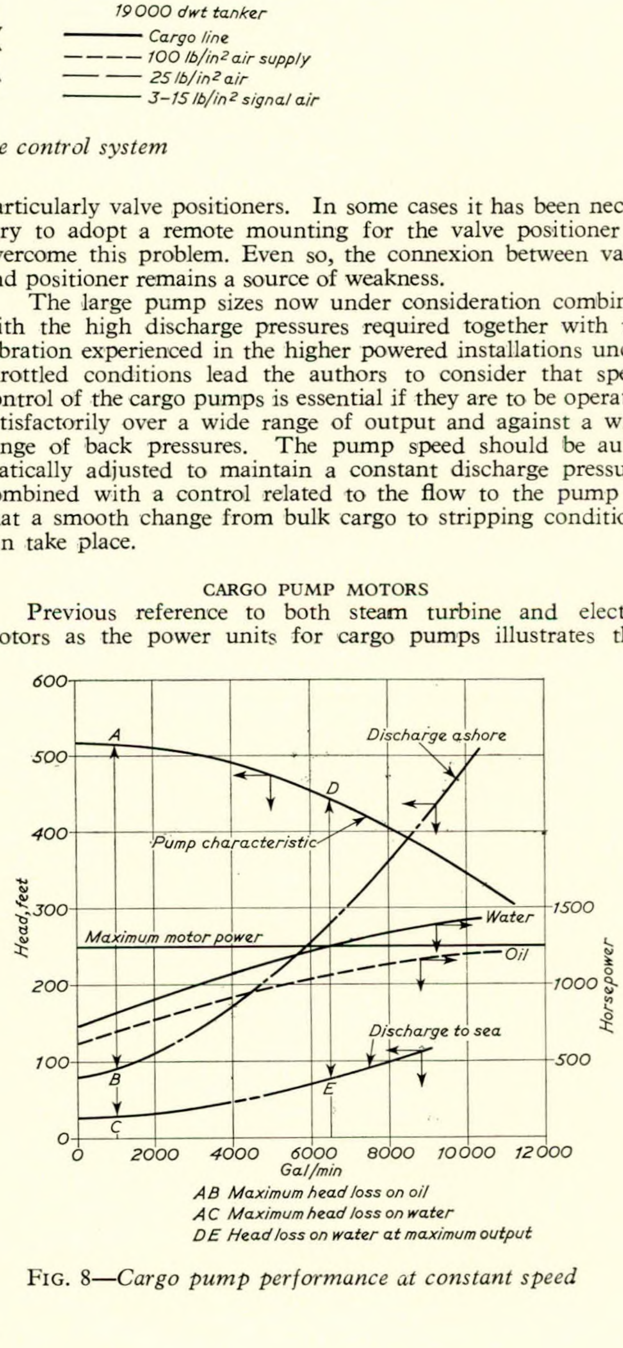
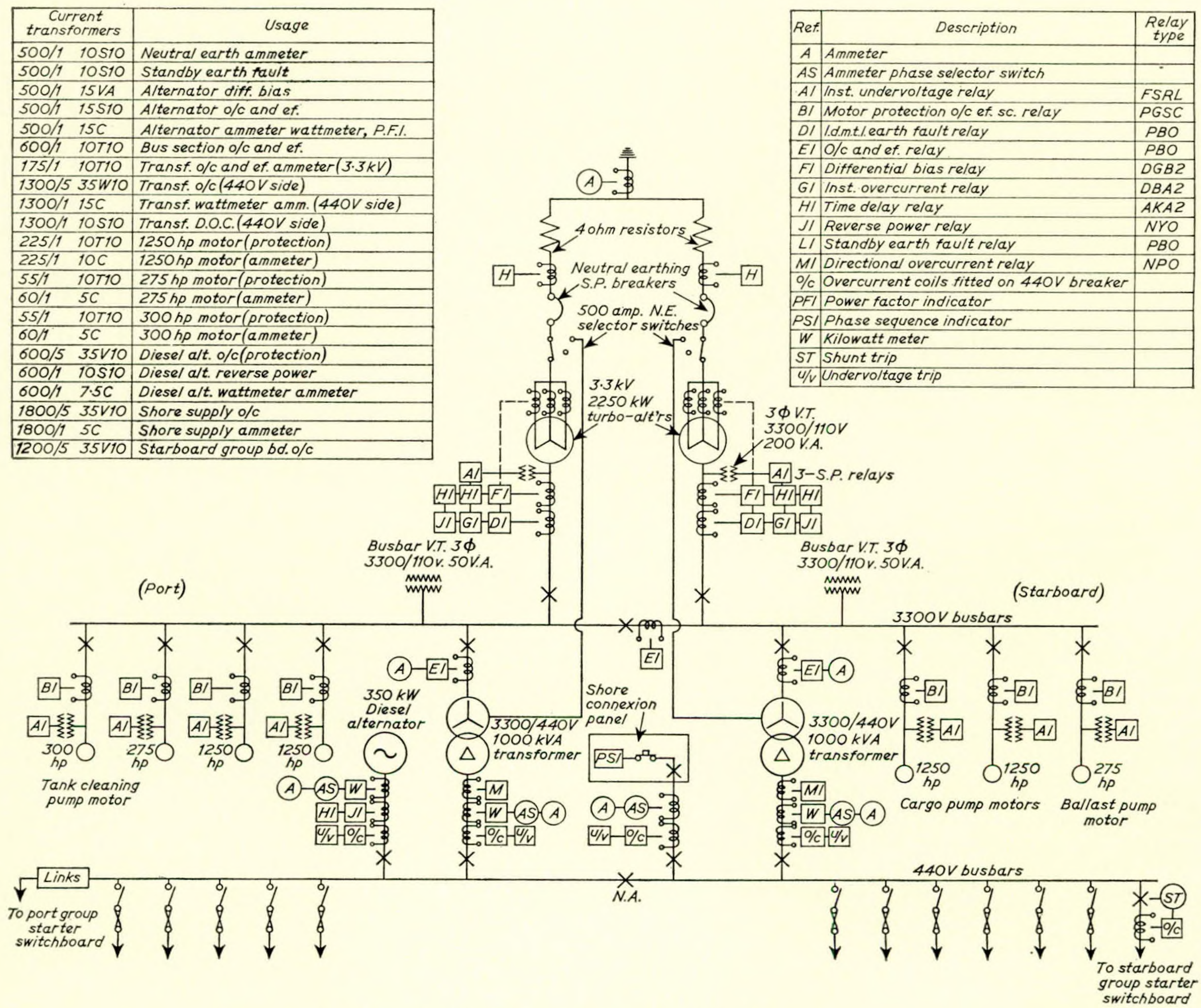


FIG. 8—*Cargo pump performance at constant speed*

TABLE V—CARGO PUMP MOTOR CHARACTERISTICS

Motor type	Steam turbine	Electric motor	Diesel engine	Gas turbine	Hydraulic motor
Drive axis	Horizontal or vertical	Vertical	Horizontal	Horizontal	Horizontal or vertical
Maximum power suitable	300-6000 hp	100-1500 hp	100-500 hp	6000 hp	100-500 hp
Output speed	1800 rev/min, variable	1800-1200 rev/min, fixed	1200 rev/min, variable	1800 rev/min, variable	720 rev/min, variable
Power availability in installation	Steam usually available as it is required for other purposes, cargo heating, tank cleaning etc	Generating capacity must be installed specifically for cargo handling duties	Diesel engine can also be used to drive an electric generator alternatively to the cargo pump	Can be used to drive an electric generator as an alternative to the cargo pump	Is an intermediate step only in the power transmission chain
Energy source	Heavy fuel oil	Heavy fuel oil	Class B marine fuel oil	Distillate fuel	Heavy fuel oil or Class B marine fuel oil
Typical cost order	4	3	5	1	2
Typical efficiency order	4	2	1	5	3
Recommended application	Crude oil and product carriers	Product carriers	Product carriers	Crude oil and product carriers	Product carriers



Current transformers	Usage
500/1 10S10	Neutral earth ammeter
500/1 10S10	Standby earth fault
500/1 15VA	Alternator diff. bias
500/1 15S10	Alternator o/c and ef.
500/1 15C	Alternator ammeter wattmeter, P.F.I.
600/1 10T10	Bus section o/c and ef.
175/1 10T10	Transf. o/c and ef. ammeter (3·3 kV)
1300/5 35W10	Transf. o/c (440 V side)
1300/1 15C	Transf. wattmeter amm. (440V side)
1300/1 10S10	Transf. D.O.C. (440V side)
225/1 10T10	1250 hp motor (protection)
225/1 10C	1250 hp motor (ammeter)
55/1 10T10	275 hp motor (protection)
60/1 5C	275 hp motor (ammeter)
55/1 10T10	300 hp motor (protection)
60/1 5C	300 hp motor (ammeter)
600/5 35V10	Diesel alt. o/c (protection)
600/1 10S10	Diesel alt. reverse power
600/1 7·5C	Diesel alt. wattmeter ammeter
1800/5 35V10	Shore supply o/c
1800/1 5C	Shore supply ammeter
1200/5 35V10	Starboard group bd. o/c

Ref.	Description	Relay type
A	Ammeter	
AS	Ammeter phase selector switch	
AI	Inst. undervoltage relay	FSRL
BI	Motor protection o/c ef. sc. relay	PGSC
DI	I.d.m.t.l. earth fault relay	PBO
EI	O/c and ef. relay	PBO
FI	Differential bias relay	DGB2
GI	Inst. overcurrent relay	DBA2
HI	Time delay relay	AKA2
JI	Reverse power relay	NYO
LI	Standby earth fault relay	PBO
MI	Directional overcurrent relay	NPO
o/c	Overcurrent coils fitted on 440V breaker	
PFI	Power factor indicator	
PSI	Phase sequence indicator	
W	Kilowatt meter	
ST	Shunt trip	
u/v	Undervoltage trip	

FIG. 9—*Key diagram of main system; 100 000 dwt vessels*

there can be no universal choice in the selection of power units for cargo pump drives. Each type of unit has its own particular merits, and in selecting the type to be installed, due consideration should be given to all mechanical and economic factors concerned with the anticipated duty cycle. Other types of motor should also be considered, including Diesel engines, gas turbines and hydraulic motors. The principal features of these alternatives are given in Table V.

In installations where the pumps are to be used frequently, such as the short haul product carrier, electric motor or direct Diesel engine drive is most attractive, combining mechanical simplicity, reliability, low operating costs and simple automation. These factors will in most cases more than offset the additional cost that these installations are likely to incur above the cheapest installation. Direct coupling of the Diesel engine to the cargo pumps, however, has one important advantage, in that the engine can alternatively be used to drive an electric generator and installations of this type can lead to both low cost, and high efficiency. It is however probable that no more than two pumps can be driven this way in any one installation, and remaining pumps must be driven by steam turbine, electric motor or other means.

ELECTRICAL DISTRIBUTION SYSTEMS AND PROTECTION

The outline distribution system in one of the vessels to which reference has been made in this paper is shown in Fig. 9. Electric power is generated at 60 cycles, 3·3 kV by two 2250 kW turbo-alternators. Each machine supplies separate halves of the main high voltage switchboard connected by a synchronizing bus-coupler circuit breaker. The "conventional" medium voltage installation is fed by two 3·3 kV/440 V three phase transformers. The 440 V switchboard is also split—the two halves of this board are also connected by a synchronizing bus-coupler. The electrical load in this installation is shown in Table VI. It can be already seen that the two main generators are near their maximum capacity on cargo discharge. A preference trip circuit is therefore provided to shed the whole of the cargo pump load should either generator become overloaded. After a short time delay, the generator itself will trip should the overload condition persist. In practice the installation has operated very satisfactorily, the only complication being the necessary provision of primary and secondary injection test facilities to prove the circuit breaker safety trips. These tests are carried out annually, together with a pressure test on the 3·3 kV system as a whole.

TABLE VI—ELECTRICAL LOAD WHEN USING CARGO PUMPS
100 000 dwt vessels: installed capacity 4500 kW

	Electrical load, kW	
	Discharge	Ballast
Base load	800	800
2 inboard pumps	1840	1840
2 outboard pumps	1600	1840
Total	4240	4480

Fig. 10 shows the much simpler system adopted in the product carriers, where all generation and main distribution is carried out at 440 V. In these installations main circuit protection is provided by conventional metal clad circuit breakers. Two of the main cargo pumps are supplied through auto-transformer starters whilst the other two are started direct to line. The auto-transformer starters are necessary because of the large starting load imposed by the 290 hp motors on the 1100 kW combined generating capacity. It also provides the facility of being able to run one cargo pump at sea when only one turbo-generator is in use. As in the case of the 3·3 kV installations described, a preference tripping system operated from contact wattmeters is provided to shed the cargo pump load automatically in the event of alternator overload. Alternator overload will not arise under normal cargo handling conditions, but could arise in a situation where an alternator is disconnected from the board when all pumps are in use. Tests carried out on the complete control systems have shown that no hazard exists to any part of the machinery installation should such high load trips occur.

THE FUTURE

The increasing size of crude oil carriers and the increased complexity of the smaller product carrier will continue to lead to a divergence of design principles and the specialized design of each type of vessel. Increasing recognition will be made of the necessity of carefully tailoring the vessels' capabilities to the anticipated trade pattern, and for the overall transport problem to be considered in conjunction with the shore installation authorities and pipeline engineers.

The steadily increasing size of crude oil ships will demand the adoption of cargo systems capable of simultaneous handling oil and water ballast without danger of intermixing and the accidental discharge of oil cargo overboard. The stricter rules on oil pollution of the sea now in force will lead to development of improved tank cleaning and oil separation procedures.

The product carrier will continue to demand a very high standard of cargo integrity as product specifications are made more numerous and rigorous, and products are loaded closer and closer to the limiting specification tolerance. The ability to load and handle multi-parcel cargoes simultaneously without contamination is an essential feature of all new tonnage and

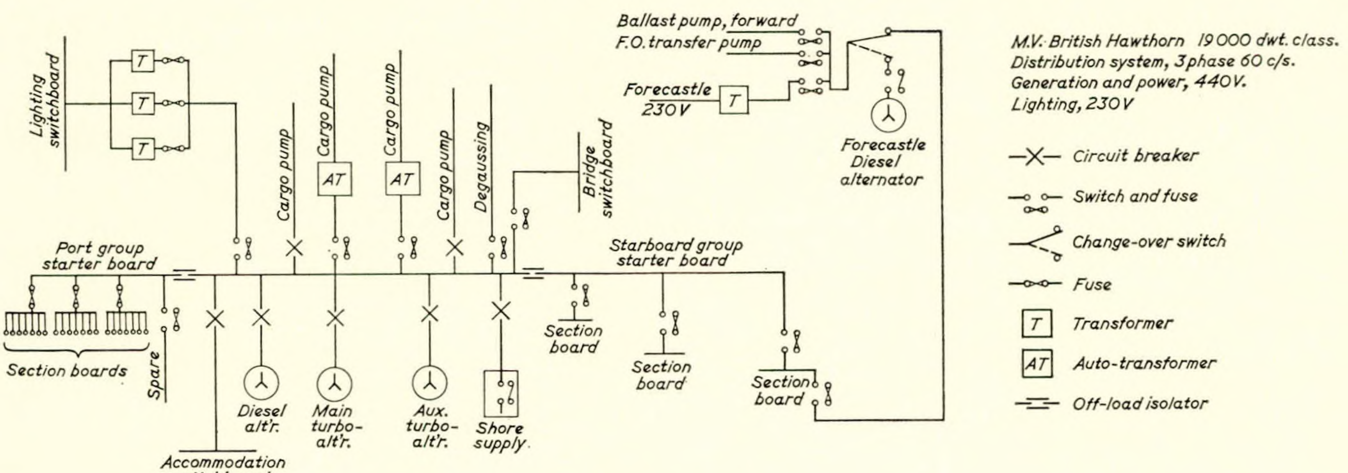


FIG. 10—*Outline distribution system*

the flexibility that is now demanded in this respect can be expected to increase.

In both categories of vessel, the progressive reduction in manning and increasing operational efficiency will lead to develoments in automation and power operation of the cargo system. The introduction of computer controlled discharge and loading systems can be expected within the next decade.

Computer programming of ships is already being developed and may shortly be a common management tool. Increasing ships operational reliability both at sea and in port are essential to the success of such a system, which will lead to more economic use of ships, of storage facilities, and reduction in the quantity of cargo in transit. This facility is likely to place increasing demands on the flexibility with which cargo can be segregated, and the manner of its loading and discharge.

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Discussion

MR. R. SCOTWICK (Member) said that, although the authors thought it preferable to have all tanks of equal capacity with one centre tank equivalent to two adjacent wing tanks, this was not necessarily correct. Mr. Scotwick's company had recently completed two 18 000 dwt vessels, and the particular trade requirements dictated that the sections of three tanks across were of differing lengths.

It appeared that any tank configuration required to be tailored around an owner's experience and requirements; what was satisfactory for one owner was not necessarily correct for another.

On cargo piping the two vessels which he had already mentioned had five sets of three tanks across, and required maximum loading/discharging flexibility, but also it was requested that the number of valves in the cargo tanks be reduced to a minimum. The vessels were to be employed mainly carrying refined products on coastal hauls and, in the event of any repairs being required to cargo valves, the additional time for gas-freeing would be unacceptable.

It was estimated that flexibility required seven suction lines but only twelve valves in the cargo tanks, a further 31 valves in the pump room being required to allow any tank to be directly connected to any one of the four cargo pumps. The cost for cargo lines plus valves was lower than that for a ring main system similar to that in the paper for the 19 000 dwt general product carrier, and had the majority of cargo valves in a preferable environment.

For these reasons it was sometimes worth while considering increasing the number of cargo suction lines to a figure above that accepted in practice.

Mr. Scotwick said that for large crude carriers the authors had described the "free flow" and "conventional piping" systems.

A compromise system which appeared more advantageous was twin ducts with sluice valves, affording the advantages of a free flow system and some of the flexibility of a conventional piping system. What were the authors' comments on such a proposal?

The authors had mentioned the success they had found in using high capacity tank cleaning machines. Although these units had advantages over the conventional portable machines, it was not always possible to use the larger units to full advantage due to stripping limitations.

Four large capacity machines in one tank would discharge 600 tons/h of washing water. Mr. Scotwick believed that such a quantity was beyond the capabilities of a normal stripping system and, in general, main suction strums had too great a bottom clearance to allow the main cargo lines to be used. For these reasons, it was often the stripping capacity that governed the time required for tank cleaning.

Referring to one possible aspect of the systems described not covered in the paper, it was possible, in theory, that, due to dissolved air in cargo, the pumping capacity might, stressing the word "might", be reduced by anything up to 10 per cent for typical crudes, dependent on suction conditions. Had the authors any experience of increased bulk discharge rates in vessels using a gas extraction type automatic stripping system?

At present, the size of the biggest crude carrier appeared to be limited to around 200 000–300 000 dwt for economic reasons. Should the economic picture change, there might be a case for building vessels of, say, three times that size. Had the authors, in their studies of cargo systems, hazarded a guess as to what sort of cargo system would be fitted in such vessels?

CDR. C. M. HALL, R.N. (Member) took the authors to task for their very minor reference to screw displacement pumps, although they had appreciated that these pumps showed considerable attraction due to their inherent self-priming and stripping ability. The type of pump required was the external bearing or geared screw displacement pump, where there was no contact between the screws, or between the screws and the casing. In Table IV—and confining his remarks to product carriers—the quoted capacity of water and oil tons/h respectively were given for 1964 as 370 and 450, and for 1968 as 990 and 750. These outputs were by no means outstripping the capacity of screw pumps, which were already manufactured and in service at 1250 tons/h.

He then showed a series of slides and drew attention to the merits of screw pumps for stripping duties. With particular reference to Fig. 7, he pointed out the amount of ancillary equipment that could be dispensed with if a screw pump were substituted for a centrifugal.

CDR. E. H. W. PLATT, M.B.E., R.N. (Member) pointed out that, twice in the paper very proper reference was made to the importance of close co-ordination between those responsible for the design of the ship and those responsible for the design of the terminals, in arriving at the proper cargo pumping installation, and as ships grew bigger, this became more important. A series of curves was given in Figs 6a and 6b, pump A was the original proposal submitted by the shipbuilder; pump B was based upon the maximum steam availability from the installed boiler plant, and necessitated the adoption of a two stage turbine design; pump C had the largest capacity that could be achieved by the manufacturer concerned with a single stage turbine. This selection was presumably the optimum economic solution to meet, within reason, the terminal requirements and, clearly, the choice of the size of this equipment became a very important management decision in the design of a ship of this size. In fact, the cargo pumping equipment could be described as the main armament of the tanker. Could the authors give more information about the decision-taking steps in arriving at the correct size of a cargo pump?

The power required for cargo pumping in these large tankers was enormous. This could be seen from Table IV. For the 200 000 ton ships, nearly 3000 hp per pump, i.e. 12 000 hp total, was required to give this pumping capacity in a ship with a propulsive power of 30 000 hp, thus using a high proportion of the ship's boiler capacity. It made one wonder about a motor vessel of this size. In the steam turbine vessel, this boiler capacity was available with a margin to give 12 000 hp for pumping. In the motor vessel it would be necessary to provide this as auxiliary boiler capacity, or to drive the pumps with some independent type of prime mover, or, as an acceptable alternative, perhaps to reduce the pumping capacity, and arrive at a suitable compromise between a maximum rate which might only be used occasionally and a slower rate which would be satisfactory at a majority of terminals.

