SELECTION OF A SHIP'S STABILIZER

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The paper is intended to deal with the practical aspects of the selection of a stabilizer for a specific type of vessel, providing the shipowner or operator with guidelines to help him in his selection.

After defining the term waveslope, or waveslope capacity, the authors describe fin and tank stabilizers and their performance.

To facilitate quick appraisal of suitable types of installation, the choice of stabilizing equipment is discussed under class of ship: passenger liners and cruise vessels; fast cargo liners, container vessels and tankers; car ferries and cross-channel ferries; naval vessels; trawlers and fish factory ships; yachts, and specialized craft.

The authors describe means of evaluating the effectiveness of a stabilizer installation and conclude with a reminder that, to obtain a successful solution to stabilization problems, each case must be examined on its merits.

INTRODUCTION

Several papers by various authors in this country and abroad have been published on the theory and mechanical and control aspects of both fin and tank stabilizers, all indicative of the fact that some form of ship roll stabilization has been accepted by the majority of shipowners and operators as a standard piece of equipment in modern vessels. A ship possesses six degrees of freedom i.e., pitch, heave, roll, yaw, sway and surge (see Fig. 1)

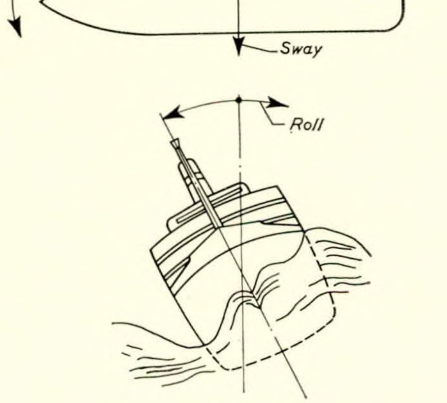

FIG. 1—*Ships motions*

of which pitch, heave and roll are probably the more important from the view of ship and cargo safety and passenger and crew morale. Of these three motions, only roll can be dealt with effectively in practice. Several attempts have been made at pitch stabilization, without success, on account of the inherently large moments involved.

This paper is intended to deal with the more practical aspects of the selection of a stabilizer for a specific type of vessel; dealing with those aspects of special interest to the shipowner or operator, and providing him with guidelines to assist him in his selection of the most effective stabilizing means for the class of ship under consideration. It is of particular value to review the question of stabilization when the ship is still at the design stage, when the selection can be made with due consideration given to all the design parameters of the ship, including cargo economics and comfort.

GENERAL

When the rolling of ships or roll stabilizers are under consideration, the term waveslope or waveslope capacity occurs over and over again, so that it is not out of place to redefine this term once more. The maximum slope of the wave (Fig. 2),

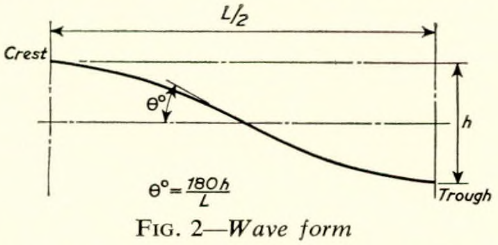

FIG. 2—*Wave form*

which has a crest to trough height h and a length $L/2$ is $\theta°$ and

$$\theta° = \frac{180 \times h.}{L}$$

The free roll of the vessel is determined by the product of the waveslope, produced by a given sea state, and an amplification factor K, which varies with type of ship, breadth to draught ratio and speed. Such data is recovered from ship model tests, but up till now very little has been published and made available to the

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general practitioner, although ship model rolling tests have been carried out in many ship model basins especially since the advent of the popularity of ship stabilizing equipment.

When a ship is rolling in a seaway, the couple applied by the sea is opposed by the couples dependent on the roll of the vessel and the derivatives of roll, velocity and acceleration. At resonance the roll angle and acceleration couple are equal and opposite, leaving the velocity couple as a damping couple to balance the sea couple. This damping couple is approximately 90° in advance of the roll angle. In a stabilized ship the stabilizer provides most of the stabilizing couple, as a small residual stabilized roll will only result in a small damping couple from the ship.

The power of the stabilizer to meet this requirement is given by:

$$\frac{\Delta\, GM\, \alpha^\circ}{57{\cdot}3}$$

where: Δ = displacement of the vessel;
GM = metacentric height;
α° = equivalent waveslope capacity, or "stabilizer power";

$$\text{or, waveslope capacity} = \frac{\text{stabilizer righting moment} \times 57{\cdot}3}{\Delta \times GM}$$

with a fin stabilizer:

the righting moment $= \frac{1}{2}\rho V^2 C_L A \times 2 \times$ righting arm;
where: C_L = lift coefficient;
A = area of one fin;
V = ship speed;
ρ = mass density;

and, with a tank stabilizer:

the righting moment $= W \times X$;
where: W = weight of fluid transferred;
X = displacement of C.G. of fluid transferred.

PERFORMANCE

Fin stabilizers are often designed for a 5° waveslope capacity, which, associated with an amplification factor of 6 to 7 for a medium size ship at an average speed of around 20 knots, would result in a total free roll of 30° to 35°. However, it should be emphasized that 5° is not a magic figure, but varies with the factors already mentioned and the service in which the vessel is engaged.

A fin stabilizer, with a power equal to the waveslope of the sea, will reduce the free roll to a residual value limited by the sensitivity of the sensing equipment. Residual rolls of 2° to 3° out to out are normal with full fin stabilization, thus yielding percentage roll reductions in excess of 90 per cent. When the fin stabilizer has a power less than the sea power, optimum results are not achievable for obvious reasons. That part of the sea power not cancelled by the stabilizer remains available to roll the vessel.

In the case of tank stabilizers, because of the limitations imposed by weight of fluid and loss of stability due to free surface effects, the stabilizer power is reduced to about 2° to 3° and therefore a lower roll reduction has to be accepted from the outset. In all cases, when percentage roll reduction is stated, the residual stabilized and the free unstabilized roll should be given, otherwise the percentage figures are valueless. With tank stabilizers in the range 2° to 3°, average roll reductions to be expected, range from 35 per cent to 50 per cent respectively.

TYPES OF STABILIZERS AND ROLL DAMPERS

The cheapest and oldest type of roll damping equipment are the bilge keels which, if fitted over 30 to 35 per cent of the length of the ship and having sufficient depth, can provide a reasonable degree of damping. This is shown in Fig. 3 for a medium sized

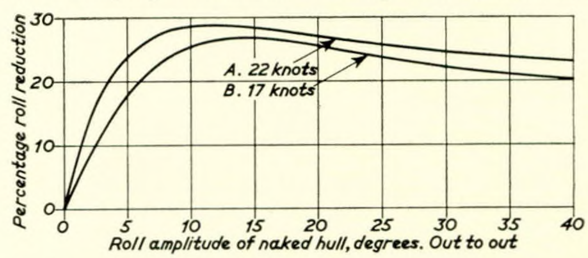

FIG. 3—*Percentage roll-reduction values for bilge keels deduced from ship model tests—Vessel A—passenger liner 630 ft length, length of keels 32 per cent ship's length—Vessel B—intermediate cargo-passenger vessel, 500 ft length, length of keels 30 per cent ship's length*

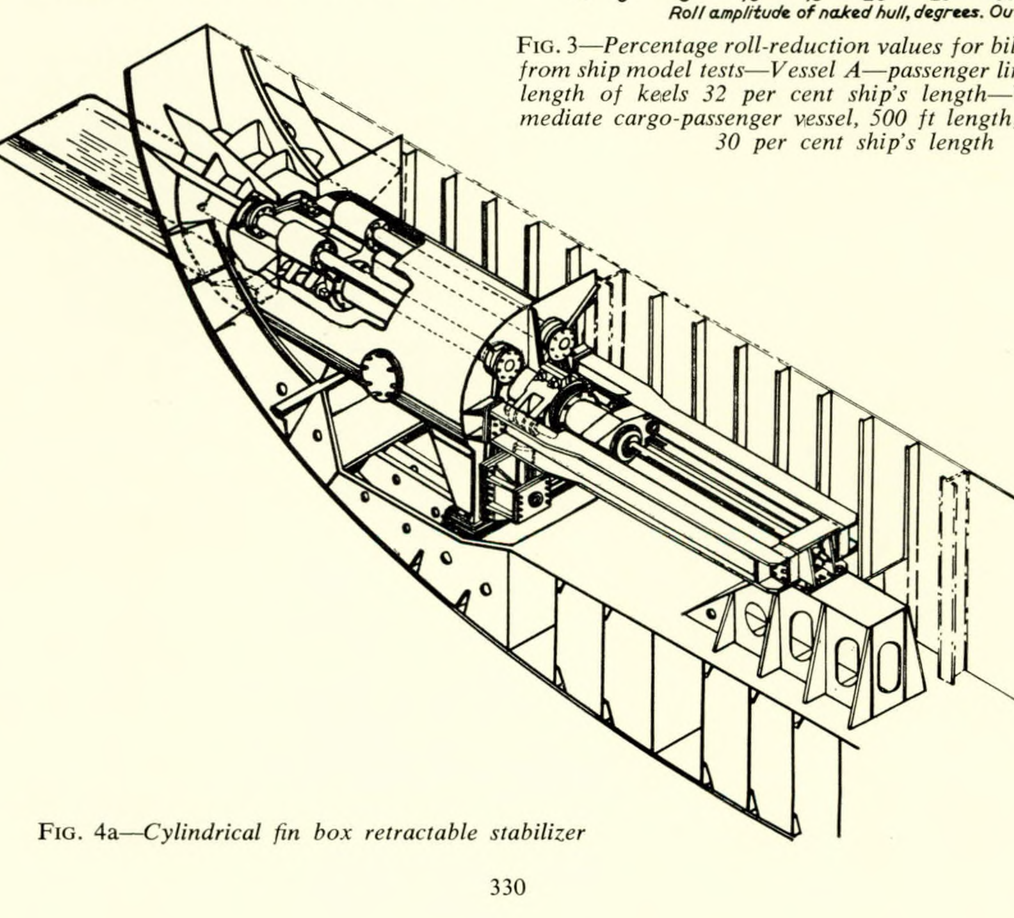

FIG. 4a—*Cylindrical fin box retractable stabilizer*

Liner A of 630 ft length, 26 000 tons displacement, 3 ft 0 in *GM* at 22 knots and for a cargo/passenger vessel of 500 ft length, 16 500 tons displacement, 5 ft 0 in *GM* at 17 knots. These values of 20 per cent to 30 per cent roll reduction were deduced from tests with 16 ft models in a ship model basin at the corresponding speeds given. The values given may be subject to a scale factor of around 10 per cent in passing from model to full size ship, but even so they would still represent from 18 per cent to 27 per cent roll reduction. Contrary to certain opinions, the retention of bilge keels, when other forms of stabilizing equipment are used, should be considered. When properly fitted in the streamlines, determined by model tests, they offer very little resistance and their resistance augmentation has little if any noticeable effect on speed.

Stabilizing equipment can be divided into two broad classes:

1) activated fin stabilizers, mostly flap type fins, where the anti-roll moment is produced by hydrodynamic lift forces on the fins;
2) tank stabilizers, in which water or any other liquid is moved from one to the other side of the ship producing the moment by the movement of the liquid.

Activated fin stabilizers may be sub-divided into two classifications:

i) the retractable type, either retracting athwartships straight into the hull (Fig. 4a) or folding (forward or aft) (Fig. 4b);
ii) the non-retractable type with flapped or unflapped fins (Fig. 4c).

In the same manner, tank stabilizers may be sub-divided into three categories:

a) activated controlled tanks, where the movement of water is aided by the action of a propeller or pump;
b) passive controlled tanks, where the passage of water is controlled by air or water valves coupled to a gyro control and hydraulic relay unit and ancillary equipment (Fig. 5);
c) pure passive tanks (Fig. 6), where the water is permanently controlled by a suitable arrangement of vertical or horizontal restrictions.

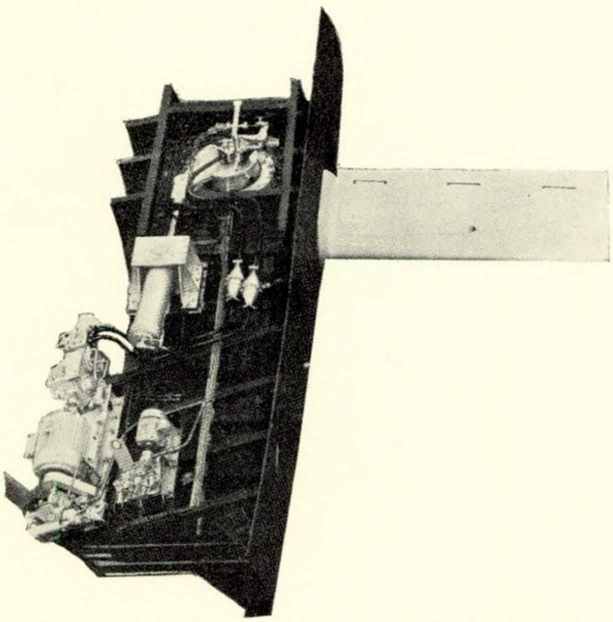

FIG. 4b—*Denny - Brown - AEG International Stabilizer—A packaged design*

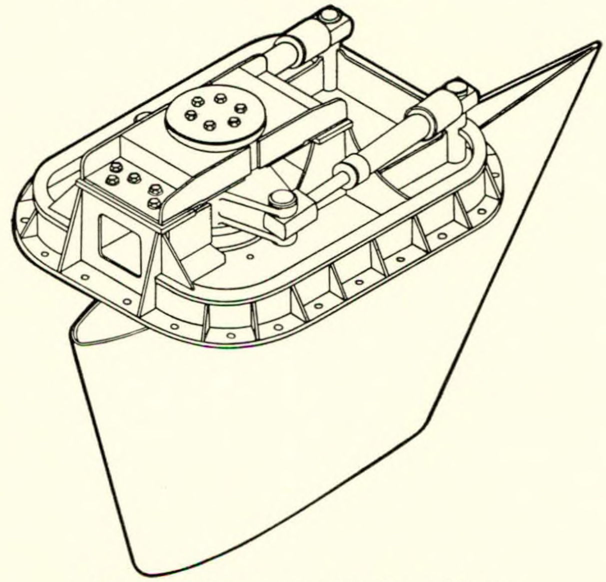

FIG. 4c—*Non-retractable fin stabilizer*

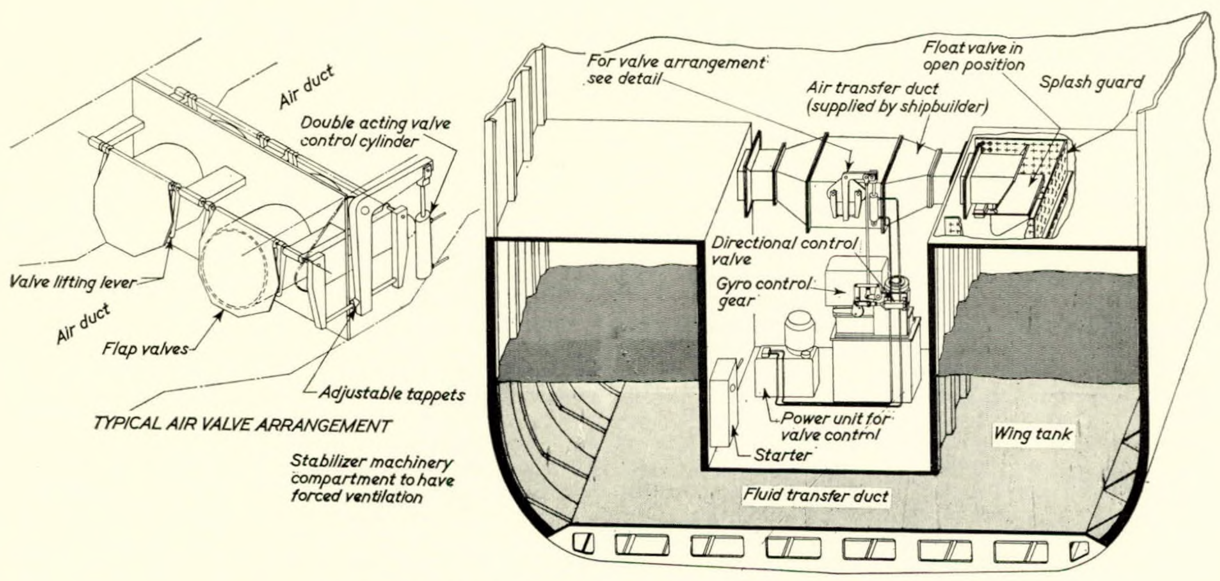

FIG. 5—*Muirhead-Brown tank stabilizer—air controlled*

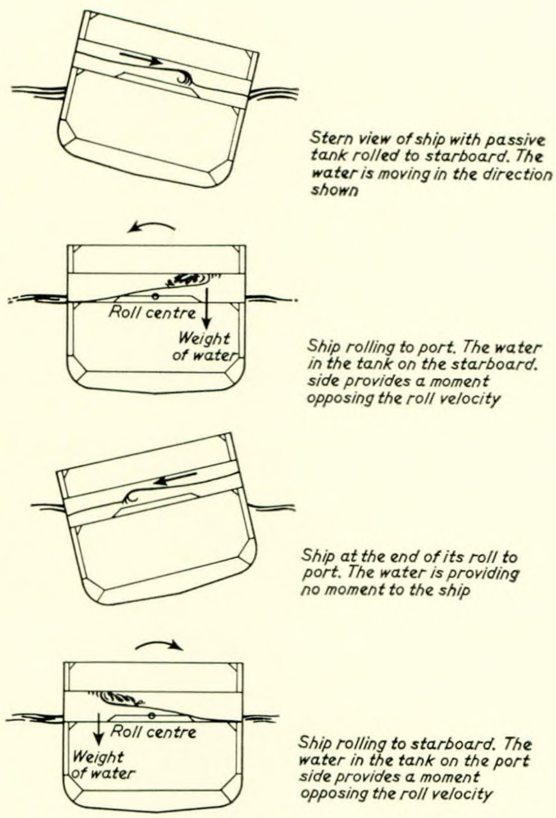

FIG. 6—*Brown-NPL passive tank stabilizer*

Tests on a small experimental vessel of 75 ft length and also on three somewhat larger vessels have shown that the advantage in roll reduction produced by an activated tank is not warranted by the extra complications and cost of fitting a propeller or pumping means and the necessary generating capacity for the driving motor. Its development has been discontinued in the meantime.

THE CHOICE OF STABILIZING EQUIPMENT

Passenger Liners and Cruise Vessels

Passenger Liners, with speeds generally from 18 knots upwards, definitely lie in the field of retractable fin stabilizers. This has been the case since the first large P. & O. Vessel *Chusan* was fitted with one set of 78 ft² fins of Denny-Brown design. At that time, this represented a large jump from the 25 ft² installations fitted up till then to various cross-channel vessels. With the advent of the *Chusan* the popularity of fin stabilizers increased rapidly, not only in the U.K., but also abroad, so that they can be regarded today as a standard equipment for a ship of this class. Until the appearance of the folding type (Sperry and Denny-Brown-AEG) they were all of the athwartship retractable type and were usually situated, either in the auxiliary machinery compartment, the boiler room or in the forward part of the main engine compartment, giving easy accessibility for maintenance. These compartments are frequently near midships, at maximum beam, where the maximum righting arm is available. The athwartship retracting type of the *Chusan* design suffers from one inconvenience that the guides and slides on which the fins move for housing and extending are exposed to the action of the salt water and hence are liable to corrosion. Today, the folding type would be preferred, occupying a limited space on either side of the ship. A complete athwartship space is not required and the unit can be readily supplied as a packaged unit. Most important, corrosion from sea water is eliminated on working surfaces as almost the whole underwater gear works in oil and is isolated from the sea. In large liners like *Queen Elizabeth 2* the product of displacement and *GM* goes beyond the possibility of installing a single gear with one large fin per side and therefore in these cases twin gears are adopted, which incidentally afford a certain flexibility in service.

When in moderate weather one gear can remain housed and adequate stabilization can be achieved on one set only. Both gears are operated by a single common control as independent controls may lead to trouble through incorrect and differing phasing. If possible, the forward unit should be abaft the forward shoulder of the hull to reduce to a minimum hull resistance increase and noise. If the fins lie in the line of the bilge keels or near to them, the latter should be cut one fin chord forward and two abaft the fins in order to avoid interference effects.

Most modern passenger liners are engaged in cruising or are specifically designed for cruising, in which case the operating speed is much lower than the normal service speed. In this case the fin stabilizer will be slightly larger, but the strength and power requirements lower. To cope with higher speeds, either automatic or stepped manual fin angle control is applied above the cruising speed. Since the lift forces vary as the square of the speed, the same lift or efficiency can be obtained with a smaller fin angle at a higher speed. This reduces also noise and erosion damage to the fin due to cavitation. Cruise vessels often lie at anchor in open roadsteads or proceed to the anchorage at very low speeds, where the fins, when left extended, could only act as a kind of bilge keel making a small, although useful, contribution to roll damping. Any additional roll reduction has to be obtained from a stabilizer tank, either controlled or passive. Some cruise vessels have been fitted with a combined system of fins and tank, but so far in practice this has not been found highly successful at speed. Such a combination has been tested on the small vessel *Second Snark* and has yielded promising results as shown in Fig. 7. The combination system should allow a reduction in the

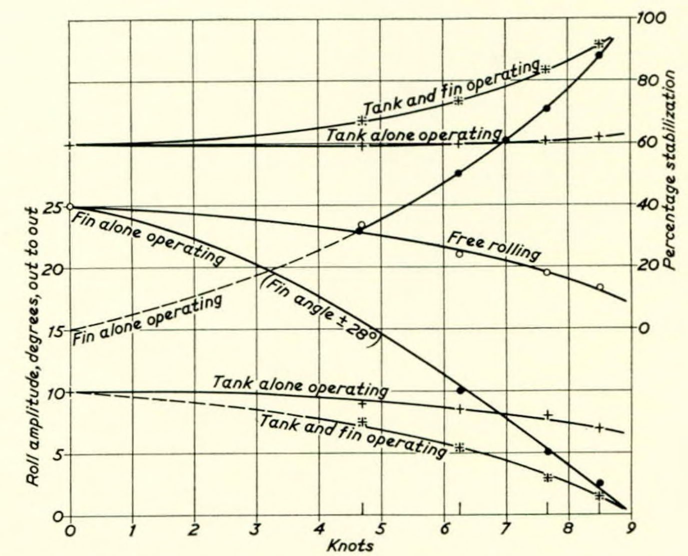

FIG. 7—*Experimental vessel* Second Snark *70 tons salt water displacement, 1·38 ft metacentric height—Curves of residual roll amplitudes for fin stabilizer, tank stabilizer and both operating together plus curves of percentage stabilization for tests carried out in synchronous seas—Sea input equivalent to 3·0 degrees waveslope—Tank capacity equivalent to 1·65 degrees waveslope—Fin capacity varying with speed*

size of the fin gear, as the tank would act as an auxiliary stabilizing medium at speed. As long as no reliable practical data is available for such a combination from larger vessels, it would be advisable at present to consider the two systems quite independently provided the space is available to accommodate both a folding type fin gear and a tank, whether controlled or purely passive, and the owners are prepared to shoulder the costs. If the owners regard the zero speed condition to be of little or no importance, it would be more economical to fit a larger fin stabilizer alone to cope also with slower speeds.

Fast Cargo Liners, Container Vessels and Tankers

Fast cargo liners lend themselves to the adoption of tank stabilizers having a capacity of around 3°, provided sufficient space is allocated for such a tank at the design stage of the vessel. Such vessels may be on long hauls, say from the U.K. to Australia, when, with the burning out of fuel and reduction in stores, they are subject to a substantial decrease in the metacentric height and displacement from start to end of the voyage. Therefore, a controlled passive tank or passive tank designed for the departure condition with shortest rolling period would offer a suitable economic alternative to a fin system. There is a limited control in a purely passive tank by altering the depth of water to take account of longer periods, while the controlled tank will cope automatically with longer periods. Since any stabilizer tank will work more efficiently when placed higher up, nearer to, or above the rolling centre, this should be taken into consideration when designing the ship. If the initial metacentric height is not high, the waveslope power of the tank must be limited to the lowest permissible metacentric height in service, due to the free surface correction of the stabilizing medium, and consequently the maximum capacity and efficiency of the tank will be limited by this factor.

So far tankers have been fitted with purely passive tanks, where, in many instances, the fluid cargo is used in selected tanks as the stabilizing medium and longitudinal bulkheads are suitably slotted to form the restriction to the passage of the liquid for the correct damping of the stabilizer tank system.

Fast cargo liners are being quickly displaced by the container vessels, for which stabilization is much more essential. Some container ships are today designed to carry up to four tiers of containers on deck, creating problems of reversal strains when rolling in heavy weather and leading to the possibility, in severe weather, of the loss of containers overboard or severe damage to their contents. They are all designed with machinery aft and usually have the container compartments separated by narrow cofferdams which do not always lend themselves to the possibility of an efficient tank stabilizer system. Effective passive systems have been built with two or three purely passive tanks, one above the other in these cofferdam spaces, and elsewhere when convenient. Care must be taken to ensure that sufficient waveslope capacity is built in at the design stage. However, container ships are becoming increasingly faster with service speeds up to 26 knots, and this moves them further into the field of fin stabilizers.

If the owners are adamant that the container space must not be encroached on at any cost, the obvious choice would be to fit the folding type into the forward end of the machinery compartment, although it must be remembered that these vessels have a relatively fine run aft and therefore finer sections in way of the fin stabilizers, which would therefore require a larger fin size to compensate for the smaller righting arm. However, efficient fin stabilizers can be designed for a suitable location near the midship part of the ship at the end of a container compartment with the minimum loss of containers, say one per side (Fig. 8).

With the constant increase of the product of displacement × metacentric height in this class of ship, it has been found that one set of fin stabilizers would be insufficient in certain instances and consequently the operator may have to accept a correspondingly higher encroachment into the container space. One class of container ship has been designed with a sufficiently wide cofferdam to take an athwartship retractable fin installation, but this can be regarded as one specific case.

