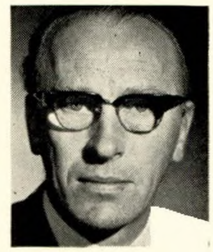
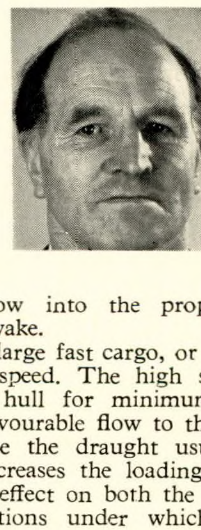
THE DESIGN AND DEVELOPMENT OF PROPELLERS FOR HIGH POWERED MERCHANT VESSELS

L. Sinclair, C.Eng., M.R.I.N.A., M.I.Mech.E. (Member)* and
A. Emerson, M.Sc., C.Eng., M.R.I.N.A.†

Mr. Sinclair





Mr. Emerson

In the last five years there has been a continuous increase in the power carried on the shaft of a single screw vessel. This has posed a number of quite serious problems not the least of which is the increasing weight of the propeller and the difficulties of handling and manufacture. The increased blade loading results in a steadily diminishing propulsive coefficient coupled with an increasing tendency towards cavitation.

It seems clear that urgent consideration should therefore be given to various methods of improving the performance and diminishing the propeller loading. To this end the paper attempts to give guidance from the propeller designer's point of view regarding the change over from single screw to twin screw propulsion and also makes the suggestion that triple screws are worthy of consideration in certain cases.

In the development of high powered vessels two distinct classes have emerged, these being the very large bulk cargo vessel or tanker where the ship size is increasing rapidly without a significant change in the ship speed and the fast cargo or container vessel where with relatively unchanged ship dimensions the speed is continuously increasing. The paper comments on the various advantages and disadvantages of single, twin and triple screws on these two ship types and also considers the influence of contra-rotating propellers, controllable pitch propellers and other propulsion devices.

INTRODUCTION

The propeller of a single screw vessel works in a wake stream which varies considerably around the disc and this means that quite large unbalanced hydrodynamic forces are developed which under certain conditions can cause hull or shafting vibration. This variation of loading on the propeller through its 360° rotation also causes high cyclic stresses in the blades such that the corrosion fatigue resistance of the propeller material becomes an increasingly important factor. Nevertheless a propeller on the centreline is able to recover some of the momentum from the ship frictional wake and it is usually more efficient to use this system rather than two or more wing propellers. In addition there are clearly economic factors favouring a single engine and a single line of shafting.

As a result of these two primary reasons, it is customary to use a single centreline shaft unless these inherent advantages are outweighed by either manœuvring requirements or by draught limitations preventing the fitting of the best propeller or by the difficulties of absorbing the required power on a single screw.

It is convenient to discuss the problems of increasing power in two parts, corresponding to the two distinct types of high-powered merchant ships that have emerged:

A) The relatively slow mammoth tanker, or bulk carrier, of ever increasing size. In this case the low speed/length ratio leads to a very full hull form and an unfavourable flow into the propeller, with wide fluctuations in wake.

B) The moderately large fast cargo, or container, ship of ever increasing speed. The high speed/length ratio requires a fine hull for minimum resistance with possibly more favourable flow to the propeller. However in this case the draught usually restricts the diameter and increases the loading on the propeller with an adverse effect on both the efficiency and the cavitation conditions under which the screw will operate.

The object of the present paper is to reconsider the propulsion, powering and propeller problem for these two special types of vessel in the light of information and techniques now available. It is also hoped to give some indication of the future trends in so far as main propulsion is concerned.

A—LARGE TANKER AND BULK CARGO VESSEL

Increase in Size and Power

There has been a considerable increase in the size of these vessels from the standard ship of 12 000 dwt during the war years to the mammoth variety now contemplated, reaching 1 000 000 dwt. On the other hand there has not been a commensurate increase in speed and this has remained relatively constant around 16 knots. Using the diagrams shown in Fig. 1, prepared from details of all ships available to the authors (some of the larger of which were design studies), it is possible to pick out a series of ships increasing in deadweight from 100 000 to 500 000 tons and then calculate hull resistance estimates based on systematic series model data (see Table I).

* Chairman, Stone Manganese Marine (Research) Ltd.
† Reader in Ship Hydrodynamics, Dept. of Naval Architecture, University of Newcastle upon Tyne, and Director, Stone Manganese Marine (Research) Ltd.

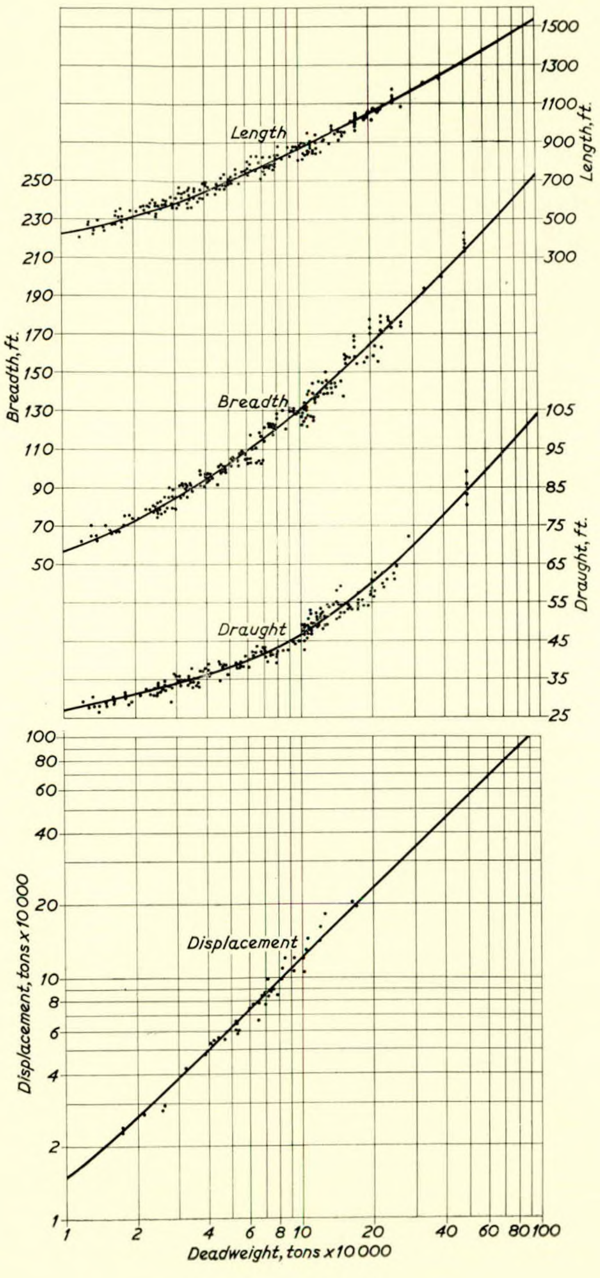


FIG. 1

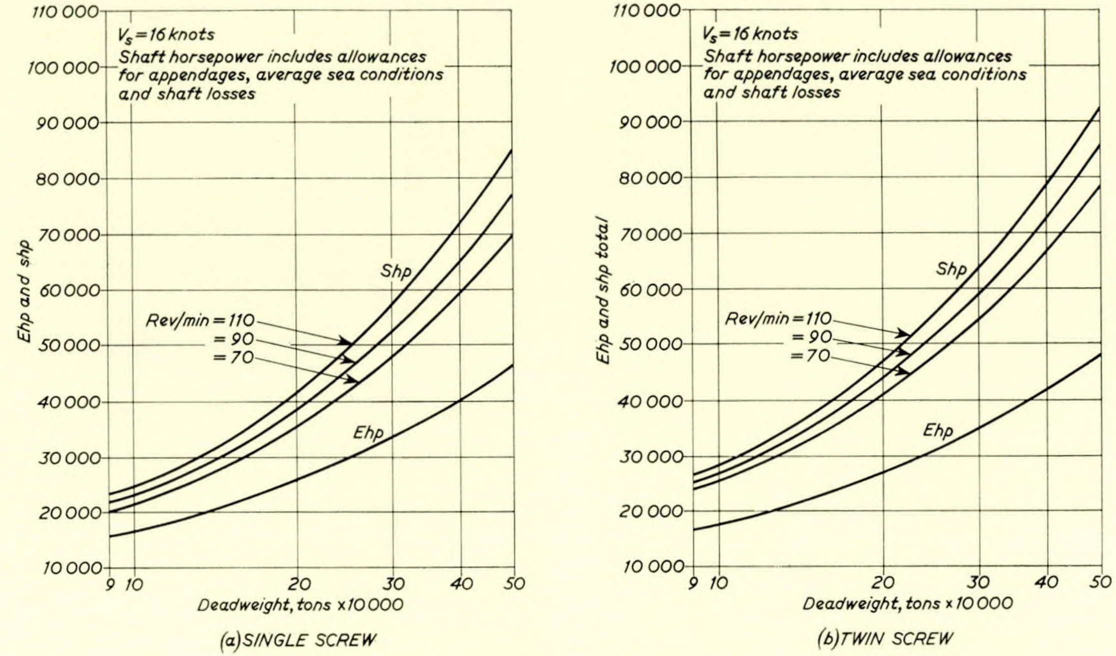


FIG. 2

TABLE I

Ship	a	b	c	d	e
Deadweight	100 000	200 000	300 000	400 000	500 000
Length, ft	865	1050	1160	1240	1315
Breadth, ft	131	165	186	202	214
Draught, ft	47	60	70	77·5	83·5
Displacement	123 250	243 540	358 160	465 900	570 660
Single Screw e.h.p. (naked) for 16K	13 360	20 744	26 890	32 225	37 129