To illustrate further the extent, complication and importance of this management decision which might have to be made, in the case where one had power other than steam driving a very big tanker, it might be prudent to take the time which might be lost in the life of the ship through reduced pumping capacity against the added cost by making a comparison on a discounted cash flow basis. Would the authors comment on this? What was the time saving estimated in the case of this 200 000 tonner for the pumping plant proposed by the shipbuilder *vis-à-vis* that finally selected?

Reference had been made to the electric drive for the pumps in the 100 000 ton ships and the fact that, with the quite high powers still required, it was considered desirable to generate at 3·3 kV for the cargo pump motors. Could the authors comment on the general reliability with this very high voltage electrical installation and say whether there had been any problems in the selection and training of personnel to operate and maintain it?

MR. G. VICTORY (Member) was sure that the authors would agree that the equipment for handling cargo, and the auxiliaries put on board tankers very often dictated the operational characteristics of the ship. Emphasis had been put on the penalties which might be incurred if any admixture of products should occur outside the specification contained in the loading charter and yet single valve separation appeared to be acceptable to the authors, who said it was extensively used in product carriers. If this was so, he could understand the importance of their other statement that the tightness of the valve must be a prime consideration, but did the saving justify the risk?

The new 200 000 ton crude carriers had ballast tanks amidships, which would probably require to be pumped out whilst the vessel was loading in the cargo tanks. Did the authors also suggest that single valve separation was satisfactory here? Mr. Victory suggested that such a practice could result in fairly extensive oil pollution and wondered if the authors had considered the penalty attached to oil pollution when considering single valve separation. They said later in the paper that stricter oil pollution rules would improve separation and he asked whether they had any improvements in sight in the near future.

The authors had referred to the use of an emulsifying agent to form a stable emulsion when cleaning tanks if the slops were to be pumped overboard. With stricter oil pollution rules now coming into force, this was a rather risky procedure.

The discussion of gas venting had been minimized in this paper. The authors were considering a discharge rate of over 16 000 tons/h, so must expect a somewhat similar rate of loading which would disperse a volume of vapour much greater than that of the oil, because of the boiling vaporizing effects when the oil was agitated. A volume equivalent to possibly $1\frac{1}{4}$ million cubic feet of flammable or explosive gas would be displaced through the vapour vent pipes. Did the authors consider that individual vent pipes, taken possibly at 7 to 9 ft above the deck, with normal PV valves, was a suitable method of dealing with this quantity of vapour? Did they consider that a vessel with individual stub pipes was a satisfactory transporter of spiked crude or would they put any restriction on the size of a vessel carrying spiked crude, particularly as in the paper they had said: "The gas/air mixture, venting from the cargo tanks, should be dispersed to the atmosphere in such a manner that dangerous concentrations about the ship are avoided".

As regards risks of fire and explosion in tankers, reports had just been received of an explosion in Buenos Aires; a tanker had blown up whilst loading petroleum, involving two other tankers in a chain reaction. It might be, as the authors said, that the incidence of serious explosions in tanks was mercifully low, but possibly the devastating results of such explosions had not been given sufficient credence. For those not in the industry, when talking of fires and explosions in tankers, the term "mercifully low" might be replaced by "with regrettable regularity".

CAPT. R. W. MORETON asked what exercise had been worked out for justifying the coating protection of product carriers. Whilst he agreed with the points raised, he wondered how management could be made to think on the same lines.

He believed that the 100 000 tonners of the authors' fleet had centralized cargo systems. Could they provide an economic justification to management for these systems? What were their accuracy and dependability? From the experience gained, was it intended to introduce them to the 200 000 tonners, in sophisticated form, and to the product carriers? If they were being introduced on the product carriers, what degree of accuracy was being looked for, particularly with such a wide range of grades handled and the great number of tanks involved?

Finally, 35 per cent had been talked about as the ballast quantity that the authors looked for. His own company would agree that 30 per cent operationally at sea under normal weather conditions was acceptable. Otherwise it was reasonable to look for something less, within harbour-berth areas, probably in the order of 20 per cent. They had carried out a number of tests to prove this point, and it was considered safe and practicable for a tanker to manoeuvre safely under normal weather conditions with 20 per cent of her deadweight as ballast.

MR. O. M. CLEMMETSEN quoted from the paper, under "The General Products Carrier": "Lighter forms of construction such as the use of corrugated bulkheads may be used with less risk of premature leakage". Was it the authors' experience that normal uncorrugated flat bulkheads with stiffeners had given more leakage than corrugated bulkheads? On smaller ships corrugated bulkheads often gave the most trouble because of the difficulty of avoiding hard spots where the horizontal girders and supporting brackets landed on the bulkheads.

Fig. 3 showing the 200 000 ton crude carrier had only four sets of cargo tanks. Usually ships of this size had five sets. Had the authors any difficulty, with this design, in obtaining satisfactory trims in ballast condition, and had they approached the maximum permissible stresses in ballast due to large shear forces in way of the ends of the tanks?

Under "Discharging", it said: "During the discharging operation it is necessary first to discharge those cargo tanks which are to receive ballast water". In the free flow system which was described later, it would appear that this could not be done, unless the tanks which were to receive the ballast water were always the forward tanks. Was that in fact the case? Water could not be put into the forward tanks under a free flow system until a late stage in the discharging operation had been reached.

Also on the same page it stated: "The provision of good drainage from the horizontal surfaces and across the bottoms of the cargo tanks is amply repaid in time saved when stripping cargo and tank cleaning. Generous cut-outs should be arranged where longitudinal stiffeners pass through transverses and large drain holes are provided through the bottom longitudinal stiffening". This problem had always been a bone of contention between the ship designer and the engineer. The ship designer wanted to keep these slots to a minimum and the engineer to a maximum. This matter was particularly important now in the case of the large tankers where the longitudinals were carrying very large loads which had to be transmitted to the webs of the transverses. There had been cases of buckling of the plating around the cut-outs. It was preferable to arrange circular drainage holes between the longitudinals and to keep the slots at the longitudinals to a minimum.

The paper mentioned the short life philosophy. It should be emphasized that this was short life in terms of the ship beginning life with full scantlings, without reduction for corrosion control, and it did not refer to the fact that some proposals had occasionally been made to give scantling reductions without the corrosion control.

With the very high pressure tank cleaning machines, had the authors had any experience of damage to paint due to the impact of the jets on the coated surfaces? He noted that the use of hot water was advocated in certain circumstances but presumably only with coated surfaces, since it had long been considered that the use of hot water for cleaning led to increased corrosion.

CAPTAIN F. W. BROAD said that it had been pointed out that there were two main types of ship, crude and general products. In the 1960s the general product carrier had itself diverged into two types, clean oil and black oil. Essentially a clean oil ship was presented in the diagrams. That type of ship, whilst it could carry black oil, was not eminently suited for it in that form. The 22 500 ton ship referred to was not fitted with heating coils and had inert gas systems. In his opinion the black oil ship was the outmoded crude carrier of a previous era, or a clean oil ship no longer quite suitable. Were there any plans for building specifically black oil ships? This would not be anything new, because in the late 1940s such ships were built.

The 1966 load line revision had had quite an impact on tanker design. The permanent ballast tanks tended to disappear and the cargo ballast tanks, with segregating line facilities, tended to take their place. Had the authors any views on the type of system which would eventually emerge? Also on the cargo ballast system, during loading particularly, it would appear from the diagram that separation between oil and water was dependent on one valve. In view of the very large line size, was that degree of separation sufficient, or did other means need to be adopted?

The 35 per cent ballast was stated to be a minimum, but he considered it to be excessive. Perhaps it was meant to be maximum, 20 per cent–25 per cent was more usual. It assisted with the berthing to have the ship on a reasonable draught and too much ballast could be an embarrassment, in that before some of the tanks could be loaded, the ballast must be taken out. In the 200 000 ton ship, the cargo ballast tanks would be the aftermost pair of wing tanks. This meant that the ballast in those tanks would be taken out on the pipeline, but in the design, a direct load line came into the pipeline at that point. Thus the initial loading of the ship could take place on three lines only instead of four, leading to an increased loading time. The timing of the ballast introduction or discharge was vital in the free flow ship. Introducing ballast too early during discharge destroyed the trim, very necessary for a quick discharge to be made with free flow, and similarly too late a commencement of ballast meant that the ballast became the critical path. Were there any guide lines for ships' crews so that the optimum discharge or loading could be made?

Mention was made of a rotary turbine type actuator for hydraulic power assistance on cargo valves in present ships. It must be borne in mind that some of these ships would trade in the Baltic and similar places, and frost would be a real danger to the system. Similarly, dry air was mentioned for the pneumatics; this was most important, not only for the actuation of valves, but also for the air system in pump rooms. Experience in the Baltic had proved that it was essential to have dry air.

Regarding cargo valve control systems in product carriers, the indication need not be continuous on all valves. On the master and cross-over valves, open and shut indication only was needed, but in suction valves, continuous indication was necessary. In crude oil ships, the open and shut indication was all that was required on most of the valves, but some essential valves, used to free flow large volumes into empty tanks, should be of a continuous indication type.

Tank cleaning was growing ever more difficult especially with large ships. In the 200 000 ton ship (Fig. 3) the suction valves which would be used in tank cleaning were sited in the centre of the centre tanks. This meant that the gradient available was the rise of the floor only, a different philosophy from that adopted in the diagram of the product carrier (Fig. 2). In the latter the suction valve was sited in the corner of the tank, making it possible to choose the gradient by listing. The product carrier had the advantage of being able to choose trim and list in tank cleaning, whereas in the big ship, where difficulties were experienced, only trim was controllable. Could the suction valves not be fitted at the corner of the large tanks instead of in the centre?

The load on top technique, one of the major innovations of the 1960s, was not mentioned in the paper. It had been adopted by most tanker operators now. There were many methods of separating the oil and the water, chemical emulsion breakers, heating coils near the interface, gravity separators, coalescers. What were the authors' views?

He was pleased to see that hot water was finding favour again, with high pressures, in tank washing. This had diminished with the arrival of chemicals, but experience indicated that chemicals were not the complete answer. Tank coatings seemed to inhibit the use of very hot water; the limiting temperatures seemed to be about 150°F (66°C). Could this be taken up with manufacturers in the hope that greater temperatures would be permitted?

In the 1960s, the International Oil Tanker Terminal Safety Group and the International Chamber of Shipping were making a determined attempt to standardize some of the safety practices and operating practices of tankers. No doubt when their publications came out they would be known more internationally. One of their recommendations would probably include closed loading, which should be more popular.

The paper could have mentioned the equipment and accessories which had been developed in the 1960s. Some of these were extremely valuable. One was the three-stud Butterworth plate, as distinct from the ten-stud in general use. In a 100 000 ton ship this saved no less than 1351 nuts. The built-in joint saved 193 joint renewals. There were now hosebarrows for tank cleaning requirements which cut the numbers employed from eight in a gang to two or three.

Eductors seemed to have found a place in the 1960s. There were hand eductors for tank mopping; eductors for permanent ballast tanks instead of pumps and lines; eductors instead of stripping pumps for tank cleaning. Why not have tappings from the cargo lines instead of the eductor equipment for mopping purposes in product carriers? What would the eventual size be for eductors to be used for cargo purposes in the large crude ships?

In the 1960s, cranes had replaced derricks in some ships. This was debatable in crude carriers, but almost imperative in product carriers frequently engaged in connecting pipelines.

The portable radio telephone had been an innovation of the 1960s. In the last month or so it had received a certificate of intrinsic safety. It would be available for cargo handling and this was essential in very large ships.

Lastly, there was pressurizing to assist the discharge of high vapour pressure liquids and in the 1960s there had been a change. The air and steam which had previously been used in their crude form had now been replaced by fans, usually motivated by air, or air movers, and the inert gas system. These three ways had built-in safeguards, in that it was difficult to over-pressurize the tanks.

MR. J. ISBESTER said that one surprising omission from the paper was any reference to the heating of cargoes. Did the authors find it necessary to make provision for the heating of cargoes in crude and product carriers? What systems of cargo heating were considered most suitable for these two specialized types of vessel? What provision was made for observing and recording cargo temperatures?

Correspondence

MR. J. R. MARSLAND wrote that the paper made no great point about the difficulty of cross-drainage within the tanks due to the limited size of limber holes through the longitudinals and transverses; with the increase in size of tankers the floor area was becoming ever greater. In view of the importance attached to quick turn-round it would appear imperative to arrange for the main cargo pumps to continue pumping at their maximum capacity, which was dependent upon the amount of liquid going to the strum for as long as possible and he would have thought it imperative that all main cargo pumps, in the 1960s, should be fitted with an evacuation stripping system. With the system referred to by the author, automatic regulation of the cargo pump capacity was obtained to eliminate a high vacuum in the suction lines and so eliminate possible crack-off of vapour during normal pumping.

Apparently, delivery pressure requirements from the cargo pumps were tending to rise, no doubt due to the shore installation requirements, and in the future steps should be taken to endeavour to obtain the worst possible discharge shore characteristic. What was the authors' opinion as to where this increase in finalization pressure would stop?

In view of the considerable discussions that had taken place apropos of proposed 500 000 ton tankers, what was the authors' opinion as to the envisaged approximate turn-round time of a ship of this size and the anticipated number of cargo pumps involved?

MR. A. W. FECK (Associate Member) wrote that, since the authors referred to his own paper,* he would be interested in what they suggested and in how to improve tank bottom drainage drastically, i.e. in view of the larger tankers under construction. Was it not correct to assume that even the most advanced cargo systems available could handle at lower tank levels only what the naval architects might consider adequate as tank bottom drainage coefficient, i.e. without risking the strength of the ship's hull design? In this respect he suggested that greater emphasis be exercised in the selection, location, spacing and sizing of limber holes, at least within the vicinity of main suction strums used for stripping purposes. The spacing of suction strums athwart ship appeared to be an associated problem too, i.e. for large crude oil carriers. The authors' point of view on these latter items was of interest, especially when considering that more than just two different grades had to be handled aboard larger ships. In some cases there were also restrictions in the ship's trim by the stern during cargo unloading operations.

The authors showed two curves, Fig. 6 (*a*) and (*b*). There was about 75 per cent of the total head in feet made up by friction. What was the authors' opinion of future cargo pump discharge pressure requirements at deck rail, which appeared to be constantly rising? In other words when did the increase in cargo pump capacity and head stop? In five, ten or fifteen years?

On several occasions the authors had indicated a certain preference for specific material selections, except for centrifugal cargo pumps. What would they consider to be an adequate cargo pump casing material specification? According to the latest Japanese publications, e.g. the cargo pumps of the Japanese super-tankers *Idemitsu Maru* and *Tokyo Maru* were not in use for more than seven to eight per cent of the total time per annum. It therefore became questionable to install high discharge pressure cargo pumps aboard ship with total requirements of 12 000–20 000 bhp, i.e. if the equipment could not be used more than eight to ten per cent of the total time. Did the authors eventually investigate any other method and how to unload large super-tankers in a more economical fashion?

* Feck, A. W. 1967. "Cargo Pumping in Modern Tankers and Bulk Carriers". *S.N.A.M.E. Paper.*

Authors' Reply

In reply the authors said that Mr. Scotwick had drawn attention to the advantages that could be taken of irregular tank capacities in product carriers. They would agree with this only if the ship was being designed for a particular trade which required the carriage of regular cargoes on each voyage.

With regard to cargo tank piping systems and the siting of cargo valves it was indeed advantageous to position the maximum number of the total valves required in the pumproom space, so that maintenance might be carried out without entering the cargo tanks. The provision of a separate line to each group of tanks however limited parcel flexibility and would become uneconomical as the requirement for numbers of separate tank groups increased. The real need was for a tank valve which would guarantee isolation with a minimum maintenance interval of two years.

Perhaps the greatest objection to twin duct cargo systems for crude carriers lay in the difficulty in washing, de-sludging, gas freeing and inspection. If greater flexibility was required than that given by the cargo transfer system the authors preference would be to increase the capacity of the single ring cargo/ballast/stripping line.

Tank cleaning with large capacity machines could only be carried out successfully if there was a stripping capacity equivalent to the delivery of the washing water so that the bottom of the tank was maintained in a reasonably dry state. For the current 200 000 dwt design of the authors' company it was intended to strip the cargo tanks with eductors powered by the cargo pumps which would also deliver the washing water. If there was a tendency for the washing water to accummulate on the tank bottom, the bulkhead valves would be opened to drain the excess water to an adjacent tank. The effect on the bulk discharge rate of the automatic stripping systems that had been installed was only marginal. The stripping systems essentially did no more than an adequate number of trained men could do operating the same basic equipment. The advantage, therefore, lay in the reduction in the number of men necessary to monitor the tank levels continuously and to take such corrective action, in regard to the pump throughput, that the pump did not try to discharge more cargo than could flow in at the inlet strum. The value of the system was at the bottom level of the cargo tank where one could expect an improvement in the rate at which the lower 12 or 18 inches in the tank was discharged over that which could be obtained by purely manual means.

Vessels of three times the current maximum, say 800 000 dwt, had not seriously been considered by the authors. If a design was to be prepared, the cargo system would not differ greatly from that illustrated but the discharge pumping rate would need careful consideration in relation to the reception facilities available.

In reply to Commander Hall, the simplicity which resulted from the screw displacement pump was well appreciated. It needed to be a vertical unit and it was not believed that one of adequate capacity had yet been designed and installed in any ship at the present time. One problem which arose in the consideration of this type of pump—which would be required to operate satisfactorily on a wide range of liquids, with a great range of viscosity, from kerosenes to heavy lubricants or fuel oils—was the clearance between the scrolls and the casing, and between scrolls themselves.

There was no simple and satisfactory solution to cover the wide range of conditions which might be required. If for example, the pump was designed with sufficiently close clearance to handle satisfactorily the very low viscosity liquids, such as kerosene, then the power required to drive the same pump when handling the more viscous liquids was substantially increased.

Commander Platt had asked about the design route in the selection of pump sizes for the big vessels. At the time that his company's terminal facilities were being designed they were not actively engaged in the design and construction of their own vessels. They had however co-operated with the terminal designers and provided information on the anticipated ships' equipment and discharge capabilities. It was an indication of the necessity for a simultaneous joint operation, that when they later started to design their own vessels and some of the information on the effects of pump capacity on cost etc. was confirmed from information obtained from the contractors and this information used in assessing the capacity to be installed in their own vessels, an adjustment was made in the installed capacity. Perhaps if they had been pursuing a simultaneous construction policy in the initial stages, the capacities might have been a little different. However, the installation which had been adopted was considered to be satisfactory from all aspects when matched up against the shore installations to which those vessels were expected to trade.

The time saving resulting from the increased capacity which they had selected, over that originally offered by the contractors, was not of itself very significant amounting to a little over two hours on each discharge.

For ships of this type, the movements of which might be dependent upon the tide, even a small reduction could mean the difference between turning round on one tide and waiting for the next.

For the motor tanker, Commander Platt was right when he said that the cargo handling system must be specifically installed for cargo handling duty, and perhaps used at full rate five or six times a year. For such vessels, it might be expected that the nominal discharge rate would not be more than 12 000 tons/h, and perhaps as low as 8000 tons/h, as the economic capacity.

A question had been asked about the electrical installation and the operation of the system. There had been only two faults of any significance in the four years of ship operation. One of those was basically bad ship design and was caused by sea water which cascaded on to the 3·3 kV switchboard, through ventilation trunking. The second fault was an unexplained breakdown on the insulation of one of the 1250 hp motors. Although a detailed investigation had been made by the manufacturers, they were unable to find any fault which might have caused it and the reason for the breakdown remained a mystery. In other respects, the system had proved extremely reliable, had not presented any operational problems and was very well liked by the ship's staff, due to its basic simplicity.

The electricians were selected from the electricians in the fleet on the basis of their general capability and experience, and were not given any special training, except a short period in their own office. It was not thought that any special training was necessary, other than an awareness that 3·3 kV was a higher voltage than 440 V coupled with a certain amount of common sense. There had been no problems in the selection and training of the shipboard personnel.

Mr. Victory had raised a very important point in that the performance of equipment that was available controlled the operational abilities of the ship. For cargo valves in product carriers, where a high degree of separation was required, it had been a common practice to fit double isolation valves, in the knowledge that failure of one of the valves was likely. It was the authors' view that isolation should be provided by a single unit of proved performance with maintenance procedures designed accordingly. To this end a variety of valves of differing characteristics had been used over the past seven or eight years. Present indications were that the best of these would not require to be serviced at less than two yearly intervals, but it was hoped to extend this period as development progressed. The reduction in the number of valves per ship, of course, made the maintenance task more acceptable.

Separation of cargo and ballast water during concurrent operations must be such as to eliminate any possibility of oil pollution. Single valves might be acceptable where there was pressure on the ballasting line and no pressure or vacuum on the cargo line, but this would require careful consideration. Where this condition was reversed or where the same conditions occurred on both sides of the isolation point, double valves with a bleed point between would be the best solution other than completely separate lines for each duty. The permanent ballast tanks amidships were of course served by a completely separate pump and line.

The reference to stable emulsions when tank cleaning referred to practices prior to the recent Oil Pollution Regulations and was not recommended.

Gas venting during loading was presently being investigated by the International Oil Tanker Terminal Safety Group (I.O.T.T.S.G.) and the authors would prefer to limit their comments to the following rather broad statements. Dispersal of gas depended upon the height of the outlet and the gas exit velocity; obviously, with higher exit velocity, lower heights of outlet were acceptable. Exit velocities could be controlled by specially designed outlets which adjusted to the loading rate. It was the practice in the authors' company to fit line venting systems with high outlets, but whichever system was adopted it must be capable of providing rapid dispersion of gas evolved from any cargo that was carried so that dangerous concentrations did not occur about the deck and accommodation.