For space saving, the multiple non-retractable fin installation is a possible solution, but as no operator would allow these fins to project beyond the building line of the vessel, this would automatically dictate the adoption of low aspect ratio fins (small outreach—large chord) with low lift qualities, because these vessels are normally hard bilged. This in turn would result in multiple fin installations with several fins per side necessitating a careful study of the interference effect of one fin on another (Fig. 9).

If, for example, the fins were spaced between centres eight fin chords apart, the first fin would have its full lift coefficient, while the second and all subsequent fins would have a lift coefficient of only 0·88 of the full coefficient, or a loss of 12 per cent. Furthermore, an inevitable constant speed loss is incurred even when the fins are not in use and set to zero fin angle. For a container vessel 700 ft, b.p. × 100 ft, moulded × 30 ft draught × 36 000 tons displacement × 3 ft 0 in GM × 21 knots service speed, the maximum permissible outreach would be 5 ft 0 in. Taking four fins per side of 50 ft² area each, spaced 110 ft apart, the estimated waveslope capacity would be around 5° with a roll reduction of 30° to 6° out to out. The eight fins

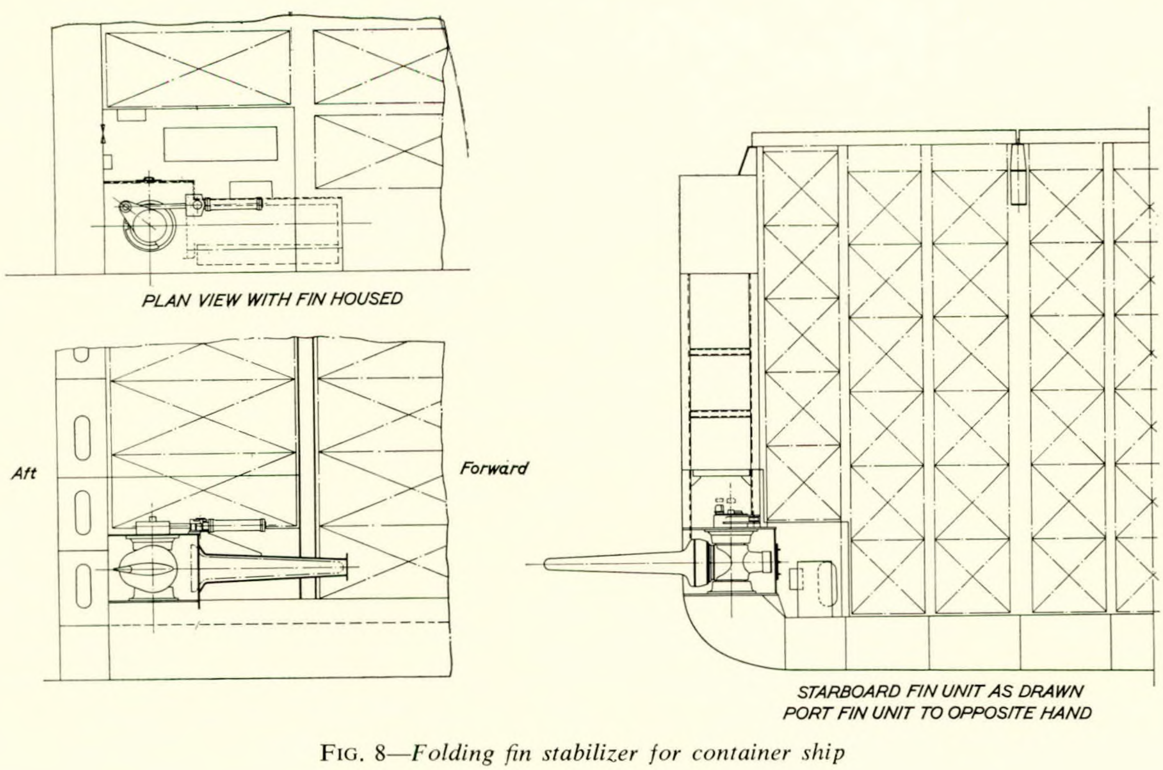

FIG. 8—*Folding fin stabilizer for container ship*

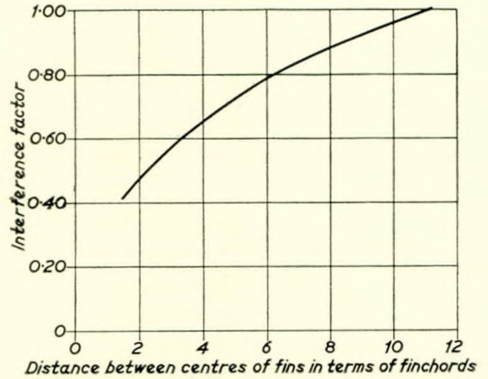

FIG. 9—*Non retractable fin stabilizers—Interference effect on lift coefficient*

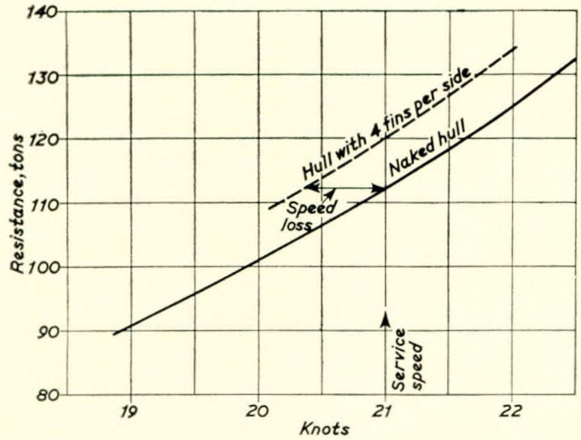

FIG. 10—*Non-retractable fin stabilizer speed loss effect—Container vessel of 36 000 tons displacement × 3 ft 0 in average G.M.—Hull resistance curve and speed loss due to permanent drag of 4 non-retractable fins per side of 50 ft² area—Each (0·50 aspect ratio, maximum permissible outreach 5 ft 0 in) set to zero fin angle—Service speed 21 knots*

would have a total drag at zero fin angle of about 8 tons and the permanent speed loss would be just under ¾ knot (Fig. 10). In addition, the fuel expended on drag would cost about £3000 per annum.

In general, container ships require a lesser degree of stabilization than that for passenger vessels. Most owners accept a maximum residual roll amplitude of 10° out to out as compared with 3° out to out for a passenger vessel and car ferries. Roll acceleration is, however, the important criterion. Tender ships with long roll periods can accept a larger residual roll than stiff ships with short periods since, in general terms, the roll acceleration is the product of the roll angle and the square of the rolling frequency.

The free roll characteristics of the fast fine form cargo liners and container vessels of today require special study as only limited information in this field exists. Much can be accomplished in the field of model testing with the model ship rolling whilst moving forward at the appropriate corresponding speed, backed by information from ships in service. The validity of the equation for the natural roll period, $\frac{CB}{\sqrt{GM}}$ (where $C = 0{\cdot}4$ to $0{\cdot}44$) applied to these vessels is in question as a number of cases have arisen where the natural roll period is appreciably less than that given by the formula.

Car Ferries and Cross-channel Vessels

Before the advent of car ferries, all cross-channel and short route passenger traffic was carried by relatively small fast passenger vessels. These were invariably fitted with fin stabilizers of the athwartship type.

This class of vessel has now been superseded almost entirely by the dual purpose ship, the car ferry, and for this class of ship, it depends entirely on the particular service, whether fin stabilizer, tank stabilizer, or both should be adopted. In some cases the vessel may travel a considerable distance to and from its destination at reduced speed, say in estuaries, when a fin stabilizer, designed for its normal service speed, to keep down the size of the gear, would have very reduced stabilizing power and in this case the tank type would take over. There are some car ferries in service with a pure passive tank only. However, in these cases the operator must accept the lower performance of the tank when compared to that of a fin installation. There has been the case where, in two Mediterranean car ferries, the passive tank was replaced by fin stabilizers. The suitable location of a tank presents problems, as all car decks have to be clear, especially in the roll-on/roll-off type, forcing the tank location down in some instances to a position just above the tank top, while the preferred position would be above the roll centre. There is the possibility, in certain cases, of fitting a controlled tank with a centre duct, which, if possible, should be located above the roll centre for optimum efficiency. This type is a better proposition for car ferries with dual service conditions, motor cars on one trip, lorries and trailers on the next, with a consequent variation of metacentric height and displacement. If the tank is designed for the worst condition i.e., highest product of displacement × *GM* and the shortest period, the control would take care of the other conditions.

For pure cross-channel traffic and long distance Mediterranean passages, without any estuary work, fin stabilizers, either of the folding or the cylindrical athwartship retractable type, should be adopted. The latter is very popular with the Italian Merchant Navy, especially on account of its simplicity and ease of maintenance (Fig. 4a).

In smaller roll-on/roll-off pure cargo ferries, the adoption of, say, two sets of non-retractable fin stabilizers should not be excluded, as the body sections are relatively finer than for large container ships and therefore fins of somewhat higher aspect ratio could be fitted, provided the operators are prepared to accept the speed loss, when they are inoperative in calm weather.

Naval Vessels

The era of the large surface warship is past. Most navies concentrate on the building of lighter craft, small and large frigates for escort and other duties, destroyers or at the most cruisers of below 10 000 tons displacement. With the increasing use of sophisticated machinery and weapons, the space problem in such vessels is acute and for this reason either a single set or a multiple set of non-retractable fin stabilizers, always designed to be within the building line of the ship, of the unflapped or flapped type, are being fitted.

Naturally, whenever possible, the spacing in multiple sets should be as wide as possible to avoid the interference or "down wash" effects between the fins abaft the foremost one (Fig. 9). Warships have a wide range of speed requirements, say, cruising at 18 knots and a maximum speed of 30 knots, which range has been further widened with the introduction of helicopters, when the vessel may have to slow down below the ordinary cruising speed for helicopter operation. In most warships, therefore, fin angle reduction with speed is essential, with fins designed and stressed for full fin angle at the cruising speed. The fin angle reduction can be either automatic, coupled to the ship's log or the propeller shaft, or can be attained by a manual stepped control on the bridge. With the stepped control a certain reduction in performance has to be accepted as a result of its simplicity and cheaper application.

In Fig. 11, lift values, speed control and estimated waveslope capacity values are given for a low aspect ratio unflapped fin. The fin shaft is stressed for a 24 tons lift at maximum fin angle of ±28°, when the vessel has an ahead speed of 22 knots. Thereafter the choice lies between the two controls; the shaded area under the waveslope capacity curve shows the reduction in performance to be expected if a stepped control is fitted. For

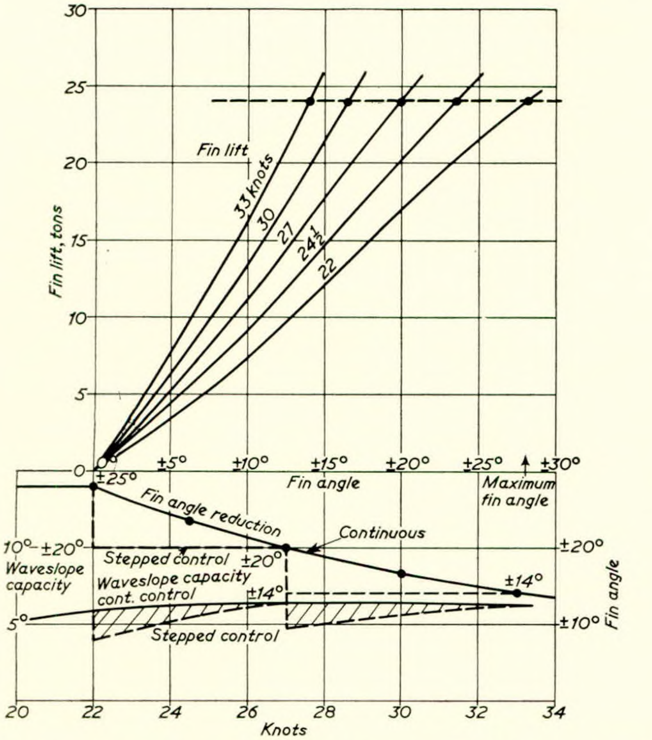

FIG. 11—*Fin angle control with ship speed—Lift curves and fin angle reduction for typical non retractable installation*

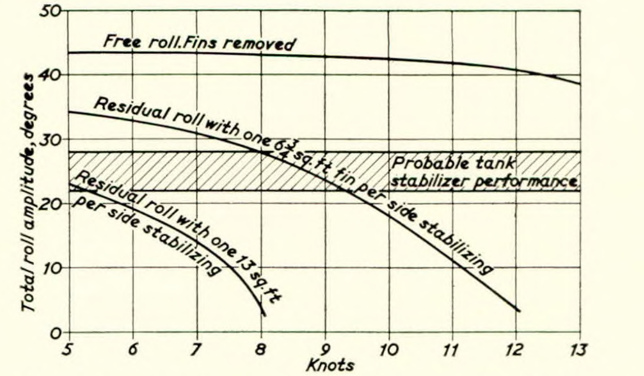

FIG. 12—*Minesweeper stabilization fins or tanks—Test on a 1:10 scale model hard chine inshore minesweeper form—Curves of free and residual rolling in synchronous beam seas—Displacement=167 tons—Metacentric height 2 ft 3½ in*

low speed helicopter operations, high aspect ratio retractable fins (say two fins per side) can be adopted, with the fins designed for the low speed and having continuous fin angle reduction. This assures a certain degree of flexibility, since, when the speed is raised to ordinary cruising speed and above, the ship can be stabilized on two sets of fins in very rough water or, alternatively, in moderate seas one set can be retracted and the ship stabilized with one set only, thus saving the drag produced by the second set.

An increased number of hydrographic survey vessels are being built by many navies, where emphasis is laid on low speed operation down to zero speed. For these types of ship, controlled or pure passive tanks are the obvious answer. This is also true for minesweepers, where sweeping speeds lie in the region of 4 to 5 knots. Both survey vessels and minesweepers are liable to have considerable space problems, which makes the post-fitting of a stabilizer tank, whether passive or controlled, rather difficult. For this reason, they should be embodied at the preliminary design stage, not only from the point of view of space, but also for the effect of the tank's presence on displacement and metacentric height. The tank stabilizer may have to be restricted in size so that the *GM* will not fall below the minimum permissible metacentric height, when the free surface effect of the tank has been taken into consideration. In 1958, a design study was made and extensive model tests were run on behalf of a foreign navy, on a 167 tons hard chine inshore minesweeper class to cover two sweeping speeds at 6 and 11 knots. The results given in Fig. 12 show that, in order to obtain over 50 per cent roll reduction at 6 knots, one set of 13 ft² retractable high aspect ratio fins were required. This would have involved staggering the installation with consequent increased fore and aft space, increased weight and cost, especially when the installation would have had to be designed on the basis of 95 per cent non-magnetic content.

Trawlers and Fish Factory Ships

With larger trawlers, travelling further afield to remote fishing grounds and often working in conjunction with factory ships for fish processing, it is rather surprising that, in general, any form of stabilization has so far evoked little interest from the fishing industry, although a few British-owned trawlers have been fitted with tank stabilizers. One would surmise that any degree of roll reduction would be welcomed by an industry which normally works under arduous conditions and where, on factory ships, a relatively stable platform for fish processing would be considered essential. Since both trawlers and fish factory ships are not subject to large variations of metacentric height and displacement, tank stabilizers would obviously improve conditions. Passive tanks at low cost for trawlers and controlled tanks for factory ships would be an appropria e choice. The controlled tank in the latter would be better able to cope with varying conditions occurring when the factory ship cruises at very low speeds, while processing and awaiting the return of the attached trawlers. At this time the vessel may be subject to variation in rolling periods in different sea directions. In addition, the controlled tank with wing portions and centre duct would provide an easier access from the for'd to the after part of the ship.

Yachts

In yachts of smaller size, the non-retractable fin stabilizer has found a popular market, because space again is of primary importance. However, small retractable fin stabilizers have also been installed in several yachts between 130 and 180 tons displacement, where the forward or after part of the motor room has been used to accommodate the units. Apart from cost, the breadth of the vessel in way of the stabilizer and the other ship parameters must be such as to permit the installation of a gear of acceptable proportions such that the installation is not staggered. Otherwise the non-retractable type is preferred. In addition, with block coefficients varying between 0·35 and 0·45, the hull sections are quite fine and this makes it easier to adopt more efficient aspect ratio fins within the building line of the vessel. Several manufacturers offer, in this category, standardized units, equipped with a simple velocity control at relatively low cost. Since, because of their size, such vessels take heavier punishment in rough weather and hence speed has to be reduced, prospective owners and designers should always state the minimum cruising speed, as it is better from the operational point of view to over rather than under-design such fin stabilizers. The larger ocean cruising yachts fall into the class of small cross-channel vessels and should be fitted with folding or athwartship type fin stabilizers according to the space available. There will be a case for a combined fin installation and tank installation, when the owners may decide to lie for considerable periods offshore. The stabilizing tank will assure increased comfort and almost constant use of the swimming pool, if this is also fitted.

Specialized Craft (Weatherships, Oil Rig Supply Vessels, Oil Drilling Barges etc.)

Roll reduction for weatherships on their trips to and from stations in the Atlantic is of secondary importance to that when on station and for this reason the controlled passive or pure passive tank is recommended. Very often such ships are ex-naval vessels, such as corvettes and frigates, and therefore the post-fitting of a tank installation presents considerable problems, requiring certain modifications to the ship's structure and layout. The degree of roll reduction possible with such an installation, for given ship parameters, is entirely dependent upon the space made available for it.

Oil rig supply vessels are relatively fast ships for their size, since crew changes on the rig and transport of material to the rig, often over a considerable distance from the supply port, has to be effected in the shortest possible time. As in weatherships the trip to and from the rig in bad weather would only affect the comfort of crew and rig staff. The problem arises once the vessel has reached the rig and unloading operations are under way, at which time damage to equipment can result from heavy rolling. A stabilizing tank would afford sufficient roll reduction to overcome such difficulties. Several companies owning oil drilling barges have installed stabilizer tanks for the good reason that, with reduced rolling, drilling operations in bad weather can be prolonged. Fig. 13 shows *Glomar Challenger* which is fitted with a controlled tank stabilizer producing average roll reductions of 50 per cent.

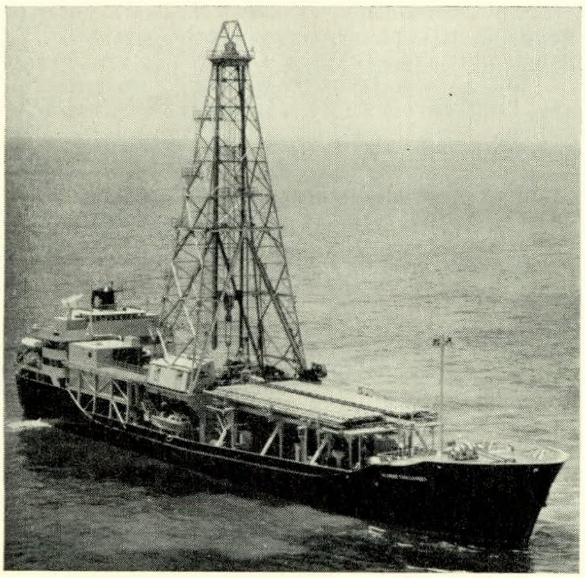

FIG. 13—Glomar Challenger *stabilized with controlled tanks*

EVALUATION OF EFFECTIVENESS OF A STABILIZER INSTALLATION

Having selected the stabilizing means and installed it in the vessel, the shipowners or operators would wish to have the effectiveness of the installation demonstrated during trial or in service. With fin stabilizers, it is relatively easy, in calm weather, to prove the capacity or power of the gear, by simply inverting the control signal and force rolling the ship in synchronism. For rough weather sea trials a method of presenting roll results has been developed during the past few years, both for fin and tank stabilizers. Its origin lies in the applied subject of oceanography. This method is based on statistical probability and offers a more useful way of appraising the merits of a stabilizing system than by a quoted roll reduction at resonance. It also offers a more convenient approach to the sea trials of a stabilizer system in that it enables comparison of the unstabilized and stabilized vessel directly under the weather conditions operative at the time. Because of the ease of obtaining the required results and simplicity of analysis of the results, trials can be carried out with the vessel in service, the long term benefits of stabilization being easily recognised by scientists and laymen alike.

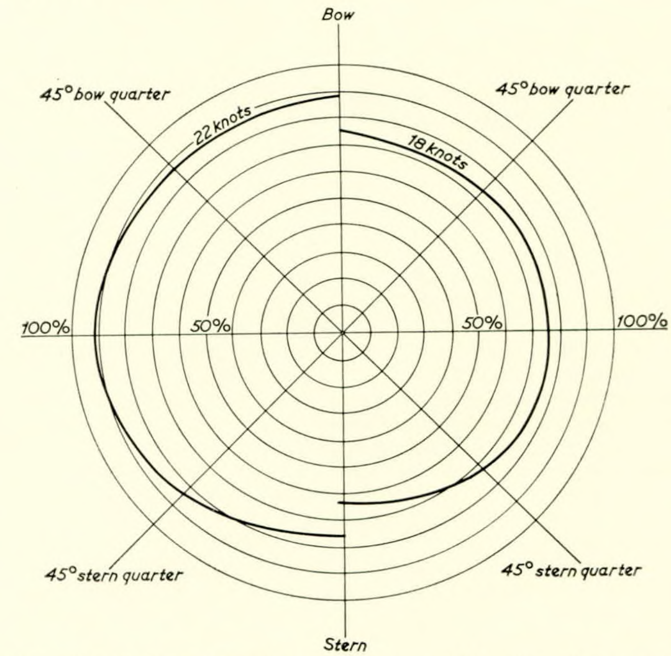

FIG. 15—*Polar plot of percentage stabilization for rough weather trials—Trials carried out at 5 ship's headings—Stabilizer fins designed for 22 knots*

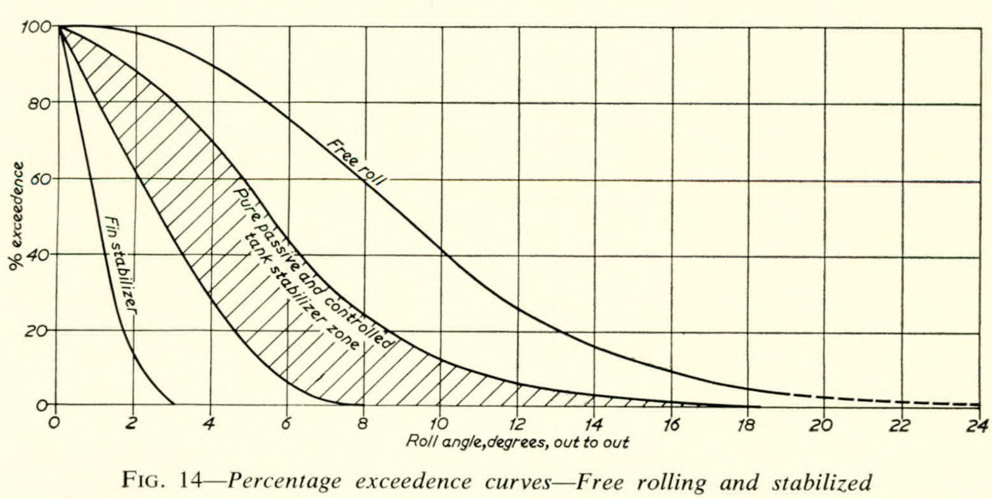

FIG. 14—*Percentage exceedence curves—Free rolling and stabilized*

Briefly, the vessel is allowed to roll, both stabilized and unstabilized, approximately 100 cycles of roll being obtained in each case. An estimate is made of the prevailing weather conditions and the results analysed to give what are called percentage exceedence curves as shown in Fig. 14. The ordinates represent the percentage number of rolls which exceed the prescribed value of the abcissa, which is roll out to out during either a 100 cycles of roll or the equivalent time under the operating weather conditions. Such trials can be further extended to cover several ship headings to the prevailing seas, although this is usually rather time consuming, since weather conditions seldom remain constant. The results can then be presented in polar form in terms of percentage stabilization (Fig. 15).

Shipowners may therefore study records at sea states prevalent throughout the year on their ship's route and then assess, in conjunction with a set of the above results, the advantages of a stabilizing system.

CONCLUSIONS

Although fin stabilizers have been in successful operation for many years and are well known in shipping circles, tank stabilizers have gained in popularity only during the last few years and therefore owners and operators may not be able to distinguish with sufficient accuracy the particular field of application for each and their relative efficiencies. It has been the intention of the authors, by means of this paper, to establish in broad outline the type of stabilizing equipment which should be selected for each class of ship and for this reason the main part of this paper was sub-divided into ship classes. Although a general picture has thus been presented, it must be remembered that each case should be examined on its merits if a successful solution is to be found.

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Discussion

MR. G. C. EDDIE, B.Sc. (Member) said that three or four years ago much was heard about anti-rolling devices and what they would do for the particular vessels his organization was interested in, not only from the direction of Edinburgh, but also from Teddington and especially from across the Atlantic. There was now a good deal of useful experience of these devices and discussion on the subject was timely.

In conjunction with Mr. P. D. Chaplin, B.Sc., he would like to comment on one of the applications mentioned in the paper, trawlers and fish factory ships. It would be a pity, however, if the authors were to be exposed only to the carpings of narrow experts concerned with one specialized type of craft or another. So that the discussion could be broadened and range over the whole state of the art he would also offer a few remarks of a more general kind. He added that Mr. Chaplin had provided the responsible comment based upon measurements at sea and that the other contributions were offered by himself.

Before reviewing the practical experience on fishing vessels he made one comment: that it was probably too late to do anything about calling tank-type devices "stabilizers". Fishing skippers were not usually versed in the subtleties of naval architecture and control engineering and their abilities in the use of the English language did not extend to fine semantic distinctions outside their own subjects of fish and fishing. These men could find themselves riding out a gale with a lot of fuel and water gone, a poor catch, slack tanks and ice accumulating on the rigging and superstructure. In such circumstances, to call an anti-rolling device of the passive tank type a "stabilizer" was asking for trouble, and it was this type that was the most likely to be found in fishing vessels, as the authors had pointed out.