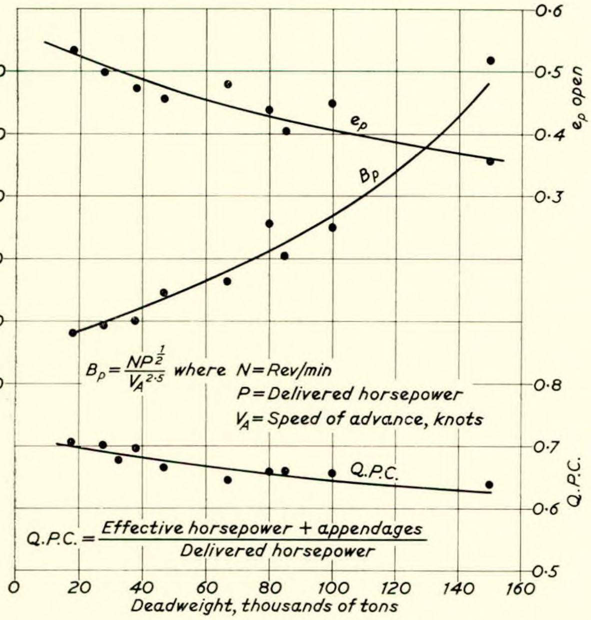


FIG. 3—*Curves showing reduction in propeller efficiency and propulsive coefficient with increasing ship size*

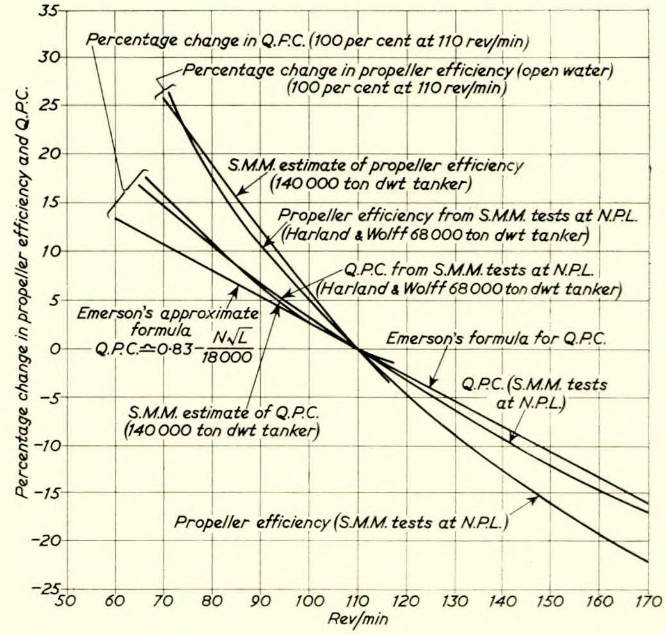


FIG. 4—*Effect of rev/min on propeller efficiency and Q.P.C.*

From these estimates of effective horsepower, using an iterative process, it has been possible to produce curves of shaft horsepower for these ships which have been plotted in Fig. 2 for both single and twin screw installations. From this will be seen the very large total power needed if such vessels are to increase in size much above 200 000 dwt while still being driven at a service speed of 16 knots. In this latter connexion, allowances on e.h.p. have been made to take account of the increase in air, weather and appendage resistance for average fair weather fully loaded service conditions. This allowance amounts to a 20 per cent increase to the trial power for a given speed.

Large Diameter Slow Turning Propellers

Until recently, the increasing size of vessel had not been matched by a commensurate increase in the size of propeller. This was in part the result of Diesel engines being developed having standard revolutions per minute (depending on type) of between 110-125 and partly because of gearing difficulties in the case of the steam turbine. Under these circumstances, the propulsive coefficient will inevitably fall with increasing ship size and this is shown in Fig. 3[(1)]. If on the other hand the propeller is more correctly designed in relation to immersion or draught and then the optimum rev/min are chosen, a marked increase in propulsive coefficient can be obtained (see Fig. 4). This is however to be achieved only with a considerable reduction in rev/min and a corresponding increase in propeller diameter and propeller weight (see Table II).

Single or Twin Screws

It is clear from the above that a single screw should be retained with increasing ship size while keeping rev/min to a minimum but there must be some upper limit to this because presumably there is a limit to the size of propeller that can be manufactured, handled and transported. To try to establish this limit, detailed calculations for a large number of propellers have been completed, this being made possible by the use of the computer programmes developed by the authors' company. The results are plotted in Fig. 5.

These curves show the diameter, propulsive coefficient and propeller weight to a base of shaft horsepower for a series of rev/min from 70-120. Both single and twin screw propellers are covered by the curves and they can be relied upon as accurate practical propellers and quasi-propulsive coefficients, providing the speed of the ship is approximately 16 knots and the mean Taylor wake fraction is 0·45 for the single and 0·20 for the twin screw ship. (These have been shown to be reasonable values in practice for full bodied vessels of this type.)

If the limit on propeller size is taken as the maximum at present considered, i.e. about 60 tons finished weight, the curves for single screws in Fig. 5 can be entered and the

TABLE II—SINGLE SCREW

Ship		a	b	c	d	e
Deadweight		100 000	200 000	300 000	400 000	500 000
70 rev/min	SHP	21 511	35 594	47 952	58 967	69 461
	Diameter, ft	30·2	33·8	36·0	37·8	39·2
	Weight, tons	44·0	70·0	93·0	113·0	—
90 rev/min	SHP	23 163	38 542	52 329	64 836	76 993
	Diameter, ft	26·8	29·7	31·6	33·0	34·3
	Weight, tons	36·5	58·5	78·0	95·0	112·0
110 rev/min	SHP	24 580	41 456	57 048	71 318	84 896
	Diameter, ft	23·9	26·8	28·8	30·1	31·2
	Weight, tons	31·5	51·5	70·0	87·0	103·5

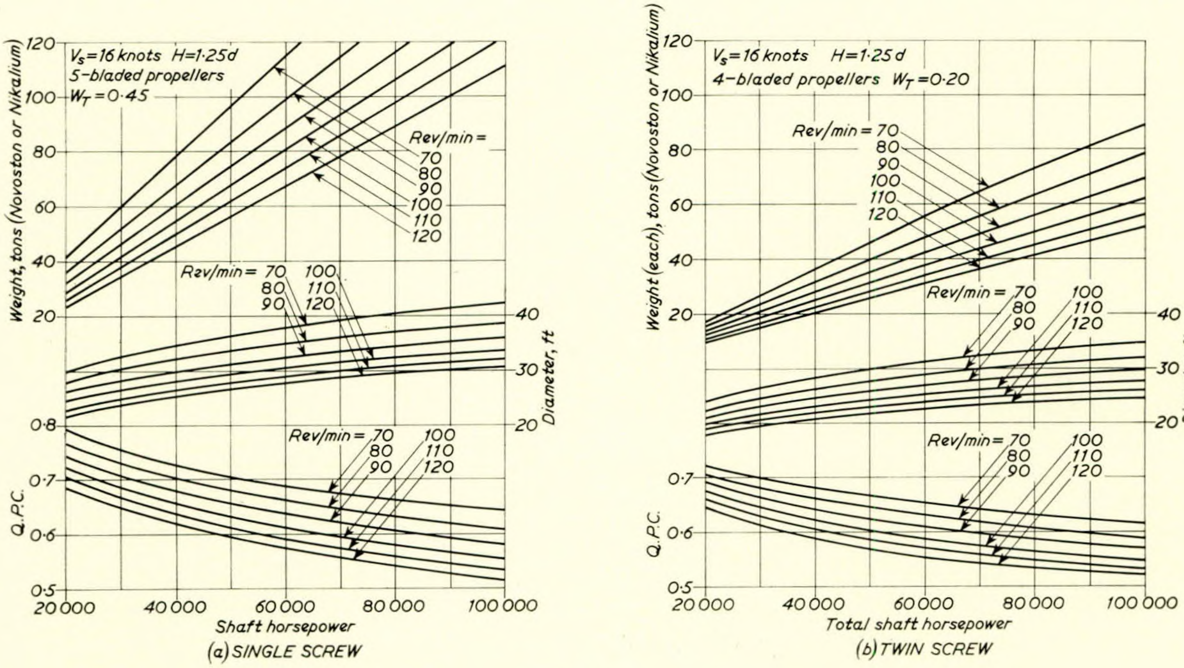


FIG. 5

maximum horsepower that can be delivered to the propeller will be seen to be about 48 500 shp at 110 rev/min with a q.p.c. of 0·615. Reference to Fig. 2 will then show that with this power and q.p.c. (the iteration process having already been embodied) the maximum deadweight that can be propelled in service can be found, i.e. about 245 000 tons. If the lowest practical rev/min limit of say 70 had been chosen, then the limiting power for the single screw would have been only 30 000 shp propelling a ship of only 160 000 dwt. Revolutions of 110 per minute were taken as the top end of the rev/min scale in common use with such powers.