Captain Moreton's question on economic justification of the coating of product carrier cargo tanks and the provision of centralized cargo control systems for large crude carriers were difficult to answer. The advantages enumerated for coated product carriers undoubtedly had an economic return and it had been estimated that this would be greater than the initial investment; by how much depended upon the effective life of the coating system. Experience in the authors' company had been that with a properly applied coating system, the breakdown after four years had not exceeded 0·1 per cent.

On centralized control systems, the principal return had been in the operational experience gained. It was now clear that the essential feature of any remote cargo control system was an accurate indication of tank liquid level which should be at least equal to the standard mechanical ullage gauge. For current large crude carrier designs it was intended to control only the pump room valves remotely; the line valves in tanks would be controlled from the cargo deck. Product carriers now building were fitted with power operation and remote control to pump room valves and in certain vessels tank valves would be power operated from the cargo deck.

Concerning the reference to 35 per cent ballast for crude carriers, also raised by Captain Broad, it was the intention to demonstrate that the system as designed would allow the vessel to berth with this quantity should it be necessary but it was agreed that this would be exceptional and that a figure between 20 to 25 per cent would be more usual.

Mr. Clemmetsen's comments on the difficulties that could be experienced with corrugated bulkhead structure were in accord with the authors' experience. Since the adoption of coated structure, however, the incidence of failure of corrugated bulkheads had reduced, presumably due to the absence of corrosion.

The arrangement with four cargo tanks in the 200 000 ton vessel illustrated posed trim problems but these were overcome by the careful positioning of the ballast only tanks and by adjustment of the longitudinal centre of buoyancy. The shear forces approached the maximum in the tank cleaning condition due to the particular ballast pattern which must be adopted. This ship would be supplied with a multi-point stress indicator and close control would be kept on all operating conditions.

The tanks selected for ballasting when discharging cargo (No. 5 wings) would be served by the larger diameter section of the ballast ring main and were not fitted with bulkhead flow valves. Additional ballast would be introduced in the forward tanks on completion of stripping.

The points raised with regard to drainage through the transverses of very large tankers were important. The cutouts in way of the longitudinals must be carefully designed to avoid possible buckling of the transverse webs.

To obtain the required free area for drainage when stripping might involve departures from the most economical structural arrangement but this must be accepted if the system was to operate efficiently at the design rate.

The structural life of any tanker could now be controlled by internal coatings. The short life philosophy for large crude carriers might be selected in the light of recent experience in which ships of only eight to ten years of age had become economically unattractive due to their limited size.

Damage to tank coatings by high pressure washing had been considered likely. Pressure was originally limited to 100 lb/in^2, this was subsequently increased to 120 lb/in^2 and washing was now carried out at 150 lb/in^2 without any adverse effects.

Captain Broad had rightly drawn attention to the black oil product carrier. The requirement for this type of vessel was generally met by the allocation of the older clean oil ship and the authors were not aware of any specific plans to build ships for this trade.

With the reduction in available capacity for permanent ballast following the 1966 load line revision, the owner had the choice of acceptance of high freeboards during loading and discharging, adopting concurrent ballasting loading/discharging or increasing the capacity for permanent ballast by increasing the depth of ship, but limiting the draught.

The lowest first cost was of course acceptance of high freeboards, but this might prove costly due to delays in service caused by bad weather. The third alternative was almost certainly the highest first cost and the increased capacity would increase the tonnage measurement and thereby increase the operating cost. The concurrent ballasting/cargo handling was the preferred solution providing that a simple and efficient system would be designed.

Concurrent ballasting/cargo handling required very careful timing, it should not retard the loading or discharge operation and guidance should be given to the operators as to the procedure to be followed. Loading lines should, as suggested, be positioned clear of the lines handling ballast water.

There was no reason why strums of centre tanks of crude oil carriers should not be sited in the corners of the tank as suggested, but it was questionable whether the drainage would be significantly improved in view of the heavy trim which could be taken and which was not always attainable on a product carrier.

With regard to "load on top" and other anti-oil pollution measures, there was a considerable amount of development and evaluation work still in progress. This, when completed, would almost certainly lead to a separate paper on this very complex subject which the authors did not feel could be covered in a small section of this present paper.

Permissible temperature of washing water in coated ships had increased with experience. Subject to satisfactory performance there was no reason why it should not be increased again to say 160°F (71°C) for those occasions when it was necessary.

The development in ancillary equipment and accessories during the period under review had contributed significantly to the handling of tanker cargoes. Though some of them were small in themselves they had improved the efficiency of the operation and allowed the reduction in crew numbers currently being made.

With regard to eductor capacity, units of 1200 tons/h were currently available but there was no reason why this should not be increased should the demand arise.

Mr. Isbester had referred to heating of cargoes. Provision was made in certain crude and product carriers, depending upon the cargoes to be carried. In the authors' opinion the bottom grid system was the most efficient for either type of vessel, though the spiral coil could readily be fitted to ships in service. Temperatures were recorded electrically from all tanks with thermocouples positioned at the mid depth.

A select sample of tanks might be fitted with additional instrumentation at bottom and top of the tank to check on the heating pattern.

Both Mr. Feck and Mr. Marsland had raised the question of bottom tank drainage already mentioned in the reply to Mr. Clemmetsen. It was indeed correct that the maximum stripping rate was limited during the final stages of drainage by the amount of bottom drainage area provided through the structure.

Adequate fore and aft drainage through transverse structure could be arranged without too much difficulty providing the required area was determined in association with a heavy stern trim throughout the stripping operation.

Athwartship drainage, particularly through the longitudinals adjacent to the strum was limited by the ability to compensate structurally for the drain hole cut in the longitudinal web. If it was not possible to provide sufficient athwartship drainage to a single strum to match the desired stripping rate, it would then be necessary to increase the number of strums and disperse them equally across the after end of the cargo tank.

One of the main advantages of the cargo transfer system mentioned on page 340 was the ease with which the ship drained. This was largely due to the increase in the number of flow paths, reducing the amount of athwartship drainage that was needed.

With regard to trim during discharge, it would be found that tankers adopted a stern trim as the ship lightened. It would be unusual for the stern trim to be such that the after draught at the lightest condition would approach the initial loaded even keel draught, but this could be controlled by the rate at which ballast water could be taken into the ballast only tanks.

With regard to discharge pressures, the authors' considered that a figure of 125 lb/in^{2}g at the ship's rail at full discharge rate was the maximum that could be economically employed in the immediate future. Pump capacity was likely to increase, but with tankers of the present power of about 30 000-35 000 shp the adoption of a high discharge rate, high head system would entail power plant equipment being specifically installed for cargo discharge purposes which was unlikely to be an economic proposition except in marginal cases. The authors, therefore, suggested that the pump total head would remain in the range of 140-160m and the total capacity of the cargo discharge system in the range 15 000 to 25 000 m^3/h. The total capacity might be made up by a number of pumps, always two or more and most likely numbering three or four. This would, therefore, indicate a pump size in the range 6000 to 8000 m^3/h at a discharge head of 140 to 160 m.

With respect to cargo pump materials, the authors' recommended the use of bronze casings and aluminium bronze pump impellers. These expensive materials could be justified by the almost maintenance free service that could be expected. Alternatively, for ships with a regular employment on long voyages and a pump operating life which might only be some ten per cent of the total ship's life, stainless iron casing and impellers could be considered, but it was thought that there could be no certainty that replacement or repair of these components would not be required during the life of the vessel.

With reference to the methods of unloading, the authors had not considered methods other than various permutations of a shipboard pump and deck line system.

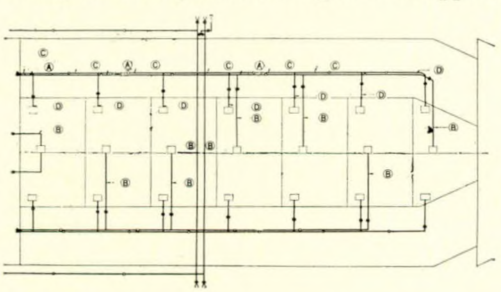
Marine Engineering and Shipbuilding Abstracts

No. 11, November 1968

New Type Combination Carrier

Makedonia is a bulk/ore/oil combination carrier of a new type built for Hellenic Bulk Transport, S.A. of Panama. Her design has been developed by Uraga Heavy Industries Ltd., with the technical co-operation of U.A. Pournaras, Inc., New York, U.S.A., so as to satisfy fully the various requirements called for in a combination carrier of this kind.

This vessel will utilize land facilities and equipment in principle for handling dry cargo, however, for the top side tank and also for the cargo hold, if need be, it is so arranged to permit a "Grain Veyor" to be carried on board and operated there to serve for grain unloading. Eight sets of "Grain Veyor" may be housed, if required, in the under forecastle space. Furthermore, in order to make it possible for the "Grain Veyor" to travel along the hatch side for cargo operation, the deck is kept clear. Accordingly, the grain loading and unloading of the top side tank can be easily done by "Grain Veyor" through the feeding hole on upper deck.



Cargo oil piping arrangement

While for the purpose of loading and unloading cargo oil, it is designed the same as a usual tanker, viz., two sets of 2300 m^3/h main cargo oil pumps and two sets of 200 m^3/h stripping pumps are installed in the main pump room. With regard to the cargo oil line, both main and stripping lines are of two independent line systems arranged in such a manner as to enable segregation into two. Furthermore, the cargo line is all installed along the pipe passage in the hopper side as shown in the figure, thereby ensuring accessibility enough for repair work. All the cargo oil valves are operated manually from deck by valve position.

Basic particulars of the main engine are as follows:

No. of sets	1
Type	Uraga-Sulzer 8RD-90 Diesel engine
Maximum continuous output	18 400 bhp at 122 rev/min
Normal output	15 600 bhp at 116 rev/min
Manufacturer	Uraga Heavy Industries Ltd.

The manoeuvring stand for control of the main engine is on the port side on middle floor level with a gauge panel for operation at one side, while at the other side is a monitoring panel wherein are set such devices as remote indicators for temperature and pressure of important parts of the main engine and auxiliary machinery, alarm device, running indicator and so forth.—*Japan Shipbuilding and Marine Engineering, January 1968, Vol. 3, pp. 29–33.*

All-enclosed Longliner for Greenland Waters

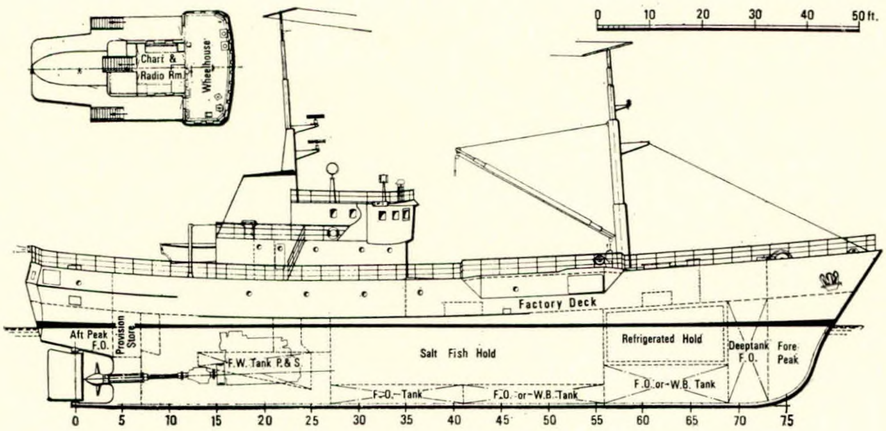
The shelterdeck longliner factory ship *Granit* has recently been delivered to her owners, A/S Granit of Ålesund.

Designed for operation off Greenland and Newfoundland, all working operations are carried out under deck; the line is shot through a hatch in the stern and hauled in through an hydraulically-operated hatch in the starboard side of the factory deck.

The vessel's complement and working deck layout permits longlining operations on a three eight-hour shift day. Fish salting will be carried out in normal daylight hours with the aid of Baader 163 beheading and a 440 Klippfish machine.

To provide the best possible working conditions a Flume stabilization system is incorporated in the engine room.

Granit is powered by a six-cylinder turbocharged MWM



Granit

type TbD Diesel engine, developing 1100 bhp at 375 rev/min, turning a Hjelseth c.p. propeller.

Electrical requirements are met by two 94 kVA alternators, each driven at 1000 rev/min by a 122 hp MWM type RHS 518S Diesel engine.

The refrigerating plant installed is of Lehmkuhl-York manufacture.

Deck machinery is of the low pressure hydraulic type by Brattvag while the steering gear is a Tennfjord electro-hydraulic installation linked to the auto pilot.

Principal particulars are:

Length, o.a.	163 ft 0 in
Breadth	31 ft 6 in
Depth	16 ft 9 in
Draught	15 ft 9 in
Deadweight	700 tons
Gross register	532 tons
Salt fish hold capacity	19 776 ft^3
Refrigerated hold capacity ...	5297 ft^3
Propulsive power	1100 bhp
Service speed	12 knots

Shipbuilding and Shipping Record, 16th February 1968, Vol. 111, No. 7, p. 239.

High Output Medium Speed Marine Diesel

Werkspoor's new four-stroke TM 410, with a bore and stroke of 410 and 470 mm respectively, is offered in versions having six, eight and nine cylinders in-line, 12, 16, 18 and 20 cylinders vee-form; at an initial rating of 500 bhp/cyl at 500 rev/min (burning residual fuel) it will cover a power bracket of 3000–10 000 bhp.

This new design incorporates the experience and knowledge gained from results achieved through the development of the older four-stroke TM engines (comparable size), the two-stroke TE 450 engines (burning heavy fuel), the higher rated two-stroke TE 290 (high thermal load) and the four-stroke, high-speed RUB 215.

Leading characteristics of the engine are as follows:

Bore	410 mm
Stroke	470 mm
Cylinder centres	700 mm
Initial rating	
Output/cyl	500 bhp
Speed	500 rev/ min
B.m.e.p.	206 lb/in^2
Maximum cyl pressure	1280 lb/in^2
Mean piston speed	1540 ft/min

Dimensions

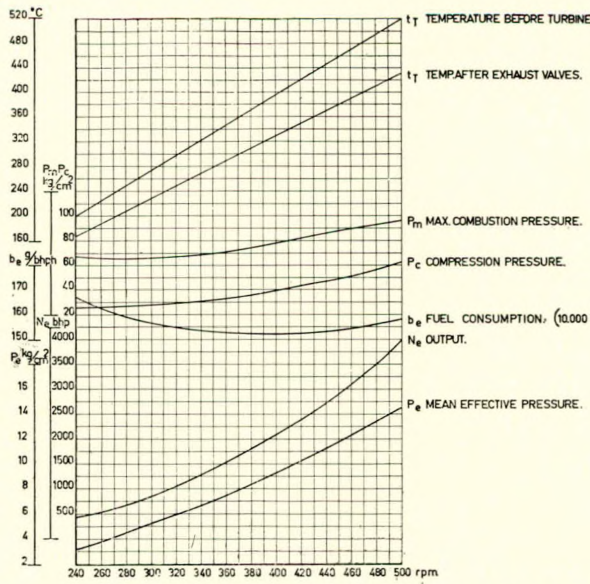
No. of cyls	*Length* (to coupling flange)	*Weight* (metric tons)
In line		
6	19 ft 9 in	52
8	25 ft 10 in	64
9	28 ft 2 in	70
Vee form		
12	19 ft 8 in	80
16	26 ft 0 in	100
18	27 ft 10 in	110
20	30 ft 9 in	125

Future development is aimed at testing the vee-form engine, the prototype of which is under construction.

The cast iron cylinder head has two exhaust valves and one large inlet valve. It also accommodates the injector and the starting air valve.

Great attention has been given to efficient cooling to avoid dangerous thermal stresses. The cooling water entering the head by drilled passages in the bottom periphery, is guided over the bottom and along the injector house by a horizontal division plate.

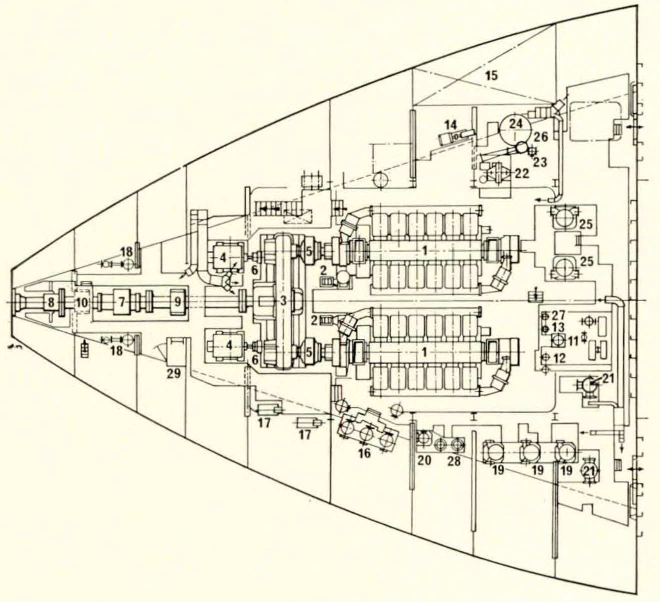
The exhaust valves have stellited seats and are mounted



Performance curves based on propeller law for eight-cylinder engine; air temperature 30°C, water temperature, before air cooler, 25°C, barometric pressure 760 mm Hg, fuel 1500 sec Redwood No. 1 at 100°F in water cooled cages. The total water flow is forced past the cooling space, which is close to the seat surface. This brings about the very low exhaust valve temperatures which are a prerequisite for extended intervals between overhauls when using residual fuels.—*Shipbuilding and Shipping Record, 5th April 1968, Vol. 111, pp. 480–482; 486.*

Forest Product Carrier

Kockums recently handed over 27 450 dwt "open"-type carrier *Malmanger,* the lead ship of a series of six vessels ordered from the yard for the joint ownership of Westfal-Larsen and Co. and the Star Shipping Group, both of Bergen. Intended primarily for the carriage of unitized cargoes such as pulp, lumber and newsprint, although equally suited to traditional bulk commodities such as ores, grain, fertilizers etc,



1) *main engines*
2) *turning gear*
3) *reduction gear with thrust block*
4) *shaft alternator*
5) *M.E. flexible coupling*
6) *alternator flexible coupling*
7) *intermediate shaft with propeller oil distribution box*
8) *sleeve coupling*
9) *intermediate shaft bearing*
10) *propeller pitch positioner*
11) *fuel oil transfer pump*
12) *fuel oil purifier supply and circulation pump*
13) *Diesel oil purifier supply pump*
14) *sludge pump*
15) *sludge tank*
16) *main lubricating oil tank*
17) *reduction gear lubricating oil pumps*
18) *controllable pitch propeller hydraulic pump*
19) *salt water/fresh water twin cooling pump*
20) *a.c. plant sea cooling pump*
21) *sea water filter*
22) *bilge pump*
23) *bilge ejector*
24) *oily water separator*
25) *ballast pump*
26) *ballast ejector*
27) *lubricating oil purifier pump*
28) *fire pump*
29) *emergency escape*

Lower platform arrangement forest product carrier

the vessels are all to be operated by Star Bulk Shipping. Two are on order for K/S Columbia and K/S Pacific, both of the Star Group.

Built under special survey of Det norske Veritas in accordance with the requirements for the highest classification ✠ 1A1"T"—the notation T signifying ore strengthening in holds Nos 1, 3, 5 and 7—*Malmanger* has a single continuous deck and features double shell construction. Machinery space and accommodation are aft.

Principal particulars are:

Length, o.a.	564 ft 0 in
Length, b.p.	530 ft 0 in
Breadth, moulded	85 ft 0 in
Depth, moulded	48 ft 3 in
Summer draught	34 ft $1\frac{3}{4}$ in
Deadweight	27 450 tons
Gross register	18 470 tons
Net register	11 246 tons
Cargo capacity, grain	1 166 642ft^3
bale	1 099 322ft^3
Ballast capacity, tanks	9329 tons
holds	8201 tons
Bunker capacity, h.v.f.	1472 tons
Diesel	183 tons
Propulsive power (continuous) ...	12 000 bhp
Block coefficient	0·778
Contract speed	15·9 knots
Trials speed at 11 650 bhp ...	17·68 knots

The two M.A.N. V6V 40/54 engines develop an aggregate 12 000 bhp (continuous rating) at 400 rev/min and turn a four-bladed KaMeWa c.p. propeller of 6100 mm diameter at 107 rev/min. The contract speed of *Malmanger* which features a ram-type bow is 15·9 knots, fully laden; during sea trials a maximum mean speed of 17·68 knots was attained on a 20 ft 4 in mean draught and the engines developing 11 650 bhp. Economy trials returned a specific fuel consumption of 161 gm/shp-h.

The drive is completed by Vulkan flexible rubber couplings and multi-disc friction clutches with reduction gearing of Renk manufacture. Drives for the two shaft alternators are taken directly from the pinion shafts of the gear unit.