Mr. Chaplin and he were particularly pleased to see that the authors had given some recognition to the effectiveness of bilge keels, since some stabilizer designers had been less than frank on this topic. There had been an unfortunate tendency for model experiments to be carried out without fitting bilge keels to the model, although these were to be fitted to the ship. Indeed, until very recently models were just not fitted with bilge keels because it was held that there were scaling problems. This led to very large angles being recorded in the unstabilized condition because of the low damping of the bare hull. The additional damping of say a passive tank system would then have a large effect, probably reducing the roll by more than 60 per cent. However, if bilge keels were fitted to a vessel the initial damping would be high and consequently the vessel would roll less than if they had been omitted. The additional damping due to a stabilizer would be the same as before but it would now be a smaller proportional increase over the initial damping, i.e. the reduction in roll due to the stabilizer would be less, probably about 30 per cent; this contrasted poorly with what the owner had been led to believe. This had become a common occurrence with respect to the anti-roll systems fitted to fishing vessels in the United Kingdom and had been a contributory factor to the disenchantment felt by U.K. fishing vessel owners. The White Fish Authority carried out evaluation trials on behalf of owners on five large stern trawlers, three fitted with passive tank systems and one with a semi-active tank system and one with an active system. In all cases the vessels were fitted with bilge keels and in three cases with bar keels as well. However, in none of the model experiments concerned had the additional damping due to these devices been considered. The result was that predictions of about 60 per cent reduction in roll were generally made. The trials showed that for the passive and semi-active systems reductions of between 15 and 40 per cent were actually achieved at sea, depending upon the severity and direction of encounter of the sea. For the active system it was found during the trials that the actuating linkages in the tank were not working in a ship roughly a year old. The White Fish Authority's trials engineer carried out extensive repairs at sea but the system did not seem to be very effective, since on one occasion roll angles of 40°, out to out, were recorded. However, since the manufacturer did not have an opportunity of checking out the system after the repairs the results were regarded as of doubtful value. This experience showed that proper maintenance was essential for the more complex systems.

In one of the trials two sister vessels were involved; one fitted with a passive tank system and the other with a semi-active tank system. Virtually no difference could be detected in the performance of the two systems. However, it was worth noting that on one vessel a survey was carried out on crew reaction by altering the state of the system from "on" to "off" or *vice versa* and asking crew members to say what had been done. The results showed that about 50 per cent of the time they were right! At first sight this seemed alarming, but it must be remembered that men who spent most of their lives at sea quickly adjusted to changes in environment. Also the crew on modern stern trawlers had generally come from smaller and older side trawlers and almost anything would be better.

In view of these results, the authors' surprise at the lack of interest in the fishing industry was a little surprising in itself, although it was added that the industry was still very interested in the possibilities, especially for the so-called inshore motor fishing vessel of 15 to 25 metres LOA (50 to 80 feet). For this class of vessel one proposed system was the heavy metal weight sliding in a tube; a type not mentioned by the authors.

Regarding large trawlers and motherships, the biggest markets were in Russia and Japan, between them possessing over five hundred of the thousand or so fishing vessels of over 1000 gross tons—a number that was rapidly increasing. At present, however, the Russians seemed to be pursuing a rather different sort of solution. They had built a twin-hulled vessel out of two old trawler hulls about 60m LOA (185 ft) and it was reported that experience was so good they were now going to build a similar new vessel from scratch. It was suspected that the authors would draw the line at attempting to provide an anti-rolling system for a twin-hulled vessel, unless the hulls were entirely below the surface and supported the deck above water on slim struts. There was an American proposal for a single-hulled vessel of this type: a submarine at about one hull diameter below the surface and supporting a pilot-house above water. One wondered if a stabilizing system would be needed and if so, of what kind.

The two or three degrees total roll, out to out, of a ship fully stabilized by fins quoted by the authors was impressive. Presumably this was not for a small ship in short, steep waves. One wondered how this performance compared with what might be expected from a hydrofoil with differential lateral controls to enable it to be "flown" in a substantially upright position, as one assumed such craft might be. If an even smaller roll amplitude and smaller accelerations than those the authors quoted became desirable, say for military reasons, would it be attainable in a displacement type ship, say by running at negative GM when the ship was at or near the vertical, upright position, and relying completely on the fins? There would be ways of bringing the ship into such a condition in a safe manner. Mr. Eddie recalled that some whale-chasers had very little righting moment near zero angles of roll, but that particular market was pretty depressed these days.

Finally, in the case of ships with fin stabilizers and hydrofoils, were there any thoughts of controlling the fins by sensing the wave profile ahead of the ship?

Mr. D. Garrett said the authors had given a clear account of many of the factors affecting the choice of stabilizer system for a proposed new ship, but had not mentioned two factors that could be important in many cases. One was cost; first cost and maintenance costs. Due to their mechanical complexity, it seemed obvious that fin stabilizers would be more expensive than tank systems on both those counts, but had the authors any relative data they could supply?

Next, it was often necessary to know the range of periods over which the various types of stabilizer were effective and the extent of their effectiveness throughout the range. This information was, of course, published elsewhere, but the authors' paper would benefit from its inclusion for the sake of completeness.

The design of a successful passive tank system depended upon an early knowledge of the minimum natural rolling period of the proposed new ship. This knowledge was not necessary for the design of a fin stabilizer system for the ship. The existing empirical methods of estimating the natural rolling period of a ship had recently proved inadequate for certain types of ship and the authors suggested that more reliable estimates could be obtained from model experiments. This raised the question of scale effects. Was it possible to apply the results of model rolling experiments directly to a new ship, or were there some correlation factors to be applied? Mr. Garrett's feeling was that there must be fundamental differences between the damping and flow régimes of models and ships, and therefore there must be correlation factors to be applied. Having seen very little data on the nature of correlation, he suggested that model and full scale experiments be carried out on existing ships to ascertain the nature of such correlation. With this knowledge, much greater confidence could be placed in estimates from models of new ships in the design stage. It seemed likely that from their wide experience of designing stabilizers the authors might have some relevant correlation data; it would greatly enhance the value of their paper if any such data could be published.

Mr. G. R. Head (Member) said that one aspect the authors had not mentioned, whilst it was not concerned in the more technical considerations affecting the choice, did assume considerable importance to the superintendent engineer responsible for the equipment which was finally fitted: that was the cost incurred and time required for maintenance.

Particularly in high speed vessels, the wear and tear of submerged parts of fin stabilizers was very heavy and the annual repair costs considerable. In the earlier retractable gears, much of this was attributable to the sliding crosshead and outboard bearing and should not arise in the present folding type, but there still remained the severe erosion of the fin material, in particular the tail fin. He had been concerned in the protection of fins with many types of high duty coatings, but in no case were these completely successful and resort had to be made to the replacement of metal by patching and welding. Did this condition still prevail, or had any successful protective fin coating been devised?

The tail fin hinges, bushes and driving gear were also a great source of expense, as repairs entailed removing the tail fin after about three years of operation to renew bushes and thrust washers. Had this gear been improved to eliminate those faults in the latest designs?

With the current trends of reducing the time spent in drydock and increasing the interval between dockings, those points assumed major importance, as it was quite often the case that stabilizer repairs governed the duration of time required in the dock. On the other hand, it would appear that tank stabilizers had every advantage in that respect, as examination and repairs could be made at any time. Indeed, they should require little more attention than a normal ballast tank.

The noise generated by fin stabilizers might also influence the choice, particularly in passenger vessels. These were generally located in the vicinity of the choice accommodation, near midships, and the cavitation rumble of the fins operating at large angles, spread vertically up the ship's side. This could be most disturbing, particularly as it varied in intensity from maximum to minimum twice in every complete roll of the ship. In addition, cases had been reported of unacceptable noise levels in vessels fitted with hinged stabilizers, when in the housed position, due to the water flow in the fin box through the access aperture.

Had the authors any information upon the comparative noise generated in tank stabilizers during operation?

One of the disadvantages of tank stabilizers was the internal volume required within the ship to achieve a given degree of moment. In his researches he had seen mention only of water or cargo oil as the operating medium. If fluids of heavier gravity were considered it would be fair to assume that the effectiveness would be increased proportionately and, hence, the space occupied reduced. He had heard of the possible use of compounds such as drilling mud, which could be made to give a density of up to 137 lb/ft^3. Had any research been carried out to explore the potential and use of such fluids or would they be precluded as unsuitable due to the inevitable increase in viscosity?

The authors had pointed out the loss of speed and consequent annual cost through drag in the case of non-retractable fins, but in a vessel fitted with any form of retractable fin stabilizers, employed in trades where stabilization was required for a significant portion of the year, this drag loss would equally apply, but no such loss of speed would be associated with tank stabilization.

MR. T. W. BUNYAN, B.Sc. (Member) said the *Chusan* stabilizers must have been in service for nearly 30 years. He was rather disappointed to find that there had not been any noticeable improvement in the efficiency of the stabilizer fin. No doubt research was being carried out to remedy this situation, because it was a critical factor in roll stabilization with fins.

A recent television programme had shown a model of a 200 000 dwt tanker using the Flettner rotor principle as a rudder, demonstrating the exciting potential of this device, which might be a fruitful avenue for improving fin performance.

A loss of speed in the region of $1\frac{1}{4}$ kn on a 26-kn ship was rather a heavy price to pay for stabilization. These figures related to a passenger liner, where a five degree capacity was necessary, a ship with two pairs of retractable stabilizers. As this fall-off represented the same order as the drop in performance which might be expected due to roughening and fouling of the hull, it was necessary to build in very large power margins in order to maintain the very tight service schedules. The same applied to most of the fast container ships. It followed that serious consideration must be given as to whether the fin stabilizer was the right type to use in these cases. A container ship, as pointed out in the paper, could manage with a three degree capacity and it was felt that the assisted tank with an air valve could be the better. Were many container ships fitted with assisted tank stabilizers? It appeared that the philosophy of the assisted tank was similar to the assisted tank using a water pump, which had however not been very successful, for reasons not mentioned in the paper. What were the reasons for the lack of success? One of the problems involved could be the rapid change in the direction of large masses of water.

Fig. 10 showed the hull resistance curve with four fins a side, giving a permanent drag with zero angle, corresponding to a seven per cent fall-off in performance. With a consumption of fuel of 150 tons/day at a cost of £5/ton, for a ship in operation for 300 days a year, the figure of the fuel expended on drag per annum was £15 000. How had the authors arrived at the figure of £3000? What was the fall-off with four fins a side in the extended position?

He was in full agreement with the authors on the matter of bilge keels. Had they any full-scale data available with and without bilge keels fitted? If the purpose of the bilge keel was similar to that of the fin stabilizer, giving an overall stabilization of 25 per cent, as described in the paper, then the effect of the drag was a matter of considerable interest.

The authors had said in the paper that: "Some cruise vessels have been fitted with a combined system of fins and tank, but so far in practice this had not been found highly successful at speed". Would they explain the reason for this?

MR. R. M. DUNSHEA (Member) in a contribution read by Mr. A. E. Franklin (Member) said that his experience with fin type stabilizers in cross-channel passenger ships was generally satisfactory. Reduction of roll amplitude was of a very high order, so much so, that a little more rolling would have resulted in more comfort. Roll reduction gave passengers possibly as much peace of mind as comfort. The authors had mentioned six degrees of freedom and in bad weather a ship indulged vigorously in the other five.

From a ferry operator's point of view, roll stabilization had the great advantage that cars did not require to be lashed, which resulted in appreciable saving in expensive dock labour. In the worst weather cars did not move in a fin stabilized ship. Since almost inevitably a ship slowed down in bad weather, consideration should be given to speed at which optimum roll reduction was required.

It would be appreciated that, with a system of fin stabilization, an accurate estimate of the period of roll was necessary. Past experience with this type of stabilization had shown that estimates were far from accurate, with the result that stabilizer assisted rolling occurred. This could be dangerous. If possible it would be desirable to investigate the period under conditions of normal loading before the ship entered service. Of other types of stabilization, he would favour passive tanks. The simplicity of the system was attractive and it gave satisfactory results in ships where reduction in roll amplitude by approximately 50 per cent was thought adequate.

A ship operated by his company had recently been converted to carry trade cars and containers, or cattle and containers across the Irish Sea. The design and model tests for the tanks were carried out by N.P.L. at a reasonable cost. On this occasion the model was fitted with bilge keels. Service tests by N.P.L. showed that these compared very closely with model tests, generally a 50 per cent reduction in roll amplitude was achieved. The behaviour with seas on the quarter when a large angle of lull could be expected was satisfactory. The crew although initially sceptical were most impressed by the reduction in rolling. Cars were carried unsecured and it had been found that they did not move in the worst weather experienced in winter on the Irish Sea.

The arrangement had much to commend it, but a disadvantage was the space taken by the tanks. The free surface affect also could cause difficulties when making a conversion. The problem would of course be easily considered at the design stage of a new ship. To reduce the space required for stabilizing tanks, had the authors considered the use of fluids having a higher S.G.

So far as controlled tanks were concerned he wondered if the added complication was justified by the further reduction with amplitude of roll, compared with pure passive tanks, was really worthwhile. Ships with controlled tanks had given a certain amount of trouble and passive tanks had the advantage of absolute simplicity.

Correspondence

MR. A. M. FERGUSON and PROFESSOR J. F. C. CONN, D.Sc., wrote that the formula for the power requirement of a stabilizer was Δ GM.α/57·3, as stated. Hence the power requirement was directly proportional to the value of GM.

Since they made no statement to the contrary, it might be inferred that the authors had used the conventional or static value of GM. The present contributors had shown* ("The Effect of Forward Motion on the Transverse Stability of a Displacement Vessel", by A. M. Ferguson and Prof. J. F. C. Conn, Institution of Engineers and Shipbuilders in Scotland, vol. 113, a paper read on January 13, 1970) that the effect of forward motion in calm water and the resulting change in waterplane characteristics due to the vessel's own wave system was to increase the value of GM. Since this increase could be quite appreciable, depending upon the type of ship, it was suggested that the power requirement might be more accurately assessed by using the actual value of GM at the designed speed.

* Ferguson, A. M., and Conn, J. F. C., 1970. "The Effect of Forward Motion on the Transverse Stability of a Displacement Vessel". *I.E.S.S.* Paper No. 1349 read on 13th January.

MR. F. V. A. PANGALILA, in a written contribution, said that a fin stabilizer was an extremely efficient technical instrument for reducing rolling motion of ships. First a sensing device measured roll, roll velocity and roll accelerations, then based on this information an analysis was performed and a decision made to obtain the optimum position for the fin to generate a counter moment to the wave exciting moments. These procedures required sophisticated instrumentation and, hence, the fin stabilizer was very expensive. The larger fin stabilizers cost approximately £100 000 and the average passive tank stabilizers about 15 to 20 per cent of this.

If one examined Fig. 14 and arbitrarily selected 6° as a reference roll angle one would find that, for 45 minutes in every hour (75 per cent exceedence), the ship experienced roll angles of

6° and greater. With a passive tank stabilizer the ship only experienced roll angles in excess of 6° for 4·5 minutes in every hour (7·5 per cent exceedence). If one considered that during the 45 minutes the significant roll angle (the mean of the 1/3 highest roll angles) of the unstabilized ship was approximately 15° and, during 4·5 minutes of the tank stabilized ship, this roll angle was 5°, one must conclude the performance of the tank stabilizer was excellent.

The performance of the fin stabilizer shown in Fig. 14 exceeded that of the tank stabilizer. The significant angle here was 2°.

The price difference between the two systems was between £80 000 and £85 000. In addition the fin equipment required maintenance and repairs amounting to approximately £3000/year. As noted in the paper the resistance of the ship increased considerably when using the fin stabilizers, especially when a full fin angle of 28° was utilized. This resistance was reflected in higher fuel cost or lower ship speeds.

Of course, there were ships that required a standard of performance as shown in Fig. 14. In this case the combination tank-fin stabilizer should be considered. It was always possible to design a tank-fin combination using a smaller fin size, which would be as effective as the larger fin alone at design speed, but would be superior to the larger fin alone at all speeds lower than the design speeds. The price of the combined system will also be lower than for the larger fin alone because the price of the fin stabilizer depended on fin size and the price of the smaller fin would offset the cost of the tank stabilizer. The effectiveness of a tank-fin stabilizer could be seen in Fig. 16 which shows actual sea trial tests conducted with such a system. If the captain of a ship generally used the fin stabilizer when wave heights were greater than 3 ft and he had a tank-fin stabilizer, the tank would reduce the rolling of the ship sufficiently at 3 ft wave heights and, therefore, only after the wave height exceeded 5 ft would the tank stabilizer need the assistance of the fins to reduce the motion.

Fig. 17 showed the percentage exceedence of wave heights. Three foot wave heights were exceeded 75 per cent of the time and 5 ft wave heights, 45 per cent of the time. The percentage of the time that the fin was not used would be increased by 30 per cent with a tank-fin stabilizer. It followed that fin maintenance, repairs and the resistance to the ship would be reduced by 30 per cent. When the tank system needed the assistance of the fin system the additional resistance of the fin was smaller than for the fin system alone, because the area of the fin with a tank-fin combination was smaller than that of the fin alone. His conclusions were therefore that, as a rule, the tank-fin combination should be preferred to the fin alone.

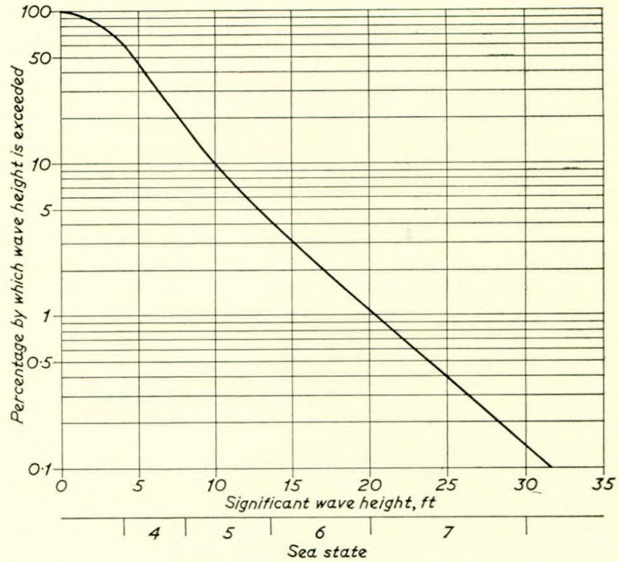


FIG. 17—*Cumulative descending wave heights for the North Pacific*

MR. W. L. S. WALLACE wrote that modern experience was showing that the original problems of maintenance experienced with athwartships retracting gears could be overcome by good detailed design and careful material choice. In effect, it had been found that retractable fins of comparatively low aspect ratio

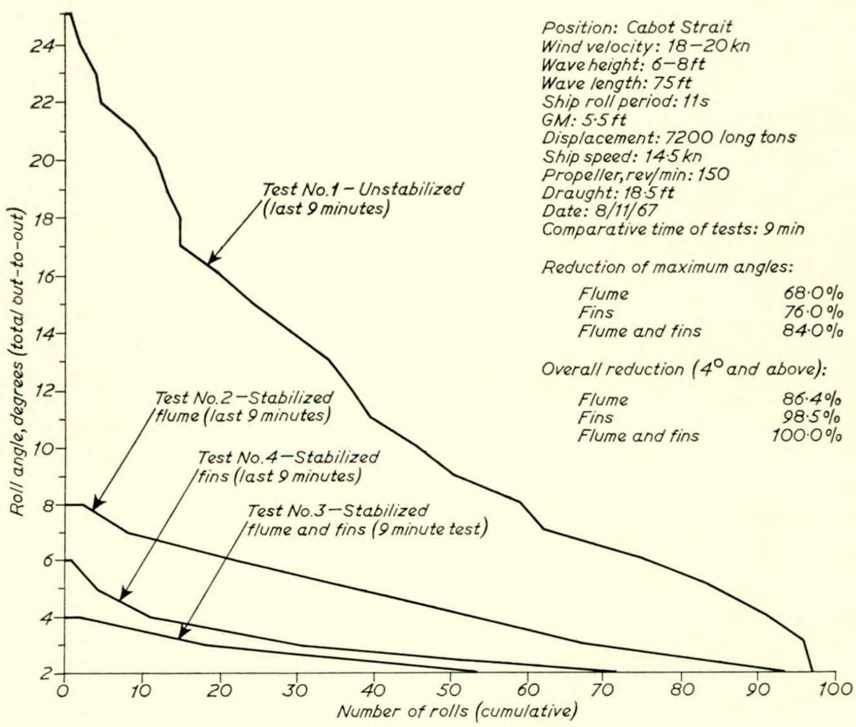


FIG. 16—*Tests No. 1, 2, 3 and 4—Quartering sea—14·5 kn*

(1 to 1·5) were quieter, equally efficient and less susceptible to corrosion/erosion problems.

The authors had mentioned container ships with athwartships retractable fins installation as specific cases. It was interesting to note that there were now complete classes of this ship from 18 000 to 57 000 tons displacement either being fitted or fitted with this type of fin. There was no loss of container space as the necessary fins required no more than the cofferdam space provided. For the larger ships, a cofferdam width no greater than 4·2 metres only was necessary. It was his opinion that a combination of fins and tank type stabilizers could be used more often than the authors considered and this might well be a better solution for the passenger ship with a large speed range between liner service and cruising.

MR. P. H. JUDD, M.Sc., wrote that during the early design stages of a vessel it would be useful to be able to assess the size as well as the type of suitable stabilizer. A difficulty arose in the case of the tank stabilizer in determining the value of W (weight of fluid transferred). Could the authors suggest a means of estimating W for a given tank and duct configuration?

When a tank stabilizer was installed in a passenger vessel it had considerable effect upon the damaged stability conditions as a result of its large free-surface when in the operating condition. Had the authors encountered any difficulty in this respect when designing tank stabilizers for such vessels? A common solution was to install large sea inlet valves to allow the tank to be flooded quickly in the event of an accident. This had the disadvantage that some action by the crew was necessary to initiate the flooding, involving a possibly lengthy time interval before the tank was full—if this were possible at all as the damaged vessel might trim and heel so that the crown of the tank was above the final waterline.

He hoped the authors would be able to accumulate reliable data for the combination of tank and fin stabilizers. In passenger cruise vessels and vehicle ferries particularly, there was often need for stabilizing power at very low or zero speed and at, or near maximum speed. These vessels could ill afford the cost and space required for two independent systems, especially remembering the effect of the tank stabilizer on the ship's ability to withstand side damage.

MR. J. A. H. PAFFETT, R.C.N.C., in a written contribution said that the authors in dealing with fin stabilizers quoted the concept of "waveslope capacity", given by stabilizer righting moment divided by displacement × GM. This could be thought of as the angle of heel which the fins ought to produce if set hard-over while the ship was steaming ahead in still water.

Twenty years ago a fin system was installed in the old cruiser *Cumberland*. They tried to measure the waveslope capacity by actually carrying out this evolution. There was a satisfying lurch as the fins went over, but the final heel at which the ship settled down was much less than that predicted from the speed, known lift coefficient of the fins and ship's GM.

The explanation appeared to be that the fins had a certain yawing action in addition to their heeling moment and, after the fins had been held over for some time, the yaw angle developed by the hull resulted in a lateral hydrodynamic force acting low on the ship, in a sense which reduced the heel angle. There was, in fact, a complex interaction between yaw and heel.

On the other side of the coin: they had recently shown N.P.L. that the operation of the rudder in a self-propelled ship model could have a marked heeling effect on the ship (he doubted if this would surprise any mariner). By suitably phased cycling of the rudder, they built up a formidable forced roll in the model. No doubt if the phasing had been reversed the rudder could have been used as a stabilizer.

This raised the question: was there any risk of the helmsman (or auto-pilot) interfering with the action of the stabilizer and ought we not to investigate the cross-linking between yaw and roll, and then control both fins and rudder together from a common "black box" much as was done in a guided weapon? The authors' comments on the possibility of future developments along these lines would be most interesting.

MR. G. H. C. BOYLES wrote that during the verbal discussion Mr. Head had expressed concern about the heavy wear and tear of submerged parts of fin stabilizers. In particular he cited severe erosion of fin and flap and also the need to replace flap hinges and bearings after relatively short service periods. The existence of unacceptable hydrodynamic noise in some stabilizer systems was also alleged to be a further source of annoyance.

Although these shortcomings were known to exist in some fin stabilizers, it was not true of all fin stabilizers.

If it was assumed, and it was thought with good reason, that the source of trouble giving rise to damage of the fin-flap assembly and to erosion of fin and flap surface, was overloading of the foil and consequential cavitation then it was clear that, to avoid these conditions, the fin should be designed to operate within the limits of acceptable loads and tolerable cavitation. This was done to good effect in his company's ship roll stabilizers.

An unique solution was implemented in the control of this system. Instantaneous forces experienced by the fin and hence lift generated by the fin were recorded and used to close the control servo-loop so that the actual lift achieved by each fin was continuously related to the lift ordered and instantaneously adjusted to the required lift up to the rated maximum of the system. Thus, the fins could not be overloaded and optimum efficiency of operational performance was ensured.

Moreover this fin stabilizer was designed to produce its maximum rated lift at an angle of attack, relative to water stream, which was commensurate with the cavitation criteria imposed by the ship/fin stabilizer system. In most cases, this angle of incidence was 16·5°. It would be noted, therefore, in referring to Fig. 18 that the fin, in producing its maximum lift at an angle of 16·5°, operated in the most favourable region of the curve with the operational angle of attack far removed from the point where excessive cavitation destroyed performance and gave rise to high levels of hydrodynamic noise.