Alternatively, the ship of 245 000 dwt could be propelled by twin screws and from Fig. 2 the total power will be 47 200 shp and the q.p.c. 0·671 the weight of each propeller being only 43·4 tons at 70 rev/min (the reasonable minimum value). Incidentally, the powering estimate includes an extra five per cent for the additional appendage resistance of the twin screw vessel and an extra one per cent for transmission losses.

It will be seen that by this criterion (obviously not the sole criterion) for a ship of 245 000 dwt, there is already a small advantage in favour of the twin screw installation although perhaps at this point, not sufficient to outweigh the other economic advantages of the single screw vessel.

The propeller manufacturer has however stated that he is prepared to produce the largest propeller required by the shipbuilder and it seemed therefore desirable to consider the influence on ship size and power of a series of limiting propeller weights: 60, 70, 80, 90 and 100 tons. These are given in Table III and clearly indicate the advantages for twin screw as size and power increases.

In summarizing this, it would appear from this approach that using the maximum weight of propeller at present available, a slight advantage will be gained from the point of view of propulsive coefficient by twin screws when the vessel reaches a deadweight of about 250 000 tons. Above this figure the twin screw installation will be found increasingly desirable from the point of view of performance as well as from other considerations such as handling, fitting the propellers, manœuvrability, etc.

Triple Screws

As will be seen from the foregoing, at around 250 000 dwt reconsideration of the shaft configuration is necessary as there may well be a case for changing from single to twin screw. If a change is contemplated however, it is obviously desirable to consider not only twin but also triple screws. The triple screw installation is more desirable in that the advantage of a smaller power per shaft is achieved while still utilizing the centreline propeller and thus obtaining some advantage from the frictional wake.

The 300 000 ton vessel has therefore been considered in detail using the curves already provided. Approximate solutions had previously shown that a reasonable distribution of power would be 50 per cent on the centreline and 25 per cent on each of the two wing screws. Using this basic consideration Table IV has been prepared.

TABLE III

	Single screw			Twin screw for same deadweight		
Limiting Weight tons	Max. shp for 110 rev/min	Q.P.C.	Deadweight for 16 Knots	Shp at 70 rev/min	Q.P.C.	Propeller Weight tons
60	48 500	0·615	245 000	47 200	0·671	43·4
70	56 800	0·597	298 000	54 100	0·661	50·0
80	65 500	0·582	356 000	61 400	0·651	56·8
90	73 600	0·568	415 000	68 600	0·643	64·4
100	82 100	0·557	479 000	76 000	0·635	69·8

TABLE IV

	Single		Twin	Triple
Rev/min	70	110	70	70
Q.P.C.	0·709	0·6	0·66	0·743
SHP	47 950	57 048	54 470	48 150
Propeller weight, tons	93	70	50·5	49 centre 21·5 wings

Again it will be noted that if a limiting propeller weight is stipulated a considerable saving is made by changing to twin screw. The gain is even more marked for the triple screw installation and the total propeller weight is less than for the twin screw ship.

As a further exercise various proportions of the total power were used in determining the power of the triple arrangement and the curves shown in Fig. 6 have been prepared from this work.

Tandem and Contra-rotating Propellers

In comparison with two wing screws, the propeller pair on the centreline (tandem) may be expected to show a gain in overall efficiency because of the frictional wake and because of the elimination of bossing resistance. By making this propeller pair "contra-rotating" there is an additional gain due to the reduction of rotational energy loss.

In the case of the tanker and bulk carrier of increasing size, there are practical limitations to the weight and diameter of propeller which at present can be made and fitted. The limited diameter propeller works in a very heavily loaded condition at low efficiency and if it is replaced by two propellers on the same shaft each works at higher efficiency. Because of the reduced loading, the extent of cavitation is much less and there is a reduction in disturbance at the stern and a reduction in blade frequency vibration. The out of balance hydrodynamic forces experienced at the moment due to the very uneven tanker wake are experienced in full by only

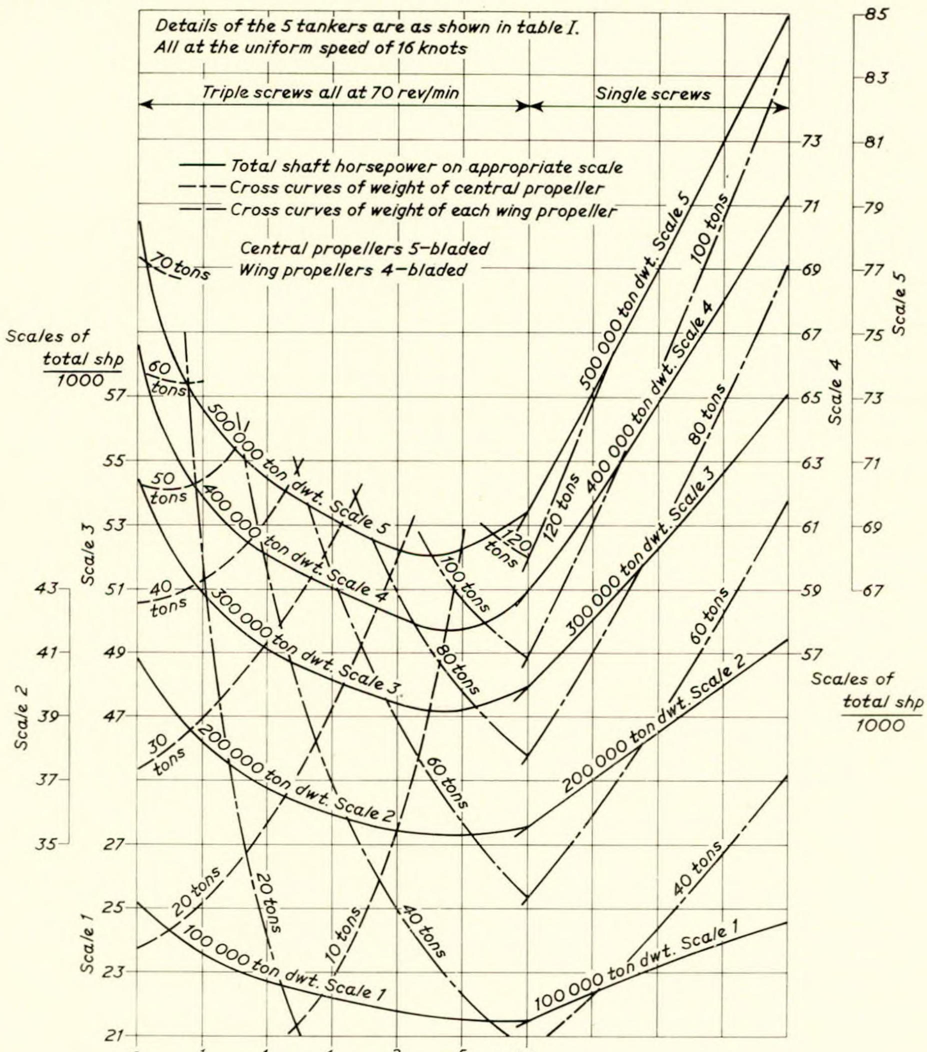


FIG. 6—*Total power required for mammoth tankers with one, two or three propellers —Details of the five tankers are shown in Table I all at the uniform speed of 16 knots*

the forward propeller of the contra-rotating pair, the forward propeller tending to smooth out the wake inequalities.

The relatively low speed tanker causes the propeller to have a low pitch-ratio and the rotational energy loss is small. In this case it may be better to use uni-directional or "tandem" propellers accepting the small loss in efficiency compared with contra-rotating propellers.

To test the possible advantages, model self-propulsion experiments are being made at St. Albans Tank as an alternative system for a 200 000 dwt tanker with an estimated shaft horsepower of 40 000 at 16 knots. For the single screw design 70 tons was taken as a reasonable maximum propeller weight and this determines the diameter and the best shaft revolutions. For the contra-rotating pair the same maximum diameter was used for the forward screw and the after propeller was made smaller to be inside the forward propeller tip vortices.

The computer programme used in this work for detailed design is based upon Morgan's development of Lerbs' original method[2]. The tandem pair was designed by an approximate method in order to obtain comparative results and on consideration it was thought that altering the aft contra-propeller but retaining the same forward propeller was sufficient for this additional experiment. The tandem propellers are spaced so that the Vortex sheet from the forward propeller passes between the after propeller blades. For this reason the tandem propellers have the same number of blades, i.e. four, but the contra-rotating pair have a different number of blades (four and five) to reduce blade frequency excitation.

Controllable Pitch Propellers

The controllable pitch propeller has made a rather dramatic entry into the larger merchant ship field during the past few years and there is no doubt that the trend will continue. Distinct advantages are derived from the flexibility of the c.p. propeller in craft such as tug boats and trawlers and for specialist applications such as ferries where manœuvrability is an important factor.

In the case of the large merchant vessel such as a tanker, c.p. propellers will be fitted because of advantages mainly concerned with flexibility of control, the unmanned engine room and with the fact that the astern turbine will not be needed. Perhaps even more important however, are the possibilities of c.p. propellers when associated with the multi-engined ship which depends upon the continued development of the medium speed Diesel engine.