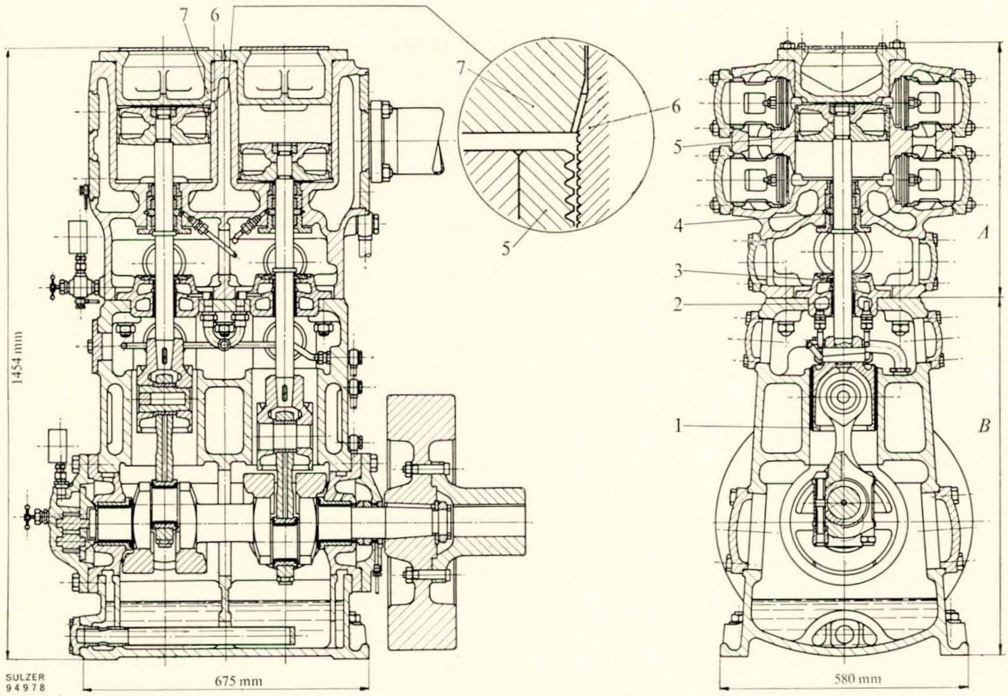
The main engines are designed to run on heavy oil of 1000 sec Redwood No. 1 at 100°F. To avoid the risk of corrosive condensation when running for extended periods at low output the scavenging air manifolds are fitted with electric heaters. Totalling 110 kW, singly or in combination they can raise the air temperature to 80°C. Switching on the heaters and shutting off the normal water flow to the air coolers is automatically effected whenever a main engine drives only its alternator.

Steam has been entirely eliminated from *Malmanger's* engine room. The heat content of the main engine's cooling system has, instead, been utilized to the fullest extent and additional heating when required is by electricity. At 80°C, engine return water maintains a suitable temperature in the fuel oil settling and double bottom tanks. Units requiring additional electrical heating are: main engine, fuel oil preheaters, oil separators, the domestic fresh water heater; air-conditioning is entirely electrical.—*Shipbuilding and Shipping Record, 26th April 1968, Vol. 111, pp. 573–579.*

Transport of Liquefied Gases by Sea

At present there are more than 180 liquefied-gas vessels in service, with a further 42 under construction. Many of these specialized ships are equipped with reliquefaction plants. Labyrinth-piston compressors are being more and more widely adopted for this duty because of their oilfree delivery and reliability in service.

Single stage designs are usually adopted for semi-refrigerated pressure tankers. They aspirate the gas at a suction pressure of about five kp/cm^2 and compress it to a



1) *crosshead*
2) *guide bearing*
3) *oil scraper*
4) *gland with labyrinth seal*
5) *piston with labyrinth grooves*
6) *cylinder with labyrinth grooves*
7) *cylinder head*

Sections through a single-stage Sulzer labyrinth-piston compressor for L.P.G. ammonia

final pressure corresponding to a condensing temperature of at most 35°C with sea water cooling.

In fully refrigerated tankers it is customary to use two-stage machines which aspirate the gas at a pressure just above the atmospheric and compress it to a final pressure usually corresponding to a condensing temperature of −10 to −20°C. As a result of the low suction temperature it is often possible to dispense with intermediate cooling between the two stages.

Sealing between piston and cylinder wall and between piston rod and gland is obtained by the use of labyrinths. Consequently no lubrication is needed in the gas-swept spaces of the compressor. The absence of any contact in the seals makes wear extremely slight and lubricating oil consumption is also kept to an absolute minimum.

The oilfree side of the compressor (*A*) (see figure), and the lubricated crank gear (*B*) are separated by oil scraper rings mounted on the piston rod. The rod also carries a collar ring which prevents the residual molecular oil film from creeping up the rod. The distance between crank gear and gland is such that the oily part of the piston rod cannot enter the oilfree gland. Should any gas leak through the gland, it is returned to the suction side. The piston rod runs in a supplementary guide bearing in addition to the crosshead.

The crankcase and separation space are kept under suction pressure. Where the crankshaft leaves the case, it is fitted with a shaft seal operating in oil. Leakages from the crankcase are in this way precluded even when the machine is not in operation. Measures have also been taken to ensure that the gas in the crankcase cannot mix with that in the separation space, and that no oil mist can find its way into the latter.

The oil under pressure needed for lubricating the crank gear is supplied by a gear-type pump driven by the crankshaft The suction volume of the compressor can be reduced in stages by keeping the suction valves open during the compression stroke. Pneumatically- or hydraulically-operated valve plate lifters are used for this purpose. In the case of hydraulic operation, the oil pressure is provided by the lubricating-oil pump.—*Kläy, H., Sulzer Technical Review, 1967, Vol. 49, No. 3, pp. 148–154.*

Clamp for Magnetic Particle Inspection

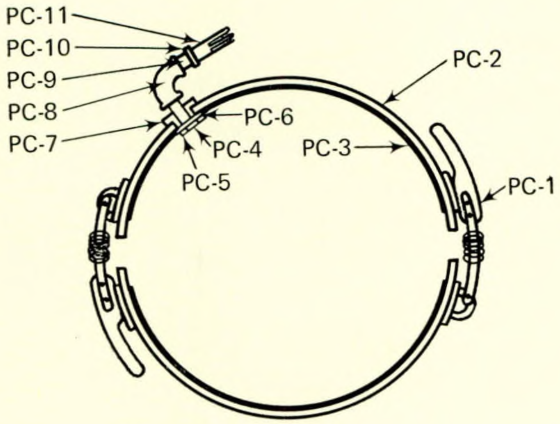
A universal prod clamp for magnetic particle inspection has been suggested by Portsmouth Naval Shipyard. The magnetic particle inspection method is now required on carbon steel pipe joints; in the past the liquid penetrant method was used for this inspection. While liquid penetrant is effective in determining defective welds and porosity on the surface of welded joints, it cannot locate a defect below the surface.

The magnetic particle method, using direct current, will locate any linear or transverse crack on the surface and also reveal any subsurface indication of a defect to the extent of $\frac{1}{16}$ in below the surface. This method is 95 per cent faster per joint and takes only about five minutes to complete inspection.

Magnetic particle inspection of carbon steel pipe joints formerly required the services of two inspectors, one to hold the two prods, while the other applied magnetic particle powder and evaluated the area involved. These areas are often located in close quarters, making the inspection procedure awkward.

Arc strikes on the piping are also a serious problem. The prods are placed upon the carbon steel pipe stradling the weld zone. If the prods are seated improperly, or if there is any dirt, grease or paint whatsoever under the prods, arc striking will occur, when the current is turned on. This leaves a mark or burning indentation on the pipe which could result in a crack. Therefore, arc striking is absolutely prohibited.

Fig. 1 shows the universal prod clamp. Fig. 2 shows a typical hookup using the clamp. One cable from the M.T. machine is connected to piece (10) (Fig. 1) for ground and one cable to piece (11) to power source side of M.T. machine.



1) *clamp*
2) *$1\frac{1}{2} \times \frac{1}{8}$-in stainless band*
3) *$\frac{1}{4} \times 1$-in rubber strip*
4) *$\frac{5}{8}$ RD brass rod*
5) *$\frac{5}{8}$-in brass nut*
6) *$\frac{1}{8}$-in bakelite washer*
7) *$\frac{1}{4}$-in bakelite washer*
8) *$\frac{3}{8}$-in brass elbow*
9) *$\frac{3}{8}$-in brass nipple*
10) *end of connector*
11) *end of connector*

Fig. 1—Universal prod clamp

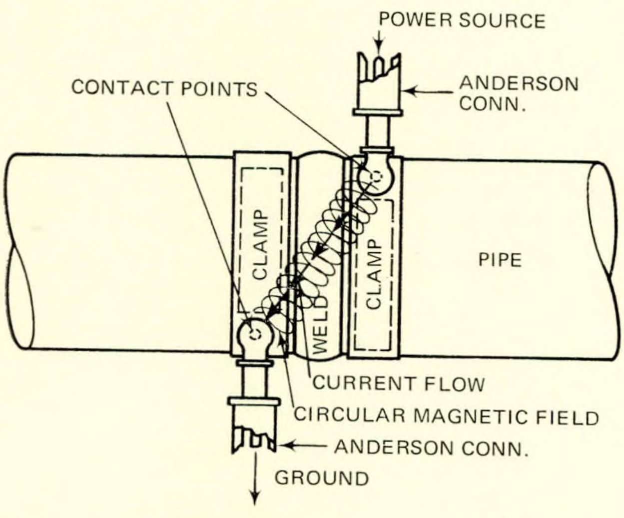


Fig. 2—Clamp hookup. Prod contacts can be located on periphery of pipe to establish desired magnetic field

Contact current is applied by the use of a hand trigger switch which was formerly attached to the Anderson connexion of the power source side.

When the current is turned on, a circular or longitudinal field of magnetism is applied to the pipe, depending on the placement of the prods. The inspector then applies the magnetic particle powder, dusts off the excess while current is on and finally evaluates the results. He has one hand free for this operation while the other hand holds the mobile prod.—*Naval Ship Systems Command Technical News, March 1968, Vol. 17, pp. 40–41.*

German-built Lifeboat for Columbia

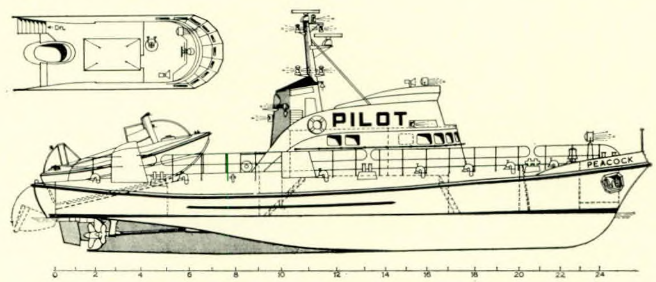
Requiring a pilot boat suited to their long working range in the difficult waters of the mouth of the Columbia River below Astoria in Oregon, the 17 members of the Columbia River Bar Pilots Association were attracted to the rescue cruisers of the German Lifeboat Institution (Deutsche Gesellschaft zur Rettung Schiffbrüchiger).

The lines are the result of a long co-operation between the D.G.z. R.Sch. and Maierform G.m.b.H. of Bremen and a model was tested at the El Pardo tank.

Peacock, as the new vessel has been named, is triple screwed. The centre screw is fixed bladed while controllable pitch wing propellers are fitted.

To ensure first class manoeuvrability in conditions of heavy ground swell and surf, a spade-type rudder is provided in the slipstream of each screw.

The whaleback form was chosen to attain a flush foredeck without forecastle. The daughter boat, designed specially for boarding and taking off pilots, is launched through a hinged, opening stern.



German-built lifeboat for Columbia

The stability of *Peacock*, as of the cruisers from which she is derived, is of the utmost importance. For all conditions of loading the vessel has positive righting levers even beyond 90° angle of heel.

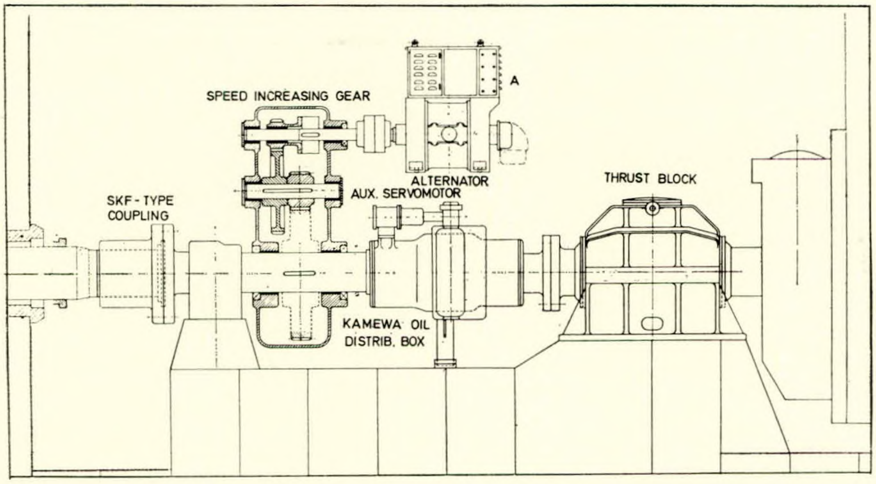
A precise calculation of unsinkability has not been made in respect of *Peacock* although such calculations do exist for similar rescue cruisers. The invulnerability of the vessel cannot, however, be in doubt as she is completely double-hulled; additionally the bulkhead spacing has been reduced compared with the rescue boats and two adjacent compartments can be flooded while still retaining adequate stability.

The deck, daughter-boat ramp, superstructure and funnel are constructed of seawater-resistant magnesium alloys (AlMg5 and AlMg3) fabricated using argon arc or Sigma shielded gas welding methods in an enclosed shop.

The propulsion plant chosen comprises a Maybach-Mercedes-Benz MB820Db turbocharged and intercooled central Diesel engine, developing 1350 bhp at 1500 rev/min, with an MB836Db turbocharged engine of the same manufacture, rated at 525 bhp at 1500 rev/min, on each of the wing shafts. All engines are resiliently mounted and indirectly cooled by way of shell plating intercoolers.—*Shipbuilding and Shipping Record, 11th January 1968, Vol. 111, pp. 51–52.*

Shaft-driven Generators

The increased use of medium-speed machinery driving the propeller via a reduction gear-box with one to four input shafts has considerably simplified the employment of shaft-driven alternators. The gear-box can readily be furnished with power take-off shafts, the speed of which can be selected to suit the alternator drive. The power take-off shafts can be directly connected to the main gearwheel and will then run with fixed position. The connected alternators can readily be paralleled and fed into the common busbar. In this case the



Arrangement of a speed increasing gear for a shaft-driven generator with a c.p. propeller driven by a slow running Diesel engine

clutches for connexion/disconnexion of the engines are mounted on the input side of the gear.

A second possibility offers an arrangement whereby the alternator is directly connected to the Diesel engine through the gear and the engine clutches are arranged so that the above connexion is not affected when the motor is disconnected from the main gear. Thus it is possible to run the alternator in harbour when the propeller shaft is stopped. With this method two alternators cannot be paralleled as the relative position of the rotors will vary each time they are engaged.

The simplest drive is when the alternator is directly coupled to the forward end of the main engine. Naturally, this method will normally lead to bigger sizes of alternator as the rev/min cannot normally be selected. With slow-running Diesel engines particularly, the size of the alternator will be comparatively large.

Another method of alternator drive is that where the rotor forms a part of the intermediate shaft and the stator is mounted round the shaft line. In this case the shaft speed will determine the dimensions of the alternator. The figure shows a gear-box which can be used when direct-coupled slow-running Diesel engines are installed. By this rather simple type of gear a suitable alternator speed can be achieved, thus reducing costs and the dimensions of the alternator. The gear must be furnished with an elastic coupling with suitable characteristics in order to prevent the combustion pulses of the engine leading to transient irregularity of the frequency.—*Larberg, L., Motor Ship, April 1968, Vol. 49, pp. 19–20.*

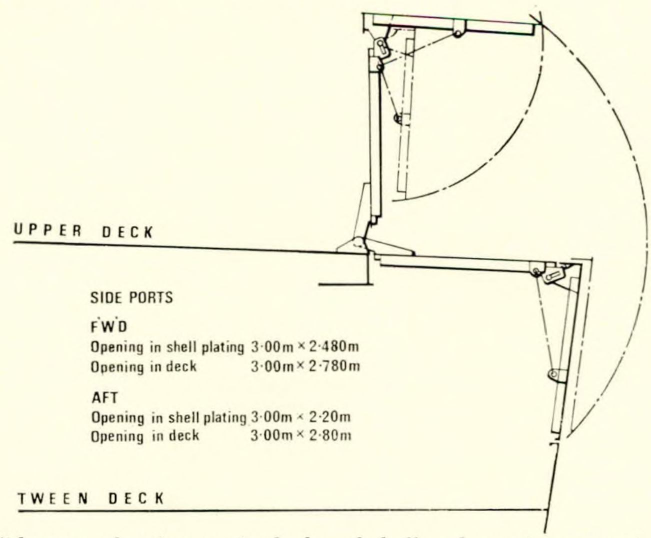
Rapid Handling of Unitized Cargo

The cargo motorship *Galila* was delivered to the Zim Israel Navigation Co., Haifa, by Orenstein-Koppel and Lübecker Maschinenbau AG, Lübeck. Designed for the rapid handling of unitized cargoes, *Galila* was built to the requirements of Germanischer Lloyd class ✠100 A4E1 MC American Bureau of Shipping ✠A1 3. The hull is strengthened to class C for navigation in ice and has a raked stem, cruiser stern, forecastle and has the all-aft arrangement of bridge accommodation and machinery.

Principal particulars are:

Length, o.a.	404 ft $2\frac{3}{4}$ in
Length, b.p.	372 ft $6\frac{3}{4}$ in
Breadth, moulded	54 ft $5\frac{1}{2}$ in
Depth, moulded (to U.D.) ...	31 ft 7 in
(to T.D.) ...	22 ft $6\frac{3}{4}$ in
Draught (open)	21 ft $8\frac{1}{4}$ in
(closed)	24 ft $10\frac{1}{4}$ in
Deadweight (open/closed) ...	5561/7027 tons
Capacity (grain)	360 937 ft^3
(bale)	334 258 ft^3
Power output	5400 bhp at 165 rev/min
Speed	16 knots
Electrical output	3 × 310 kVA 400 V a.c.

The hatches are closed by steel covers with the sections stowed forward and aft of the coamings in the open position. As can be seen from the accompanying illustration the side port is cut into the shell and deck, is hinged upward and is hydraulically operated. Inboard of the opening is a ramp, hinged at the foot and with vertical travel up the sides of the opening to adjust for differing quay height or varying state of the tide for truck-to-truck operations.



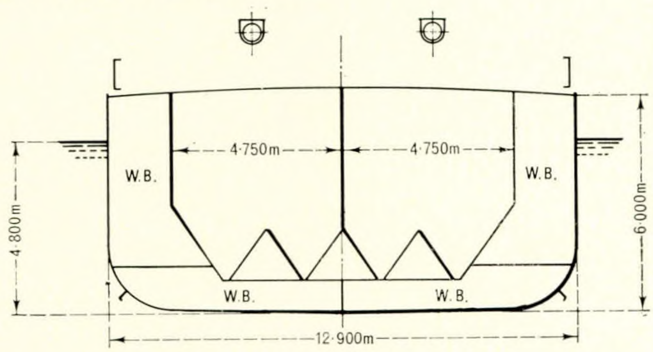
Side ports showing cut in deck and shell and opening operation

Main propulsion is by a six-cylinder M.A.N. type K6Z.60/105D turbocharged Diesel engine developing 5400 bhp at 165 rev/min and designed to operate on heavy fuel oil with a viscosity of 3500 sec Redwood scale at 100°F. The unit is directly coupled to a four-bladed propeller.—*Shipbuilding and Shipping Record, 9th February 1968, Vol. 111, pp. 196–198.*

Self-loading/discharging Bulk Cement Carrier

Planet, a bulk cement carrier, was recently completed by the Aarhus Flydedok and Maskinkompagni for AB Aalborg Portland-Cement-Fabrik.

Built to the highest requirements of Lloyd's Register for this class of vessel, the hull has a forecastle and poop, raked stern and cruiser stern, is specially strengthened for navigation in ice and has the all-aft arrangement of bridge, accommodation, and machinery.



Cross section of Planet *showing configuration of hold floor arranged as four vee-channels*

Five transverse bulkheads divide the hull into the following main compartments: fore peak and chain locker, forward deep tank and bow thrust unit, fore hold, after hold, engine room and after peak. The holds are completely squared off by longitudinal bulkheads on each side forming wing tank compartments for water ballast, fresh water and are also sub-divided by a centreline bulkhead.

To meet the requirements of high manoeuvrability needed by the ship, she is fitted with a KaMeWa bow thrust unit situated in the forward deep tank, which is remotely controlled from the bridge. Cathodic protection by sacrificial zinc anodes is provided around the propeller and, at intervals, along the length of the hull. The cargo holds, on the other hand, are unpainted.

Main propulsion is by a 12-cylinder unidirectional B. and W. type 1226-MTBF-40V Diesel engine, developing 1980 bhp at 600 rev/min. Power is transmitted to a KaMeWa controllable pitch propeller through a 1:2·5 Renk reduction gearing. The propeller is strengthened for ice and pitch and rotations can be controlled from the bridge. Instead of the usual practice of carrying a spare propeller, extra blades for the c.p. unit are substituted.—*Shipbuilding and Shipping Record, 8th March 1968, Vol. 111, pp. 334–335.*

Uprating of Maybech Mercedes-Benz Engine

The introduction of exhaust gas turbocharging to the MB 518 engine was intended not only to compensate for the power lost in the Rootes blower (approximately 300 hp) but also to increase the indicated cylinder output.