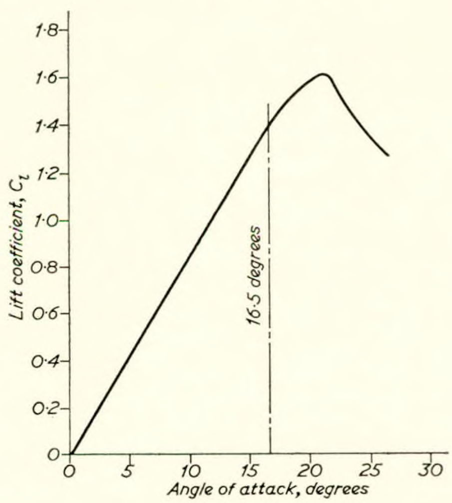


FIG. 18—*Typical curves of lift coefficient versus angle of attack for a fully flapped fin (Ref. Taylor model basin report No. 950)*

Although the fin produced its maximum lift at ±16·5° relative to the water stream, it was capable of rotating through ±25°. Because the fin, in seeking to produce the desired amount of lift, would be angled to the water stream it would accommodate a minimum of 8·5° of false angle which arose from the other motions of a vessel in a seaway and local flow conditions in the region of the stabilizer position. Since the lift produced by any particular fin installation was dependent upon V^2 (V = ship speed) it would readily be appreciated that in installations in ships where the ship speed was in excess of the speed at which maximum rated lift of fin was achieved, the fin angle of attack would be considerably reduced, hence further increasing the false angle capability of the system and the overall efficiency of the installation.

Noise generated when the fins were stowed, could be diminished to an acceptable level not only by judicious siting of the fins but, probably and more important, by ensuring that the fin aperture was hydrodynamically clean and as small as practicable. With an aft folding fin configuration it was possible to close the large portion of the keyhole slot in way of the rigging axis and virtually fill the remaining slot by the fin itself. This ensured the least disturbance of the water flow past the fin box aperture.

When comparing the loss of ship's speed due to fin stabilizers and to tanks, one must consider the respective merits of each system as effective stabilizers. A ship's resistance was increased when rolling and in rolling greater rudder activity was necessary to maintain course, which itself augmented hull resistance. Fin type stabilizers provided a high degree of roll stabilization over the full range of disturbance frequencies. Passive tank systems provided a much lesser degree of roll stabilization over a narrow band of disturbance frequencies around the ship's natural rolling frequencies and in sea states of a much lower magnitude. In some conditions tank systems augmented free rolling. Therefore, although tank systems themselves might not appear to contribute directly to increased hull resistance they did so indirectly. It was therefore thought that the two systems suffered equally in producing increased resistance.

Considering Mr. Eddie's closing remarks concerning a device to sense wave formations ahead of the vessel, he could envisage considerable practical problems in conceiving a satisfactory solution. However, his company's control system instantaneously sensed the disturbance at the ship and the "lift control" feature discussed earlier monitored variations in flow conditions around the fin and compensated them in the course of providing a degree of stabilization not obtainable in angle control systems.

Authors' Reply

In replying to the discussion, the authors fully agreed with Messrs Chaplin and Eddie on the "misnoma" of the word "stabilizers" in fishing vessels, since any anti-roll device fitted to this class of vessel consisted of some sort of tank system, whether pure passive, passive controlled or in one instance activated. A better name would be "roll damping device" and not "stabilizer". They had always advocated the fitting of bilge keels to the model, when a tank system was being tested in a ship model basin, because otherwise, as the contributors correctly pointed out, an entirely false and misleading idea of the expected roll reduction was being given. Moreover, the model should be tested at the appropriate speed and not stationary.

The idea of sensing the wave profile ahead of the ship was not new. One of the leading hydrofoil designers and builders, apparently carried out some tests with such a device to control the blade angle of a hydrofoil. Whether this device was successful and was exploited commercially was not known to the authors.

It was fully appreciated from Mr. Garrett's contribution that shipowners and operators had to weigh up the cost and relative merits of fin stabilizers versus tanks. It must be emphasized that a fin stabilizer above a certain speed range would be a far more efficient roll damping device than a tank. Ship speed as such did not enter into the tank design parameters, but was an important factor in a fin stabilizer. Hence the cost ratio between tank and fin varied from ship to ship and no hard and fast figure could be given.

It was true that, for the successful design of a passive tank system, the minimum natural rolling period of a proposed vessel should be known at an early stage.

The simplest empirical approximation was the well known formula for period:

$$\frac{k \times \text{Breadth of Ship}}{\sqrt{GM}}$$

where k was a factor depending on type and size of vessel.
An attempt had been made recently to compute all the values available from actual ship trials and model experiments and to plot the k factor against the term

$A = \frac{D}{(L/100)^3} \times \frac{B}{H,}$ where D = displt. tons s.w., L = length of ship on w.l., B = breadth and H = draught (see Fig. 19).

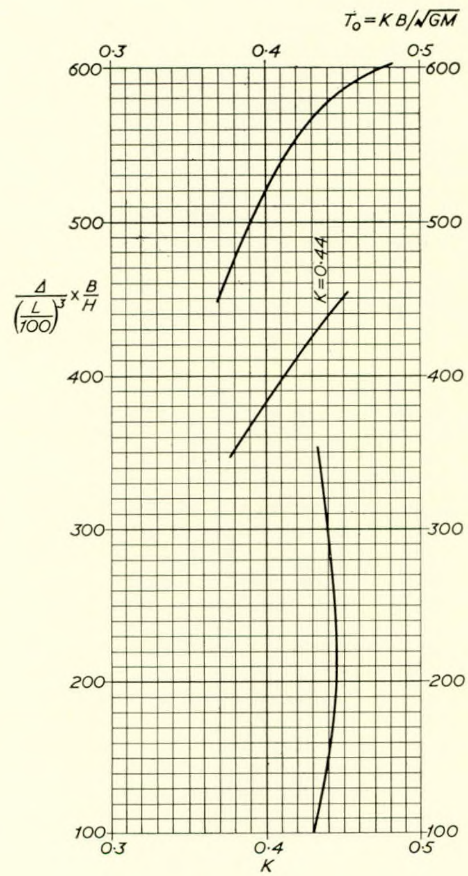


FIG. 19

With regard to applying rolling periods obtained from model tests to full scale ships, several years ago such a comparison was made for vessels such as the original *Queen Elizabeth*, *Chusan*, *Media*, *Saxonia* and a large tanker *J. P. Gettie*. The difference was not more than five per cent which was considered quite satisfactory. Recently similar rolling tests were carried out on a class of large container vessels, but some time would elapse before these vessels went into service, to check the model/ship comparison.

Mr. Head had raised the question of maintenance of fin stabilizers. The modern athwartship retractable gear with cylindrical fin box no longer had slides and guides subjected to heavy wear and tear, while the folding type with all the mechanical parts working in oil should also have little or no maintenance requirements. The problems of protective fin coating was constantly under review, but unfortunately up till now no entirely successful material had been found. The best solution would be for the owners to shoulder the higher initial cost of adopting stainless steel fins or clad larger fins with stainless steel plating, as was done in the French liner *France*.

Certainly improvements had been effected over the years on tail fin hinges, bushes and driving gear, which had greatly reduced maintenance costs. There was no information available on the noise generation in tank stabilizers, because the majority of vessels so fitted did not carry passengers. The use of fluids of heavier gravity in order to increase the tank's effectiveness was theoretically correct and drilling mud was actually tried in the tank of the research vessel *Second Snark*. It was very difficult to keep it suspended in the liquid, one was never certain how much remained in the bottom, thus density and viscosity changed continually and the successful tuning of such a system was extremely difficult. Any improvement on *Second Snark* proved to be only marginal and it was not found worthwhile continuing this investigation.

Mr. Bunyan could rest assured that continuous research was being carried out to improve the efficiency of fin stabilizers and to look into new ideas.

As to the general concept of container ship stabilization, owners were becoming more inclined to shoulder the higher initial cost of fin stabilizers, although in the past some were fitted with pure passive tanks of the flume type. Apparently one container vessel abroad was fitted with a combination of tank and fin, but without great success. Mr. Bunyan might have put his finger on the problem of successful combination of fin and tank, i.e. the change in direction of the large masses of water and the correct phasing of fin and tank in various loading conditions. This of course applied to cruise vessels, which Mr. Bunyan found somewhat worrying. It was not inferred from this that it was impossible to hit upon a successful combination; however, with the present state of the art it might be prudent to consider the two systems as separate entities.

In connexion with Fig. 10, which showed the penalty for fitting non-retractable fins to a large container vessel, the annual cost figure of £3000 p.a. for extra fuel was arrived at by taking a more pessimistic view than Mr. Bunyan of the time per annum spent at sea plus a better value of fuel consumption per hp/h. The authors found general agreement on the matter of bilge keels, but unfortunately had never come across comparative full scale data for the same vessel with and without bilge keels, since it would be very difficult to find a shipowner, who would be prepared to run such full scale tests over a period of time in a properly scientific manner.

The authors agreed with Mr. Dunshea that when designing fin stabilizers consideration should be given to using a slower speed than the normal service speed in the design. This was usually done especially for vessels on an arduous service like those plying in the Irish Sea. However, it would be of great help to the designers, if the operators could state, from their practical experience, the mean speed for which stabilizers should be designed. In all due fairness to shipowners, some stated the speed, at which full stabilization should be achieved, but unfortunately they were in the minority. With a view to reducing the space required for a tank stabilizer, experiments were carried out in *Second Snark* with drilling mud in the water, but the results did not warrant any further investigation.

Dr. Conn and Mr. Ferguson were quite correct in their assumption that the static value of GM was usually taken for the power requirement of a stabilizer. Very little was known on the effect of forward motion on the GM, until Dr. Conn and Mr. Ferguson recently presented their very informative paper. Once more was known of this particular problem, a corrected GM value would have to be used; this correction will certainly vary for various types of ships and speeds.

Mr. Judd had raised the question, whether a tank stabilizer could have an adverse effect on a passenger vessel in the damaged stability condition. So far no difficulty had been encountered, quick dumping of the water into the double bottom had been allowed by the classification societies in the few cases when tanks were fitted to passenger ferries. In one proposed scheme for a large passenger liner, the use of compressed air was seriously considered to hold the water against the heel of the ship, but the provision of a large number of compressed air bottles and appropriate valve system would have been rather expensive.

For a very preliminary assessment of tank sizes and weight of water for a tank stabilizer of the Brown—NPL type, the following method could be adopted:

Take a vessel of say 60 ft breadth, 5000 tons displacement, 3 ft 0 in GM and assume the tank extended over the full breadth of the vessel:

1) under normal circumstances a reasonably effective tank should produce between 30 and 40 per cent free surface loss, i.e. in this case about 12 in;
2) the height of the water in the tank should be about 2/3 GM = 24 in. in this case;
3) therefore the free surface effect = 1 ft 0 in

$$= \frac{\text{length of tank} \times \text{breadth}^3}{420 \times \text{displacement}}$$

$$= \frac{L \times 60^3}{420 \times 5000}$$

$$L = \frac{420 \times 5000 \times 1{\cdot}0}{60^3} = 9{\cdot}75 \text{ ft};$$

4) amount of water in tank $= \dfrac{9{\cdot}75 \times 60 \times 2}{35{\cdot}9} = 32\tfrac{1}{2}$ tons.

The authors were very grateful to Mr. Pangalila for publishing the actual voyage analysis of *William Carson* with a fin/tank combination. The results looked very satisfactory, but unfortunately it was the "odd man" out compared with some others, where such a combination had not been as successful. Results, similar to those for *William Carson* and shown in Fig. 14, were from continuous graphical roll records obtained in the experimental vessel *Second Snark* of only 70 tons displacement. It would be too early to make a definite statement for or against the fin/tank combination and for this reason the authors were recommending that owners in the meantime should consider the two systems independently.

Mr. Wallace had made a plea for the athwartships retracting fin gear, which with up-to-date design ideas could be very efficient with low maintenance problems. More than one firm in the stabilizer business has been pursuing this quite successfully, but low aspect ratio fins had to go to a greater angle, say to about 20° to 30°, to produce the same lift as a high aspect ratio fin at lower fin angle. Therefore erosion/corrosion problems would also appear on the low aspect ratio fins. Usually container vessels had rather narrow cofferdams and the vessels which Mr. Wallace mentioned were probably already designed with a somewhat wider cofferdam to house an athwarthship fin gear.

It was admitted that the adoption of lift control, which Mr. Boyle had strongly advocated for fin stabilizers, was theoretically correct. Pressure transducers or strain gauges to sense the correct lift must be fitted on the fin itself or on the fin shaft, in both cases in somewhat inaccessible places. There was no guarantee that such sensing devices would, over a period of time, give almost the same answer and, as they were not readily accessible, they could not be easily checked or, in the worst case, renewed. Therefore, it might quite possibly happen that the lift control had to be put out of action and one had to revert to the ordinary angle control.

In the full scale trials of the old cruiser *Cumberland*, mentioned by Mr. Paffett, the smaller heel obtained in practice was more due to a counter moment produced by the forward shoulder effect than due to yaw. Actually Mr. Paffett produced a trial method, from which the correct waveslope capacity could be deduced with certain assumptions. This method had been tried out quite successfully on many occasions, when the true waveslope capacity of a fin stabilizer had to be demonstrated on a full scale trial. The late Mr. Bell of Messrs. Muirhead investigated some years ago the complex problem of interaction between roll and yaw, but came to no definite conclusion. This certainly presented a field for further research and development, which could only be pursued on full scale trials. However, opportunities for such extra trials were seldom afforded to people engaged in the stabilizer design business.

Marine Engineering and Shipbuilding Abstracts

No. 9, September, 1970

Reversing Propulsion Gear

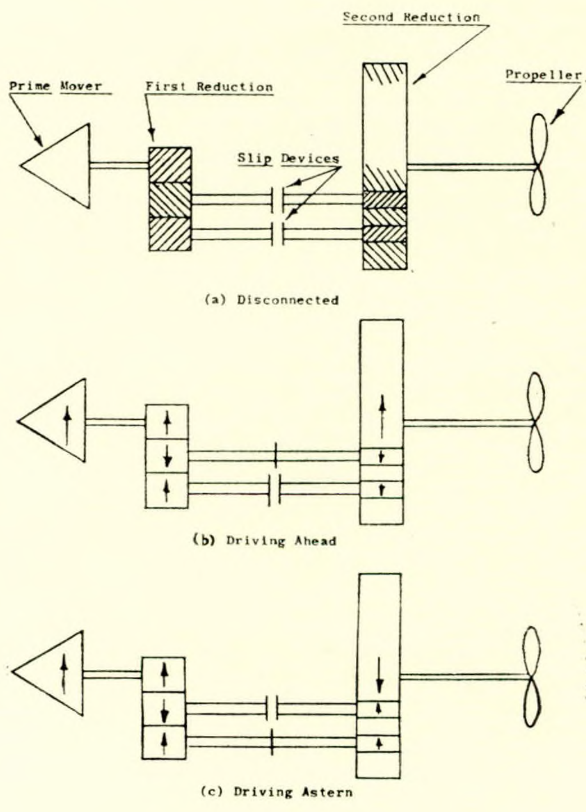
A vessel in service requires astern propulsive effort for only a small proportion of its useful life. It must, however, have a reasonable ability to manoeuvre and to avoid collision. Realizing that this requirement is subject to legal interpretation, performance equivalent to that of a comparable steam-turbine-powered vessel (with reversing stages) ought to be a reasonable objective for the reversing gear-equipped vessel.

Reversal of propeller rotation to reverse the effective propulsive effort on the ship may be realized as the cumulative result of a number of discrete operations affecting the entire propulsion system. These steps are summarized in terms of their gross effects:

1) decelerate system to idle condition;
2) decouple the initial power path;
3) couple the alternate (reversed) power path;
4) accelerate the reversed power train.

The sequence in which these steps are performed is important to the success of objective function.

The torque loads imposed on reversing gears are no different from those for standard reduction gears of similar rating, but there may be need for additional derating factors to accommodate the reversing transient. The possibility of impulse loading resulting from improper sequencing of operation or from erratic operation must be considered. Also, the ratio split in double reduction gear sets may be influenced by the type of slip device selected. In some cases, it may be desirable to provide a large final reduction so that the slip device may have the greatest possible mechanical advantage over the propeller. Similarly, the speed at the slip device shaft may effect its torque capacity —for better or for worse. Some friction clutches are speed sensitive, tending to open under the influence of centrifugal forces. In such a case, the final reduction should be small



Reversing propulsion gear

to permit the clutch to operate in the speed range where it has the desired torque capacity. At the same time a fluid coupling generates torque in proportion to the square of shaft speed. For such a device it is desirable to operate at the highest speed consistent with desired torque and stress limitations.

Special considerations must be given to the bearings on those shafts which are active reversing elements. In addition to the change of direction of reactions encountered in reversal, the function of one or more shafts may change; in offset gearing, one element may become an idler, in which case the reaction magnitude may be doubled.

A reversal of rotation occurs at each gear mesh in an offset gear train. As a result it is a simple matter to achieve reverse rotation: alter the torque path to include one more (or one less) mesh when the direction of rotation is to be reversed. The schematic arrangement of the figure shows how this may be done.—*Stahly, C. L., Maritime Reporter/ Engineering News, 5th February 1970, Vol. 32, pp. 20–21.*

Ro-Ro/Container Ships for North Sea Services

Evidence of the tremendous growth rate in container, trailer and (seasonally) private car traffic across the North Sea is provided by the recent entry into service of Triport Shipping Company's new Ro-Ro vessel *Tor Mercia*, to be followed later by her sistership *Tor Scandia*.

Access to the main deck is by a 40 ft long stern loading ramp. Its width (23 ft) allows ISO containers to be taken aboard by fork lift trucks in the athwartships position, so simplifying stowage arrangements. The stern ramp has hydraulically-operated locking devices and is capable of being lowered and raised 10° from the horizontal to accommodate tidal differences. The hydraulics for the stern ramp are of Brodr. Bauer-Nielsen A/S supply.

An interesting point is that no bow door has been incorporated to permit straight-through drive. North Sea operations have convinced the owners that weather conditions can be too severe for a bow door to be incorporated.

There is space on the main deck for 125 × 20 ft containers, stacked two high, or 50 semi-trailers, or a combination of both. Parking space for 70 private cars is provided on two portable car decks, port and starboard in the forward section of the main deck. Access to these decks is by portable ramps. The shelterdeck aft of the superstructure can be utilized for 34 empty containers.

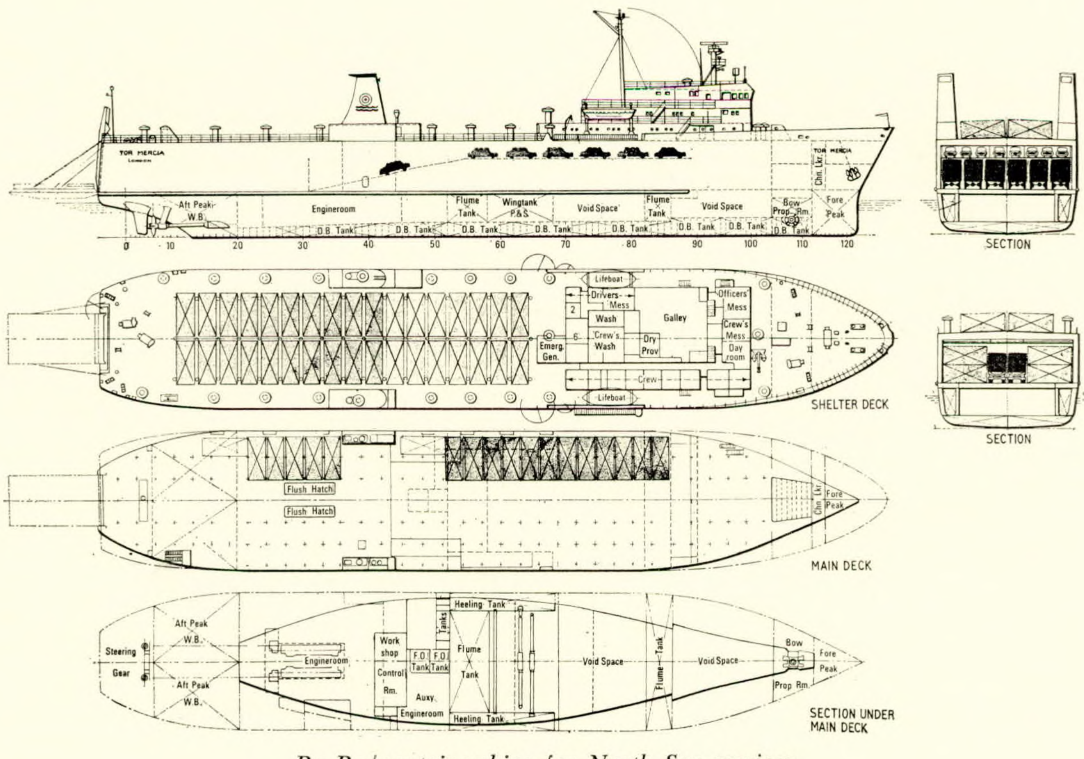
Principal particulars are:

Length, b.p.	100·00 m
Breadth, moulded	19·20 m
Depth (main deck)	6·20 m
Depth (shelter deck)	13·00 m
Draught	4·95 m
Gross register	1600 tons
Deadweight	2500 tons
Propulsion output	2 × 3500 bhp
Speed (trials)	18 knots
Speed (service)	17 knots
Capacity (20 ft containers) ...	125
or (semi-trailers)	50

Tight schedules and unmanned engine-room operation have resulted in the choice of an installation in *Tor Mercia* incorporating considerable automation, but not over-elaborate simply for automation's sake. The outfit has been designed for two-man operation unmanned for 16 h periods. As is normal, however, for the first six months a chief and three engineers will be carried.

Main propulsion is by two Lindholmen-Pielstick type 8PC2L Diesels giving a combined continuous output of 7100 bhp at 500 rev/min. This output will be increased to 8000 bhp when modifications have been carried out to the turbo-blowers.

Lohman and Stolterfoht type GVA630B 2:1 reduction gears connected through Spiroflex couplings transmit the engine output to twin KaMeWa four-blade propellers. Twin hanging-type rudders are fitted. Manoeuvring capability is enhanced by a KaMeWa 600 hp bow thruster.—*Shipbuilding and Shipping Record, January 2nd 1970, Vol. 115, pp. 19–21.*



Ro-Ro/containerships for North Sea services

The Hamburg–America 'Omni' Ships

The conventional cargo ship on a liner trade is dead. The cellular container vessel, without boxes, is quite useless. Lifts, pallets and bulk liquids, not to mention refrigerated capacity, require the gear to handle them. But with the delivery from Howaldtswerke-Deutche Werft of the 15 550 dwt *Ludwigshafen*, Hamburg-America Line are confident they have a vessel that will handle boxes, bulk, pallets and lifts with equal facility.

Flexible enough to serve Transatlantic, Central American or Far East/Australasia services alike, *Ludwigshafen* is the first of four sophisticated cargo liners to be built for the company by HDW. Known now as *Omni* ships, all four are scheduled to be in service by the autumn of this year.

Built with a bulbous bow and transom stern, *Ludwigshafen* is sub-divided by seven watertight bulkheads into six dry cargo holds with a bale capacity of 780 000 ft³. Hatch 1 has two decks with cargo bulk oil tanks in the lower hold. Hatches 2 and 5 each have two 'tween decks and a deep lower hold, suitable for grain and bulky cargoes, hatch 2 being suitable for long loads. The remaining hatches have a standard 2·7 m deck. All decks are obstruction free and have flush coamings for the use of fork lifts. With shelter deck doors open, fork lifts can travel the full length of the upper 'tween decks working several holds at once. Holds No. 3, 4 and 5 are fitted with three sets of hydraulic hatches and little dragging of cargo into the wings will be necessary.

Principal particulars are:

Length, o.a.	165·10 m
Length, b.p.	155·00 m
Breadth, moulded	24·50 m
Depth to weatherdeck	14·50 m
Speed	23 knots
Bunker capacity	2450 tons
Range	14 500 miles
Draught (closed)	10·70 m
Draught (open)	9·75 m
Deadweight (closed)	15 550 tons
Deadweight (open)	12 650 tons

No. 6 hold immediately abaft the superstructure island consists of 30 000 ft³ of refrigerated capacity on two decks, served by two small hatches. Cooling is provided for a range from +12°C to a deep frozen −25°C.

A loaded operational sea speed of 23 knots is expected from the single acting two-stroke cross head engine from HDW-MAN. Designed for a maximum continuous output of 22 500 hp at 122 rev/min the direct-coupled turbocharged K9Z86/160F engine is suitable for heavy oils and a bunker capacity of 2450 tons is carried, mainly in the deep tank under the after part of the ship. This will give *Ludwigshafen* a range of about 14 500 miles.—*Shipbuilding and Shipping Record, 20th and 27th February 1970, Vol. 115, p. 36; 38.*

Brittle Fracture Strength of Welded Joints

In the majority of brittle fracture casualities in welded structures it has been reported that the fracture initiated at a pre-existing or elongated weld defect or crack along the bond or in the weld metal rather than in the base metal. Sometimes the brittle fracture is influenced by the superposition of welding residual stress.

This paper describes the results of studies of the brittle fracture initiation characteristics of welded joints for various high strength steels including 80 and 100 kg/mm² high strength steels and the low temperature structural steels including 9 per cent Ni steel. These were welded with various heat inputs and were evaluated by using the deep notch test specimens with welded joints. The heat input control is needed not for HT60 but for HT80.

In addition, the effect of super-imposition of residual stress on the increase in brittle fracture initiation temperature for various high strength steels was investigated theoretically by means of fracture mechanics concept.