With regard to the performance of such vessels, full scale trials have repeatedly shown that because of wake changes the power/rev/min correlation on a large tanker is unaffected by changes of draught. A fixed pitch propeller suitable for the loaded condition is therefore also suitable for the ballast draughts. In addition, it is also true that the increased resistance due to adverse weather produces insufficient change to the operating slip to affect seriously the performance of the propeller or to warrant a pitch change. This is particularly so bearing in mind that heavy weather would in any case involve a reduction in ship speed for reasons of safety. In fact the comments made by the late Professor Burrill[3] seem to be substantially true to this day, i.e. that a fixed pitch propeller has a very flat efficiency curve on a base of operating slip and there is thus little hydrodynamic gain to be achieved by pitch adjustment.

It must also be remembered that the larger propeller boss, the relatively restricted blade root design and the unfavourable blade root angles that are given at pitch settings remote from the design condition, must inevitably involve efficiency losses for the controllable pitch as compared with the fixed pitch arrangement. With good design this loss can be restricted to around two to three per cent which, even so, can be important on a large vessel of the type under consideration.

Up to the limit of the single screw tanker postulated above, i.e. around 30 000 shp, there is a good case for the use of the multi-engined installation and here a very considerable advantage can be shown for the fitting of a c.p. propeller. This is illustrated in Fig. 7 which has been prepared for a hypothetical vessel having four engines driving a single propeller through a gear-box. The fixed pitch propeller curve is superimposed on four straight lines representing the torque output from one, two, three or four power units respectively. The gain in available power can easily be seen, for example, with three engines on line, the fixed pitch propeller would absorb 15 600 bhp at 78 rev/min while an adjustment to pitch of the c.p. propeller would enable the full power of 18 000 bhp to be developed at the rated 90 rev/min. As a matter of interest the c.p. propeller for this exercise was tested in the cavitation tunnel and the Bp-δ diagram for the whole range of pitch settings and conditions is shown in Fig. 8.

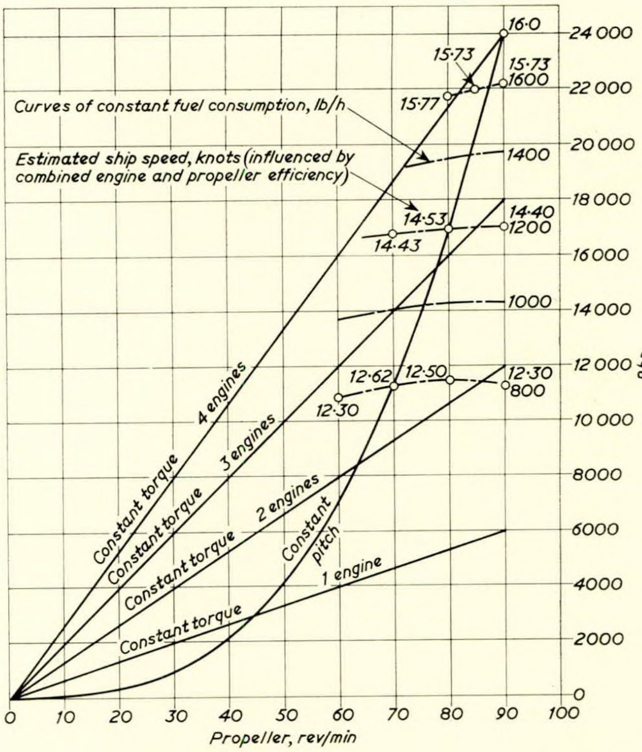


FIG. 7—*Estimated performance with controllable pitch propeller*

The advantage of the c.p. propeller in this application is well known but it is only fair to point out that some penalty, although possibly a small one, might be incurred. This results from diminishing propeller efficiency coupled possibly with the engine operating with higher specific fuel consumption than would be experienced with the straightforward solid propeller.

On very large vessels the problem of stopping the ship is one of considerable importance. In this connexion there is no doubt that the use of a c.p. propeller has a profound effect on the "head reach" and this fact in itself may well prove a very good reason for the use of a c.p. propeller on large tankers. In the authors' experience the influence on stopping time by the slow application of reduced pitch on a c.p. installation is spectacular.

Overlapping Propellers

In an interesting new stern arrangement suggested by Pien and Strom-Tejsen[4], the centrelines of twin screws are placed about 35 per cent of the propeller diameter from the centreline of the ship so that the blades of both propellers overlap in the region of the high central wake field. It is claimed that this combines the advantages of the single screw with the reduction in power loading due to twin screws.

In the central region of high frictional wake, it is said that each propeller will increase the local inflow velocity to the other and thus reduce the angles of incidence and the local peak loads, which on a large single screw tanker are

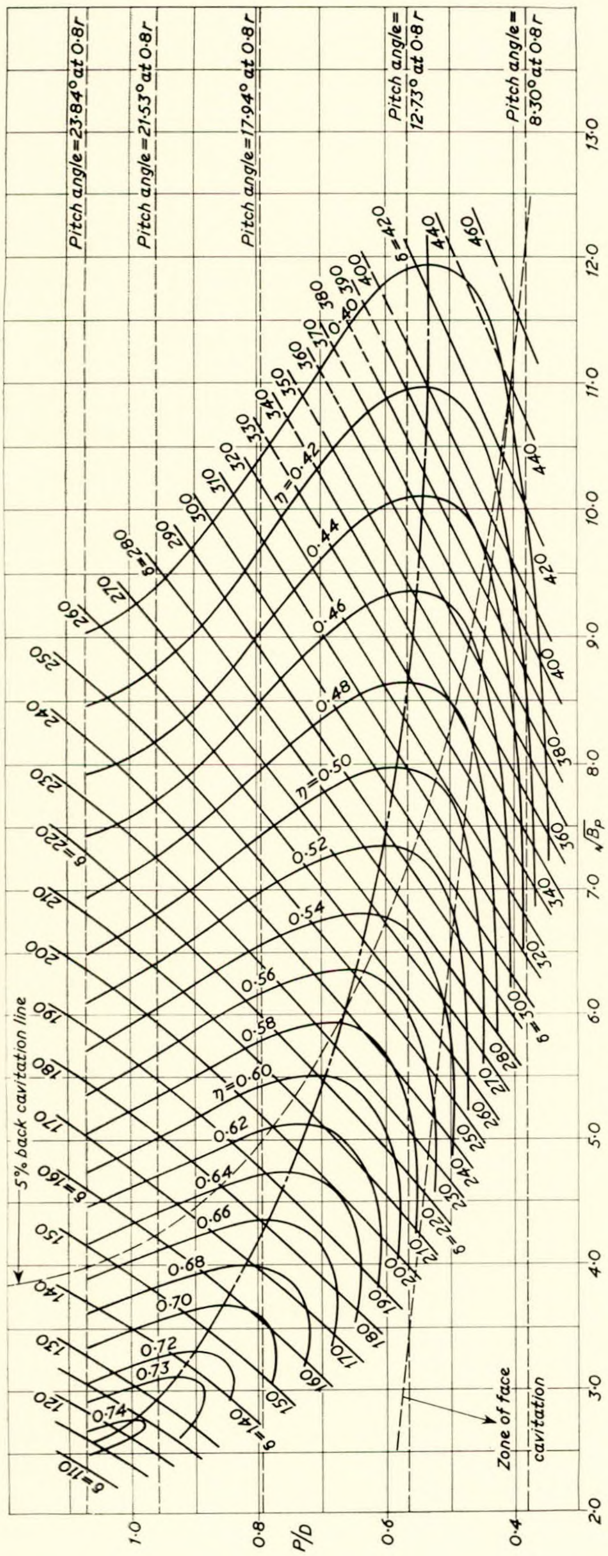


FIG. 8—*K.C.C.8: B.A.R. 0·525: 4 blades—Bp~δ diagram: σ = 11·0*

normally very high in this region. If the extreme angles of incidence are nearer to the mean angle for which the camber of the section must be designed, the tendency to both face and back cavitation should be reduced. Similarly, if the local peak loads are reduced, the fluctuations in thrust and torque should be diminished on each blade and on the whole shaft system with possible advantages with regard to fatigue and vibration.

The results given in the reference quoted are not very extensive but the arrangement seems sufficiently attractive to warrant further investigation.

Ducted Propellers

At the high propeller loading associated with the large single screw tanker there is a considerable incentive to try to improve the performance by all possible means. Reducing rev/min and increasing the propeller diameter is the most obvious solution but c.r.p. and tandem propellers offer reasonable alternatives.

The ducted, shrouded or nozzle propeller is also worthy of consideration and a number of investigators have published useful information on this subject[5]. It would appear from published work that, at the loading involved, gains of five to six per cent can be secured by a well designed propeller/nozzle system on a tanker and this must be an attractive solution if the structural difficulties can be overcome. It is also possible to use the nozzle as a steering device but here again there are practical difficulties involved.