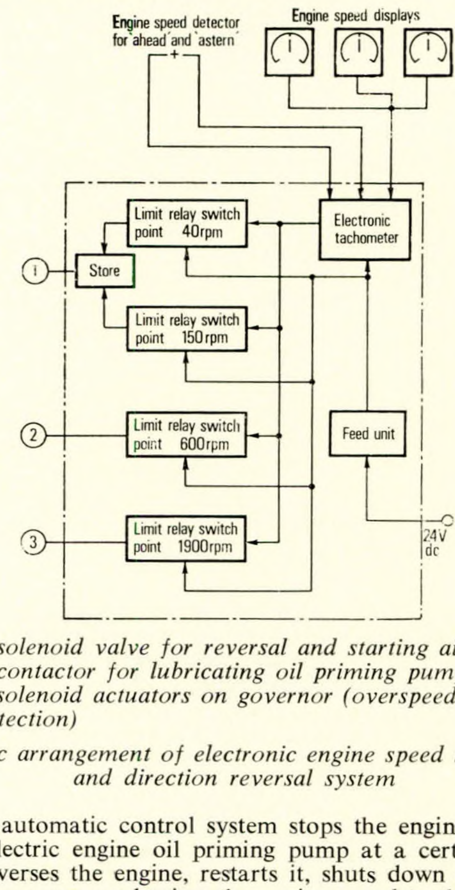
Based on preliminary investigations, the maximum output of the standard engine designed for exhaust gas turbocharging was fixed at 3500 hp at 1700 rev/min, despite the results of test series carried out at 3600 hp at 1720 rev/min.

The exhaust gas manifolds are located external to the vee of the cylinder banks—arranged at an angle of 40°. The engine is therefore equipped with two exhaust gas turbochargers—Hispano-Suiza HS 419 BX1 units, each having five turbine inlets—mounted above the gear-box.

The impeller shaft of the turbine is horizontal, the blower and turbine wheel being cantilevered. The plain bearings of the turbochargers are lubricated and cooled with engine oil. In order not to impair cylinder cooling, cooling water for the blowers is obtained from the cooling water manifold downstream of the cylinders where the water temperature is approximately 80°C. Each blower delivers combustion air—via a cooler—through an independent inlet manifold in the vee of the engine; thus, in the event of unequal loading of both engine sides and therefore unequal turbocharger speeds, there will be no interference between the two turbochargers which operate in parallel.

It is chiefly for reasons of weight that it was decided to continue to construct the engine in reversible form although it could of course be used without direct reversal facilities in association with a reversing gear-box.

The shortest reversal time with the most efficient air consumption for reversal and starting is achieved in all cases irrespective of personnel skill. The manoeuvre now depends only on the deceleration effect of the engine and of the propeller and on the resultant reduction of shaft speed. All other control functions are achieved more quickly.



1) *to solenoid valve for reversal and starting air supply*
2) *to contactor for lubricating oil priming pump*
3) *to solenoid actuators on governor (overspeed protection)*

Schematic arrangement of electronic engine speed monitoring and direction reversal system

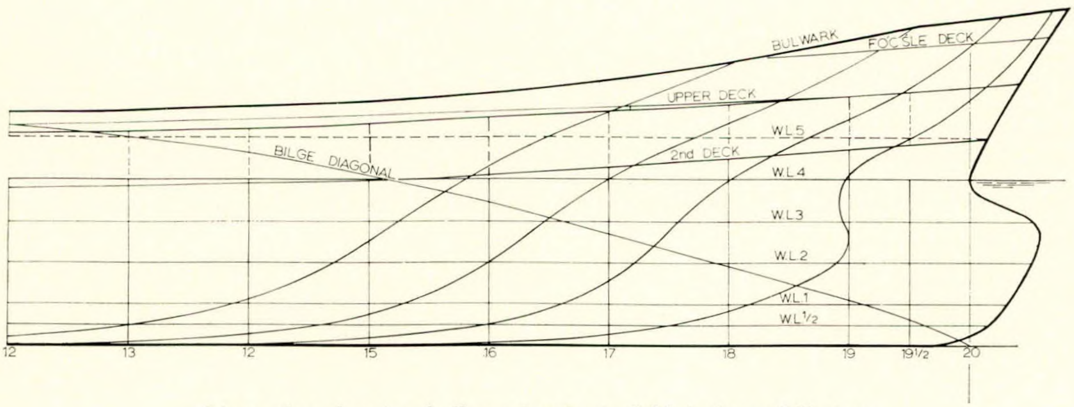
The automatic control system stops the engine, switches on the electric engine oil priming pump at a certain engine speed, reverses the engine, restarts it, shuts down the engine oil priming pump and raises the engine speed to the specified value. There is also a speed-governed automatic change-over from underwater exhaust to above-water exhaust during shut down from ahead running or for re-opening the underwater exhaust after restarting ahead. Running astern is possible only with the above-water exhaust.—*Shipbuilding and Shipping Record, 29th March 1968, Vol. 111, pp. 442–444.*

British-built Vessel with Maierform Hull

A cargo ship with a Maierform hull design has been built in Scotland for German owners and is now in service. This vessel, *Louise Bornhofen*, 8906 dwt, has been designed for the carriage of general cargo.

Although of conventional layout and design, one feature of this medium sized vessel's hull form is the distinctive bow contour.

The S.V. bow follows an S-shaped profile but gives a



Lines plan showing bulbous forefoot of Maierform S.V. type

bulbous V in the sectional view; which is a departure from the conventional globular shape of bulbous forefoot normally applied to present day designs. Details of the hull form can be seen on the accompanying body lines.

As with all hull design work, such refinements offer the best results when designed for a special hull at the tank testing stages. A saving in excess of 18 per cent has been recorded on known tests; favourable results have also been registered by fitting the bow to an existing vessel—in one case obtaining an increase of 0·6 knots at 7200 bhp. Most gains, in the case of existing ships, have been obtained in the ballasted condition.

Up to the present time only two vessels under construction in the U.K. have this feature, the second ship being the 13 000 dwt passenger/cargo liner being built by the Burntisland Shipping Co. Ltd. for the East and West Steamship Company of Karachi. The ship described in this article has been built by the Caledon Shipbuilding and Engineering Co. Ltd., Dundee, for Robert Bornhofen Reederei, Hamburg.—*Shipping World and Shipbuilder, February 1968. Vol. 161, pp. 427–431.*

Norwegian-designed Stern Trawler for Greenland

At the beginning of 1965 an agreement was entered into by the Royal Greenland Trading Co., Copenhagen, and the technical development and research department of Bergens Mekaniske Verksteder that the latter should undertake for the former an investigation into the possibilities for trawl fishing off the coast of Greenland.

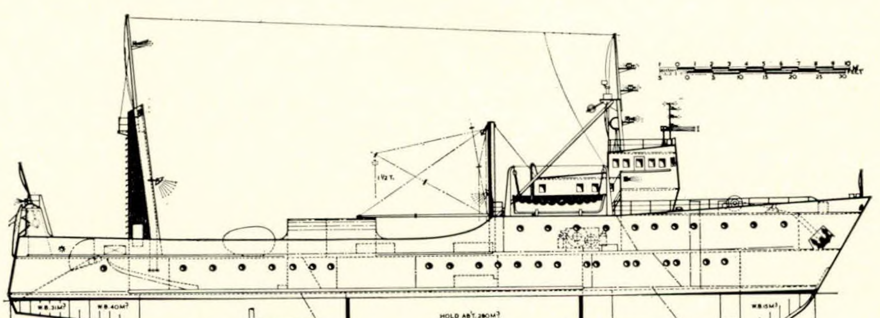
During the first half of 1965 a considerable amount of basic data was collated and analysed with the aid of a computer. The results from the economic model, BMV Project 9006, were submitted on 1st June 1965, and concluded with a recommendation for an open type stern trawler of just under 500 tons gross.

Principal particulars are:

Length, o.a.	164 ft 0½ in
Length, w.l.	152 ft 6¾ in
Moulded, breadth	31 ft 0 in
Moulded, depth (to U.D.) ...	22 ft 1¾ in
(to M.D.) ...	15 ft 1 in
Draught	14 ft 1¼ in
Gross register	499 tons
Power output	2 × 1100 bhp at 750 rev/min
Speed	14½ knots

To meet the necessary power requirements two main engines, each of approximately 1100 bhp, proved the most favourable solution. After investigating the many different existing types of engines and the space they occupy, the recommendation was made to install two Normo type LSM8 Diesel engines each with continuous output of 1100 bhp at 750 rev/min which reduces to approximately 195 rev/min at the variable pitch four-bladed propeller. Under normal service conditions the vessel's speed is calculated at approximately 14½ knots with both engines operating and approximately 12½ knots with one engine only.

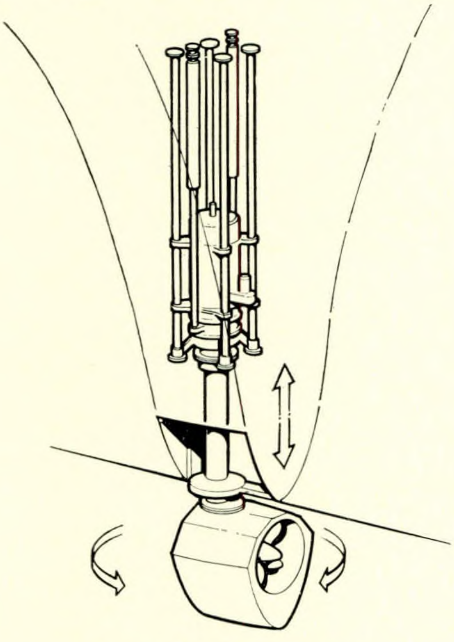
The recommended trawl winch was of Normo manufacture and is powered by two hydraulic pumps which are driven by power take-offs from each main engine. With two hydraulic pumps—that is to say with both main engines operating—the winch will exert a pull of 16 tons on the centre drum with a winding-in speed of 262½ ft/min; but with one hydraulic pump in operation while the pull is the same, the winding-in speed is consequently halved.—*Shipbuilding and Shipping Record, 19th April 1968, Vol. 111, pp. 542–543.*



Norwegian-designed stern trawler for Greenland

Retractable Bow Thruster

A retractable bow thrust unit will enhance the station-keeping capability of the 230 ft research stern trawler building at Ferguson Brothers for the Ministry of Agriculture, Fisheries and Food. This Diesel electric vessel is to be used for a programme of investigation into the movements of fish around the deep water fishing grounds and, as such, it is essential that she should be capable of remaining on station under all conditions.



Retractable bow thruster

The ship will be fitted with a specially designed bow thrust unit which will be retractable inboard when not in use. Fitted into a nozzle built within a section of the hull structure, the unit can be lowered on hydraulic rams from inboard to its operating position below the keel of the ship. Exposed under the ship it is then capable of rotating a full 360° so that from whichever direction the wind or tide flows, up to and including gale force nine winds, the thrust from the propeller unit can be used to equalize the other forces and maintain station. Since the propeller, complete in its hull section, can revolve, it can cope very rapidly even with gusting from different directions.

An ingenious system of bridge indication allows the unit to be controlled remotely from the bridge. Red and green lights indicate whether the unit is fully drawn up inside the hull or is protruding beneath the ship's structure, and a needle/dial indicator shows the angle at which the propeller in its tunnel is turned. This latter has a dual function since the unit can only be retracted when in the fore and aft position.

The propeller diameter is 4 ft 4 in but because of the shape of the hull the total lift of the unit is 8 ft. Powered by a 350 hp motor giving a propeller speed of 480 rev/min, it provides a fully variable thrust up to five tons.—*Shipbuilding and Shipping Record, 3rd May 1968, Vol. 111, p. 615.*

Holland-built Products Carrier

The products carrier *Texaco Ghent,* which has been constructed by the N.V. Koninklijke Maatschappij "De Schelde", Flushing, a member of the Rhine-Scheldt Group of companies, for Texaco Panama, Inc., New York, is one of three sisterships built in the Netherlands for the same owners.

The ship, her machinery installation and equipment, have been constructed to the American Bureau of Shipping class +A 1 E "Oil Carrier" +AMS. In addition, she fulfils the requirements of the International Loadline Convention, SOLAS 1960, the U.S. Coast Guard, the U.S. Public Health Service, Panama Canal Zone Regulations, Suez Canal Rules of Navigation and AIEE No. 45—Recommended practice for Electrical Installations on Shipboard. The ship's hull form was tank-tested at the Netherlands Ship Model Basin, Wageningen, and the ship is provided with a straight-cylindrical bulb having a diameter of 4·90 m. The curves of the ship's hull and the laying-off of the shell plating was carried out with the help of computer calculations.

Principal particulars are:

Length, o.a.		592 ft $1\frac{3}{4}$ in
Length, b.p.		555 ft
Breadth, moulded		78 ft
Depth at side		43 ft
Summer draught		32 ft $11\frac{7}{16}$ in
Corresponding deadweight ...		22 200 tons
Tank capacity (100 per cent) ...		30 019 m^3
Tonnage (International) ...	14 579·79 grt	8 872·63 nrt
Tonnage (Panama)	15 602·53 grt	12 207·84 nrt
Tonnage (Suez Canal) ...	15 607·87 grt	11 707·86 nrt
Main propulsion machinery: Steam turbine installation developing 15 500 shp at 105 rev/min of propeller		
Speed		17·8 knots

All deck machinery is of Hydraulik Brattvaag type, driven by two sets of three hydraulic pumps installed, respectively, in the forecastle space and under the poopdeck. Each of the pumps is driven by a 57 hp NEBB electric motor with a speed of 210 rev/min.

Since the vessel is to be used for the simultaneous carriage of a number of oil products, the cargo section of the vessel consisting of 22 tanks is subdivided into eight groups, each having its own cargo handling system, so that cargoes can be handled entirely separately and without the danger of contamination. Six different kinds of cargo can be handled simultaneously.—*Holland Shipbuilding, April 1968, Vol. 17, pp. 60–64.*

Advantages of Aluminized Wire Rope for Sea Water Service

As part of an investigation by the American Chain and Cable Co, Inc. into the comparative resistance to corrosion in sea water of aluminized, galvanized and stainless steel wire ropes, an extensive series of tests under a variety of environmental conditions at different coastal points in the U.S.A. has been carried out on stressed and unstressed samples implanted in the mud bottom and suspended at sea level and at intermediate depths.

The results of these tests have led to the conclusion that aluminized-steel wire ropes provide longer service in sea water than those which are galvanized or made of stainless steel. Thus, it was found that the aluminized wire rope tended to weaken after exposure far less quickly and severely than the stainless steel material and that, although galvanized wire rope resisted corrosion well enough at intermediate depths, it was not so satisfactory at sea level and in the mud bottom. By contrast, the corrosion resistance of aluminized wire rope was relatively good at all levels and could be very greatly improved by using a swaged rope, rather than an unswaged rope. A finding of particular interest is that, even after its aluminium coating is badly damaged or completely corroded, an aluminized wire rope will still resist corrosion, owing to the aluminium-iron layer remaining on its surface.—*Materials Engineering, March 1968, p. 30; Engineers' Digest, June 1968, Vol. 29, p. 73.*

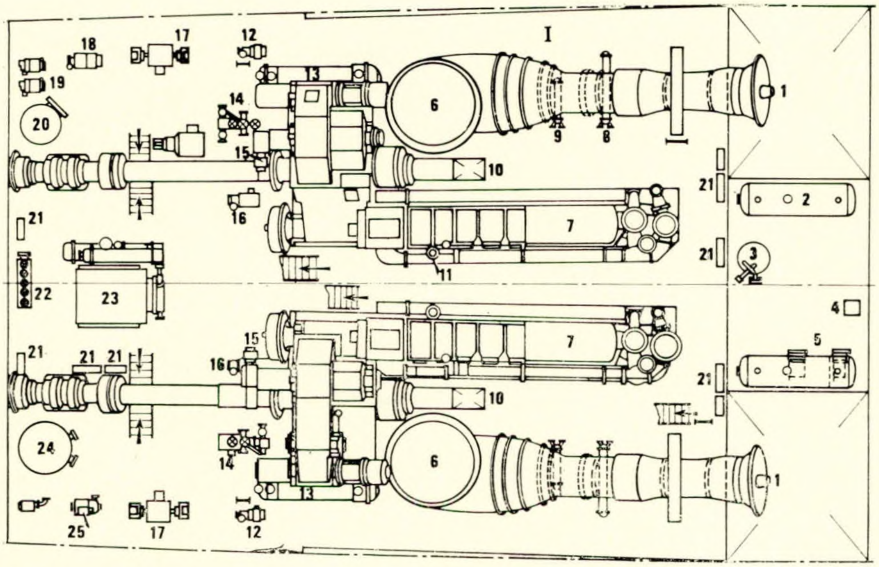
New U.S. Coast Guard Cutters

A large replacement order for high and medium endurance cutters was put in hand in 1964 by the United States Coast Guard because of block obsolescence in these categories.

Hamilton class vessels are powered by a combined plant, comprising gas turbines for full speed and Diesel engines for cruising, driving twin shafts with c.p. propellers. Combined reduction gears with two-stage turbine synchronizing and Diesel disconnecting clutches are used to allow three modes of operation: Diesel, Diesel and gas turbine during change-over and gas turbine. Owing to the wide range of operations required by CODOG plants and for high and low speed manoeuvrability, c.p. propellers were selected for their higher overall performance.

of $8\frac{1}{2}$ in and a stroke of 10 in and the weight of each engine is 49 095 lb (wet).

The main reduction gear consists of two self-contained dual input drives, both directly coupled to the gas turbine and Diesel engine. The double reduction turbine gearing has a ratio of 15·314:1, the first reduction being at 4·465:1 and the second at 3·43:1 for a propeller speed of 235 rev/min. The starboard gear drive incorporates an idler in the first reduction train to give the required propeller rotation as both gas turbines turn in the same direction. The single reduction Diesel engine gearing has a ratio of 6·034:1 with a resulting propeller speed of 149 rev/min. The turbine clutch will synchronize the shaft to the prime mover regardless of which is moving slower, within a differential speed range of up to



Key to engine room
1) *turbine intake*
2) *air receiver*
3) *oily water separator*
4) *transformer*
5) *air compressor*
6) *gas turbine*
7) *Diesel engine*
8) *lubricating oil cooler box*
9) *free turbine*
10) *servo control box*
11) *water conditioner*
12) *alternator cooling and sea water pump*
13) *lubricating oil cooler*
14) *lubricating oil pump and strainer*
15) *jacking gear*
16) *lubricating oil cooling and sea water pump*
17) *hydraulic pump for c.p. propellers*
18) *potable water transfer pump*
19) *potable water service pump*
20) *potable water pressure tank*
21) *power panel*
22) *fuel oil transfer and ballast manifold*
23) *evaporator*
24) *hot water tank*
25) *hot water circulating pump*

New U.S. Coast Guard cutters

The gas turbines are Pratt and Whitney type FT4A-6 units, each rated at 19 700 shp (maximum) at 3600 rev/min, and comprise a multi-stage reaction turbine driven by hot gas flow from an axial flow gas generator consisting of multi-stage compressors powered by multi-stage reaction turbines. The hydraulic starting system was supplied by the New York Air Brake Co. and is made up of six major items: one variable delivery hydraulic supply pump, one pump unit powered by a 100 shp electric motor, two variable displacement hydraulic starters, one accessory package and a $\frac{1}{2}$ hp pump motor unit. In operation, hydraulic fluid at 4000 lb/in^{2}g from the pump H.P. outlet port is applied through the inlet port of the starter to its nine pistons which causes the starter elements to rotate. The fluid is returned from the output port of the starter through the L.P. line directly to the pump inlet port.

The 12-cylinder Fairbanks-Morse type 38TD-1/8 turbocharged uni-directional Diesel engines each have a continuous rating of 3690 bhp at 900 rev/min. The cylinders have a bore

2250 rev/min, after which a dental clutch—operating in parallel with the synchronizing clutch elements—solidly couples the driving shaft to the driven shaft. No power is required to keep the dental clutch engaged or disengaged.

Reversing and manoeuvring is by two Liaaen type D125/4 four-bladed c.p. propellers.—*Shipbuilding and Shipping Record, 15th March 1968, Vol. 111, pp. 361–365; 368.*

Welding Low Alloy Steel Castings with the Flux-cored Process

Until recent years, the welding of steel castings was almost an unspeakable subject. Before high quality welding was available, castings were over-designed to allow for the discontinuities which are inherent to the process. Since the availability of high quality welding, however, castings have been redesigned to save material and weight. The combination of good casting practice and quality welding has resulted in more efficient and more reliable castings.

Following recognition of the efficiency and the economy achieved by the use of the flux-cored carbon-dioxide shielded-arc process to weld low and medium carbon steels, it was natural to project the use of the process to the welding of low alloy steel castings. The successful welding of low alloy engineering type castings offered the possibility of greater returns from welding with the process because of the greater value of the castings. Not only were monetary savings expected, but shipments of castings would be expedited by the faster flow of work through the welding department.

Preliminary test and development work was necessary to determine the best chemical composition of the electrode to be used with each class of low alloy castings. It was also necessary to conduct studies of the efficiencies of the flux-cored process compared to the manual covered electrode process previously used in order to justify the installation of equipment for the new process.