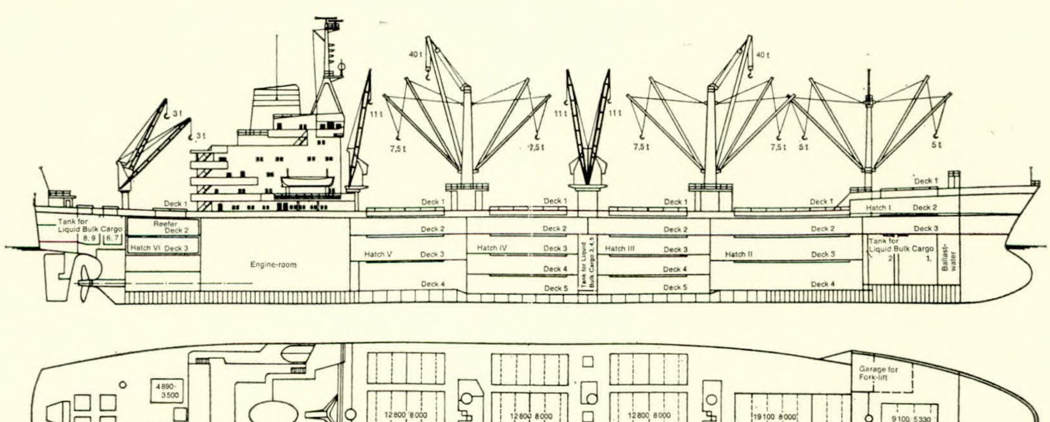
Finally, the characteristics of the brittle crack path in the region of the welded joint for various kinds of steel were observed. The brittle crack propagates along the welded joint when the weld metal is deposited by submerged-arc welding, or the applied stress is large for HT80.—*Ikeda, K. and Kihara, H., Welding Journal, March 1970, Vol. 49, pp. 106-s–114-s.*

Bridge Control System for Propulsion Machinery with c.p. Propeller

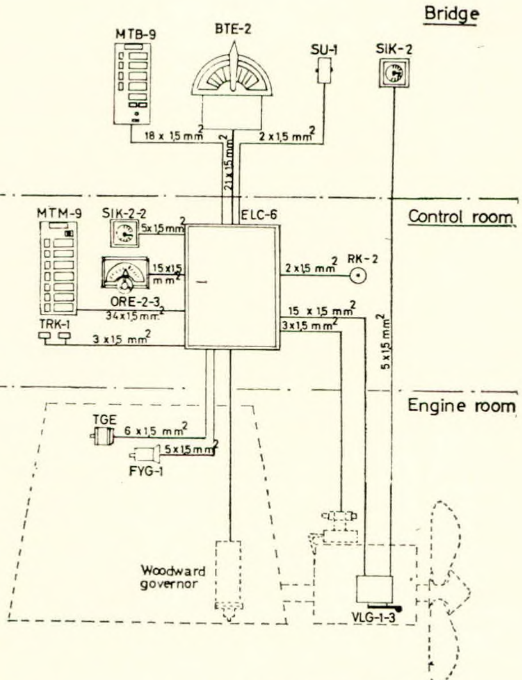
The FAMP-2 has been designed so that it can be adapted to different types of engine as well as c.p. propeller systems.

The system is operated by means of a switch on the control panel MTM-9 in the control room, the positions being "O", "manual control" and "bridge control".

In the "O" position the entire system is shut off and void of current. In the "manual control" position the bridge telegraph and order repeater are engaged in the manner of a conventional engine-room telegraph system. In this position the propeller pitch angle and engine speed can be set manually



The Hamburg-America Omni *ships*



Bridge
Bridge telegraph BTE-2
Control panel MTB-9
Buzzer SU-1
Pitch indicator SIK-2

Control room
Central control unit ELC-6
Order repeater ORE-2-3

Control panel MTM-9
Pushbuttons (2) TRK-1
Pitch indicator SIK-2-2
Alarm bell RK-2

Engine room
Angular position indicator VLG-1-3
Tachogenerator TGE
Fuel injection transmitter FYG-1

Bridge control system for propulsion machinery with c.p. propeller

in the control room by means of pushbuttons (increase/decrease).

In the position "bridge control" the system is engaged for automatic control from the bridge. Acceptance of control is acknowledged by personnel on the bridge by means of an acknowledgement pushbutton on control panel MTM-9. At this point the servo transmitter in the bridge telegraph is turned on. The servo transmitter is of the synchro type and is connected mechanically to the control lever of the bridge telegraph. It supplies the electrical reference signal for the setting ordered from the bridge.

The electrical signal from the transmitter in the bridge telegraph is fed to the central control unit ELC-6, where it is transformed in the electronic combinator calculator. The calculator emits two electrical signals corresponding to the desired values for engine speed and propeller pitch.

Depending on the make of the main engine and propeller system, the mechanical adaptation required for rev/min and propeller pitch settings varies somewhat, although the principle is the same for most current makes.

A control transformer of the synchro type is mounted in direct connexion with the propeller pitch setting mechanism. The control transformer senses the actual angle of pitch of the propeller mechanically; and if the reference value of the calculator does not agree with the electrical value from the control transformer, the central control unit delivers an electrical control voltage to actuate the propeller's setting mechanism to obtain the desired propeller pitch.—*Shipbuilding and Shipping Record, 9th January 1970, Vol. 115, pp. 21–22.*

Shock Testing Machine for Heavy Installations and Parts

For testing pumps, turbines, couplings, air conditioning units, converters, rectifiers etc. for shock resistance, the Netherlands Institute for Applied Scientific Research (TNO) has developed a shock tester, which can give an object weighing two tons an acceleration of 600 times its own gravity, at a speed of 0·3 m/sec. This testing machine comprises essentially a moving crosshead with top plate, a number of sets of Belleville springs, a fixed frame and eight hydraulic cylinders, the piston rods of which are connected to the crosshead.

This design is believed to be unique, since the power is provided by the sets of Belleville springs, while the force which compresses the sets of springs is provided *via* a fracturing bolt by the hydraulic cylinders.

The force is determined by the compression of the set of Belleville springs. By determining the depth of compression in advance by means of a fixed pressure ring, the fracturing bolt can subsequently be fractured by the build-up of a greater force, while the force of maximum 1200 ton, with which the object is accelerated, remains constant.

With steeply climbing acceleration forces of this kind, vibrations with various frequencies, which can influence the measuring signals and the properties of the object to be measured can occur in complicated structures such as this shock testing machine. To prevent this, damping layers of self-adhesive tape have been mounted in the Belleville spring sets and in the top plate.—*Holland Shipbuilding, February 1970, Vol. 18, p. 48.*

Residual Stresses in Heavy Welded Shapes

Residual stresses can have a significant influence on the load-carrying behaviour of structural steel members subjected to compressive loads. Previous experimental research on residual stresses and the strength of columns was related to small and medium-size shapes. In today's large structures, increasingly heavy shapes are employed. While heavy column shapes are being used extensively, very little information has been available on the residual stresses and strength of such members.

This paper presents the results of the first phase of a major investigation into the residual stresses in, and the behaviour of, thick plates and heavy shapes used in compression members. The shapes considered in this initial study are a 15H290 shape and a 23H681 shape, as well as two loose component plates, PL16x2 and PL24x3½. For the smaller shape, comparative tests were carried out for different manufacturing conditions of the component plates (universal-mill and flame-cut plates), different weld type (penetration) and different yield strengths of the material.

The results of residual stress measurements carried out in this first phase of the study indicate the following for heavy fabricated members:

1) all phases of the manufacture and fabrication procedure generally affect the formation of residual stresses;
2) the weld type (penetration) and the yield strength of the steel are not major factors in the formation of residual stresses;
3) the geometry of the plates and shapes is one of the important variables affecting the residual stress magnitude and distribution;
4) the variation of residual stress across the thickness of plates more than 1 in thick can be considerable;
5) the welding residual stresses in portions of the cross section other than the weld area tend to decrease with increasing size of the member, probably because the weld area, and consequently, the heat input, is relatively smaller in heavy plates and shapes as compared to light members;
6) the initial stresses can be of a higher magnitude than the welding residual stresses.

The relationship between initial residual stresses in component plates and welding residual stresses implies that efforts to limit the magnitude of residual stresses in heavy welded shapes should be directed towards the manufacture of the component plates. Thus, by using flame-cut plates in heavy welded shapes, there is a prospect for an increase in strength when compared with lighter members at the same slenderness ratio. There is even a possibility that such welded shapes may be stronger than their rolled counterparts.—*Alpsten, G. A. and Tall, L., Welding Journal, March 1970, Vol. 49, pp. 93-s–105-s.*

Inert Gas Generators

Recent explosions on board mammoth tankers have again aroused interest in the supply of inert gas on board ships, because in a hold protected by inert gas the danger of explosion can be ruled out, no matter what the cause of past explosions may have been.

The amount of inert gas required for this purpose is determined by the pumping capacity of the ship, because the liquid pumped out is not to be replaced by air, but by inert gas. As a result, the capacity required for large tankers may well be as much as 15 000 Nm^3/h. If it is assumed that inert gas generators having a capacity of 5000 Nm^3/h can be built, it will be necessary to install three of such inert gas plants.

The advantages of the inert gas generator are:

1) gas of a high purity is produced at all times;
2) the gas generator operates completely independently and thereby increases the safety;
3) the installation can be mounted in various parts of the vessel;
4) light marine Diesel oil with a low sulphur content, can be used as a fuel.

Smit-Nijmegen builds series of large inert-gas generators having capacities of 500, 1000, 1500 and 2000 Nm^3/h. These generators are used in chemical tankers, L.P.G. carriers (LPG / NH_3-tankers), etc.

The figure shows a generator for 1500 Nm^3/h being tested. This generator is intended for installation in a chemical tanker where the purity of the inert gas must fulfil stringent demands. The maximum oxygen content, for instance, is 0·3 per cent and that of hydrogen and carbon monoxide only 0·05 per cent, each.—*Holland Shipbuilding, March 1970, Vol. 19, pp. 54–55.*

Propeller Vibration Forces

Described is a modified technique for measuring vibration forces on self-propelled models and test results for Series 60, 0·6 block coefficient, single-screw models with a variety of 4, 5, and 6-bladed propellers with varying propeller and rudder clearances and other parameters.

The 10-ft model used, consists of an aluminium skeleton model to which is attached a changeable outer shell of fibreglass plastic skin with foam plastic filler between the skin and the skeleton. The skeleton was designed for 10-ft series 60 models of parent V or U form. It has an 0·6 block coefficient and 16-in beam, but can be used for any wider or fuller model. The model is designed to have all of its natural frequencies well above the highest blade frequency with a six-bladed propeller, so that it acts substantially as a rigid block.

Forces are measured by a comparison of the amplitude and phase of vibration of the model produced by propeller forces with the amplitude and phase vibration produced by known forces of the same frequency.

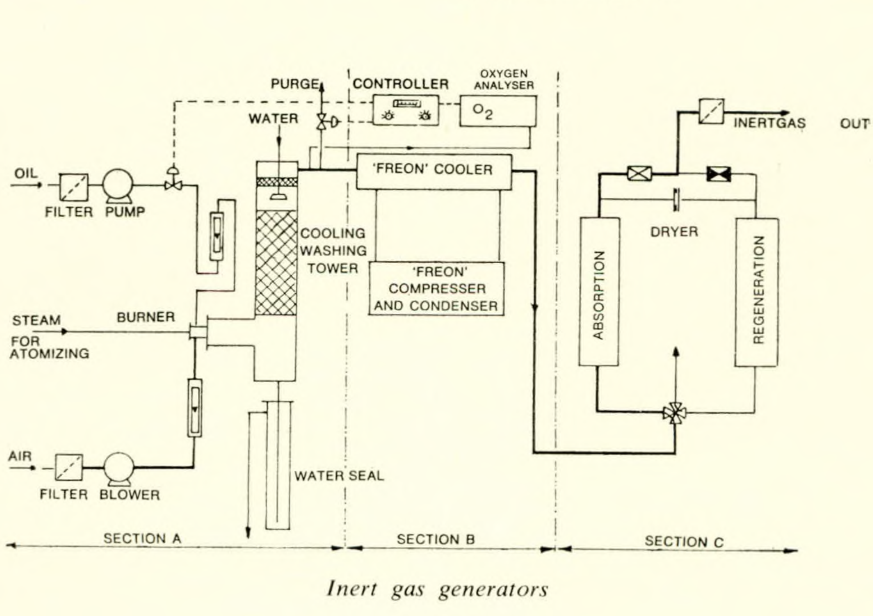
With a propeller mechanically and hydrodynamically balanced and steady wake, forces are periodic with harmonic components of blade frequency; i.e. the rev/sec times the number of blades, or multiples of blade frequency. An erratically varying wake, such as may occur in a very full ship, will produce erratic forces. A steady wake is assumed in this work.

It is evident that there is little to choose between the 4, 5 and 6-bladed propellers insofar as vibratory force is concerned, provided they are operated at the optimum points. At the present time the optimum points for any other combination of hull, rudder, and propeller can be determined only by test.

The tests show that many unknowns exist in our knowledge of the phenomena. No adequate theory exists for the surface forces on axial-clearance surfaces or the bearing forces produced by such surfaces with or without a variable wake.

No adequate theory exists for the rudder forces. The rudder forces have two components, one produced by the bound vortices of the propeller and the other by the trailing vortices. It is the trailing vortices that produce the periodicity of the total horizontal force.

For many things found by these measurements not even a qualitative explanation can be given at this time.—*Lewis, F. M., Maritime Reporter/Engineering News, 1st February 1970, Vol. 32, p. 18.*



Inert gas generators

UCS Launches Seagoing Hopper Dredge

Upper Clyde Shipbuilders have launched from their Govan Division the sea-going hopper dredge *Pacific*. The vessel is being built for the D.O.S. Dredging Company Ltd.

Pacific is the largest dredge of this type so far built in the United Kingdom and is the first of a new design of twin hopper suction dredges developed by Simons-Lobnitz Ltd., the dredge-building subsidiary of Upper Clyde Shipbuilders.

Pacific is a twin screw, twin side pipe, trailing suction hopper dredge with the following main dimensions:

Length, o.a.	432 ft (131·67 m)
Breadth, moulded	63 ft (19·20 m)
Depth, moulded	33 ft (10·06 m)
Hopper capacity	9250 cm^3
Speed approximately	13 knots

The dredge is capable of dredging to a depth of 80 ft below the light waterline with provision for an extension of the dredging depth to 98 ft. All the dredging gantries, machinery and lifting appliances are designed to accommodate the extended dredging depth.

The principle of the two hoppers is primarily to give a much better disposition of hull loading throughout the length of the ship, so that longitudinal bending moments and shear forces are reduced to acceptable values.

Two dredge pumps of the centrifugal single inlet type, each driven by a Diesel engine developing 2000 bhp are fitted with an overhung impeller and suction and discharge branches of 1000 mm and 900 mm bore respectively. Bow thrust is also effected from the dredge pump by means of a branch pipe led forward and branches fitted port and starboard to an aperture on the shell. The thrust obtainable using one dredge pump is 5 tons.

The main propelling machinery consists of twin Smit-Bolnes Diesel engines driving twin screws and developing a total of 8800 bhp.—*World Dredging and Marine Construction, February 1970, Vol. 6, pp. 24–25.*

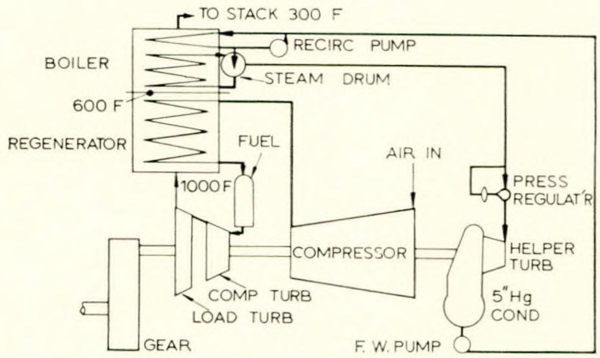
Marine Use of Industrial Gas Turbines

The use of gas turbines for the propulsion of merchant ships has been a topic of dicussion between marine engineers for many years. So far few shipping companies have accepted this form of prime mover mainly because it has not been an economically viable proposition. Shell's tanker *Auris* was an early experiment with gas turbine machinery, but this vessel was scrapped about 12 years ago. The American experimental ship *John P. Sergeant* went into service in 1956 and logged some 9700 h before being taken out of service for economic reasons. Another American vessel, *Adm. Wm. P. Callaghan*, the first commercial vessel to be designed from the start for gas turbine propulsion, was built in 1967 and is still in service. Two container ships are to be built in Germany for delivery in 1971.

Large tankers are in a horsepower range where the gas turbine can substantially alter the economics of the conventional Diesel or steam turbine, but are operated by owners with long classical propulsion plant experience. A completely different situation prevails in some of the more specialized types of carriers that are recently becoming so popular.

The principal objective in designing a plant for containerships has been to get very high powers in a very short engine room, permitting the maximum number of containers to be carried. Dicussions with various shipowners planning containerships suggested that a horsepower range of 20 to 30 000 shp would cover the majority of ships being planned.

The cycle shown in the diagram is a combined gas-steam arrangement that employs waste heat generated by the gas turbine exhaust to provide steam, which in turn drives a small steam turbine mounted on the gas turbine base. This steam turbine is coupled to the gas turbine compressor, and by "helping" the compressor turbine, transfers output capability



Marine use of industrial gas turbines

to the load turbine without any increase in fuel consumption. The overall effect is to reduce the fuel rate to a very attractive level. A supplementary benefit is also achieved in that the effect of ambient temperature variation on fuel rate is virtually eliminated. That is, the fuel rate is constant across a wide range of temperature variations—a very important feature when it is considered that many of these ships will be operating from or through the tropics.—*Balfour, T. T., Shipping World and Shipbuilder, June 1970, Vol. 163, pp. 787–790.*

Retractable Pelorus

The Canadian Westinghouse Co. Ltd. has developed a pelorus sighting device that retracts into the deckhead. Designed to save deck space, the device is ideal for vessels with small or crowded bridge configurations. Installation is straightforward and inexpensive whether as original equipment or as a replacement for conventional equipment on existing vessels.

Units consist of a retractable mount and a sighting head. The mount is a telescopic tubular arrangement with a latching height adjustment and tension springs for upward lift.

A standard gyro repeater compatible with most master gyros is housed in the sighting head and transparent windows in the sighting tunnel carry alignment lines that constitute a "V" sight. These lines as well as the compass card can be illuminated by control edge lighting when necessary. The entire head assembly is gimbal-mounted allowing for 20° of pitch and roll.—*Shipbuilding International, April 1970, Vol. 12, p. 29.*

Container Ship for Sea-land Feeder Service

At the end of December 1969, N.V. Scheepswerf Gebr. van der Werf, of Deest-Nijmegen, a member of Rijn-Waal Scheepswerven, delivered the shelterdeck vessel *Black Swan*, which has been constructed for operation in the Sea-Land Feeder Service, Rotterdam.

Principal particulars are:

Length, o.a.	97·15 m
Length, b.p.	90·0 m
Breadth, moulded	17·20 m
Depth to maindeck	7·75 m
Depth to 'tweendeck	3·85 m
Draught, summer approximately	3·80 m
Dwt capacity approximately ...	2500 metric tons
Container capacity	123
Fuel capacity	260 tons
Fresh water capacity	50 tons
Water ballast capacity	1150 tons
Gross rate tonnage	994
Reduction gear L and S	500/250 rev/min
Trial speed	14·75 knots

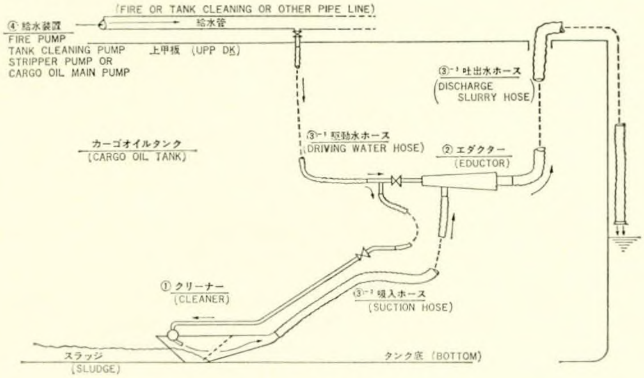
Black Swan has been constructed as a shelterdeck vessel with large hatchopenings on shelterdeck and main deck. The engine room and the accommodation for a crew of 20 are arranged aft. The ship has been designed to carry 123 standard-size containers with dimensions 35 × 8 × 8 ft $6\frac{1}{2}$ in, each having an average weight of about 20 tons. Among these there are 28 refrigerated containers which can be connected to the ship's electrical system by a fixed system of plugboxes.

The propulsion machinery consists of two 4-stroke, 6-cylinder, direct reversible, air-started fresh water cooled MWM Diesel engines of the type TbRHS 345 SU. The engines have a continuous uprated output of 2000 hp at 500 rev/min. The engines drive, through a Lohmann and Stolterfoht gear-box with a 2:1 reduction ratio, a 4-bladed Ni-Alu propeller, giving the ship a trial speed of $14\frac{3}{4}$ knots.

Electricity for power and lighting is supplied by two generating sets.—*Holland Shipbuilding, February 1970, Vol. 18, pp. 44–46; 48.*

Slurry-Type Sludge Lifter

Mitsubishi have developed a sludge lifter to facilitate the troublesome work of lifting sludge in the tanks of oil tankers and ore/oil carriers.



Slurry-type sludge lifter

The device is based on the application of slurry transport techniques to sludge lifting, with a view to reducing both the labour and time required for this operation.

With the Mitsubishi-developed unit, the sludge sucked into a cleaner is immediately turned into slurry with high-pressure jet water and a vertical flow in the mixing chamber. The slurry is then discharged overboard by an eductor.

Sludge lifting capacity is about 1–2 m^3 h. Weight of the cleaner is 3·5 kg.—*Tanker and Bulk Carrier, May 1970, Vol. 17, p. 22.*

Tanker Explosions

An urgent programme of research is in progress by Shell to investigate the possible cause of the explosions on board the tankers *Marpessa* and *Mactra*. It is now a matter of building up data and eliminating possibilities as rapidly as possible.

The trials conducted so far have provided certain pointers, for example, the possibility of a static electricity spark having caused the explosions, cannot be discounted. Preliminary tests in the electrostatic field, showed that it would be worth pursuing this line of investigation further and it was decided to carry out detailed tests during simulated tank cleaning operations.

So far tests have been carried out using uncontaminated water and a clean tank. Tests are under way now to see what effect the introduction of controlled quantities of oil into the water has on the generation of static electricity within the tank.

The fatal spark might have been caused, not by the gradual build up of a static charge, but by the impact of an object dropping on to the bottom of the cargo tank. To test the likelihood of this happening, Thornton Research Centre have had a 90 ft tower of scaffolding erected near one of the laboratory entrances and are conducting impact tests.—*Tanker and Bulk Carrier, March 1970, Vol. 16, p. 537.*

Fuel Bills Cut by Burning Slop Oil

Reductions in operating costs are the aim of every shipowner: the introduction of equipment which made residual grade fuel suitable for burning under boilers and in Diesel engines marked one of the major developments in this field.

There are a large number of occasions where slops from these residual fuels and cargo oil could usefully be burned but where contamination with water makes this a difficult procedure. For this reason a number of companies have worked at producing equipment to separate the water from the oil. Centrifugal purifiers are, of course, the most usual means but as the s.g. of the oil and the water approach unity, separation becomes increasingly difficult.

To overcome this problem the engineering staff of Cities Services Tankers, of New York, together with the Todd Organization and Combustion Engineering have developed a mechanical-electronic system which enables either crude oil, waste slop or Bunker C fuels to be burned with no manual intervention.

Known as the Nopol system, it provides the following advantages:

1) prevention of oil pollution;
2) reductions in tanker operating costs;
3) arrival at loading ports with clean ballast;
4) the tapping of crude oil cargoes for fuel supplies.

This ability to burn bunker C crude or waste slop oil offers tanker operators a choice of fuel to meet any situation. It has been estimated that about 600 barrels of slop oil can be recovered on a normal ballast voyage by a vessel of some 70 000 dwt. On voyage to shipyard, where vessels must dry-dock with clean tanks, recovery can be as high as 1200 barrels.

The burning of this slop oil, in addition to economic benefits, provides a solution to the pollution of the seas problem, since the burning of the oil requires the separation of water from the oil, thus eliminating the discharging of oil into the sea.

The process involves three distinct stages: first, clean water is drawn off from the tank washings and pumped overboard, limiting the amount of oil/water and emulsion to be treated. Crude oil and salt water from tank cleaning operations tend to form very stable emulsions which are difficult to break down by ordinary means, so chemicals or emulsion breakers are used to change the surface tension, causing the water and oil to separate in the fastest time possible.

The second stage reduces the volume of the remaining oily water mixture by the process of natural gravitation separation, which is assisted by a chemical process.

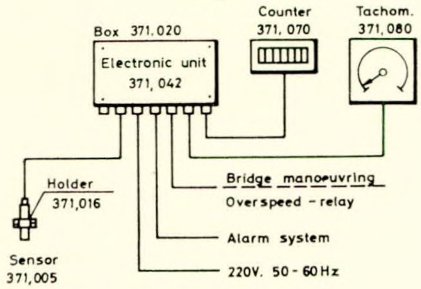
Finally, the remaining water is removed by a special procedure, leaving oil of a quality that can be pumped direct to the ship's boilers, where it can be burned without preheating.

Where, for reasons of economy or emergency measures, crude oil is used as fuel, this is conveyed directly from the tanks and pumped to the boiler-room, where the crude oil line is connected to the fuel burning system by means of a two-way change-over valve.—*Shipbuilding and Shipping Record, 17th April, 1970, Vol. 115, pp. 47.*

Monitoring and Control of Large Diesel Engines by Electronic Tachometer

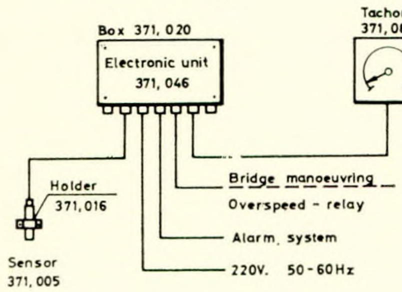
The Copenhagen electronics firm Soren T. Lyngso A/S has recently introduced on the marine market their tacho system type 371 for use in monitoring and control of large Diesel engines.