(B)—FAST CARGO OR CONTAINER VESSEL

As mentioned in the introduction to this paper, this type of vessel differs from the bulk carrier by virtue of its increasing

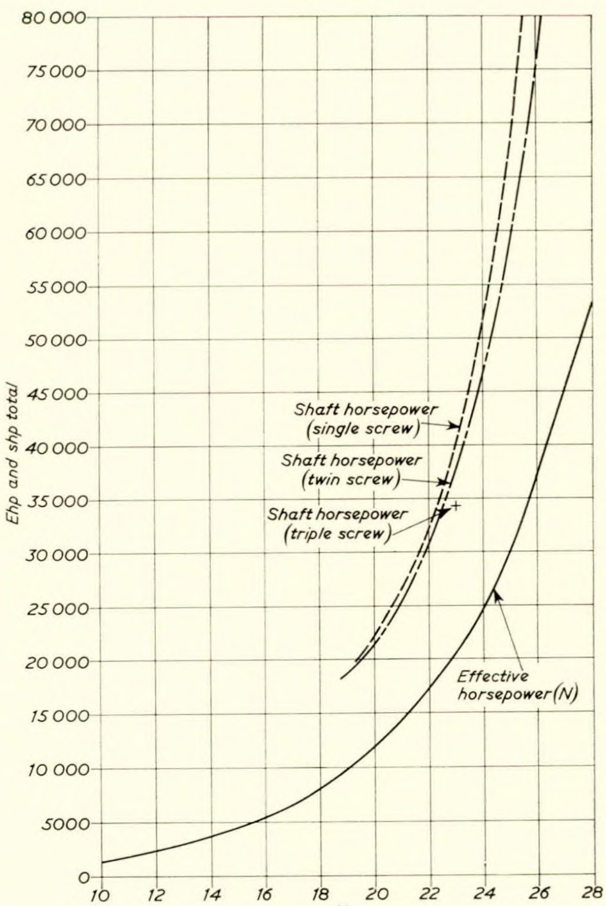


FIG. 9—*Power estimation—single and twin screw high speed cargo vessel*

speed with a relatively small increase in length and virtually no change in draught. This imposes a heavy loading on the propeller with relatively low efficiency. Because of the restricted draught it is impossible to reduce the loading by increasing the diameter and reducing the rev/min, and in fact rev/min as high as 140 to 150 are indicated to give a diameter providing sufficient immersion of the propeller. As a result of this, consideration to increasing the number of propeller shafts must be given rather earlier along the scale of increasing power than for the tanker or bulk cargo vessel.

Single or Twin

At the moment the sort of ship that is under consideration is the fast vessel, possibly for containerized cargo. A typical vessel may have a length b.p. of about 680 ft, a beam of about 95 ft on a draught of approximately 30 ft. The vessel would be fairly fine, the block coefficient being approximately 0·61. On this basis a resistance curve has been prepared using standard systematic series hull model data. (see Fig. 9).

From this curve, using a suitable ship/model correlation and allowances for appendages and average weather in service, power curves for single and twin screws have been prepared over the speed range from 20 to 26 knots. The rapid increase in power with speed will be noted and it will be seen that very high powers will be needed if speeds are to advance beyond 24 knots when single screws might become impracticable. An exercise has therefore been completed considering the alternatives of single, twin and triple screws for a ship of the above dimensions running at 23 knots, at the revolutions appropriate to propeller diameters suitable for the draught and beam for the various shaft configurations.

It will be seen that at this speed a power in the region of 40 000 shp will be needed. The single screw suitable for a centreline shaft would give a q.p.c. of 0·645 and would weigh 37·5 tons. Alternatively the twin screw installation would give a q.p.c. of 0·724, each propeller weighing 22·2 tons. This illustrates that a point has already been reached at which twin screws would be the preferred arrangement from the propeller point of view.

Triple Screws

As an additional exercise a triple screw installation was considered using one third of the power on each shaft. This would lead to an overall q.p.c. of 0·803 and would require a centre propeller weighing 17·5 tons and wing propellers of 16·9 tons each.

In each case, single twin and triple, it has been assumed that the propeller diameter would be retained at 21 ft as this is appropriate to the draught, and there would seem adequate beam to give reasonable cover, even with the triple screw arrangement.

Table V summarizes the results.

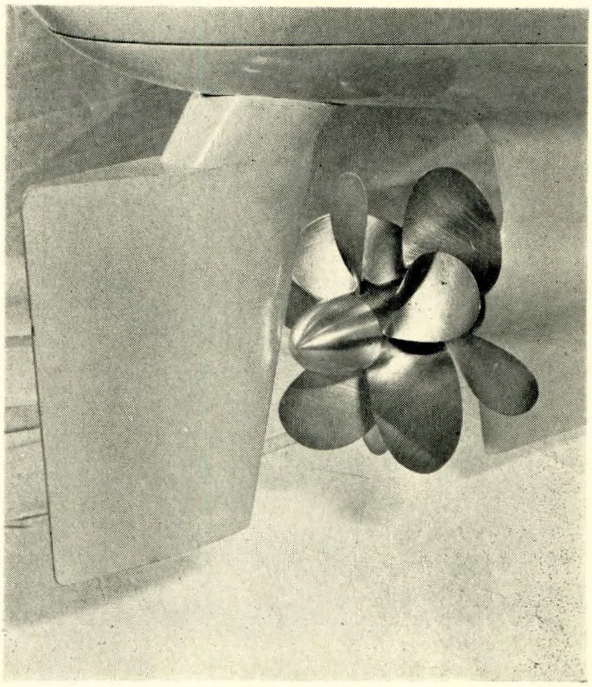


FIG. 10

Tandem and Contra-rotating Propellers

In this case as compared with the tanker, the diameter of the propeller is restricted by draught and this means a heavily loaded propeller with low efficiency as well as a difficult cavitation problem. As the pitch ratio is no longer small and the rotational losses high, the tandem propeller has not been considered for such a vessel but experiments have been made with c.r. propellers as a further alternative to the single screw vessel.

The first results available are for the c.r. pair designed according to the method published by Glover(6). Self propulsion experiments on such a vessel have shown a clear improvement to the propulsive coefficient of ten per cent achieved over a good conventional design with a single propeller. The stern arrangement for these tests is shown in Fig. 10. This preliminary result should probably be improved upon by adjustment to design in an effort to find the optimum solution to the problem. In this case there is some loss due to the increased thrust deduction factor with the c.r. pair and the actual propeller efficiency improvement is greater than ten per cent. This contra-rotating pair of propellers are shown

TABLE V—FAST CARGO VESSEL

Length, b.p.	680 ft			
Breadth	95 ft			
Draught	30 ft			
Displacement	33 770 tons			
Block coefficient	0·61			
Speed	23 knots			
	Single	Twin	Triple	
			Centre	Wings
Power, shp	40 400	2×19 050 38 100 total	11 450	2×11 650 34 350 total
Rev/min	142	100·5	87·5	86·5
Diameter	21 ft	21 ft	21 ft	21 ft
Surface area	335 ft^2	221 ft^2	188 ft^2	177 ft^2
Number of blades	4	4	4	4
Propeller weight	37·5 tons	22·2 tons each	17·5 tons	16·9 tons each
Q.P.C.	0·645	0·724	0·803 (overall)	

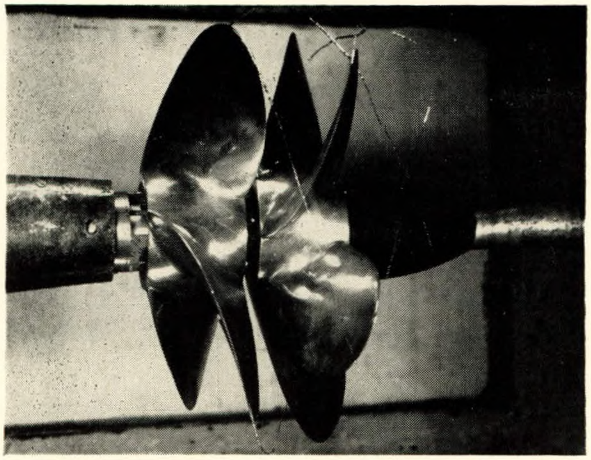


FIG. 11

in Fig. 11 in the cavitation tunnel at Newcastle, and the marked absence of cavitation at the operating condition will be noted.

It must be emphasized that these data represent only the start of this investigation and it is thus difficult to generalize from the results.

Controllable Pitch Propellers and Other Devices

The controllable pitch propeller has begun to infiltrate into this class of ship but here again the reasons are concerned with considerations other than propeller efficiency. Very high powers on a limited propeller diameter pose special problems for a c.p. propeller particularly bearing in mind the limitation to blade area ratio if the propeller is to be fully reversing. The possibility of the ease of control in the case of an unmanned engine room and the possible introduction of the multi-engined installation will still provide a very good case for the use of the c.p. propeller.

At the high speed of such ships, ducts are not quite so attractive and it does not appear that very significant gains can be achieved on such vessels. So far there has been no serious attempt to employ ducts on the fast cargo ship.

PROPELLER PERFORMANCE WITH INCREASING SIZE

In dealing with vessels of very large power it would be difficult to avoid making some remarks on the influence of cavitation on performance and also of propeller excitation particularly on the single screw vessel. Some remarks on these are now given.

Cavitation

The cavitation tunnel is now increasingly used as an aid to design and with the possible exception of the U.K., cavitation tunnel testing is part of the normal routine design process. This is primarily due to the increasing power on a single propeller and also to the problems associated with the non-uniform wake existing behind the very full bodied single screw ship.