Three low alloy compositions of castings were investigated: a triple alloy of chromium (Cr), nickel (Ni) and molybdenum (Mo); $\frac{1}{2}$ per cent Cr–$\frac{1}{4}$ per cent Mo; and $\frac{1}{2}$ per cent Cr–$\frac{1}{2}$ per cent Mo. In this particular study, one flux-cored electrode composition and one low-hydrogen covered electrode composition were used to weld all three of the casting compositions. The flux-cored electrode was nominally $\frac{1}{2}$ per cent Cr–$\frac{1}{2}$ per cent Mo and the covered electrode was $\frac{1}{2}$ per cent Cr–$\frac{1}{4}$ per cent Mo.—*Wick, W. C. and Lee, R. K., Welding Jnl, May 1968, Vol. 47, pp. 394–397.*

Japanese Distant Water Factory Trawler

Nippon Suisan Kaisha, Tokyo, took delivery of two large stern trawlers suitable for fishing the distant water grounds in the North Pacific, Australian and African areas. The lead ship, *Fuji Maru*, was completed at the Fujinagata yard (Osaka) of Mitsui Shipbuilding and Engineering Co.

Built to the requirements of Nippon Kaiji Kyokai NS✠ MNS✠ RMC✠ the hull is flush decked with a curved stem, transom stern into which is cut a 13 ft $1\frac{1}{2}$ in wide ramp extending from the upper deck to the waterline and has standard sheer and camber. Except for the longitudinally framed double bottom, transverse framing is otherwise employed for the hull which, below the main deck, is divided by transverse watertight bulkheads into the following main compartments: fore peak for fuel oil or water ballast, two deep tanks, similarly arranged for fuel oil or water ballast, three fish holds, fish meal plant and hold, fish oil deep tank, engine room, and the after peak.

The flush deck was adopted to secure the maximum space for fish processing plant and accommodation, and the engine casing was dispensed with in order to utilize fully the main deck which is mechanically ventilated.

Principal particulars are:

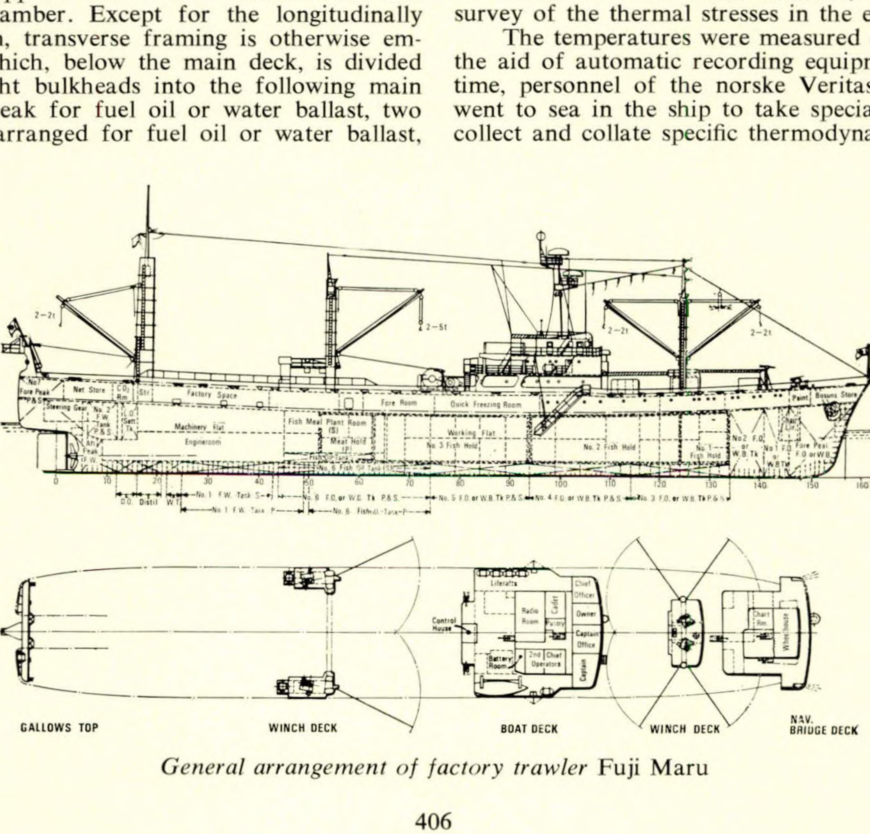
Length, o.a.	335 ft $7\frac{1}{4}$ in
Length, b.p.	308 ft $4\frac{3}{4}$ in
Breadth, moulded	52 ft 6 in
Depth, moulded (to U.D.) ...	32 ft $1\frac{3}{4}$ in
(to M.D.) ...	23 ft $11\frac{1}{2}$ in
Draught	19 ft $8\frac{1}{4}$ in
Deadweight	3825 tons
Capacity (bale)	22 195 ft^3
(liquid)	8069 ft^3
Power output	4400 bhp at 248 rev/min
Speed	$13\frac{3}{4}$ knots

Main propulsion is by a nine-cylinder B. and W. type 9M42CF Diesel engine, constructed under licence by Mitsui with a continuous rating of 4400 bhp at 248 rev/min but which will be operated at 240 rev/min in service with an output of 4050 bhp. A centralized watch on the main engine can be maintained from the machinery control room and control can be exercised from this position or from the bridge. Sufficient bunkers are stowed for an endurance of 24 300 miles.—*Shipbuilding and Shipping Record, 19th April 1968, Vol. 111, pp. 541–542.*

Measuring Combustion Chamber Temperatures in Large Engines

From time to time damage through overheating occurs to engine parts surrounding the combustion chamber. Among these are cracks in the cylinder head, piston and liner as well as burning of the pistons and heavy wear of the cylinder liners. In order to establish the temperature conditions adjacent to a combustion chamber, Det norske Veritas have, in association with other societies, carried out a detailed survey of the thermal stresses in the engines of a ship at sea.

The temperatures were measured over a long period with the aid of automatic recording equipment and, from time to time, personnel of the norske Veritas Research Department went to sea in the ship to take special measurements and to collect and collate specific thermodynamic data. A very com-



General arrangement of factory trawler Fuji Maru prehensive series of temperature readings was submitted to a careful statistical analysis for purpose of comparison.

Piston temperatures afford the best indication of the thermal stress in an engine, especially the peak temperatures at the piston crown and the general temperature in the piston body. As expected, these temperatures are dependent on the engine's surroundings, on the volume and temperature of the scavenging air passing through, on the efficiency of the cooling system and also on the occurrence of high temperature rises not associated by these factors. Such temperature rises, comparable with what in the human body would be called "bouts of fever", are as a rule connected with defective fuel valves and poor fuel quality. Unusual causes of overheating include the use of oil with a high water content and unstable oils in which the heavier asphaltenes have coalesced and agglomerated.

During such periods of overheating, piston rings are inclined to slacken which can lead to gas leakage and distortion while, as a result, bearing surfaces of the cylinder become overheated, possibly leading to cylinder lubrication failure.

As a rule, an engine is able to accept such temporary extra stresses without damage, but this depends on the quality of the material and the accurate workmanship of the individual parts.—*Veritas, Oslo, May 1968; Marine Engineer and Naval Architect, July 1968, Vol. 91, p. 286.*

Diagnosis of Engine Trouble

Investigators of International Research and Development Co. Ltd. have helped shipbuilders by determining the causes of faults in ships' engines and recommending preventive measures. In one case the failures, at short intervals, of a series of fuel pump cams on a Diesel engine caused successive stoppages in mid-Atlantic of a single-engined ship. By the end of the journey nearly all the cams carried as normal spares had been used.

Two types of failure had occurred, in the one the cams were found to be chipped and worn, in the other they had deformed and the work face flattened. Both defects caused engine failure and loss of power and required replacement of the damaged parts. Samples of the failed cams and an unused replacement were flown across the Atlantic to IRD. Investigations showed that the cams had come from two different suppliers and were of two types.

The first had been over-carburized causing brittleness of the surface hard casing; the second had had a heat treatment omitted altogether and consequently the surface was soft and readily deformed. As well as detecting these faults, IRD recommended that changing the type of steel used for the cams would increase the strength of the cam core. Shortly afterwards, the foreign licensor of the engine issued a circular specifying material, heat treatment and hardness data following very closely the IRD recommendations.—*Marine Engineer and Naval Architect, August 1968, Vol. 91, p. 316.*

Powerful American Tugs

Town Point, the newest addition to Curtis Bay Towing Co.'s powerful tug fleet, is now in operation at the Port of Norfolk and her sister tug, *Drum Point,* is scheduled to join the Blue Diamond fleet at Baltimore this month.

Both *Town Point* and *Drum Point* were designed by Marine Design, Inc. of New York City, to meet the American Bureau of Shipping's requirements for ✠ A-1 coastwise towing classification.

Principal particulars:

Length, o.a.	99 ft 10 in
Beam, moulded	27 ft
Depth, amidships	16 ft 4 in

The tugs are of welded steel construction. Unusually heavy plating and rugged construction were incorporated in their design to enable them to transmit their tremendous power without undue vibration. The bows are reinforced with additional plating for winter ice conditions.

Each tug is powered by a 12-cylinder model 12-645-E5 General Motors turbocharged Diesel engine rated at 2360 hp on a single shaft. This hp, high in relation to the length overall, was made possible by the installation of what is the most noticeable feature of the tugs in drydock—a giant propeller measuring 11 ft 6 in in diameter.

These unusually large five-bladed propellers were designed specifically for *Town Point* and *Drum Point*. Their 11 ft 6 in diameter is equivalent to nearly one eighth of the entire length of the hull. The propellers are designed to permit maximum utilization of engine power at the low speeds encountered in ship docking.

Shipboard electrical power is supplied by a 25 kW belt-driven generator hung below the deck beams and utilizing a power take-off from the main Diesel engine. An auxiliary Detroit Diesel-driven 30 kW generator set is also available to meet the ship's d.c. current demands.—*Maritime Reporter/Engineering News, 15th March, 1968, Vol. 30, pp. 7–8.*

50 000 shp c.p. Propellers

Recent years have seen many design studies and a few applications of high powered gas turbines, either alone or in combination with other prime movers. One of the most pressing needs for such applications is a minimum weight and cost and highly efficient and reliable means of providing a reversal of propeller thrust with the unidirectional gas turbine. For most applications the c.p. propeller can meet this need.

The future use and size limit for c.p. propellers are difficult to predict. It does appear that the larger power sizes may be closely related to the future use of the unidirectional gas turbines and perhaps to reheat steam plants.

It is recognized that there has been prejudice against the use of c.p. propeller. Some of this may have been valid at one time. More recent questions on the use of c.p. propellers relate to efficiency, cavitation noise and reliability.

Propulsion and propeller efficiency of c.p. propellers frequently has been stated to be less than f.p. propellers. This, of course, is partially true if an excessive size of hub is used.

Recent David Taylor Model Basin model tests, both open water and self-propelled, comparing a 40 000 shp f.p. propeller having a 20 per cent hub with a c.p. propeller having a 28·8 per cent hub indicated the c.p. propeller efficiency to be 2·9 per cent lower and the ehp/shp to be 3·3 per cent lower at rated power. At 35 000 shp these values were 1·45 per cent and 1·64 per cent lower, respectively. This shows the effects of reduced loading. The c.p. propeller had a 28·5 per cent higher blade-thickness factor, 1·3 per cent more pitch at nominal design (the foregoing test points) and 8·3 per cent less expanded area ratio to permit the five blades to pass during reversing. These differences in efficiency resulted in one per cent less ship speed at the same maximum shp.—*Maritime Reporter/Engineering News, March 1968, Vol. 30, pp. 24–25; 27.*

Structural Analysis by Finite Element Method

It is now possible to predict accurately stresses and deflexions in highly complex structures by routine procedures. The geometrical complexities of modern structural assemblies precludes a rigorous analysis for the stresses and deflexions sustained under loading. However, with the advent of high-speed computers, a so-called approximate analysis yields accurate answers.

One of the procedures is known as the finite element method. Since its original introduction, this method has been applied to a wide variety of problems with gratifying success. Its greatest asset is perhaps its versatility in that the same general technique is applied, regardless of the type of elastic continuum and both loading and boundary conditions may be completely abitrary.

Matrix algebra is used extensively since this simplifies and generalizes the mathematical expressions of the basic concepts of the theory. This should not limit the circle of readers since it is only necessary to be familiar with the rules of matrix multiplication and addition, and the definition of the inverse of a matrix.

The finite element method is essentially an extension of standard structural analysis procedures which allows one to calculate stresses and deflexions in two and three dimensional structures using the same methods which are applied to ordinary frame structures such as trusses or rigidly jointed frameworks.—*Tolefson, D. C. and Brand, L., Maritime Reporter/Engineering News, 1st May 1968, Vol. 30, p. 58.*

World's Biggest and Fastest Containership

Ponce de Leon is the world's biggest and fastest containership. It is the largest roll-on/roll-off trailership. It is the first ship to have a General Electric MST-14 power plant. It is the first large dry cargo ship to be built since World War II in an American shipyard without government assistance.

The owners feel that the high sea speed of the vessel and the ability for fast turn-a-round of this ferry-like ship more than offsets the loss in full cubic utilization which previously was the major drawback to roll-on/roll-off ships.

Principal particulars:

Length, o.a.	700 ft 0 in
Beam, moulded	92 ft 0 in
Beam, maximum	105 ft 0 in
Depth, moulded	60 ft $1\frac{5}{8}$ in
Draught	27 ft 0 in
Shaft horsepower	32 000
Speed	25 knots
Displacement	24 100 tons
Deadweight	13 100 tons
Crew	37

The key to the economical operation of *Ponce de Leon* is the General Electric Co.'s MST-14 reheat steam power plant which was built by the Marine Turbine and Gear Department.

The cycle finally selected has a fuel rate of 0·433 lb/hp-h at 32 000 shp and with the main generator and main feed pump powered by the main turbines.

Steam is generated in the steam drum at 1650 lb/in²g. It passes through the superheater where its temperature rises to 955°F. At the outlet from the superheater the steam flow is divided into two streams: part flows through the desuperheater supplying steam to various services and the main flow goes to the propulsion turbines. It enters the H.P. turbine at 1450 lb/in²g and 950°F where it expands down to 320 lb/in²g and 628°F. From the H.P. turbine it is returned to the reheat section of the boiler where its temperature is again raised to 955°F. The steam is then put through the I.P. turbine and continues on to the L.P. turbine, finally exhausting to the condenser. There are no extraction points in the H.P. and I.P. turbines and the first bleed occurs at the crossover from the I.P. to the L.P. turbine.—*Maritime Reporter/Engineering News, June 1968, Vol. 30, pp. 7–10.*

Polish-built Stern Trawler for British Owners

The largest British stern trawler to be operated as a wet ship, *Boston Lincoln,* is the first of a series of four ships ordered in Poland by British owners. The following three vessels will all be freezer trawlers, the second ship being *Boston York* which will enter service in the near future, while two further ships are to be delivered to Boyd Line Ltd., of Hull. Most of the equipment for the series of four ships is supplied by British companies and includes the refrigeration equipment of 35 tons/day capacity which will be installed in each of the freezer vessels.

Boston Lincoln is a prototype variant of the builders' B427 freezer trawler design which is a development of the builders' 195 ft-long B28 class, an example of which, *Saint Martin,* is owned by Glaciers Lejeune of Boulogne, an associate company of the Boston Group. If required, the new ship can be converted to a freezer trawler in a very short time thus providing a flexibility which will enable the owners to examine the merits of operating one ship wet and the other as a freezer trawler.

Boston Lincoln can carry about 300 tons of fish and 140 tons of ice.

Principal particulars:

Length, o.a.	212 ft 0 in
Length, registered	195·23 ft 0 in
Length, b.p.	182 ft 1 in
Breadth, moulded	39 ft 4 in
Depth to upper deck	25 ft 1 in
Depth to lower deck	17 ft 11 in
Mean draught	16 ft 5 in
Gross register	845·95 tons
Net register	284·91 tons
Fishroom capacity	21 200 ft³
Fuel capacity	370 tons
Endurance, approximate ...	60 days
Trials speed	14·5 knots
Crew	26 + 2 spare

Main propulsion is by a Mirrlees KLSSMR8 unidirectional engine developing 2500 bhp at 400 rev/min. The drive is transmitted via a 2:1 reduction gear to a Liaaen c.p. propeller. The gear-box has built-in fluid couplings from the main and two lay shaft drives. Each lay shaft is coupled to a Laurence, Scott d.c. generator, one of 315 kW for the trawl winch and the other of 350 kW for the ship's services.

The main machinery can be controlled either from the bridge or from a soundproofed control room located in the machinery space. The main engine is automatically prevented from overspeed by a propeller pitch limiting device.—*Motor Ship, June 1968, Vol. 49, pp. 136–137.*

Screw Pumps for Tanker Cargo Handling

Owing to the increased demand for screw pumps as cargo pumps, important progress has been made in their construction. Until fairly recently the maximum capacity of external bearing screw pumps was about 600–700 ton/h but it is now practical to provide screw pumps having a capacity of 1500 ton/h.

Because of improved bearing installation design and screw profile, the normal design pump is capable of a maximum discharge pressure of 200 lb/in² and, in special designs, 350 lb/in². The improved profile gives a very good efficiency irrespective of whether the viscosity is high or low. Developments in the design of mechanical seals make it possible to produce a pump which is free from shaft leakage under all conditions of temperature, vapour pressure and viscosity.

The pump position is determined by the cargo specific gravity, vapour pressure and degree of toxicity as well as by the size of the vessel.

Screw pumps are used as cargo pumps for a variety of small tankers and on river tankers, for example, the pumps are usually situated on deck and are driven from the main engine by suitable transmission equipment.

For discharging products having a very high vapour pressure, it is imperative that the pump should be placed within the tank; this will ensure that the permitted low suction lift is not exceeded.

To achieve this a pump combination has been developed in which a small axial flow centrifugal booster pump is mounted at the foot of the screw pump suction pipe and delivers the product to a vertical screw pump mounted on deck.

The booster pump is driven by an intermediate shaft extending downwards from the screw pump. When each tank is provided with this pump combination the following advantages are obtained.

Products having a vapour pressure of 11·5 lb/in² and a viscosity of up to 3000 sec Redwood No. 1 can be pumped. The expense of pump rooms is avoided and toxic risks are reduced to a minimum, also the number of suction and discharge lines and valves is small. A further advantage is that since each tank has its own pump there is only a remote chance of grade contamination taking place.—*Bleijenberg, G. W., Motor Ship, July 1968, Vol. 49, pp. 187–188.*

Damaged Tanker Drydocked with Own Pumps

Heavy bottom damage sustained by a 58 350 dwt tanker, *Tank Countess,* posed a difficult and unusual problem. When the damage occurred, the ship was *en route* to Stavanger, where after berthing, it was found that so much sea water had entered the cargo tanks that the limited shore facilities could not cope with the output rate of shipboard pumping required to exceed the inflow of water through the cracks in the hull and thus empty the cargo tanks. The vessel then proceeded to Rotterdam but again it proved impossible to empty the tanks of contaminated cargo. After discussion, the best method of solving the problem appeared to be to drydock and discharge the ship at the same time.

The No. 4 drydock of the Netherlands Dock and Shipbuilding Co. was chosen as the dock is of sufficient size to take the 58 350 dwt ship and could also handle the large quantities of contaminated water. At the time there was still some 10 000 tons of cargo oil in the ship and approximately 20 000 tons of leaked water and water ballast (the latter—in the undamaged starboard wing tanks—had been used to correct a 12° list).

Nitrogen had already been blown into the damaged tanks in Norway to prevent an explosive gas-air mixture forming and further large quantities of the gas were added during final discharging. Another effective precaution was to cover the oil in the tanks with a thick layer of foam. This method was later used in the drydock itself.

The area around No. 4 drydock was fenced off and special identity cards were issued to personnel working in the area. Electricity was switched off, which meant that crane assistance was not available and that lighting had to be provided by safety lamps. Bronze tools only were used. A 14-in diameter pipeline was laid from the dock wall to an adjacent tank cleaning station with an additional branch from the dock's stripping pumps.

Tank Countess entered the drydock and was put on the blocks by pumping water from the dock with the large pumps of the dock. As soon as the ship was firmly settled, the ship's pumps took over. Contaminated cargo was pumped to the API separator of the tank cleaning station. At a later stage oil, though contaminated with water, was pumped direct into tank barges.

Finally, when the water level in the dock had been lowered to the top of the keelblocks, an appreciable amount of oil leaked into drydock. This was pumped, with the remaining water, to the API separator by the dock's stripping pumps. —*Motor Ship, July 1968, Vol. 49, p. 192.*

20-knot Refrigerated Cargo Liner

With a refrigerated cargo capacity of more than 546 000 ft³ the New Zealand Shipping Co.'s cargo liner *Mataura* ranks among the largest vessels of this class in service. Recently delivered by the Mitsui Shipping and Engineering Co. Ltd., from the Tamano shipyard, the ship is the first of two ordered by the owners—a P. & O. Group company—and is also the first ship of this type to be built by Mitsui. While so much attention is paid to the containerization of traffic on the U.K.–Australasia services it is noteworthy that *Mataura* and her sistership are designed for the traditional perishable products high-speed services which have for so long been the main feature of this trade. However, attention has been paid to the possibility of carrying unitized or containerized cargoes both below and above decks.

Principal particulars:

Length, o.a.	540 ft 0 in
Length, b.p.	505 ft 0 in
Breadth, moulded	74 ft 6 in
Depth, moulded	46 ft 2¾ in
Full load draught	30 ft 1¾ in
Corresponding deadweight ...	11 731 tons
Gross register	9504·39 tons
General cargo capacity	78 241 ft³
Refrigerator cargo capacity ...	546 319 ft³
Liquid cargo capacity	7283 ft³
Service speed	20·1 knots

In normal service the ship will carry meat, fruit and other perishables on the Australia/New Zealand–U.K. route and in the off-season she will be employed in carrying general cargoes on the U.S. and Canadian coasts and the New Zealand/Australia routes.