Fig. 1



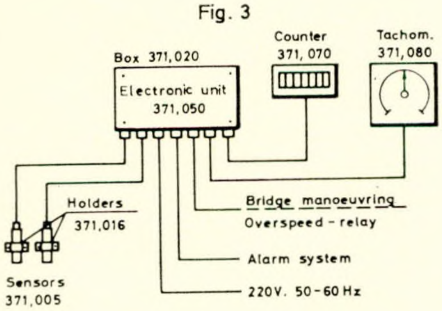
One-way tacho with rev. counter

Fig. 2



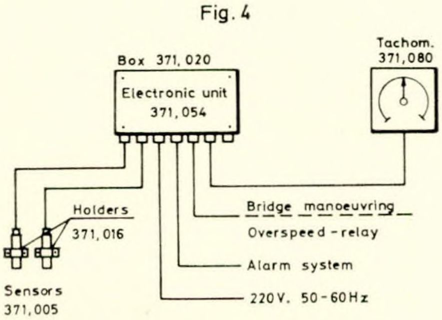
One-way tacho without rev. counter

Fig. 3



Two-way tacho with rev. counter

Fig. 4



Two-way tacho without rev. counter

Monitoring and control of large Diesel engines by electronic tachometer

There are four different versions, i.e. a one-way tacho with/without rev/counter and two-way models also with/without counters (see Figs. 1, 2, 3, and 4).

The two-way tachos have a built-in wrong-way relay while the versions with rev/counters have output for control of slow turnings. The principle of the system is an electronic sensing of the movement of the turning wheel past a sensor. Should indication of direction be desired, two identical sensors are employed. The electronic plug-in unit is built into a water-tight silumin box and can be situated at the engine. The number of teeth in the unit, passing the sensor per second, is converted to a voltage which in turn is proportional to the engine rev/min. The voltage can be used as input to the bridge manoeuvring system or to the overspeed relay. A specially designed filtering circuit ensures a ripple-free signal at low rev/min and a fast response at overspeed level. A separate output of constant current for instrument circuits is standard and at the same time serves as supervision of the voltage output. Up to 40 tacho instruments can be connected to the output.

If the tacho system is to be used in conjunction with a bridge control system or an overspeed relay of Lyngso manufacture, or similar equipment of another make requiring a voltage higher than the 5V output of the tachometer, a booster amplifier is added to the electronic unit.

In the case of reversible engines, the electronic unit is equipped with a direction relay for signal of the wrong way alarm.—*Shipbuilding and Shipping Record, 3rd April 1970, Vol. 115, p. 34.*

Double Reduction Gear Used in Conjunction with Pneumaflex Couplings

The container ship *Black Swan,* which was recently delivered by Scheepswerf Gebr. van der Werf at Deest-Nijmegen, is equipped with Lohmann and Stolterfoht gear for propeller drive. The gear consists of a double reduction marine gear unit of the type NaviLus GVA 1000. The distance between the input shafts of this unit is 2000 mm. The gear unit transmits the power of two Diesel engines, each developing 2000 hp at 500 rev/min, to the propeller which rotates at 250 rev/min. The engines and the gear unit are connected by highly elastic friction clutches.

The gear-box is of the slide bearing type with three gears: the input pinions mesh directly with the output gear wheel. Special surface treatment of the gearing enabled the helical spur gears to be kept as compact as possible. In spite of the high performances required, the dimensions and weight of the gear unit have been kept to a minimum. The propeller thrust is absorbed by a built-in Mitchell type thrust bearing with tiltable thrust pads.

The highly elastic Pneumaflex KAA 260 clutches are used in this case, instead of the former fluid couplings, to protect the installation against torsional vibrations. They shift the critical speed ranges to below the operating speed. Also the reversing time of the turbo-charged Diesel engines can be shortened—i.e. if the Pneumaflex clutches are disengaged at full speed, and the engines then are reversed and started again unloaded. Should quick manoeuvring be required it is possible to use contra-rotating engines while one Pneumaflex clutch is engaged or the other disengaged. By manoeuvring in this way, the additional strain on the Diesel engines caused by blowing coldstarting air into the hot cylinders is avoided.

The pneumatic remote control of the Pneumaflex clutches is interlocked in such a way that both clutches can be engaged only if the main engines have the same direction of rotation and the same speed. Control of the clutches is from a sound-proof cabin.—*Holland Shipbuilding, March 1970, Vol. 19, p. 55.*

Japan's First Ship with Extensive Computer Control

Currently under construction in Japan is a 138 370 dwt tanker, being built for the Sanko Steamship Company by Ishikawajima Harima Heavy Industries Co. Ltd. (IHI). This vessel, *Toko Maru,* is Japan's first extensively computer-controlled ship, and represents the practical implementation of the results of a co-operative research project initiated in the autumn of 1967 by the Japanese Ministry of Transport. The principal tasks to be controlled by the computer on *Toko Maru* will be navigation, cargo handling, loading, engine monitoring, data logging, engine control and crew illness diagnosis. The object of this project is to investigate the technical feasibility of extensive computer control and to obtain data for application to ships of the future.

Completion of this vessel, which has a single screw and is powered by a Sulzer 10 RND 90 Diesel engine having an output at m.c.r. of 28 000 shp at about 121 rev/min, is scheduled for September 1970. For her first year in service she will be operating on an experimental basis with various adjustments being made to the system and their effects assessed. An overall assessment of the practicability of the computer-controlled ship will be made at the end of this trial period.

A TOSBAC 3000 computer developed by Tokyo Shibaura Electric Co. (Toshiba) will form the backbone of the installation, and this company has been closely involved with IHI on ship operation by computer (SOC) research since 1967. This particular computer has a 16 bit/word and 16 k-word core memory, with 80 k-word drum memory. It is claimed that it can be operated by crew members having only a moderate knowledge of computers and electronics:

1) an anti-collision system;
2) position fixing system by satellite;
3) dead reckoning by DRP calculator;
4) navigational calculations;
5) loading/unloading control;
6) ship's condition calculation;
7) calculation of optimum loading condition;
8) engine room fault detection;
9) data logging of engine room machinery;
10) torque control of the main engine;
11) computerized medical diagnosis for crew.

All of the above-mentioned functions can be handled simultaneously by the computer which with its peripheral equipment has been designed to function satisfactorily in a marine environment. In addition, measures have been taken to minimize the risk of the computer being damaged by inadvertent mishandling. A standby power supply is available and a power failure of up to 10 minutes' duration will not affect its operation.—*Shipping World and Shipbuilder, April 1970, Vol. 163, pp. 557–558; 560.*

Petroleum Products Carrier for Canadian West Coast Operation

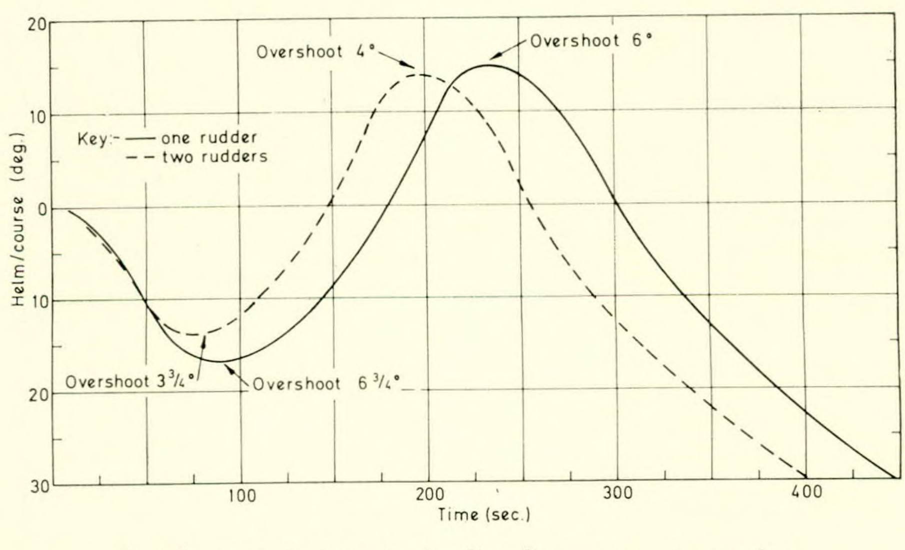
A new vessel for operation off the West Coast of Canada has been delivered to her owners, Imperial Oil Limited, Toronto, by the Burrard Dry Dock Company Limited, of North Vancouver. This vessel, *Imperial Skeena,* can carry petroleum products in tanks.

Imperial Skeena's full form has been the subject of an investigation to find the most suitable shape for incorporating the twin-screw, twin-rudder arrangement, and yet still remain within the dimensional limitations imposed by ports of call. The twin-screw arrangement was adopted on the strong recommendation of the West Coast operating staff, who felt this essential because of this vessel being very large for some of the restricted areas she visits, and because there is a high risk of propeller damage due to the presence of numerous semi-submerged logs.

Principal particulars are:

Length, o.a.	300 ft 0 in
Length, b.p.	290 ft 0 in
Breadth, moulded	53 ft 0 in
Depth, moulded	22 ft 0 in
Draught, loaded	18 ft $1\frac{7}{8}$ in
Displacement, loaded	6420 tons
Deadweight	4856 tons
Light ship	1564 tons
Cargo capacity	1 322 572 gall 212 332 ft^3
Packaged cargo	70 tons
Machinery output	300 bhp at 750 rev/min
Speed on trials	12·54 knots
Output on loaded trial	2630 bhp at 212 prop rev/min
Block coefficient	0·805

An accompanying figure gives the results of a zig-zag manoeuvre performed during the steering trials. This shows that the initial response to the helm is practically the same



Petroleum products carrier for Canadian west coast operation whether one or two rudders are used. At larger helm angles the two rudders showed better response, but these angles are of less significance as regards course keeping.

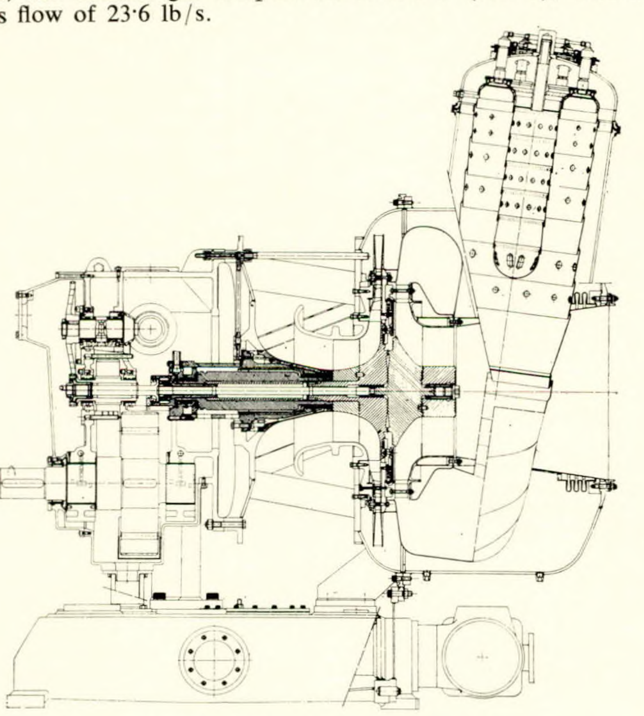
Propulsion machinery comprises two British Polar type SF 112 VS-B four-stroke, 12-cylinder Vee-form turbocharged Diesel engines, each having an output of 1500 bhp at 750 rev/min. Coupling of each engine to a Liaaen gear-box having a reduction ratio of 3·56 : 1 is through a Vulcan Twiflex coupling. Controllable pitch propellers, also of Liaaen make, are fitted. Pneumatic controls for the main engines are of Kobelt design, and are arranged so that the engines can be controlled from consoles in the wheelhouse and the engine room.—*Shipping World and Shipbuilder, June 1970, Vol. 163, pp 767–770.*

Kongsberg All-Radial Gas Turbine

This turbine has been planned for industrial, i.e. static or mobile and marine duties, requiring marked ruggedness, reliability and simplicity in design. With a maximum output of 1700 bhp at 20°C (68°F) inlet air temperature, normal rating 1600 bhp, the weight is 5500 lb and the overall dimensions 75 in × 67 in × 87 in.

Specific fuel consumption is given as 15 750 Btu/hp h. Taking Diesel fuel of 18 540 Btu/lb calorific value the rate quoted would be 0·85 lb/bhp h at rated load.

Fuel may be either liquid, Diesel oil, kerosene or natural gas; the exhaust gas temperature is 997°F (536°C), with a gas flow of 23·6 lb/s.



Kongsberg all-radial gas turbine

An unusual feature of the Kongsberg is that it can be obtained with its output shaft either horizontal or vertical according to the driven unit. The latter is particularly advantageous for pump drives in tankers.

An installation favoured in Norwegian turbine tankers of 220 000 dwt is for Kongsberg emergency generating sets to be housed together, with all their auxiliaries in a generator room on the poop deck, aft of the boiler casing. Turbine inlet air used for oil cooling, starting is by compressed air, and a 6 ton fuel tank is provided, fed from the ship's double-bottom tanks.

An order for Kongsberg emergency and stand-by service sets has been received from the Danish Odense Staalskibsvaerft A/S. It is for six of these Turbosafe sets, one in each of six 286 000 dwt steam turbine tankers for A. P. Moller. On each vessel the 1200 kW output is sufficient to take care of electricity demands in case of complete failure of normal energy sources.

A standard 1500 kVA three-phase generator outfit with exciter and automatic voltage regulator is supplied on a common baseplate. The weight is 7·5 tons.

The turbine runs at 17 100 rev/min but the generator shaft can be geared to 1500 or 1800 rev/min according to the generator cycle concerned. For other driven units the output shaft can be geared as required. Air consumption is 25 lb/s and the exhaust gas temperature 535°C (995°F). The alternator is of 95 per cent efficiency and of 380 or 440 V. Maximum steady-state frequency variation is 4 per cent, of a maximum transient variation 0·25 per cent.

In outline the machine is a simple open-cycle, single-shaft design, with a compressor section, a diffuser, a combustor and a turbine which drives the compressor and provides useful power.

Air enters through a sleeve-form screen between the turbine section and the reduction gear housing, flowing to the compressor through a contoured inlet casing; this compressor has a ratio of 3·6 to 1, embodying an inducer and a radial impeller. Discharge is into a vaned diffuser. From there the compressed air passes into a pressure vessel and thence to the single combustor. Fuel enters through six nozzles. Gas with a maximum T of 750 to 800°C flows via a scroll to the nozzle guide vanes and the radial turbine. An exducer directs the exhaust gases through a diffuser in line with the rotor axis.

The reduction gear is a Kongsberg-MAAG assembly, reducing speed from 17 100 rev/min of the rotor to 1200, or 1500 or 1800 rev/min, according to the application. Directly connected to the reduction gear are the starter drive, fuel pump and lubrication pump. Starting power is either an air motor, electric motor or other energy source.—*Marine Engineer and Naval Architect, May 1970, Vol. 93, pp. 225–227.*

Evaluation of Effectiveness of Propeller Shroud

The effectiveness of the shroud fitted to the propeller of *Pyatidesyatiletie Komsomola,* was evaluated by a series of full-scale investigations carried out during the ship's trials and maiden voyage. Model tests have shown that the combination of shroud and bow bulb should increase the ship speed during trials (deep, still-water and newly-painted hull) by 1·1 knots.

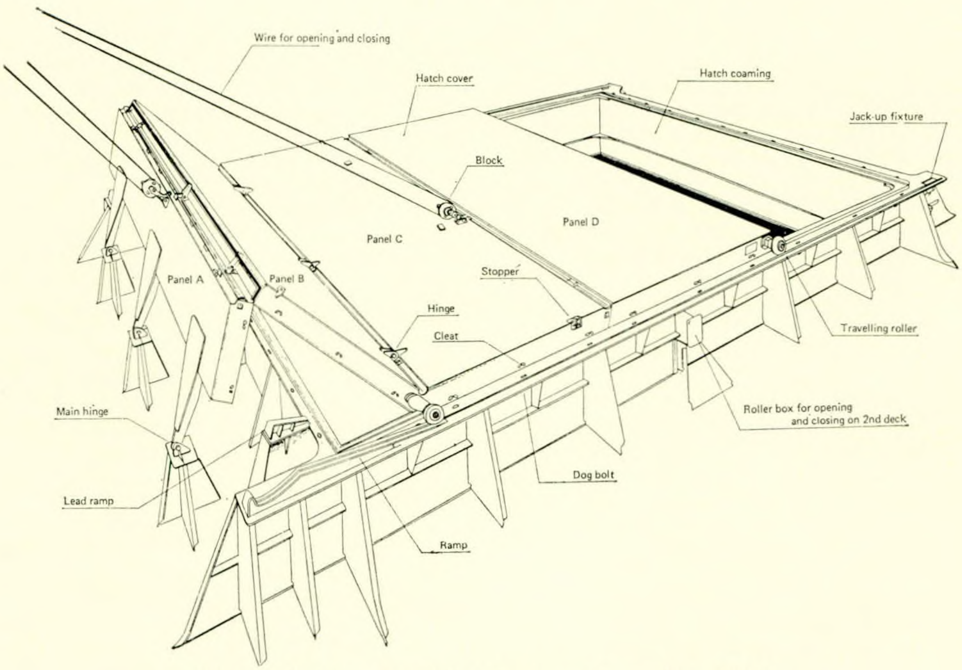
The full-scale investigation showed that the incorporation of the shroud increased the ship speed by 0·5 knot in still water and up to 1 knot in sea state 5–7, reduced the stresses in the propeller, and reduced blade-frequency hull vibrations to a minimum.

The investigation confirmed the ship's good course and seakeeping qualities. The minimum diameter of the ship's steady turning circle was 4·5 ship lengths, and the speed loss against head waves in a sea state of around 5 did not exceed two knots. Measurements of the shaft loading showed that stresses in it did not exceed 200 kgf/cm², and blade-frequency hull vibrations were unnoticeable.

It is noted that the use of non-axisymmetric propeller shrouds on future large tankers, which characteristically have high propeller loading, might be particularly advantageous.—*Turbal, V. K., Shpakov, V. S. and Stump, V. M., Sudostroenie, 1969, No. 10, pp. 9–10; Jnl. BSRA, January 1970, Vol. 25, Abstract No. 28 611.*

Economical Range of Hatch Covers from Mitsubishi

The Kobe Shipyard and Engine Works of Mitsubishi Heavy Industries has developed a number of different types



'Economical' range of hatch covers from Mitsubishi

of hatch covers, of which the most recent examples are claimed to be economical units for general cargo ships. Two multi-purpose Kobe-built cargo vessels, the 15 200 dwt sister-ships *Nautilus* and *Nautic*, are the first vessels to be fitted with covers of the latest designs.

These units are Mitsubishi's double-folding weatherdeck and hing-up 'tweendeck covers.

The new design consists of four hinged panels and appropriate blocks. In operation, the ship's winches or cranes are employed to pull a wire cable to lift and fold the covers which are then rolled open in the direction of the main hinge at the end of the hatch opening.

Closing the hatch is an equally simple operation, i.e. when the wire is loosened the two leading edge panels slide under their own weight into the horizontal position on rollers on the leading panel. This action pulls down the rearmost panels which push the others ahead of them into the closed position. The rollers on the leading panel are then secured in place and dog bolts around the hatch are tightened to complete the closing operation. Metal gaskets between panels are automatically compressed when the panels are laid out over the hatch and keep the panels aligned with each other, thus eliminating any need to fasten the panels to one another.—*Shipbuilding and Shipping Record, 15th May 1970, Vol. 115, pp 29; 31.*

Gas Turbine Engines to Power 100-ton 80-knot Surface Effect Ship

Pratt and Whitney Aircraft's Turbo-Power and Marine Department will supply three aviation-type gas turbine engines for a proposed novel high-speed captured air bubble vessel which would skim over the water at 80 knots on a bubble of air.

This 100-ton Surface Effect Ship (SES) test craft programme is being implemented by Textron's Bell Aerosystems Co., of Buffalo, N.Y., under an incrementally-funded contract announced by the U.S. Joint Surface Effect Ships Programme Office which is composed of elements of the United States Navy and the Commerce Department's Maritime Administration.

The three P and WA FT12 gas turbines each develop 4500 shp.

Surface Effect Ships literally ride on a bubble of air that is created when air is forced down beneath the craft's hull where it is captured and contained by catamaran-like side hulls and bow and stern seals of flexible rubberized material.

The three aft-mounted P and WA FT12 gas turbines are designed to be connected to a reduction gear system driving two super-cavitating propellers. The vessel would be able to drive its twin propellers with either one, two, or all three gas turbines. At high speed, the craft's hull would be no more than a few inches below the water surface.

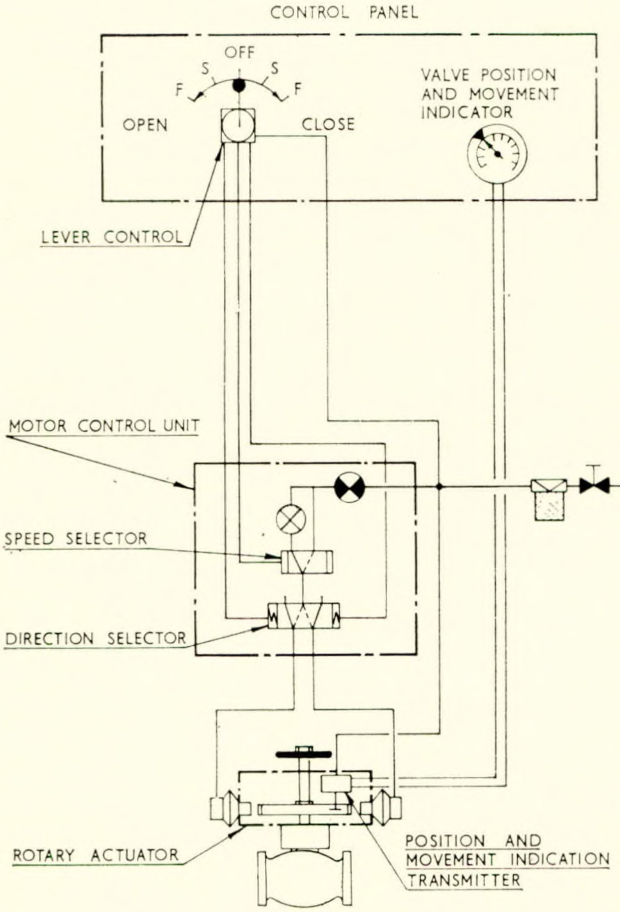
Larger P and WT gas turbines are in commission in a number of high-endurance United States Coast Guard cutters, a large roll-on/roll-off merchant ship, a 70-knot Canadian hydrofoil, and are scheduled for installation in several containerships and Canadian destroyers. Other P and WA marine engines are operating with Danish Navy frigates.—*Maritime Reporter/Engineering News, 15th January 1970, Vol. 32, p. 48.*

Standard Control Systems for Remote and Automatic Control

Basic control systems in conjunction with pneumatic actuators can be adopted to meet the requirements of marine remote and automatic control systems. Comprehensive centralized control systems installed in ships normally comprise a multiplicity of simple basic systems operating individually or interconnected to form a sequentially interlocked or fully automatic system controlling a particular section of the ships plant and services.

The success of the overall control scheme is dependent on the correct choice of basic systems and equipment that meet the important marine requirements of simplicity, reliability, standardization, ease of maintenance and availability of emergency manual control in the event of component or air supply failure.

The systems described here have been specifically designed for use with Telektron rotary actuators, which, inherently, embody manual operation and position locking characteristics when used as valve actuators. Thus considerable simplification



Standard control systems for remote and automatic control

of the basic control circuits to meet the overall requirements of marine control is possible.

Centralized control systems often provide the facility for an operator to position a valve remotely at any precise point in its travel. One of the most exacting applications of this type is the control of main propulsion turbine manoeuvring valves. To obtain the degree of control required, the system must be capable of high resolution. For rapid emergency manoeuvring good response and speed is essential.

The diagram illustrates the type of system that is required for this arduous duty.

The control system provides the operator with two speed control of the manoeuvring valve by means of a 5-position lever control valve. Movement from the central 'off' position to either of the two inner positions of the gate adjusts the position of the valve in slow speed in the appropriate direction, whereas selection of the outer positions of the gate adjusts the valve position in fast speed for rapid manoeuvring.

System response is of prime importance on this type of application and additional valves are added to the system adjacent to the manoeuvring valve to control the power air to the motors with the shortest possible pipe runs, and to provide the speed/flow selection function. The pilot control lines between the panel-mounted lever control valve and the pilot operated valves are operated at reduced pressure to further increase response.

The resolution of this type of system is extremely high; for example a 10-turn valve fitted with an actuator toothwheel with 76 teeth can be moved one tooth producing an increment of movement of 1/760 of the total valve travel or 0·13 per cent.—*Dowdall, D. S., Shipping, April 1970, Vol. 59, pp 20; 22; 24.*

Container Feeder Design for Hungarian Export Order Book

Tonnage measurement conferences notwithstanding, the 499-ton parameter is clearly in mind with the specification for a small container feeder ship recently designed by the Hungarian Shipyards and Crane Factory of Budapest.

A container capacity of 93 units has been achieved, the 13 m beam enabling a stowage of five containers wide on the weather deck, under-deck alleyways being provided for foc'sle access. Twenty-nine laden containers can be stowed on the weather deck and a top layer of 24 light boxes may be stowed above these. The lower hold contains room for 16 containers and 24 are to be stowed in the 'tweendeck. Special supports at the weather deck bulwarks are arranged to transfer stress from the containers on deck and hatchcovers, fully load bearing, are to be fitted with ISO approved lashing arrangements.

Main deck hatch openings will be of 39 m × 10·6 m with a continuous longitudinal coaming construction and a smaller hatch forward of 6·6 m × 5·4 m.

Principal particulars are:

Length, o.a.	75 m
Length, b.p.	68 m
Breadth moulded	13 m
Depth to maindeck	6·05 m
Depth to 'tweendeck	3·80 m
Draught at summer freeboard ...	3·76 m
Deadweight	1200 tons
Crew	about 12 men
Engine output	1800 hp
Service speed (90 per cent output)	13·00 knots

The hull is of all-welded construction and sub-divided with five watertight bulkheads, built with transverse framing and alternate plate floors in the double bottom.