Initially, it was believed that because of the varying wake stream on the larger tanker, it would be impossible to avoid the effects of cavitation and consequent erosion of the blade surfaces. However, so far, the ship propeller has shown little effect from the increase in loading and although some slight blade tip back erosion has been suffered and in certain cases slight face leading edge erosion, the performance of the single screw propeller fitted to tankers has not altered materially from the 8000 shp vessel in the early 1950s to the 25 000-30 000 hp propellers made in the last five years.

Unfortunately there is no clear correlation between model and full scale and Figs. 12, 13 and 14 show the model and full scale performance of a tanker propeller working in the wake field shown in Fig. 15 (24 000 shp at 108 rev/min). It

FIG. 12

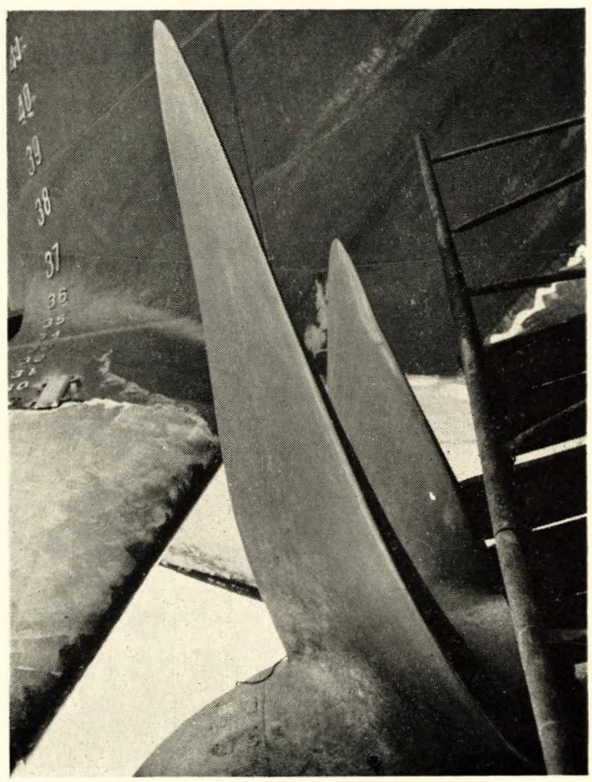


FIG. 13

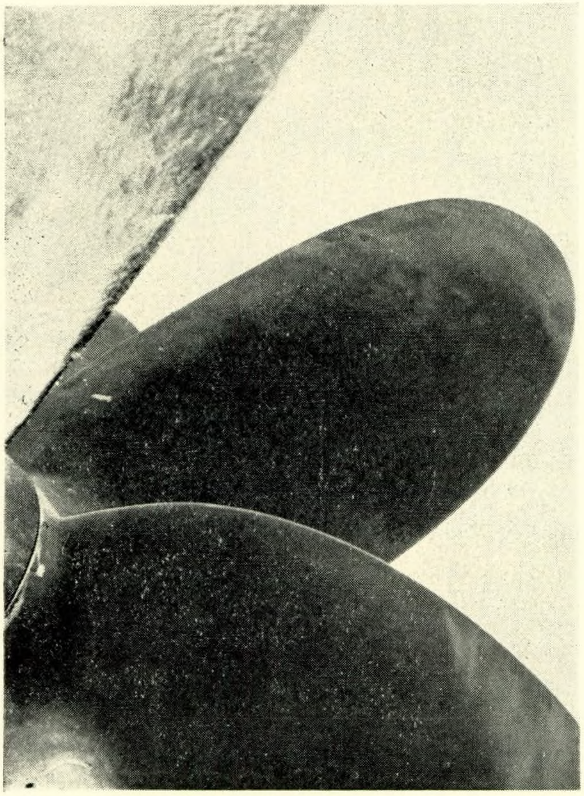


FIG. 14

will be seen from these data that although substantial back cavitation was present on the model, in fact, little evidence of this was shown on full scale although face erosion of a minor order was experienced.

Work is therefore needed to determine the wake on the actual ship, as it would appear extremely likely that the model and ship wake distribution differ substantially as does the mean value. Meanwhile it is the opinion of the authors that cavitation tunnel testing in a simulated wake field on a small model is of qualitative rather than quantitative value and it

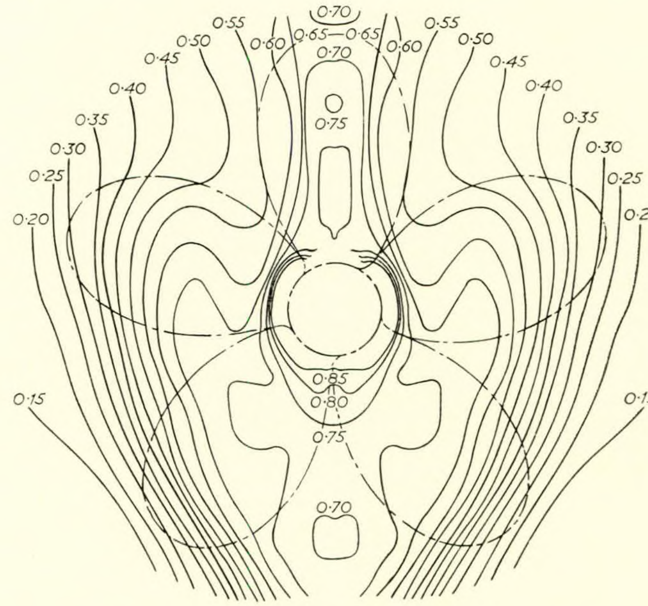


FIG. 16—*Wake distribution—Single screw vessel*

would seem better to use the uniform stream to represent local conditions as this permits easier and more accurate control of water speeds.

Cavitation tunnel tests in a simulated field on a small model sometimes give frightening pictures and it is fortunate that the actual propeller on the ship pays little attention to these. When a reasonable design process is followed, cavitation erosion is not yet a very serious problem using the propeller materials now available and, in the authors' experience, no merchant ship propeller has yet been removed as defective for this reason. On the other hand, damage apart, propellers in bronze and the proprietary alloys now available, can still be expected to last the life of the vessel provided reasonable maintenance and repair facilities are applied from time to time.

Varying Wake Stream and Blade Number

Perhaps the most important obstacle in way of the development of the single screw vessel to higher powers is the con-

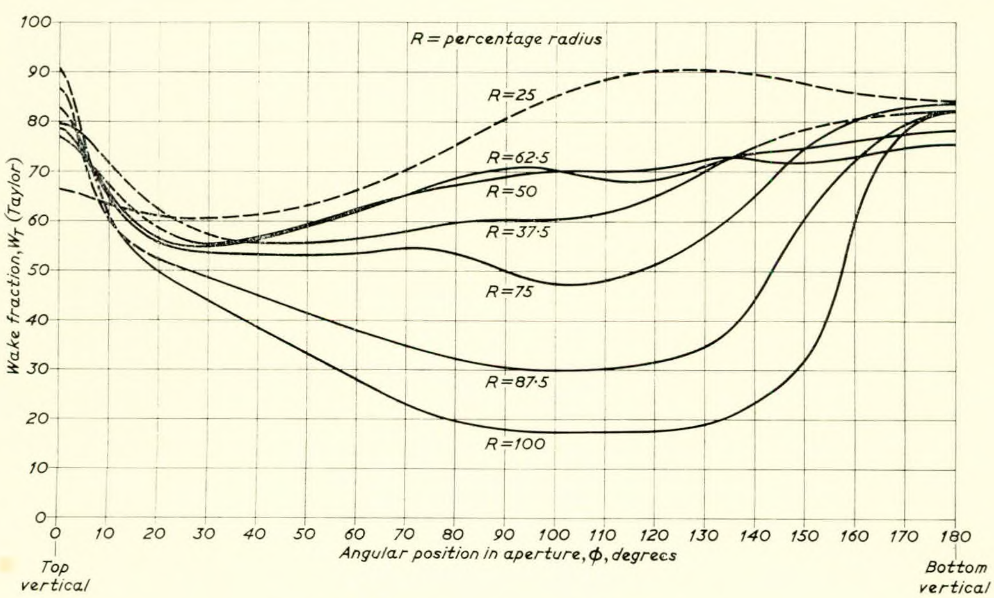


FIG. 15—*Circumferential wake distribution*

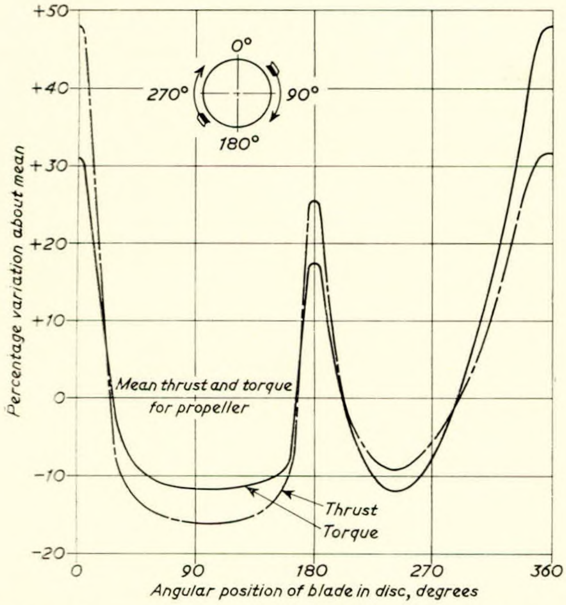


FIG. 17—*Fluctuation of propeller thrust and torque working in a varying wake stream*

siderably varying wake stream in which the propeller works. A wake variation diagram is shown in Fig. 16 for a typical tanker having an open water stern. The considerable changes of velocity involved, produce marked changes of blade section angle of incidence which occur twice per revolution with quite substantial resultant alterations in thrust and torque through the 360° rotation. As a result it is found that:

a) The cavitation problem is considerably more difficult to resolve as compared with a propeller working in a uniform stream.
b) The propeller becomes a considerable source of excitation of the various manifestations of hull and shafting vibration.
c) Very severe cyclic forces are induced in the propeller with the result that the fatigue, or more properly the corrosion fatigue resistance, becomes the most important criterion on which a propeller material must be based. See Fig. 17.
d) Because of this wake variation and the asymmetric nature of the flow, the centre of thrust is invariably markedly off the shaft axis and on a large tanker this can be as much as 12 inches away from the shaft centreline in the starboard upper quadrant of the propeller disc.