Mataura is a five-hold vessel with four holds forward of the machinery space and one aft. No. 1 hold is non-insulated and refrigerated cargoes are carried in Nos 2–5 holds. In addition, two stainless steel-lined liquid cargo tanks are arranged aft of No. 5 hold.

Propulsion of *Mataura* is by a Mitsui-Burmeister and Wain engine of the 948-VT2BF-180 type, rated at 20 700 bhp at 114 rev/min and which permits a service speed of 20 knots to be maintained. Bridge control is provided for the main engine but there is also a well equipped control station on the port side of the engine room upper platform. Equipment in this station includes a data logger made by the Japanese Tokyo Keiki Co. for monitoring main and auxiliary machinery and the refrigeration installation—*Motor Ship, July 1968, Vol. 49, p. 177.*

Polish Remote Control System

Polish engine builders are licensees for some of the leading West European-designed main propulsion engines and the recent development, by the Gdansk shipyard, of a machinery remote control system can be seen as a logical extension of the engine building process.

The system is primarily intended for Burmeister and Wain machinery and the first application has been made to a B. and W. unit of the 562-VT2BF-140 design. Development of the control system was undertaken by the Polish shipbuilding industry's Design and Research Centre, Gdansk, in close collaboration with the marine engine construction department of the Gdansk shipyard.

Operation is electro-pneumatic; the pneumatic section provides transmission of the manoeuvring orders, the basic logic circuits and the final control elements. Engine and turbocharger speed, signalling and change-over of manoeuvring position are electrically operated.

Air at 6 kg/cm² is employed for the reference pressure elements, two-position and actuation servomotors; logic elements are of the pneumatic, electrical and mechanical control types. Order transmission is effected by the control unit via a conventional engine order telegraph. Speed adjustment for half ahead and full ahead can be made by reference elements in the control circuit.

When starting the engine to a definite order, say slow ahead, the remote control system successively effects the following operations:

1) A servomotor moves the engine reversing lever to the desired direction.
2) The fuel control lever is moved to the start position.
3) The control signal value is given by a reference element.
4) Starting air is admitted.
5) When the engine and turbochargers attain the correct speed on air the fuel control lever is set to the position "starting fuel".
6) The engine speed is increased to the required revolutions. Should the initial starting attempt be unsuccessful, a further automatic attempt will be made. When reversing the engine a controlled amount of braking air is admitted to slow and stop the machine.

Change-over from either of the remote control positions to hand control may be effected at any time without affecting the engine performance. Emergency stop facilities are incorporated in the remote control facilities.

During testbed trials of the system on the B. and W.-type engine, recordings were made of various dynamic characteristics for different manoeuvring conditions. An analysis of these recordings showed that the remote control system enabled minimum times to be attained for the different manoeuvres with the maintenance of optimum conditions for operation. Air consumption with the remote control system proved to be slightly lower than that attained by careful hand control.—*Motor Ship, August 1968, Vol. 49, p. 244.*

Starting Sequence Controllers for Gas Turbines

The function of the start-up sequence control equipment is to make it possible to start the gas turbine power plant simply by pressing a button at the machinery control room console or on the bridge. For this to be possible it is necessary for a number of checks to be made before the starting sequence is initiated and as it takes place. If any of the pre-start checks reveals that something is wrong, the starting sequence cannot be initiated and an appropriate indication is given on the console. If the checks are satisfactory the pressing of the button starts a 30-second sequence which proceeds automatically until the engine is running at normal idling speed. If any of the steps in the sequence is not completed satisfactorily, the starting attempt is automatically terminated.

Actuation of functions on the engine is generally pneumatic with electrical control signals—which can be expressed as "pneumatic muscles and electric nerves". The proving of functions is generally by microswitches, pressure transducers and tachogenerators. This panel operates with conventional electro-magnetic relays of Ministry of Defence approved design. When the starting sequence is complete and the engine is running satisfactorily, it comes under manual throttle control and interruption of and subsequent establishment of the electric supply does not affect its operation. This arrangement, which is appropriate where the engines are in unskilled hands, and unwarranted shut-down under emergency conditions might be disastrous, is described as a "fail-set" system. The start up sequence cannot be initiated again until a stop button has been depressed.

If the gas generator fails to light up within 30 seconds, the attempt to start is automatically abandoned and the effect is the same as if a stop button had been depressed.

While the gas turbine unit is running, a number of important functions are monitored continuously and arrangements made for faults to be indicated in the form of a master flashing light and audible warning, with an individual fault indicator light when pre-set limits are exceeded. There is provision for alarms and trips to be accepted, in which case the master warning light stops flashing and becomes steady and the audible warning is muted while the fault is investigated. If the fault is cured the reset button is operated and the panel returns to a dark presentation. Operation of the reset button against an accepted but unrectified fault will cause the fault display to show again.

The Industrial and Marine Controls Division of the Vosper Thornycroft Group have supplied relay panels for the complete start up sequence control of the Rolls-Royce Olympus gas turbine units for a number of vessels building for foreign navies.—*Reed's Marine Equipment News, June 1968, Vol. 12, p. 18.*

Multi-purpose Dry Cargo Ship with Geared Machinery

Built by the Govan Division of Upper Clyde Shipbuilders Ltd. to the order of Sir William Reardon Smith and Sons Ltd., *Welsh City* is a sophisticated and highly versatile form of what is generally termed a tramp ship. The design of this vessel and a sistership, at present under construction at the Govan yard, has resulted from the close co-operation between the owners and builders. The initial requirement was for a closed shelter deck type vessel of around 15 000 dwt, to be achieved on a 1930 Convention draught of 28 ft 6in. However, since the placing of the order, the new 1966 load line rules which permit an increase in deadweight as shown in the table below have come into force.

Principal particulars:

Length, o.a.	499 ft 6 in
Length, b.p.	475 ft 0 in
Breadth, moulded	72 ft 0 in
Depth, moulded to upper deck ...	40 ft 5 in
Depth, moulded to main deck ...	30 ft 0 in
Load draught (1930 Convention)	29 ft 0⅞ in
Corresponding deadweight ...	15 570 tons
Load draught (1966 Convention)	29 ft 10⅝ in
Corresponding deadweight ...	16 223 tons
Corresponding displacement ...	21 223 tons
Gross register	10 790·18 tons
Net register	7053·38 tons
Speed, service	16 knots
Speed in ballast condition ...	17·9 knots

During the design stage of the vessel, alternative plans were drawn up on the basis of slow-speed direct drive and medium-speed geared drive installations, bearing in mind the strict limitation on the cost of the ship, believed to have been about £1½ million. After careful consideration of the technical and commercial factors involved, it was decided that a medium-speed installation would be the most advantageous, giving considerable savings in weight and space: it is claimed that an estimated 70 000 ft^3 of additional cargo space has been made available at the aft end of No. 5 hold.

The main propulsion engines of *Welsh City* are two nine-cylinder units of the Ruston AO design and main particulars are as follows:

Maximum continuous rating ...	4500 bhp
B.m.e.p.	150 lb/in^2
Corresponding engine speed ...	450 rev/min
Overload setting	4527 bhp
Temporary load and no load speeds	400 and 480 rev/min
No load maximum speed ...	470 rev/min
Idling speed	180 rev/min
Cylinder bore	14¼ in
Piston stroke	18½ in
Fuel cons. (fuel not less than 17 500 Btu/lb)	0·355 lb/bhp-h
Total weight of each machine approximately	39 tons

The engines, which are of the two-stroke, uniflow-scavenged type, are bolted directly to the mounting plates. Each engine drive is taken, through a flexible coupling, to a Renk twin input/single output gear-box which incorporates oil-operated two-stage clutches connected to the primary pinions. The overall reduction ratio is 4·85 to 1, giving a propeller shaft speed of 93 rev/min at full speed and thus the designers were able to take advantage of the higher efficiency offered by the low shaft revolutions.—*Motor Ship, August, 1968, Vol. 49, pp. 234–238.*

World's Largest Refrigerated Cargo Liner

The new Port Line twin-screw motor refrigerated cargo liner *Port Chalmers,* recently delivered by the Linthouse (Alexander Stephen) division of Upper Clyde Shipbuilders, is the largest by far and one of the fastest vessels of her type to go into service.

Designed particularly for the carriage of palletized and unitized loads, every effort has been made to make cargo space as regular in shape as possible and to avoid obstructions which could hamper fork-truck operations. All decks are parallel to the baseline, except for the fore end of the upper deck, and without camber, again with the weather deck as the only exception.

Apart from Grace Line installations, *Port Chalmers* is the first ship to be equipped with Cargo Dynamics "Screw-Torq" hatch covers. Of the vessel's 17 hatches, all but six weather deck openings, fitted with Trans-Roto mechanized covers, have this new marque constructed of AH2 high-tensile steel; apart from No. 7 second deck hatches are fully insulated.

The 11 sets of "Screw-Torq" covers are arranged as 33 pairs, with single or double pairs stowing at either end of the hatches which measure up to 43 ft in length and 26 ft in width. One section of each pair contains the small electric motor and gear-box used for driving the two halfshafts which are in turn connected to two worm gear jacks. The other section of the pair is connected to the worm gear jacks—and to the other panel—by means of a special linkage arrangement. When the jacks are extended the hatch covers fold upwards and backwards into the stowed position, the opposite applying when the jacks are retracted.

Port Chalmers is powered by a twin Clark-Sulzer type RD90 Diesel installation, each unit of which has an m.c.r. of 13 000 shp and a normal service output of 12 000 shp.

An automatic remote control of the propulsion plant from the bridge is provided by way of a development of the shipbuilder's Mahout system. Improvements include simplification of the mechanical controls on the engines which are only for emergency use and a starting air valve control device which has shown 4 per cent savings of starting air in recent installations aboard the recently completed *Majestic* and *Britannic*.

Woodward synchro-phasing gear is fitted to ensure the engines run at exactly the same speed and in a fixed phase relationship in order to minimize vibration—massive shaft bracket construction has allowed the natural bossing vibration frequency to be tuned to a point well above that of propeller excited forces.—*Shipbuilding and Shipping Record, 21st June 1968, Vol. 111, pp. 853–858.*

Electromagnetic Log

With this log a bronze probe, similar in size to that of the pressure log, extends below the hull. An electromagnet is embedded in a glass fibre shell at the head of the probe and is energized by a 50 Hz a.c. which sets up a magnetic field in the vicinity. As sea water starts to move past the rodmeter it cuts these lines of force and generates a small voltage in the water. The small voltage is picked up by two bronze electrodes, in contact with the water and fitted on each side of the rodmeter head, and fed to the speed and distance transmitter. The faster the water moves past the rodmeter the greater the voltage generated: about $\frac{1}{2}$ μV/knot.

The speed and distance transmitter unit consists of a cast aluminium cabinet which includes the power supply. The output from the sensing head is fed to two amplifiers and thence to the speed motor which drives a worm shaft at one rev/min-knot which, in turn, drives:

a) the speed indicator;
b) the speed transmission system;
c) the distance potentiometer;
d) the resetting device which cuts off the speed motor when it reads the correct amount.

The output of the distance potentiometer, proportional to the ship's speed, is fed to an amplifier and thence to the distance motor which mechanically drives the following:

a) the distance tachometer to control the movement of the distance motor in relation to the amplitude of the signal from the distance potentiometer;
b) the distance indicator;
c) the distance transmitters.

The compact speed and transmission unit is powered by 115 V or 220 V 50/60 Hz a.c. and a stabilized d.c. for transmissions.

Since the flow characteristics of the hull influence the accuracy of the system each log must be calibrated for a particular ship.—*Shipbuilding and Shipping Record, 28th June 1968, Vol. 111, p. 893.*

Danish-built Twin-screw Ferry

Built by Aalborg Vaerft A/S, the ferry *Christian IV* has recently been delivered to her new owners, A/S Kristiansand Dampskipsselskap, of Kristiansand, Norway. She is intended as a versatile ferry capable of transporting trains, cars and passengers on a service between Kristiansand and Hirtshals. Four B. and W. Diesel engines give the ship a service speed of about 18·5 knots.

Christian IV has been constructed to comply with the rules of Det norske Veritas and is strengthened for navigation in ice. The ship complies fully with the requirements of the International conference for safety of life at sea, 1960, and features a cruiser stern and a raked stem of the *Clipper* type. There are two continuous decks, with a three-storey deck-house situated above the uppermost continuous deck and extending over three-quarters of the length of the ship. The ship is divided by 12 watertight bulkheads which are continuous up to the level of the car deck. The funnel amidships is used solely for ventilation purposes. For removal of exhaust gases from the main and auxiliary engines there are two funnels mounted above the engine casing, port and starboard.

Principal particulars:

Length, o.a.	285 ft 5½ in
Breadth	51 ft 8 in
Draught, loaded	13 ft 4 in
Depth to car deck	17 ft 10½ in
Free height on car deck	13 ft 9½ in
Normal car capacity	100 cars
Peak load car capacity	145 cars
Machinery output	1980 bhp at 600 rev/min
Speed	about 18·5 knots

The propelling machinery in *Christian IV* consists of four B. and W. vee-type 1226-MTBF-40V 12-cylinder four-stroke single-acting trunk Diesel engines with exhaust gas turbocharging. In normal service each engine develops an output of 1980 bhp at 600 rev/min. Maximum continuous output is 2180 bhp at 620 rev/min. The engines are coupled in pairs by Pneumaflex couplings and drive two KaMeWa variable-pitch propellers through Lohmann and Stolterfoht double-reduction gear-boxes. The gear ratio is 1:0·25. Twin spade rudders are fitted. For good manoeuvrability at the ferry berths, the vessel has been fitted with a bow thruster unit driven by a 600 hp motor and capable of developing a thrust of 7 tons.—*Shipping World and Shipbuilder, May 1968, Vol. 161, pp. 839–841.*

German-built Bulk Cement Carrier

Towards the close of last year, Deutsche Werft AG, Hamburg, delivered the bulk cement carrier *Cementia* to Cement Tankers S.A., Panama, for their service between storage dumps on the East African mainland and the offshore islands.

Built to the requirements of Lloyd's Register class ✠100.A1, the hull has a raked stem and transom stern, forecastle and poop and has the all-aft arrangement of bridge, accommodation and machinery.

For loading and discharging, hoses are used to connect the shore installation with the fixed pipe system in the vessel. The cement/air mixture flows along the high velocity pipe system terminated in the deckhead of each hold compartment and is disposed by a distributor. The surplus air used in delivery is constantly extracted through a filter and, during loading, the holds are continuously kept at lower than atmospheric pressure, the difference in working pressure corresponding to a water head of 50 mm. By this means the cargo is de-aerated to such an extent that subsequent settling is hardly noticeable.

Discharging is effected by mixing air with the bulk cement so that it possesses fluid properties and can be passed through a pump system. For this purpose there is the normal longitudinal vibrator, with a jute cover to allow the passage of air, and the cargo flows to the central tunnel containing the discharging pumps. From here the aerated cement is passed through the vertical pipes to the deck and to the shore.

Main propulsion is by two eight-cylinder Deutz type RBV8M358 turbocharged Diesel engines, each with an output of 2000 bhp at 300 rev/min, driving a single shaft through a twin-input/single output gear-box. The propeller was supplied by Zeise and has a diameter of 13 ft 9¼ in.

Both the main engines and the disc clutches and shaft brake are pneumatically controlled from the local control position in the engine room. Hydraulic disc clutches enable the main engines to be disengaged so that one can run ahead and the other astern during manoeuvring and the shaft brake is applied during this operation. The couplings driving the alternators can only be engaged from the start position as they are otherwise blocked against the disc clutches.—*Shipbuilding and Shipping Record, 5th July 1968, Vol. 112, pp. 9–12.*

Passenger/Car Ferry

Claimed as the fastest passenger/car ferry on the Irish Sea service, the new 4230 grt vessel, *Munster*, is in service between Liverpool and Dublin for her owners, the British and Irish Steam Packet Co., of Dublin. With a reserve service speed of 22 knots, she will be able to clip three hours off the run during peak holiday periods and complete ten round trips each week.

The ship's stabilizing system designed by the architect is a controlled roll-damping tank. Full air conditioning has been provided throughout, operated from the control room on the bridge deck with individual control in sleeping quarters.

The four main engines, V9V M.A.N. Diesels each of 2840 hp at 428 rev/min are turbocharged and intercooled. Each pair of engines drives through Vulcan elastic couplings and Renk multi-plate clutches, twin gear-boxes connected by line-shafting to the twin KaMeWa propellers.

In the central engine control room the engines are fitted with overspeed and overload governors which work in conjunction with the pitch-setting devices in the propellers. The bow-thrust propeller, 600 hp with seven tons thrust, is also of KaMeWa manufacture and is driven by a Siemens motor.

Ballasting, trimming and bilging operations throughout the ship are carried out from the control room. It also contains the instruments for checking the level conditions in the double-bottoms of the ship and her two sewage tanks.

Munster carries 1000 passengers and 220 cars or a combination of cars and heavy vehicles. The vessel is scheduled for a one-hour peak period turnabout at the two terminals, both of which are also designed to handle large numbers in minimum time.—*Shipping World and Shipbuilder, June 1968, Vol. 161, pp. 921–923.*

Fluidic Bow Thruster without Moving Parts

A bow thruster with no moving parts was recently installed in a 35 ton, 30 ft long 12 ft wide steel barge. Full thrust reversal was achieved in three seconds with no evidence of water hammer. There was no measurable flow from the unused discharge port.

The Y-shaped thruster is four feet long and three feet wide and produced a bollard line pull of 1700 lb at a flow rate of 1000 gal/min, which was the limit of the test pump's capacity. With higher water flow rates the test unit is capable of producing 2500 lb of thrust.

The unit used in the tests was designed to be used in small craft such as towboats, offshore supply boats, buoy tenders or any craft where there is a requirement for a thruster.

The thruster consists of a Y-shaped fitting, rectangular in cross section, with one inlet leg and two discharge legs. When water flows equally from both discharge legs the thrust vector is zero. Thrust from either discharge leg is increased by increasing the water flow out of the leg. As the thrust from one leg increases the other decreases. Switching the flow from the neutral position to either port or starboard is achieved by a simple air control system—no mechanical controls are involved.

The division of flow is achieved as the jet strikes the flow splitter located downstream from the inlet nozzle. As the water jet issues from the nozzle, air is entrained on either side. This entrainment causes replenishment of air from the atmosphere through control lines on both sides. When one of the control lines is blocked, the replenishment air supply is cut off and continued air entrainment by the water jet produces an area of reduced pressure near the mouth of the closed control port. The pressure difference causes the jet to move to one side. Thus directional control is achieved.—*Undersea Technology, May 1968, Vol. 9, p. 99.*

Heat Transfer in Long Tube Vertical Falling Film Evaporator

Two slightly differing mathematical models were developed to describe the heat transfer in the long tube vertical falling film process. The process was investigated experimentally with a ¾-in diameter tube for various lengths up to 13 ft, for flow rate Reynold's numbers from 1000 to 13 000, for temperature differences of the order of 20°F and for vacuum conditions down to 16°F. A comparison of theoretical and experimental results was made and was found to be in good agreement. i.e. within 10 per cent.—*Kroll, J. E. and McCutchan, J. W., Trans. A.S.M.E. Jnl Heat Transfer, May 1968, Vol. 90, pp. 201–210.*

Determination of the Viscous Drag of a Ship Model

In recent years, experimental results obtained by means of the wake survey method have indicated that the viscous drag of a ship model varies with the Froude number; later studies have tended to confirm these findings. The present work was undertaken to establish the validity of this variation. In order to avoid the principal sources of inaccuracy of previous work, modifications to the equipment were made. Also, to obtain viscous drag, the authors applied a refinement of the Betz-Tulin formula.—*Tzou, K. T. S. and Landweber, L., Jnl of Ship Research, June 1968, Vol. 12, pp. 105–115.*

Crankshaft Stress Analysis and Bearing Load-carrying Capacity

The complex problems of stresses and deflexion in crankshaft elements and the changing conditions of bearing oil film are reviewed. A treatment of their evaluation is presented. It includes a number of previously unsolved complications, such as unsymmetrical cranks, flexibility of supports and oil film thicknesses.—*Porter, F. P., 12th–16th May 1968, A.S.M.E. DGP Conference and Exhibition Paper No. 68-DGP-8.*

Fore and Aft Vibration of Superstructure

Most of the very large vessels recently built in Japan are of a type with the entire superstructure aft and their bridge has to be positioned as high as possible for navigational reasons. On these ships, troublesome fore and aft vibration of the superstructure has sometimes occurred due to resonance with the so-called blade frequency. In order to avoid this, it is necessary to investigate the vibratory characteristic of superstructures of this kind. The article describes the vibratory characteristics obtained from full scale measurements.—*Hirowatari, T. and Matsumoto, K., Japan Shipbuilding and Marine Engineering, March 1968, Vol. 3, pp. 11–21.*

Special Design Features, OSS o1 Oceanographer and OSS o2 Discoverer

Special construction features required by oceanographic research ships are described in this paper. They are grouped into categories relating to the following: atmosphere, surface (mid-water, bottom, sub-bottom) and general. Other overall construction features which assist ships in their performance in the foregoing activity fields are also described. These features are position fixing, position holding and control, operations control centre, stability, open deck, anti-roll tank, etc.—*Powell, A. L. and Stover, H. B., Marine Technology, July 1968, Vol. 5, pp. 207–229.*