The propulsion plant contained in the specification is a 4-stroke 6-cylinder MWM Diesel type TBRHS 345 SU, giving 1800 hp at 500 rev/min and driving a four-bladed propeller through a Vulkan coupling and reduction gear. The engine is provided with exhaust turbocharger, intercooler and two 1000 litre air starting vessels, and bunker capacities will give a range of 3000 miles at service speed. The machinery may be simply controlled from the bridge.—*Shipbuilding and Shipping Record, 9th January 1970, Vol. 115, pp. 18–19.*

Vibration and Strain Analysis by Means of Holography

Fluctuating stress and vibration of components in an engine are usually measured by strain gauges or electronic probes. It is also important to measure individual vibration frequencies and nodal patterns of components, in particular, blades and discs. Current methods employing sand patterns, piezo-electric probing, stroboscopic and electron-probe scanning are briefly reviewed.

A new technique known as holography has been added to these existing methods. It is a method by which information concerning the shape of an object illuminated by coherent light is recorded as an interference pattern on a photographic plate, and can then be reconstructed as a true three-dimensional image. This technique, invented in 1948, has become of value in stress and vibration analysis problems, now that the laser is available as a source of powerful coherent light.

An account is given of the application of holography in the laboratory, and holograms are given for compressor blades and turbine discs up to 10 in diameter. Details are then given of a complete holographic system for routine analysis of objects up to 24 in. in diameter. The unit can be used equally well for either vibration mode-pattern detection or static deflexions for strain analysis.

The average time taken for completing a hologram test, including photographing the reconstruction, is approximately 15 minutes.

Future developments may reduce the exposure times and simplify operation of the system.—*Hockley, B. S. and Hill, R. J., Jnl. BSRA, August 1969, Vol. 41, pp. 6–12; November 1969, Vol. 24, Abstract No. 28 414.*

Radar Antenna Location

Very little has been published to guide the naval architect in the proper placement of a cargo ship's radar antenna. Obstructions in the path of the radar beam cause blind spots and false targets. Therefore, in order to get a clear picture, the radar antenna must be above the many kingposts, cross platforms and cargo booms. In most cases, the pilothouse is roughly in the middle of the vessel, and usually the radar antenna is on a mast above the pilothouse. In many of the new designs, it is even more difficult because the pilothouse has been moved further aft.

Radar energy travels in space very much like a searchlight beam. The radar beam may be obstructed or reflected just like the light beam. If a mast or boom is in the path of the light, the object directly on the opposite side will not be properly illuminated. Also, the radar beam may be reflected similar to light. If one is on the deck below a searchlight and there are no objects being illuminated nearby, the deck is dark; but if the light beam hits booms and masts, and cross trees, the reflected light illuminates the deck. The same principles apply to radar. The booms and masts cause shadows and the deflexion from such objects cause false indications on the picture when large objects are within a few miles of the ship.

Fig. 1 shows a sketch of a ship equipped with radar, showing how the shadowed area caused by heavy-lift kingpost may obscure a ship as (C). Fig. 2 shows how a reflection from a large heavy lift gear may produce an echo from the ship at (A) and make it appear at (B) dead ahead.

A radar beam is narrow in the horizontal plane. For 10 cm radar using a 12 ft radiator, this is usually around 2 degrees. The higher frequency, 3 cm radar, will have roughly one-third the beam width for the same size radiator. Thus, for a 6 ft radiator, the beam width is about 1·3 degrees and for a 12 ft radiator, the beam width is around 0·6 degrees, Fig. 3.

The vertical beam width is wide to produce a return signal even though the ship rolls and pitches within certain limits. This beam width is usually around 20 degrees. With the ship trimmed, the centre of the beam should point toward the horizon, Fig. 4.

Beam widths are specified at the 3 db point. This is where the radiated power is 50 per cent of the maximum. Thus, a horizontal beam specified as 2 degrees is actually wider at lower points. For example, at 1/100 of the maximum power, the width is approximately double, or in this case 4 degrees. Likewise, the vertical beam is much wider than the specified 20 degrees. Since antennas are in many cases well over 100 ft above water, it is this wider part of the beam that allows the radar to see down and pick up close targets.

Usually there are many kingposts, cross platforms and booms; and the exact boom causing the false target is unknown. Also it is difficult to determine the actual object producing the echo. An example of such a condition might be found in entering the bay at San Francisco or in going up the Hudson River opposite Manhattan.

If there is a large steel forest in the path of the radar beam producing many deflexions, the false targets add up to smears and may inhibit visibility.—*Moore, C. E., Maritime Reporter/Engineering News, 15th March, 1970, Vol. 32, pp 18; 20.*

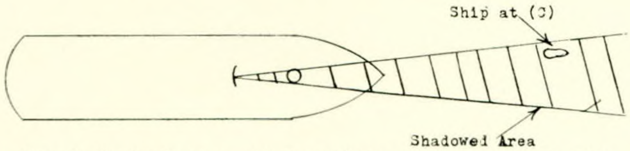


Figure 1—Shadowed area caused by heavy-lift kingpost may obscure a ship at (C).

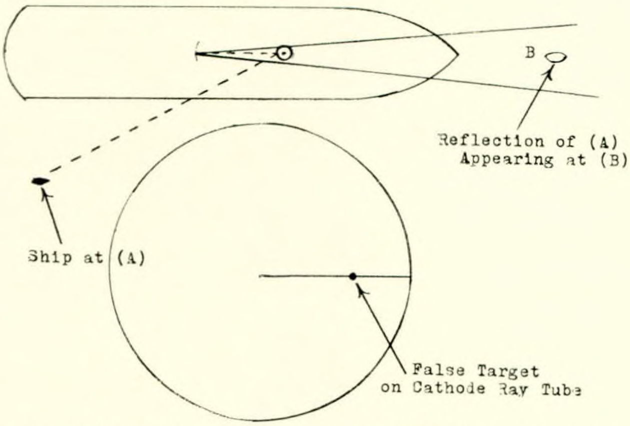


Figure 2—Reflection from heavy-lift gear may produce an echo from ship at (A).

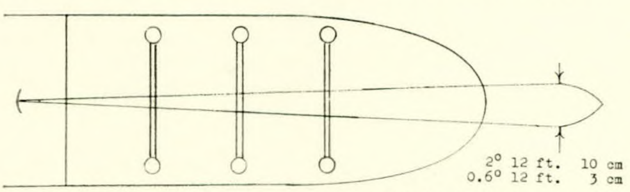


Figure 3—A radar beam is narrow in the horizontal plane.

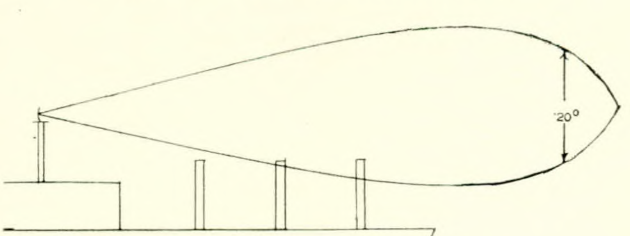


Figure 4—The vertical radar beam is wide to allow for ship's motions.

Radar antenna location

Stabilizer Types

There are many devices used to combat the rolling motions of vessels but no one device can solve all the problems. All the methods currently in use have some benefit to offer, and all have limitations as to the degree of benefit achievable, and the conditions under which the benefit is possible. In some cases benefit exists under certain conditions, but the same device in other sea or ship conditions may in fact make the ship rolling motions greater.

The pure passive tank stabilizer is typically a rectangular tank across the width of the ship, one or two decks in depth and about one fiftieth of the ship's length. The tank is about one fifth full of sea water, and this water is seen to behave approximately as yet another spring, mass and damping system. It is possible to design the tank size and shape, and to put in a definite amount of water, such that the tank exhibits the same natural frequency as the ship in which it is placed. When the sea causes the ship to roll at its natural frequency, the water movement in the tank is maximum, and its timing relationship is in time quadrature with respect to the ship, which in turn is in time quadrature with the disturbing waves. The rolling moment exerted on the ship by the tank water is thus in direct opposition to the rolling moment due to the waves, and the ship rolling is thereby reduced.

Since the time quadrature relationship between disturbance and movement is confined to frequencies at or very near to the natural frequency, the tank water only provides a restoring moment over a narrow band of frequencies. At higher frequencies of disturbance both ship and tank water motions diminish rapidly, but at lower frequencies ship motion is increased by the tank water, which then exerts a moment aiding the sea disturbances. At very low frequencies and in listing conditions, the tank water increases the ship roll.

A controlled passive tank improves upon the performance of the plain passive system at the expense of additional machinery by fitting controlled baffles in the water or air ducts between wing tanks, to restrict the flow of tank water when this is not opposing the disturbing waveslopes. The figure again shows a typical idealized response in which the increase in ship motion due to the tank is removed except for the very long periods of encounter. Ship motion is held to about three times the applied waveslope over much of the range, but there is some increase in ship motion to the shorter encounter periods.

Controlled passive and pure passive tanks are activated by the ship motion, and improvement upon this performance is possible only by some other form of activation.

When the vessel is under way, a hydrofoil surface, or fin, protruding from the hull can be used to exert a rolling torque upon the vessel. If the angle of such a fin in the water streamline is controlled, it may be used to combat the rolling motion of the ship. Stabilizer power is limited only by the size of fin which can be accommodated, and the cost, weight, and space occupied by the positioning machinery.—*Bennet, D. B., Shipbuilding and Shipping Record, 13th March 1970, Vol. 115, pp. 33–34.*

Japanese-built Bucket Dredger for Russia

For service in the Far East, the Black Sea and the Baltic under Russian ownership, is the self-propelled bucket dredger designated *3ER*. This vessel is part of a three-ship order placed with Nippon Kokan Kabushiki Kaisha (NKK) by the Soviet National Ship Export-Import Corporation (Sudoimport).

3ER has been built for classification by the Register of Shipping of the U.S.S.R. and the rules have been applied in accordance with an agreement between Nippon Kaiji Kyokai, the Japanese classification society, and the Russian society. This is one of the first instances in Japan of ships being built under Russian rules.

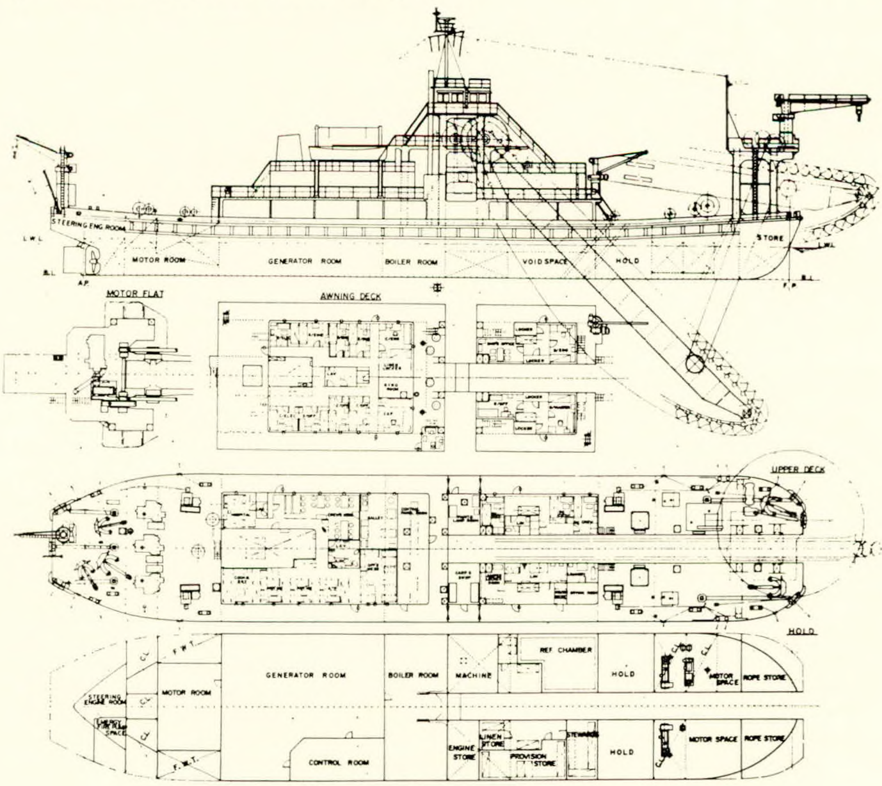
Steam piping fitted to the belt conveyors can melt any ice which forms, and measures have been taken to prevent moisture accumulation on the clutches and brakes of deck machinery. By these means it is possible to continue dredging operations at temperatures down to −15°C. A maximum dredging depth of 18 m, with 12 m as normal, at a capacity of 750 m^3/h has been designed for, and operation is possible in sea states up to Beaufort Force 5.

The engine room is divided into three compartments for the main generators, boiler room and propulsion motor room respectively, and these are divided by watertight doors to comply with the single compartment flooding requirement. Propelling machinery comprises a MAN G8V 30/45 ATL Diesel engine, built under licence by Mitsubishi, which has an output of 1700 shp at 500 rev/min. This gave the vessel a speed of 8·5 knots on trials.

Principal particulars are:

Length, o.a.	71·50 m
Length, b.p.	67·50 m
Breadth, moulded	14·00 m
Depth, moulded	5·10 m
Draught, loaded	3·10 m
Gross tonnage	1500
Machinery output	1700 shp at 500 rev/min

—*Shipping World and Shipbuilder, June 1970, Vol. 103, p. 784.*



Japanese-built bucket dredger for Russia

Phenomenological Investigations of Stress-Relief Embrittlement

Phenomenological characteristics of stress-relief embrittlement were investigated for a 5Ni-Cr-Mo-V steel. Studies included the cumulative nature of stress-relief cycles, the effects of cooling rate and strain on stress-relief embrittlement, and the reversibility of stress-relief embrittlement. Charpy V-notch properties were utilized as the toughness parameter.

Repeated stress-relief treatments are shown to be cumulative, resulting properties of air-cooled specimens being dependent on time at temperature. Stress-relief treatments at 950°F result in a degradation of toughness that is dependent upon time at temperature and cooling rate, longer times at temperature and slower cooling rates producing greater embrittlement. Embrittlement obtained during cooling constitutes the major portion of degradation after short times at temperature, but only the minor portion of the degradation after long times at temperature.

Stress-relief at 1050°F causes both softening at temperature and embrittlement on cooling, and the resulting properties are dependent upon time at temperature and cooling rate. The presence of strain results in a degradation of toughness during stress-relief that is greater than the embrittlement produced by time and temperature on unstrained material. The loss of toughness due to stress-relief embrittlement may be recovered by retempering to a lower strength level.—*Rosenstein, A. H., Welding Journal, March 1970, Vol. 49, pp. 122-s–131-s.*

Optimal Rounding of the Corners of Rectangular Cut-Outs in Ships' Hulls

The results of theoretical and experimental investigations into stress concentrations in the region of rectangular cut-outs with circular, elliptical, and parabolic rounding of the corners are presented. Plates of both finite and infinite breadth are examined under uniform tension and pure shear.

N. I. Muskhelishvili's method is used to solve the plastic-theory equations for determining the stress concentration factors for the infinite plate.

In order to overcome the deficiencies inherent in the plastic material usually used in the tests, an epoxy resin material was used to simulate the plate. A special test rig was also constructed to ensure the uniform loading of the test piece. The test plates measured 1400 × 600 mm.

It was found that under tension the stress-concentration coefficient in a plate with parabolic rounding of the corners is 10 per cent greater than in a plate with circular corners, and that under pure shear the stress-concentration factor is 30–40 per cent greater in the plate with parabolic corners than in the plate with circular corners. The stress concentration factor in the plate with elliptical rounding of the corners of the cut-out was found to have an intermediate value.

From these results it is concluded that the optimal form for the corners of rectangular cut-outs, when the plate is subject to either tension or shear, is circular.—*Anikin, E. P. and Rodygin, V. V., Sudostroenie, 1969, No. 7, pp. 7–9; Jnl. BSRA, November 1969, Vol. 24, Abstract No. 28 311.*

First of Ten Container Ships for Sea Containers Ltd.

M.V. *Minho* is the first of a series of ten container ships being built for Sea Containers Ltd., London. Built by A. Vuyk and Zonen's Scheepswerven N.V., Capelle a.d. IJssel, the ship was officially handed over to her owners on the last day of 1969.

Minho has been built to Lloyd's Register of Shipping Class 100 A1, Ice Class 2, LMC, UMS and complies with the regulations of the British BoT, Class VII.

The vessel is a single-screw ship of the open shelterdeck type and has two continuous decks, while the accommodation for a complement of 14 and engine room are arranged aft. Containers can be carried in the hold, which has no transverse bulkheads, as well as on the pontoon hatchcovers. At the same time *Minho* can also carry other than container cargo in the hold when the guide rails are removed. The latter can be placed so that 20-, 30- and 40-ft containers can be loaded.

Principal particulars are:

Length, o.a.	85·80 m
Breadth, b.p.	78·84 m
Breadth, moulded	13·70 m
Depth to maindeck	4·22 m
Depth to shelterdeck	6·05 m
Gross tonnage	930 r.t.
Net tonnage	492 r.t.
Speed	15·5 knots

The propelling machinery consists of Werkspoor Diesel engine type 6TM410. This 6-cylinder engine, which has a bore of 410 mm and a piston stroke of 470 mm, develops 3200 bhp at 515 rev/min and drives, through a Tacke gear-box type HSV 630, a four-bladed Lips c.p. propeller type 4 LH 890 with a diameter of 2815 mm. The Tacke gear-box reduces the engine speed from 515 to 275 rev/min at the propeller, which is provided with a type V320 pitch control mechanism. Electric power is supplied by two generator sets. —*Holland Shipbuilding, February 1970, Vol. 18, pp. 36, 37, 40.*

Vessel Evaluation

Many vessels have been compared solely on speed attained, or on the basis of e.h.p. *versus* speed, displacement *versus* resistance, lift/drag ratio, or similar values. While these factors are all useful in the developmental stages of design or redesign, they may not provide a comprehensive resultant value of system effectiveness for the final vessel.

Therefore, an approach using power as a function of weight-velocity has been developed, along the lines of the cost per ton-mile economic study method. This method offers several advantages for the evaluation of dissimilar vessels and vehicles. First, it is placed in the basic terms of speed times tons divided by power which is used to arrive at a basic number of effectiveness, ton-knots per h.p.

When using these terms, they result in a numeral which expresses the actual effectiveness power attained in transport.

The area of investigation is in the high-speed small vessel field, where military vessels and passenger vessels predominate, and many planing or near-planing vessels have found employment. However, it is equally applicable to other vessels. The speed-length ratios of these vessels exceed 6·0 in a number of cases, and their principal design goal is speed.

Non-conventional vessels or vehicles come in several forms and are chiefly distinguished by their high cost of performance in most cases. Despite the impressive speed and capability of hydrofoils, their high cost and complexity make them non-feasible for most civilian operations.

Surface-effect vehicles, again are high power, low payload vehicles, despite their high speed capability in most cases.

This discussion has been based upon investigations into the relative efficiencies of various high-performance vessels in the range from ten to 200 tons displacement, and having speeds ranging from 15 to 60 knots. However, the methods employed are equally suitable for examination and comparison of vessels outside these general limits. Initially, the methods presented were employed in order to provide a common denominator in comparison of widely diverse vessel types and sizes, but proved useful in also establishing family characteristics of classes of vessels.—*Bond, C. W., Maritime Reporter/Engineering News, 1st February 1970, Vol. 32, p. 23.*

Some Stabilization Problems and Solutions

The long-standing and now almost traditional use of fin stabilizers for passenger liners and roll-on/roll-off ferries has now been extended to ships of new design and concept, notably container ships, which are designed primarily to increase the earning power per ship.

The operation of these vessels poses certain special problems:

i) standard cargo units;
ii) mechanical handling of units;
iii) computerized sorting of units for sequential loading;
iv) cellular stowage of units;
v) drive on/off facilities for vehicular cargo, reducing non-productive but expensive time and labour in harbour.

There may still be some disagreement between designers and owners on the stability aspects of container ships, but several new facts have emerged from sea experience and loading practice, particularly in relation to the cellular stowage of containers:

a) most operators agree that a list angle of three degrees or more during loading can impair the achievement of optimum turnround;
b) most operators also agree that roll angles of plus or minus six degrees during voyage can jam or damage containers in their cells to the point where man-handling is eventually required during discharge operations.

Both problems can be minimized by the introduction of an effective automatic list control system brought into use during loading/discharging and by fitting the best and most cost-effective roll-stabilizing system to minimize roll throughout the speed range, from zero speed through to service speed.

An advanced fin-stabilization technique, known as lift control, is now available which goes a long way to achieve minimum particle disturbance. This system ensures that each fin continuously carries the load required, up to the lift limits of the system, at any ship speed; that fins are never overloaded, and never stall or enter the critical cavitation zone.—*Palmer, R., Shipping, February 1970, Vol. 59, pp. 36–37.*

Ship Propulsion Systems

In order to be thoroughly successful, automation and rationalization must be linked together when designing the propulsion package. Rationalization and simplification are of the utmost importance but can be considered only within the contest of the whole set of machinery. Unless these aims be achieved, the reliability of an unmanned engine room cannot be assured. All the major Classification Societies make recommendations or enforce regulations governing the points to be monitored in an engine room for unmanned operation and each installation must be approved on board before insurance is granted.

Realization of all essential requirements for a successful system of the type referred to here is conveniently illustrated by the practice adopted with Vickers propulsion package concept. In this, Vickers, in conjunction with Negretti and Zambra, have produced a modular system for sale with the engine. All sensing devices, i.e. transducers for temperature, oil pressure and the like are installed on the engine at the construction stage and wired to a common junction box. Indicators and alarms are fitted into specially designed cases or modules which can be mounted together in stacks without additional support. Thus apart from installing transducers on external equipment, the shipbuilder has only to run cable from the engines to the modules, so notable savings can be made in engineering costs. Vickers are also working with Westinghouse on a remote pneumatic control system.

The modular system of instrument panels, arrived at by Negretti and Zambra in collaboration with Vickers, is included in a propulsion package comprising a prime mover which may be selected from a range of Vickers MAN medium speed engines, developing up to 18 000 hp from a single engine coupled to an epicyclic gear-box of Vickers design and manufacture together with a simplified fuel system, other auxiliary systems in modular form and remote control and instrumentation systems.

The standard series of co-axial epicyclic gear units offers striking reduction in size and weight as contrasted with the normal gear and pinion unit; no less striking is the economy which may be effected in installed cost. Also associated with the package is a fuel system in which heavy fuel oil is processed to ensure complete combustion, thus giving, in effect, reduced fuel consumption and cutting fouling rates.—*Braddyll, J. R. G., Shipping, February 1970, Vol. 59, pp. 20, 29, 30.*

Jungner System for Automatic Control of Diesel Alternators

The AHJM automatic system Jungner has developed for controlling auxiliary machinery is designed for the automatic operation and monitoring of three Diesel-powered auxiliary generators plus a shaft or turbo generator. The system can be readily adapted to another number of auxiliary generators.

The present installation includes, for instance, three USSA start/stop units and three UEG-A electric regulator units, i.e. frequency maintenance, load distribution. In other words, one unit per auxiliary generator.

The present installation carries out automatic starting, synchronization and phasing of Diesel generators. It also controls frequency maintenance and load distribution as well as monitoring, disengagement and stopping. An automatic pre-lubrication control mechanism is also included. Engagement and disengagement are effected in accordance with the load so that the generators are not subjected to overloading or operation with unnecessarily low output. If one auxiliary engine fails, it is disengaged and replaced by another. Large-scale loads have been programmed and are not switched in until sufficient generator power is available.

The main functions may be listed as follows:

1) pre-lubrication;
2) start/stop;
3) synchronization and phasing;
4) frequency maintenance and load distribution;
5) load control;
6) monitoring;
7) programme function including mains failure programme.

The pre-lubrication pumps of the auxiliary Diesels are switched on for about five minutes every four hours. The engagement of the different pump motors is phased in such a manner that only one motor is engaged at a time. An alarm circuit senses the oil pressure and blocks starting of the Diesel if the pressure is too low. When the Diesel is started, pre-lubrication is blocked automatically.

When more power is required by the ship's mains or if one of the generators in operation is disengaged because of faulty performance, a starting order is given automatically to the Diesel generator next in line. The starting order for the Diesels is selected on the control panel.

In case of a power blackout, the programme units UPEA and UPEB of the automatic control system register the positions of the circuit breakers and send out orders in accordance with the power blackout programme.—*Shipbuilding and Shipping Record, 23rd January 1970, Vol. 115, pp. 26–27.*

Largest Vessel Yet Built by Scott-Lithgow

An oil tanker of 134 400 dwt has been built at the Scott Lithgow Kingston Yard for the Nestor Shipping Co. SA, Panama, an Anglo Norness Group company. This vessel, *Naess Enterprise,* is the largest ship yet built by Scott Lithgow, and also the largest oil tanker so far built on the Clyde.

Principal particulars are:

Length, o.a.	900 ft 3 in
Length, b.p.	860 ft 0 in
Breadth, moulded	137 ft $10\frac{5}{8}$ in
Depth, moulded	73 ft 6 in
Draught, summer	55 ft $7\frac{1}{2}$ in
Deadweight	134 000 tons
Lightweight	22 370 tons
Displacement	156 770 tons
Gross tonnage	67 442·52
Cargo capacity	6 072 750 ft^3
Designed speed	16·20 knots

The engine is a turbocharged two-stroke, single acting, crosshead type Diesel of Burmeister and Wain design. The 12-cylinder unit has a bore and stroke respectively, of 840 mm and 1800 mm and develops 25 200 bhp (M) at 110 rev/min with a maximum continuous rating of 27 600 bhp (M) at 114 rev/min. The unit has also been designed and arranged for an optimum condition on the service rating of 25 200 bhp (M) and the propeller to absorb initially 25 200 bhp (M) at 113·5 rev/min to allow for roughening of the hull, etc.

The four-bladed solid nikalium alloy propeller, supplied by Stone Manganese Ltd., has a diameter of 7300 mm, a pitch of 4845 mm, a 25·10 m^2 blade surface area and weighs 34·65 tons.

The B and W automatic manoeuvring system effects automatically starting, stopping and reversing by electric impulses transmitted from a set of contacts built into the bridge telegraph; local engine control is also possible.

A mechanical contactor system in the speed setting for the hydraulic governor prevents overloading of the engine during accelerating periods and ensures a sufficient supply of oil from the fuel pumps during crash-stop manoeuvres. The system also ensures against free-wheeling of the propeller which might occur during crash manoeuvres.