In propeller design there is little that can be done about this as one can only design for the mean velocity at each radius of the propeller and this is the practice followed in designing the so-called "wake adapted" propeller. However, to some extent, harmful conditions of resonance can be avoided by changing blade number. The effect of blade number on vibratory forces was well illustrated by Van Manen and Wereldsma[7] to whom thanks are due for the diagrams in Fig. 18 which show the variations of thrust, torque and bending moment with four, five and six bladed propellers. The designer also has some control of clearances, rake and throw-round; factors which have an important but somewhat less marked effect on the smoothness of the thrust and torque variations.

From the point of view of propeller excitation, it would seem that the larger number of blades is the right choice and although in the past systematic series results on standard propellers have shown one to two per cent diminution in efficiency with each additional blade this has not been supported by recent tests on modern propeller designs.

From all the available evidence it appears that increasing the number of blades up to six raises no serious problems with regard to propeller efficiency, cavitation and strength; while the increase in cost is relatively small. There is little doubt that the fluctuating forces and the avoidance of harmful resonances are the most important factors in the choice of blade number.

In general, increasing the number of blades should result in lower levels of vibration and an increase in human comfort.

With regard to vibration however, it must be remembered that this is not only a function of the source of excitation, but is dependent on the dynamic characteristics of the ship, machinery and shafting and therefore blade number is usually outside the control of the propeller designer and must be based on a knowledge of the characteristics of the whole installation. Meanwhile it is clear that efforts should be made to provide by hull design, a more compatible environment for the propeller, permitting a much better propeller design and a more efficient and trouble free installation.

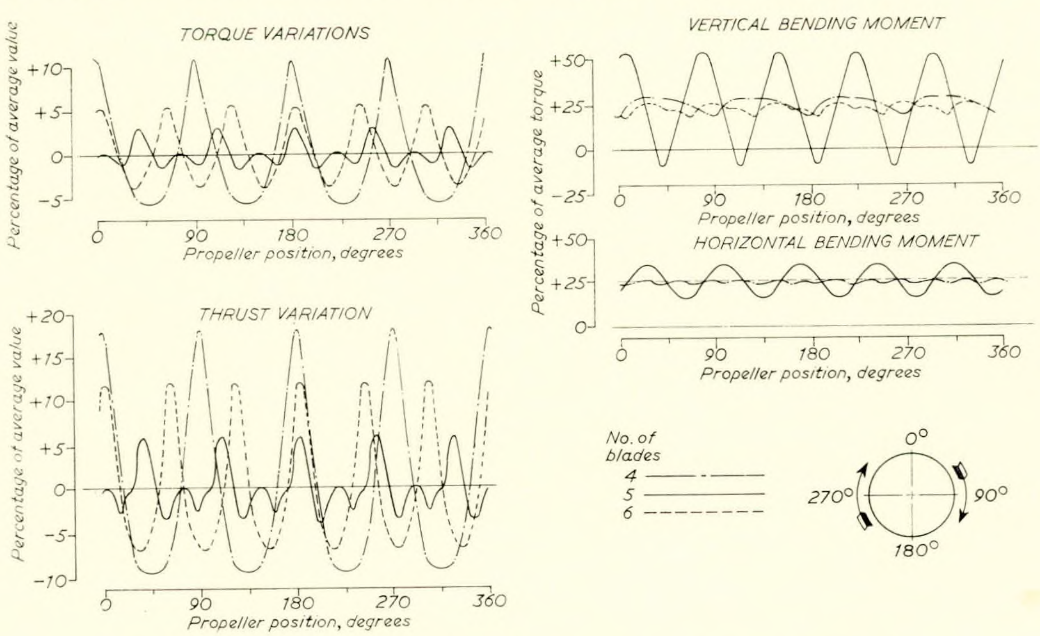


FIG. 18—*Effect of the number of blades on the dynamic performance*

CONCLUSIONS

It is difficult to draw clear conclusions from a paper such as this, but there are a few points which possibly emerge.

1) Twin screws and possibly even triple screws with a use of relatively low rev/min might be successfully applied to the large tanker if power requirements continue to rise.
2) Twin or triple screws must be employed on fast dry cargo ships because of draught limitations if speeds rise higher than those currently used, although contra-rotating propellers may here have a useful field of operation.
3) Serious attention must be paid to hull form, particularly on the single screw vessel, to give more reasonable flow conditions into the propeller.
4) Perhaps insufficient use of the cavitation tunnel is made in the U.K. but nevertheless experience has shown that these data must be firmly related to full scale experience before a great deal of attention is paid to them.

ACKNOWLEDGEMENTS

This paper is published with the permission of Stone Manganese Marine Ltd. and is concerned generally with work from the Company's research programmes. Some of the published information is the product of experimental work carried out on the Company's behalf by the University of Newcastle Cavitational Tunnel and the Vickers, St. Albans, Experimental Tank. The authors would like to express their appreciation of the close co-operation received from these two establishments.

REFERENCES

1) S.M.M. Newsletter No. 11. October 1965. "Large Diameter, Slow-running Propellers".
2) MORGAN, W. B. 1960. "The Design of Contra-rotating Propellers Using Lerb's Theory". *Trans. S.N.A.M.E.*, Vol. 68, p. 6.
3) BURRILL, L. C. 1943. "Developments in Propeller Design and Manufacture for Merchant Ships". *Trans.I.Mar.E.*, Vol. 55, p. 148.
4) PIEN, P. C., and STROM-TEJSEN, J. 1967, May. "A Proposed New Stern Arrangement". Naval Ship Research and Development Centre Report 2410.
5) VAN MANEN, J. D. 1957. "Recent Research on Propellers in Nozzles". *Journal of Ship Research*, Vol. 1, No. 2, p. 13. 1963. "Effect of Radial Load Distribution on the Performance of Shrouded Propellers". *Trans. R.I.N.A.*, Vol. 105, p. 59.
6) GLOVER, E. J. 1967. "Contra-rotating Propellers for High Speed Cargo Vessels: A Theoretical Design Study". *Trans. N.E.C.I.E.S.*, Vol. 83, p. 75.
7) VAN MANEN, J. D., and WERELDSMA, R. 1960. "Propeller Excited Vibratory Forces in the Shaft of a Single Screw Tanker". *International Shipbuilding Progress*, Vol. 7, No. 35, p. 371.

Discussion

MR. C. A. LYSTER, B.Sc. said that Fig. 10 showed the stern arrangement of a container ship fitted with, as far as was known, the first contra-rotating propellers tested in the United Kingdom designed and tested at his company's St. Albans tank, and it was necessary to provide new self-propulsion dynamometers. This equipment was designed and constructed specially for the work described in the paper. As could be seen in Fig. 19, there were two separate dynamometers for measuring independently the torque applied to each of the two propellers, and the thrust developed in consequence by each. The first dynamometer operated on the solid inner shaft which carried the aft propeller of the contra-rotating pair. The second dynamometer, the more unusual one, operated on the tubular outer shaft which carried the forward propeller of the c.r. pair. The drive for this was brought from the motor through gear-boxes which could be of fixed ratio, or infinitely variable.