Gas Turbine Propulsion for High-speed Small Craft

This paper presents the results of marine gas turbine propulsion studies related to small craft of the high-speed planing hull type. These include appraisal of craft performance requirements for both transportation and patrol mission profiles. A substantial increase in both service speed and craft productivity may be realized with gas turbine propulsion.—*McCoy, A. W., Marine Technology, July 1968, Vol. 5, pp. 232–248.*

Analysis of Assumed Mooring Arrangement for Maritime Class Ship

An assumed but routine mooring arrangement for a standard Great Lakes ship—in this case a *Maritime* Class vessel—is analysed to ascertain what wind conditions would be sufficient to establish the sequence necessary to cause parting of one line, followed by parting of the second and third lines and, finally, by full failure of the mooring arrangement. Wind tunnel tests on a model of the ship are reviewed. —*Horton, J. L. and Yagle, R. A., Marine Technology, July 1968, Vol. 5, pp. 257–266.*

Theoretical Predictions of Characteristics and Cavitation Properties of Propellers

In this report the status of the vortex theory of propellers is briefly outlined and some examples are given of the rate of success attained when applying calculation schemes of different degrees of accuracy to the solution of different design and analysis problems. The possibilities of predicting the relative rotative efficiency by calculations is discussed and the results of some quasi-steady calculations of thrust and torque variations are compared with experimental results. —*Johnsson, C. A., 1968, Swedish State Shipbuilding Experimental Tank Pub. No. 64.*

Computer Definition of Ship Characteristics

Computer application is a new cost element in ship design; thus, consideration should be given to developments that will render the computer a useful and cost-effective tool to the industry. Cost-effectiveness compels the analyst and the designer to distinguish between mere use which they should avoid and usefulness which they should seek. The increasing trend to employ systems analyses to ship design is greatly facilitated by using computers.—*Boumis, T. P., Marine Technology, July 1968, Vol. 5, pp. 288–306.*

Study on Boiler Efficiency Control

In this report, the practical application of the Gradient Method to the boiler efficiency control is treated both experimentally and theoretically. The experimental studies were made on a monotube boiler equipped with a digital computer which was programmed to measure the boiler efficiency and keep it automatically at its maximum value.—*Terano, T., Kurosu, K., Murayama, Y., Okumura, K., Wada, T. and Kobayashi, M., Report of Ship Research Institute, Tokyo, January 1968, Vol. 5, No. 1.*

Experimental Verification of a Design Method for Ducted Propellers

A method for the design of ducted propellers has been developed at the Swedish State Shipbuilding Experimental Tank. Starting from known values of total thrust, number of revolutions, propeller diameter, blade number and form, distribution of blade circulation, duct vorticity, minimum cavitation margin, etc., the method determines propeller efficiency, blade area, duct thrust, shape of duct and pitch camber and thickness of propeller blade sections. A series of tests in homogeneous flow has been carried out.—*Dyne, G., 1968, Swedish State Shipbuilding Experimental Tank, Pub. No. 63.*

Britain First with Computerized Fisheries Research Vessel

The Department of Agriculture and Fisheries for Scotland is to equip its fisheries research trawler F.R.S. *Explorer* with an Elliott-Automation Hydroplot shipboard computing system for experiments with fishing gear and other marine research applications. It is claimed to be the first such vessel in the world to have a computer on board. The Hydroplot system will automatically record information from more than sixty sensing devices attached to fishing gear, the engine and the vessel's navigation system.—*Measurement and Control, June 1968, Vol. 1, p. 225.*

Underwater Power from Aluminium

A thermochemical power system having the highest theoretical performance (hp–h/ft^3 propellants) and particularly suited for underwater applications (power, heat and propulsion) is described. In this system aluminium is reacted exothermally with sea water to produce gaseous exhaust products, hydrogen and steam. These are expanded through a prime mover or a combustion process can be added before the prime mover to convert the hydrogen to steam.—*Bobb, F. H., Trans. A.S.M.E., Jnl Engineering for Industry, May 1968, Vol. 90, pp. 255–260.*

Toothed Couplings—Analysis and Optimization

The paper is a theoretical analysis of a toothed coupling. It shows the inherent necessity of backlash and its optimal value; it presents a critical comparison of all the existing designs and those proposed by the author, based upon:

a) maximum and total lag angle between the shafts and a function of misalignment;
b) maximum backlash that may appear between the teeth;
c) angles at which the teeth are in contact;
d) power losses;
e) type of contact stresses.

Moked, I., 12th–17 November 1967, A.S.M.E. Winter Annual Meeting Paper No. 67-WA/DE-3.

Refrigerated Sea Water Trawler

Bolsönes Verft recently completed the motor fishing vessel *Selvag Senior* for Alfred og Odd Söheim. This vessel incorporates several unusual features in that it is virtually a combination of a purse seiner and tanker and is equipped with two transverse thrust units, one forward and one aft.

The catch is stored floating in sea water refrigerated at a temperature of 0°C(32°F), this method of storing offering the advantages of eliminating crush damage to the fish during transportation. Experience of this storage system in Canada and the United States of America has shown that the fish will stay fresh from 8–20 days without any deterioration of quality. There are nine storage tanks having a total capacity of 27 192 ft^3 and the refrigerated sea water is circulated by a system designed and developed by the builders.

Propulsion is by a 12-cylinder Nohab-Polar type SF112VS Diesel engine, rated at 1800 bhp at 750 rev/min, driving a Liaaen controllable pitch propeller at 300 rev/min through a Lohmann ad Stolterfoht gear-box.—*Shipbuilding International, May 1968, Vol. 11, p. 16.*

Gas Turbines v Steam Reliability Analysis for Warship Propulsion Plant

Methods of mathematical analysis are reviewed for defining and numerically computing the reliability of warship propulsion systems. Experimental analysis is made of a twin-screw all gas turbine plant design for a small warship and the results are compared with computed figures previously published for a geared steam turbine plant.—*Benn, D. H., 17th–21st March 1968, A.S.M.E. Gas Turbine Conference and Products Show Paper No. 68-GT-9.*

Hydromechanical Fuel Control for Portable Gas Turbine Generators

Controls for small lightweight gas turbines present some unique design problems. The requirements for small size, light weight, ability to rotate at high speeds to save reduction gearing and low production cost conflict with the requirements for reasonably accurate control of very small fuel flows and the scheduling of a wide range of hydrocarbon fuels over a wide range of ambient temperatures. The author discusses the design of such a control and the satisfactory results obtained.—*Parker, G. E., 17th–21st March 1968, A.S.M.E. Gas Turbine Conference and Products Show Paper No. 68-GT-43.*

Robertson Crack Arrest Test

For almost twenty years, the Robertson test, which determines the threshold conditions of nominal stress and temperature for brittle fracture of steel plates, has been developed in various laboratories in the U.K. Although the principles of the test and the methods used to assess results are widely known by steelmakers and users, there has been some confusion as to the interpretation and there have been variations in test procedure. The present report which has been sponsored by the Navy Department Advisory Committee on Structural Steels, Group C, describes the test methods used.—*British Welding Jnl, August 1968, Vol. 15, pp. 387–394.*

Toughness in Electroslag Welds

The embrittlement associated with the coarse grained regions of heat affected zones of electroslag welds has been investigated with regard to the extent to which toughness can be recovered by post weld treatment. Preliminary results show that post weld normalizing greatly improves the toughness but in steels of a composition suggesting a high susceptibility to burning this recovery is only partial.—*Bentley, K. P., British Welding Jnl, August 1968, Vol. 15, pp. 408–410.*

Measurements of Boundary Layers of Ships

This paper is the second report on the results of experimental investigations initiated by the Research Institute for Applied Mechanics, Kyushu University. The work described was carried out on the fishing training ship *Nansei Maru*. Similarly to the Wakasugi tests, the three kinds of experiment undertaken were:

1) measurement of velocity profiles in the boundary layer;
2) measurements of turbulence in the boundary layer;
3) visualization of the wake including the separated flow.

Bulletin of Research Institute for Applied Mechanics, Kyushu University, 1967. No. 28.

Strain, Stress and Flexure of Two Corrugated and One Plane Bulkhead Subjected to a Distributed Lateral Load

Two corrugated and one plane bulkhead subjected to a distributed lateral load were tested and most of the results are given and discussed. An extension of the beam model is introduced and verified for certain cases. The assumption underlying the method of calculating the transverse bending moments and stresses in corrugated panels is verified with the aid of experimental data gathered both in the laboratory itself and from previous papers by other authors.—*Jaeger, H. E. and Van Katwijk, P. A., International Shipbuilding Progress, July 1968, Vol. 15, pp. 223–250.*

Hydrodynamic Coefficients for Swaying, Heaving and Rolling Cylinders in a Free Surface

The hydrodynamic coefficients of two-dimensional cylinders are determined for various cross-sections by forced oscillation tests and by theoretical computations. The purpose of this study is to check the theoretical basis of the computation for all three possible modes of motion and to establish the influence of section shape in this respect.—*Vugts, J. H., International Shipbuilding Progress, July 1968, Vol. 15, pp. 251–276.*

Measurement of Transient Vibrations

A method for employing commercially available solid state telemetry transmitters and receivers as a torsiograph for the measurement of transient vibrations is described. Typical traces are shown and equations are given for their interpretation.—*Herrman, A. S., 12th–16th May 1968, A.S.M.E. DGP Conference and Exhibition Paper No. 68-DGP-13.*

Evaluating Wear of Cylinders and Piston Rings by Quick Spectrographic Sampling Method

Small samples of the oil which lubricates the cylinder wall, piston and rings were drawn off through a small hole in the cylinder wall. The samples then were analysed spectrographically. This method of evaluating cylinder and piston ring wear proved to be quick, simple and inexpensive. It was found useful especially in cases of drastic differences in wear rates.—*Wisniowski, H. U. and Jackson, D. R., 12th–16th May 1968, A.S.M.E. DGP Conference and Exhibition Paper No. 68-DGP-1.*

Stratification and Combustion in Reciprocating Engines

The major differences in the combustion characteristics of the Otto, Diesel and stratified charge cycles are discussed. For the Otto and Diesel cycles, there is a definite fixed fuel requirement for each. The Otto cycle utilizes thermally stable gasoline and the Diesel cycle requires a less stable and easily ignitable fuel. In contrast, the stratified charge engine displays a high degree of insensitivity to fuel quality and can be operated on most commonly available hydrocarbon fuels.—*Wizky, J.E. and Clark, J. M., 12th–16th May 1968, A.S.M.E. DGP Conference and Exhibition Paper No. 68-DGP-4.*

Experimental Correlation Between Rate of Injection and Rate of Heat Release in a Diesel Engine

The primary objective of this paper was to obtain a relationship between the rate of injection and the rate of heat release. Results computed from experimentally obtained engine data are presented. These data have been correlated and expressions for these correlations are given.—*Shipinski, J., Uyehara, O. A. and Myers, P. S., 12th–16th May 1968, A.S.M.E. DGP Conference and Exhibition Paper No. 68-DGP-11.*

Criterion for Evaluating Diesel Engine Performance

By measuring engine power output, fuel consumption and air temperature to the engine, a quality factor can be established for evaluating the performance of a Diesel engine. A simplified diagram and equations are developed for the ideal limited pressure cycle and are applied to test data.—*Froehlich, K. F., 12th–16th May 1968, A.S.M.E. DGP Conference and Exhibition Paper No. 68-DGP-15.*

Matching of Exhaust Turbochargers to Two-cycle Diesel Engines

The mean effective pressure of marine Diesel engines is apparently increasing. Along with the improvement in the technique of matching turbochargers to Diesel engines, this paper describes the step-by-step calculation method of engine performance using the digital computer in considering the characteristics of the exhaust turbocharger.—*Izumi, S., Omotehara, I., Yano, T. and Kushiyama, T., 12th–16th May 1968, A.S.M.E. DGP Conference and Exhibition Paper No. 68-DGP-9.*

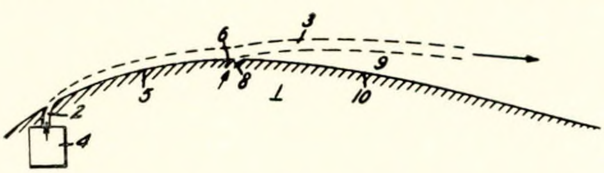
Engineering Design of Oil-free Internal Combustion Engines

Development of engineering design approaches for the use of solid lubricants in mechanical systems is described. The outlined experiment primarily concerned initial work related to translating basic research findings to critical areas of an internal combustion engine. The findings also enable a more precise insight into the numerous technical advantages that can be realized from solid lubricants, especially where lubricating greases and fuels are non-functional.—*Cerini, J. P., Devine, M. J., Lamson, E. R., Bowen J. H. and Abbott, R. L., 12th–16th May 1968, A.S.M.E. DGP Conference and Exhibition Paper No. 68-DGP-16.*

Patent Specifications

Drag Reduction of Water-borne Vehicles

The invention consists of a method for reducing drag on water vehicles and comprises the steps of adding a drag reducing liquid to the water proximate the surface of the vehicle. In the figure is shown a cross-sectional view of a ship employing this method. A suitable distribution outlet (2) or other port is positioned adjacent the front end, bow, of a water vessel, such as the ship (1). Drag reducing additives, premixed with a water solution, are stored or prepared in a



suitable tank (4), or other container which communicates with the opening (2). Particularly suitable additives are guar gum, locust bean gum, carrageenan or Irish moss, gum karaya, hydroxyethyl cellulose, sodium carboxymethyl-cellulose, polyethylene oxide, polyacrylamide and polyvinyl-pyrrolidone. Underwater as well as surface vehicles can make effective use of the drag reduction techniques. With surface vehicles, the drag reduction would take place along the underwater surface areas and with underwater vehicles along all the surface areas. Even vehicles such as hydrofoils and others, where only a slight area is in the water, can make effective use of drag reduction in the underwater surface areas.—*British Patent No. 1 119 152 issued to General Electric Co. Complete specification published 10th July 1968.*

Watercraft Propulsion Pump

An axial flow watercraft propulsion pump is designed to change the velocity of water passing through the pump to provide a force tending to move the boat. The velocity of the water is changed by increasing the total pressure head of the water as it moves through the pump and then by causing the water to be expelled through an opening of restricted size. Referring to Figs 1, 2, 3 and 4, a boat (10) includes a hull (11) provided with an intake opening (12) communicating between the bottom of the boat and a propulsion pump (13). The pump is driven by an engine (14) and pumps water from the intake (12) to the stern (20) of the boat. The water is formed into a jet stream by a converging surface (17) of an annular water discharge member (18) being fixed to the pump (13) and to a booster housing (21). The pump (13) includes a shaft (22) which is connected by a universal joint (24) to the shaft (15) and which rotates rotors (25) and (26). A spacer tube (27) acts as a bearing for the shaft (22). Fixed to the spacer tube (27) is an annular member (31) of frusto-conical external shape which tapers from a smaller upstream diameter to a larger downstream diameter at its end (36). The rotor (26) includes the hub (32) of frusto-conical external shape with the forward end (35) of the same diameter as the end (36). The rotor (26) also includes blades

(37) of the configuration as shown in Fig. 3. A conical fairing (111) is mounted on a member (101) by means of a fluid motor (115) with a piston rod (116). The conical fairing can be retracted to the illustrated solid line position for normal operation of the pump to form a jet stream. The conical fairing can be moved by the motor to the projecting dotted

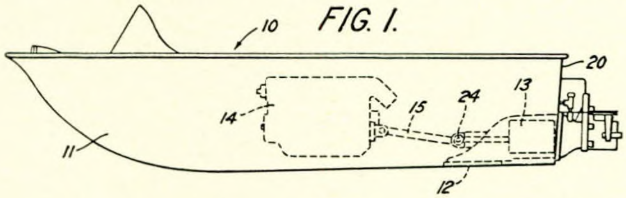


FIG. 1.

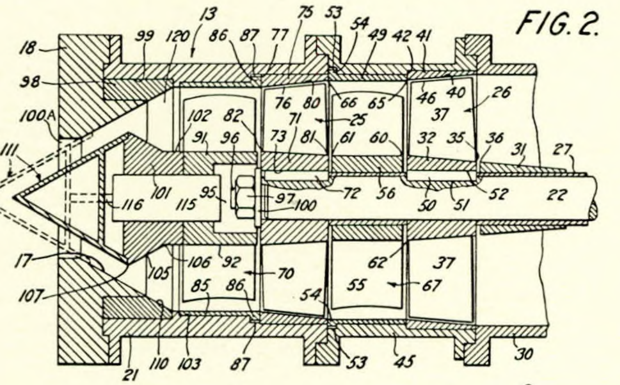


FIG. 2.

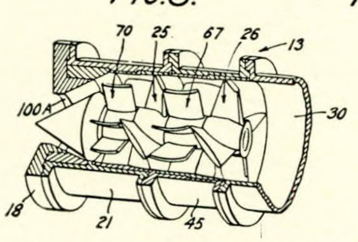


FIG. 3.

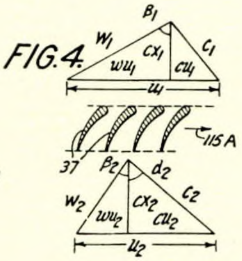


FIG. 4.

line position to shut the orifice (100a) and cause the pump to produce water for various applications such as fire fighting. In such an application, the pump would be provided with an outlet opening leading from the water passage area (120) possibly through the wall of the booster housing (21).—*British Patent No. 1 119 687 issued to Buehler Corp. Complete specification published 10th July 1968.*

Underwater Suction Anchors

This invention relates generally to anchors and more particularly to manoeuvrable anchors which are attached to submerged objects. In Fig. 1, the anchor comprises a housing (10) which encloses a motor (12) in a watertight chamber between bulkheads (14) and (16). The motor (12) is connected to a centrifugal pump impeller (18) in an impeller housing (20) in the forward portion of the device through a pump shaft (22) mounted in a bearing (24) in the bulkhead (16). An annular elastomeric seal (30) is concentrically disposed on the forward wall of the housing (10) around the pump inlet (26) and serves to provide a suction attaching means for the anchor. In the operation of the device, the anchor is lowered into the water from a mother ship (54) (Fig. 2) and the motor (12) is energized. Water is drawn in through the inlet (26) and is expelled through the outlets (28) in the form of high velocity jets. The anchor (8), propelled through the water by the jets, is manoeuvred by change in orientation of the tail vane assembly by energizing the

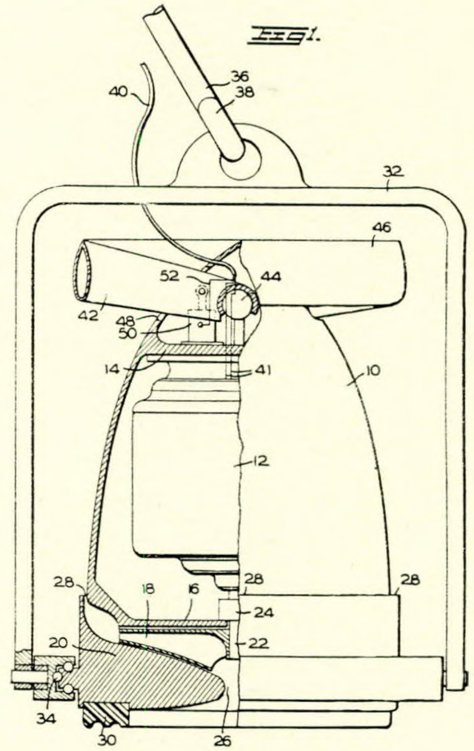


FIG. 1.

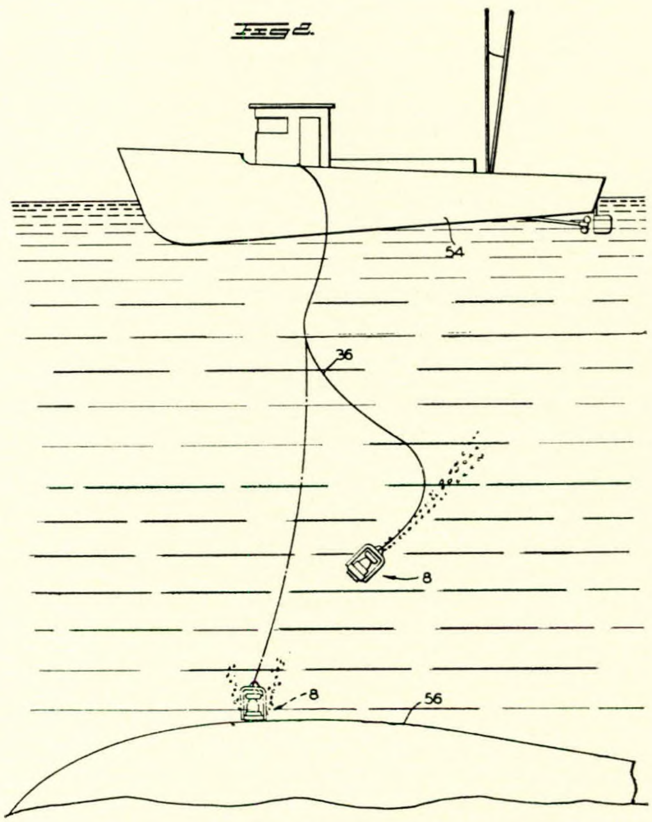


FIG. 2.

actuator (50) as required. When the seal (30) makes contact with a smooth surface, such as the submerged ship (56), for example, the pump, in withdrawing water through the intake (26), creates a low pressure region with the space defined by seal (30) and holds the anchor firmly on the surface.—*British Patent No. 1 120 914 issued to Hydronautics Inc. Complete specification published 24th July 1968.*