Two marine watertube boilers are fitted complete with superheater. The boilers are arranged for burning heavy fuel oil in an open furnace under forced draught and are arranged to work in conjunction with the economizer.

The design particulars for the boilers are:

Maximum evaporation	85 000 lb/hr
Design measure	300 lb/in^2
Steam outlet pressure	250 lb/in^2
Steam outlet temperature	510°F
Feed water temperature	120°F
Boiler heating surface	4537 ft^2
Superheater heating surface	891 ft^2

Each boiler is a simple two drum unit, the generating bank consisting of five rows of 2-in o.d. tubes followed by 12 rows of $1\frac{1}{4}$-in o.d. tubes. A superheater is arranged at the boiler outlets. It consists of 36 elements, six tubes high, manufactured of $1\frac{1}{2}$-in o.d. tubing.—*Shipping World and Shipbuilder, April 1970, Vol. 163, pp. 517–620.*

Strength of Large Tankers

The paper considers the evolution in tanker size and the development of methods of assessing primary structural scantlings. The various aspects of structural strength are considered and the increasing importance of transverse strength criteria is discussed. The method currently employed by Lloyd's Register of Shipping for direct calculation by computer of structural behaviour is illustrated in detail. Consideration is also given to detail design of girder webs in respect of plate buckling, stiffener requirements and the effect of perforations. Future research activities and developments of analytical systems are also discussed.—*Roberts, W. J., 19th January 1970. Paper presented at a meeting of the North-East Coast Institution of Engineers and Shipbuilders.*

A New Approach to Gas Venting of Tankers

The aim of this paper is to analyse the benefits which can be obtained by ejecting the gases, displaced when petroleum is loaded into cargo tanks, at a high velocity so that they reach a height well above the ships' working areas. In addition to the advantages of pushing the gases away from the outlets, increased efflux velocity will increase the rate of air entrainment and the concentration of the gases will reduce rapidly. Thus the dependence on atmospheric turbulence is proportionately reduced. It has now been established that a minimum efflux velocity of 100 ft/sec. will provide adequate dispersal and it is proposed that this velocity is achieved by installing a variable nozzle in each outlet in which tank pressure is used to provide a vent area proportional to the loading rate.—*Martin, W. S. (1969). Paper submitted to R.I.N.A. for written discussion; paper No. W10.*

Application of Exhaust Gas Ejector to Engine Cooling

Although exhaust gas ejectors have been little utilized in engines, their application is worthy of consideration because of their basic simplicity. An application for engine cooling has been investigated using a four-stroke cycle Diesel engine and an air model. The following results are obtained:

1) the cooling air for an engine may be supplied only from the ejector which is well designed. Fuel consumption can be decreased by using the ejector together with a fan for engine cooling;
2) flow rate of air induced into the ejector becomes larger with an increase in engine load. However, flow rate per unit horse power becomes smaller;
3) flow rate of secondary fluid for the ejector, the driving fluid of which is intermittently supplied, is much larger than that for the ejector of steady state, because the inertia effect of fluid in the ejector is a useful factor to draw the secondary fluid.

—*Nago, F., Shimamoto, Y., Shikata M. and Toyofuku, H., Bulletin of the Japan Society of Mechanical Engineers, Vol. 12, No. 53, pp. 1153–1162.*

Numerical Control of Plate Forming and Associated Problems

In the past few years, the definition of the hull shape in a mathematical form has been successfully achieved. As a result, it has become possible to define any part of the ship's hull mathematically and, therefore, very accurately. In order to make use of this achievement, numerically controlled machine tools, for the production of the different parts of the ship-hull, have been the subject of much research work. The first achievement was the numerically controlled flame cutters which are developed to a satisfactory degree of operation and are now in use. The second application of the numerical control technique was to the frame bending machine.

The impact of using these two machines on control of shipyard production, planning and economy will be much improved when plate forming operations are also controlled numerically. This will eliminate completely the mould loft and the use of templates in addition to dispensing with the skilled labour required for these operations. However, the largest savings are likely to accrue from increased accuracy of individual parts which will reduce assembly times and rectification costs.—*Shama, M. A., Shipbuilding and Shipping Record, 16th January 1970, Vol. 115, pp. 21, 23.*

The Stopping of Large Tankers and the Feasibility of Using Auxiliary Braking Devices

The very rapid increase in the size of tankers, accompanied by a significant decrease in the ratio of installed power per ton of displacement, has emphasized the need to examine the stopping ability of this type of ship. Part I of the paper puts forward a method of calculating stopping distances which appear to be satisfactory when compared with the limited data at present available. This is on the basis of braking using the available astern thrust of the propeller. Part II examines the use of auxiliary devices to reduce the stopping distance of a ship with particular reference to a 165 000 dwt tanker in the loaded condition.—*Paper by Clarke, D. and Wellman, F., presented at a meeting of The Royal Institution of Naval Architects, 23rd April 1970, Paper No. 4.*

Design and Construction of the Dynamically Positioned Glomar Challenger

This paper is a description of the dynamically positioned drilling ship, *Glomar Challenger,* referring to novel and unusual practices which should be of interest to marine and petroleum industries. Even today a seagoing ship with a 142-ft derrick is an oddity, although such derricks have been used in this capacity for over ten years. Probably the most noteworthy technological advance of this ship is found in the dynamic positioning system. To be able to hold a fixed position steadily—and within 100 ft of that position—and to be able to maintain position and orient the ship throughout the compass azimuths makes exciting work and oceanic exploration quite practical.—*Graham, J. R., Jones, K. M., Knorr. G. D. and Dixon, T. F., Marine Technology, April 1970, Vol. 7, pp. 159–179.*

Higher Mode Vertical Vibration of Giant Tanker

For the purpose of clarifying the nature of the hull vertical vibration in the higher mode, the preliminary analysis has been made in which the ship hull is simulated by the two parallel beams model elastically connected. Each beam represents the side shell and the longitudinal bulkhead and the connecting springs the transverse structural members. The result obtained shows that the relative deformation of longitudinal bulkhead to the side shell is remarkable, in the modes higher than the 5- or 6-noded modes which means the elastic deformation of the cross section of ship is significant, contrary to the assumption in the beam-theory calculation. As the mode becomes higher than these modes, there appear new modes in which the deflexion of the longitudinal bulkhead is out of phase by 180 degrees to that of the side shell.—*Ohtaka, K., Kagawa, K. and Yamamoto, T., Jnl of the Society of Naval Architects of Japan, June 1969, Vol. 125, pp. 157–188.*

Bracket Effect—an Analysis by the Elastic Theory of Plate Bracketed Beams

A plane stress problem of bracketed beams is investigated by the method of elastic analysis employing the airy stress functions. The problem is finally reduced to the solution of a set of 100 simultaneous linear equations which are numerically solved on a computer.

The stress distribution inside the bracketed region is found to be non-linear unlike the beam theory, and the effect of the bracket on the stress distribution over the central region of the beam is studied and qualitatively compared with approximate methods of structural analysis. The principle of viewing the effect of a bracket as introducing additional constraining moment at the beam and seems largely justified from this investigation.—*Ghosh, Roy M. K. 1969. Paper submitted to R.I.N.A. for written discussion, paper W12.*

Effect of Residual Stress on Brittle Fracture in Steel Structures

It appears to the authors that the theory of crack opening displacement based on a dislocation model is useful for the analysis of brittle fracture initiation, which necessarily follows the local yield at the tip of the pre-existing crack.

The authors carried out deep notch tests with and without welding residual stress and investigated the effect of the residual stress on brittle fracture initiation in steel structure. They also examined the effect of pre-loading on the initiation using same specimens.

The test results were analysed using the crack opening displacement theory. Calculation results are in good agreement with the experimental ones.—*Akita, Y., Yada, T. and Sakai, K., Jnl of the Society of Naval Architects of Japan, June 1969, Vol. 125, pp 227–235.*

The Thermal Fatigue Resistance of Copper-to-Steel Weld Joints

An investigation has been undertaken to evaluate various steel-to-copper weldments under conditions involving thermal cycling. The filler metals compared were aluminium, bronze, copper and nickel and two different levels of base metal dilution.

Thermal fatigue cracks tended to initiate and propagate preferentially in and through the coarse-grained heat-affected zone of boron-deoxidized copper. Crack growth was intergranular and was accompanied by oxidation along the crack boundaries. Weld metal embrittlement occurred in aluminium bronze deposits if iron pickup was excessive. Embrittlement did not occur in either nickel or copper deposits.

The data indicate that copper filler metal is the best choice for the dissimilar metal welds subjected to thermal fatigue conditions in service.—*Howes, M. A. H. and Saperstein, Z. P., Welding Journal (N.Y.), December 1969, Vol. 48, pp. 543-s–550-s.*

An Experimental Study of the Effect of Beam Variation and Shallow Water on Thin Ship Wave Predictions

This paper describes a new method of correlating ship wave theory with experiment and examines its application to a study of the effects of beam variation and shallow water using three mathematical hulls.

Equivalent source arrays are determined which according to linear theory would generate the waves actually measured behind the three models over a range of speeds in deep water. It is shown that the analysis offers valuable insight into the interpretation and validity of thin ship wave calculations. Also the equivalent arrays determined from deep water experiments yielded reasonably accurate predictions of the corresponding wave resistance and wave spectra derived from experiments in shallow water.—*Everest, J. T. and Hogben, N. 1969. Paper submitted to R.I.N.A. for written discussion, paper No. W11.*

Stress Corrosion Testing of 7079-T6 Aluminium Alloy in Seawater Using Smooth and Pre-Cracked Specimens

Stress corrosion cracking of a 7079-T6 aluminium alloy in seawater was investigated by bend tests of smooth and pre-cracked specimens. It was found that there was a general agreement between results of the two methods. The susceptibility of this alloy to intergranular cracking was shown to depend on its directional characteristics. The use of pre-cracked specimens for stress corrosion testing is discussed.—*Chu, H. P. and Wacker, G. A., Trans ASME, Jnl of Basic Engineering, December 1969, Vol. 91, pp. 565–569.*

Mammoth Ships

Ships provide the cheapest means of transporting bulk products where speed of delivery is not of major importance. Increase in ship size leads to a decrease in the cost of transportation per ton mile. The various technical problems which arise from greater ship size are reviewed. It is concluded that it is technically possible to construct vessels larger than the largest built to date. Choice of dimensions depends upon economic feasibility and cost optimization. Further research in various branches of naval architecture is needed since practice has outstripped the advances in knowledge from theoretical and experimental investigations which are necessary if mammoth ships are to operate efficiently.—*Conn, J. F. C. 24th February 1970. Paper presented at a joint meeting of the Institution of Engineers and Shipbuilders in Scotland and the Royal Institution of Naval Architects.*

Brittle Fracture Characteristics of New Mitsubishi Bainitic Steel

The author succeeded in developing a new type of M.B.H.T. (Mitshubishi Bainitic High and Ultra-High Tensile Steel) of the 100 kg/mm^2 class, which can be used for high tensile steel welded structures subjected to explosive loading under low temperature circumstances.

In this paper, the results of Charpy-V notch and press notch impact test, deep notch test for brittle fracture initiation characteristics, Esso test with temperature gradient for brittle fracture arresting characteristics, N.R.L. type drop weight test and crack starter explosion test are reported for two kinds of this M.B.H.T. The results proved that these two kinds of M.B.H.T. have superior characteristics for brittle fracture in comparison with the HY-80 steel used in U.S.A.—*Suzuki, K., Mitshubishi Technical Bulletin, June 1969, No. 58.*

Thrust of Poppet Valve

Analysis and experiments were performed to study the static characteristics of the thrust of poppet valves.

Valves with cone angles between 40 degrees and 90 degrees were tested under different conditions.

Experimental results reveal that the momentum theory is valid for the estimation of the thrust. For poppet valves having lappings with valve seats, pressure integrals on the seats are required for the estimation of the thrusts.

The boundary layer theory was applied to the flow in the clearances between valves and valve-seats, thence pressure integrals were calculated.—*Urata, E., Bulletin of the Japan Society of Mechanical Engineers, 1969, Vol. 12, No. 53, pp. 1099–1109.*

Model Icebreaking Experiments and their Correlation with Full Scale Data

The mechanism of icebreaking is discussed and in particular the part played by broken pieces of ice sliding along the hull sides is mentioned. The techniques involved in producing an analogue of the situation by means of models are described and it is shown how an appreciation of the deficiencies of the use of modified wax to represent the icefield, has led to the development of a new material which provided an encourgaging degree of correlation with the results of full scale trials. The full scale trials are described in some detail. Methods of representing icebreaker performance data are discussed and a simple theory is derived to link model data with the full scale results.—*Paper by Grago, W. A., Dix, P. J. and German, J. G., presented at a meeting of The Institution of Naval Architects, 22nd April 1970; Paper No. 3.*

Fuel Drop Vaporization Under Pressure on a Hot Surface

The evaporation lifetime of liquid drops in contact with a hot surface is investigated at pressures of up to 69 atm. Modes of evaporation and drop behaviour at sub-critical and supercritical conditions of temperature and pressure are described. It is shown that at any sub-critical pressure the lifetime is a minimum at a surface temperature just above the saturation temperature; the lifetime at all supercritical pressures is a constant minimum when the surface temperature is 60°C or more than the critical temperature of the liquid.—*Paper by Temple-Pediani, R. W., submitted for written discussion to The Institution of Mechanical Engineers, Paper No. P38/70.*

Simulation of Steering and Manoeuvring Qualities of Ships

Simultation of ship manoeuvring qualities by means of free-running models and a manoeuvring simulator is discussed briefly.

It is shown how the use of a manoeuvring simulator has advantages when the human operator is an important part of the system and when automatic control systems and navigating instruments have to be developed and adjusted. The use of a simulator in ship design handling is shown.

A description is given of the new N.S.M.B. manoeuvring simulator.—*Van den Brug, J. B., International Shipbuilding Progress, January 1970, Vol. 17, pp. 4–14.*

H.M.C.S. Bras D'Or—an Open Ocean Hydrofoil Ship

H.M.C.S. *Brass D'Or* is a 200 ton 50–60 knot hydrofoil ship designed for anti-submarine operations in the open ocean. This paper outlines the hydrodynamic principles on which this novel design has been based, describes the main features of the ship and reports progress of initial trials. While her evaluation is far from complete, early results are promising and some preliminary thoughts are presented on prospects for this type of ship in both military and commercial roles.—*Paper by Eames, M. C. and Jones, E. A. presented at a meeting of The Royal Institution of Naval Architects, 22nd April, 1970; Paper No. 2.*

Nautical Aspects of Ship Handling in Simulation

When ship handling is simulated on a manoeuvring simulator either for training or for research projects, where the human being, handling the ship, is involved, a careful analysis has to be made of the nautical aspects of handling a ship in a simulator.

Discussions between designers, users and people with practical knowledge of situations and manoeuvres to be simulated, are essential to ascertain an optimum use of the simulator facilities.—*Don, C., International Shipbuilding Progress, January 1970, Vol. 17, pp. 14–19.*

The Structural Design of Supertankers

This paper discusses the structural design problems of supertankers and describes a finite element programme written for their stress analysis. This programme written completely in Fortran minimizes data preparation and computer time and makes practicable the three dimensional analysis of the entire cargo length. The need for three dimensional analysis is demonstrated and it is shown that a correct analysis need involve no more work than cruder approximations. The need for using correct loadings and wastage allowance and for an understanding of collapse mechanisms is emphasized.—*Paper by Kendrick, S. presented at a meeting of The Royal Institution of Naval Architects, 23rd April, 1970; Paper No. 8.*

Economics of the Containership Sub-System

The effects of a container operation on a traditional steamship company are discussed, and the need for planning emphasized. The "optimum" containership is defined, and the naval architect's role in its determination is outlined. The two basic approaches to optimizing a system are described, and the limitations of each approach stipulated. An algorithm which estimates both capital and operating costs for containerships is presented, together with relationships for capacities, weights and dimensions.—*Miller, D. S., Marine Technology, April 1970, Vol. 7, pp. 180–195.*

The Development Trend of Marine Diesel Engines

The author, after examining the evolution in the course of time of the basic characteristics of the 4-stroke medium speed engines built by Fiat, and after illustrating the trends of the present production in this field, describes some researches and experiments for the increase in performances and use of heavy fuel oil. Finally, the author presents some examples of propelling sets with engines type A 230—B 300 and C 420.—*Fiat, G. M. Technical Bulletin, January–March 1969, Vol. 22, pp. 1–14.*

Study on Separation of Ship Resistance Components

This paper describes theoretical and experimental studies on the separation of ship total resistance into its components with physical meanings, namely viscous resistance and wave resistance. The theoretical analysis gives the asymptotic formulas of both components with linear approximation. The experimental data are analysed by use of the asymptotic formulas. The sum of both components is in good agreement with the total resistance derived by dynamometer. This study also indicates that the interaction term of these components can be neglected as a higher order quantity.—*Baba, E., Jnl of the Society of Naval Architects of Japan, June 1969, Vol. 125, pp. 9–22.*

Qualifications for Shipbuilding

This paper considers the qualifications available to those who intend to work in and manage modern British shipyards and suggests a salary scale for the various stages in the management hierarchy. It also attempts to forecast the pattern of shipbuilding and the work of a naval architect in the future.—*Paper by Baxter, B., presented at a meeting of The Royal Institution of Naval Architects, 22nd April 1970, Paper No. 5.*

Patent Specifications

Marine Propeller

Referring to Figs. 1 to 3, the stern portion (10) of a vessel is provided with a cylindrical opening (11) sized to receive rubber staves (12) forming a water lubricated rubber bearing bush (13). A stuffing box (14) is bolted on the forward side of opening (11). The assembly (14) provides a radially extending surface against which one end of each of the staves abuts. Adjacent to the other end of the cylindrical opening (11) is a heavy duty segmented ring secured in place by bolts (17). The ring (16) co-operates with assembly (14) to provide axial compression of the staves (12) to secure the bearing staves in position by radial expansion. A one-piece propeller (18) of non-corrosive material has a hollow hub (19) from one end of which extends a number of propeller blades (21).

The hub portion (19) is faired at its rearward end (22), and its forward portion extends through the bush (13), past the stuffing box (14), seals (16) being provided which are compressed by ring (16a). The stuffing box assembly (14) can be moved forward along the shafting to permit inboard removal and replacement of staves.

The hub portion (19) has an axially extending opening (26) which extends with substantially uniform diameter from a closed end at (27) to a substantially conical inner face (28) joining the main opening (26) with a reduced diameter forward opening (29). Adjacent to the forward end of hub (19), forward of the conical surface (28), is an annular groove (31) defining the rearward side of flange (32). Extending inward from the periphery of the forward side of flange (32) is a radial face (33) fitting against a mating end face of flange (39) on the ship's internal drive shaft (36). The end face (33) is provided with axial projection (37) which fits a mating recess (38) on shaft (36) to assist in laterally locking the joint.

The flange (32) and flange (39) have bolt holes through which bolts (41) extend. The annular groove (31) provides access for the nuts (42).

The adjacent end of the internal drive shaft (36) is supported about its longitudinal axis by bearing (43) secured to the stern portion (10) of the hull. The design shown com-

FIG.-1

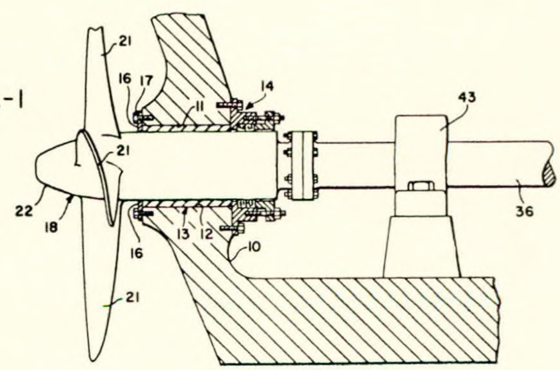
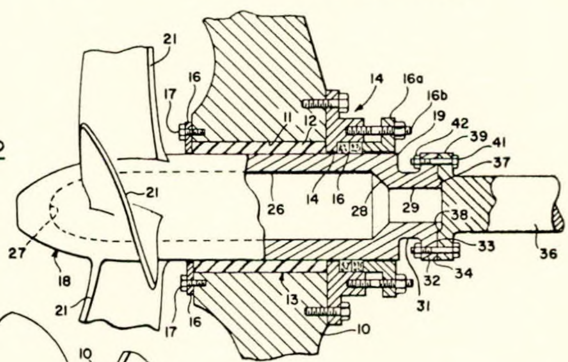


FIG.-2



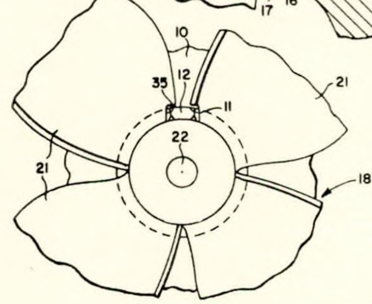


FIG.-3

pletely eliminates the conventional tail shaft and its corrosion resistant sleeve. Instead it provides a large diameter, one-peice tubular hub. The hub because of its large section modulus is capable of efficiently absorbing substantially all of the torsional and bending stresses encountered even when it is formed of non-ferrous metal. It does not provide objectionable points of stress concentration, so that fatigue problems are reduced or virtually eliminated.—*British Patent No. 1 75 763 issued to Satterthwaite, J. G. and Booth, J. Complete specification published 23rd December 1969.*

Bulbous Bow Hull

The present invention relates to bulbous bows.

Fig.1.

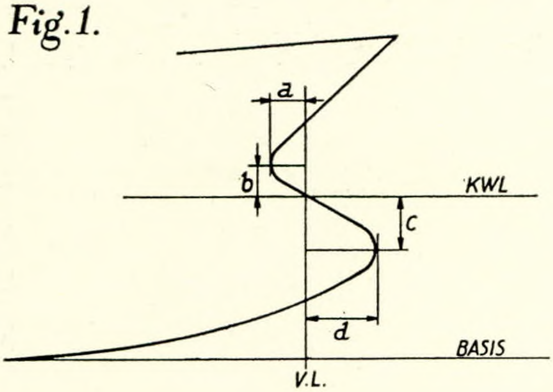


Fig.2.

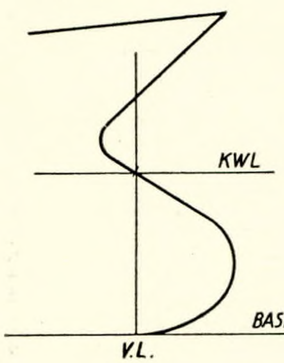
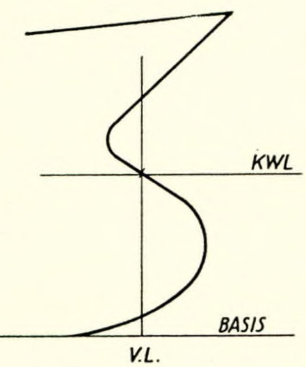


Fig.3.



Figs 1, 2 and 3 show various bulbous bows according to the invention.

The bulb of the bulbous bow extends, beneath the designed waterline KWL, forwardly of the forward perpendicular V.L., i.e. the perpendicular line drawn through the point at which the stem intersects the designed waterline. The bulb extends upwardly above the designed waterline and, above the designed waterline, rearwardly of the forward perpendicular.

With a known shape of boat it is possible to calculate the maximum height of the bow wave, when the ship is at full speed. The bulb according to the invention is designed so that the dimension *b* is as large as the height to which the maximum bow wave reaches on a ship otherwise the same but having a conventional bow.

The other dimensions *a*, *c* and *d*, have to be determined for each ship to obtain the most advantageous dimensions. —*British Patent No. 1 176 189 issued to Veb Schiffswerft Neptun Rostock. Complete specification published August 3rd 1967.*

A Ship with Ice-Breaking Attachment

This invention relates to ships with ice-breaking attachments of the type having a vibratory mechanism for crushing the ice.

Referring to Fig. 1, an articulated connexion of the ice-breaking attachment (1) having a vibratory mechanism (2), to the ship (3) is ensured by means of a coupling (4) rotatably mounted on a shaft (5), the coupling being fastened in the stern of the ice-breaking attachment. In this arrangement, the centre line of the shaft (5) lies in the fore-and-aft plane of symmetry of the ice-breaking attachment, while the coupling (4) is in engagement with a guide (6) located in a vertical position in the stem of the ship (3). The movement of the coupling along the guide ensures linear movement of the ice-breaking attachment and the ship in relation to each other along their respective vertical axes, while the rotation of said coupling enables angular movements of the ice-breaking attachment and ship in relation to each other with respect to their fore-and-aft axes. Angular movement of the ice-breaking attachment and the ship in relation to their transverse axes, said transverse movement being actually negligible, is made possible by the provision of the necessary play between the coupling (4) and the guide (6).

Mounted at the sides of the stern part of the ice-breaking attachment are two rollers (7) which rotate freely on the shafts (8). These rollers take up the thrust and the torque imparted by the vessel's screws and rudders through vertical girders (9) installed at the bow of the ship. Upon movement of the ice-breaking attachment and the ship in relation to each other the rollers (7) roll along the contact surfaces of the vertical girders (9).—*British Patent No. 1 179 667 issued to Tsentralnoe Tekhniko-Konstruktorskoe Bjuro Ministerstva Rechnogo Flota Rsfsr. Complete specification published January 28th 1970.*

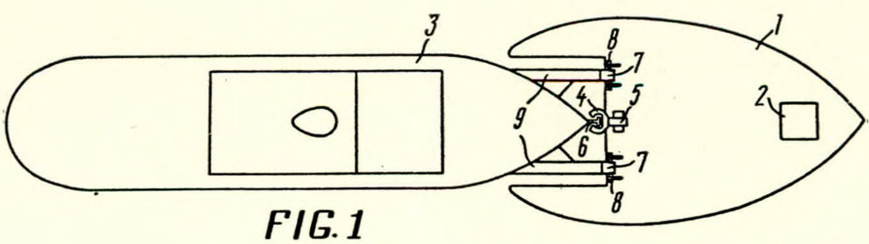


FIG. 1

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