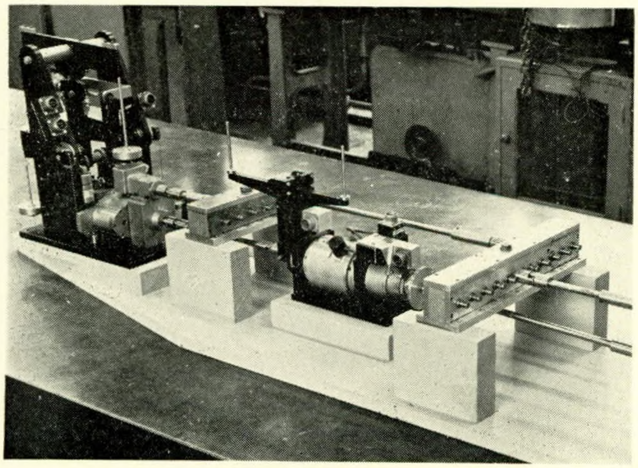


FIG. 19

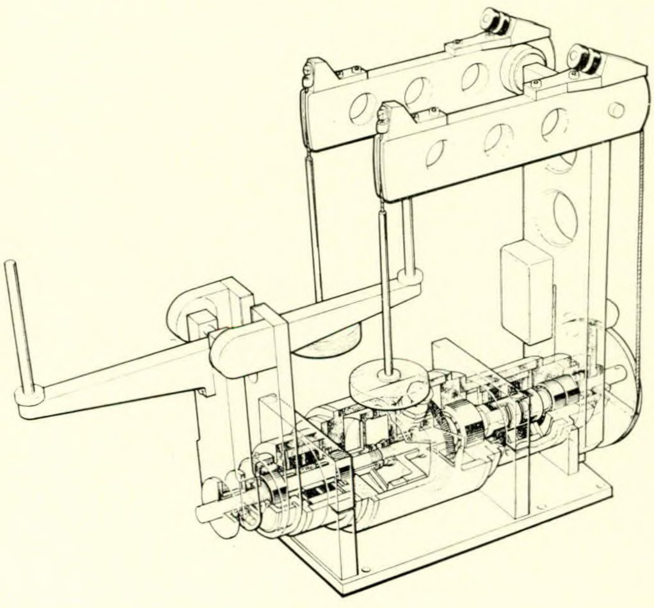


FIG. 20

The direction of rotation can also be selected at will so that the two propellers could be driven at the same or different rates of rotation in the same or opposite directions. Most foreseeable variations of the c.r. or tandem propeller arrangements were thus provided for. Both propellers were powered from the same motor.

Fig. 20 showed the first self-propulsion dynamometer. Torque and thrust were both measured by electrical sensing of the movement of magnetic core pieces fitted to the shaft.

Fig. 21 showed the contra-rotating propeller apparatus.

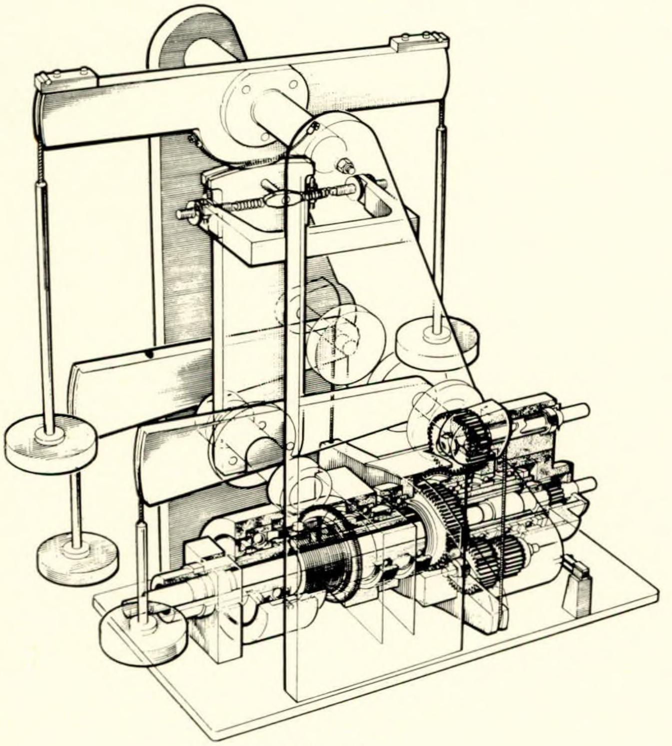
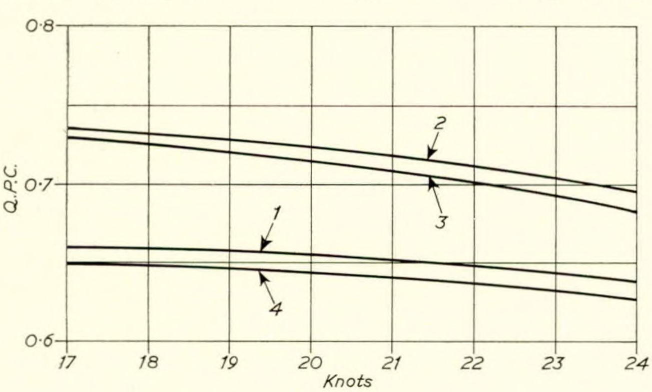


FIG. 21

The drive, necessarily, brought in from off centre and passed through a trunnion-mounted spur differential gear-box where the shaft speed was changed and the resulting reaction force balanced to give a measurement of torque. The thrust force was resisted by a fork and ball-race arrangement and balanced to give a measurement of thrust. Both balance movements were sensed electrically and recorded together with the signals from



1. Normal single screw
2. Contra propellers based on Glover theory
3. Contra propellers based on Lerbs theory
4. Single propeller in contra propeller aperture

FIG. 22

the other apparatus. This apparatus had worked well throughout the tests.

Since the paper was written the results for the container ship with four stern arrangements had become available. The quasi-propulsive coefficient obtained was shown in Fig. 22. The line (1) was for the basic design with a normal single screw. The hull and propeller resulted from extensive tests, and were considered extremely good examples of modern design practice. Lest anybody conclude that this basic Q.P.C. was lower than the standard promulgated by Moor at the North East Coast in 1964, it should be pointed out that this container ship had an extremely light draught and high power, as stated in the paper. This automatically resulted in a lower efficiency than would be obtained with the more orthodox 500 ft cargo ship with which Moor was dealing.

Line (2) was obtained with the first pair of c.r. propellers mentioned in the paper, designed according to the method developed by Glover. The improvement was of the order of $10\frac{1}{2}$ per cent at 17 knots and $8\frac{1}{2}$ per cent at 24 knots. Hydrodynamically, this was a most impressive first result.

Line (3) gave a result for a second contra-rotating propeller based on Lerbs' theory. The results were again far better than the single screw, but worse than Glover's design. The two sets of c.r. propellers were of course fitted into a modified stern since they would have had quite inadequate hull clearances in the normal single screw stern. For a comparison of the propulsive performance behind the hull, unaffected by the actual shape of that hull, the single screw was also fitted in the c.r. propeller aperture.

The results in line (4) were about $1\frac{1}{2}$ per cent worse than the original single screw design, so that it might be concluded that either set of c.r. propellers gave an improvement in performance of the order of 10 per cent, throughout the complete range—a very worthwhile result.

MR. P. W. AYLING said that the paper provided an interesting assessment of the limitations to be placed on the conventional single screw arrangement. But, he was disappointed that the various possible alternatives were not more fully developed.

Referring to vibratory forces, he said that they arose from two main sources: the reactions on the hull and appendages of the fluctuating pressure field developed by the rotating and thrusting blades of the propeller, and the reactions at the shaft supports arising from the fact that the propeller operated in a non-uniform flow. The first of these, commonly known as the pressure or surface force, was not mentioned in the paper.

Work carried out in the U.S.A. on an articulated model of a Series 60, 0·7 block coefficient form, indicated that about 50 per cent of the total vertical excitation and nearly all the horizontal excitation was pressure transmitted. Admittedly, for the full form ship all the evidence suggested that the wake induced or bearing forces predominated. There were some recent indications that the magnitude of the pressure fluctuations increased under cavitation conditions. Could the authors comment?

He questioned the inference that little could be done from the point of view of propeller design to reduce bearing forces, or wake forces, other than by variations in blade number. Suppose a shipowner was sufficiently concerned about propeller excited vibration in a new design to bear the costs of special tests to determine the wake field behind the model; would the propeller designer be in a position to estimate the unsteady propeller forces likely to arise? Did the authors' company possess methods which could, for example, estimate the forces as a function of the angle of skew of the blades and so determine the optimum angle for minimum excitation? He believed that skew might have been neglected in merchant ship design, although not in naval vessels, as a possible means of reducing the magnitude of bearing forces. He would agree that, generally speaking, the higher the number of blades the better from the point of view of propeller excited vibration.

It was doubtful, however, whether harmful hull resonances could be avoided by changing the blade number since, in the main, they were not predictable with the required degree of accuracy. Sooner or later it would be necessary to measure or check the wake of an actual ship for comparison with the model wake. It was difficult to foresee any method of predicting unsteady propeller forces which did not involve a model wake in one form or another. There were, however, many complicating factors in such a comparison, including the effects of separation, particularly on full form ships, and the absence of a propeller in model wake measurements.

Would the authors care to comment on the possibilities of relieving the difficulties of handling massive single screws by adopting a sophisticated design of built-up propeller, possibly involving flanged blades?

DR. J. W. ENGLISH questioned the authors' approach in fixing the speed of the large tanker, probably with conventional screws, at 16 knots, in calculating the power required for propulsion. An alternative and probably more practical approach would be to fix the power and revolutions consistent with present-day engines and developments likely to take place in the foreseeable future. In the case of Diesel engines, this might be relatively simple, but no doubt more complicated with steam turbines due to the larger range of revolutions available with the introduction of triple reduction gear. Nevertheless, some practical constraints must exist in this direction. If this approach were followed the ship's speed would result for a given displacement, and the owner would then decide whether or not this was satisfactory. The propeller weight in which the authors were mainly interested would also follow from the solution.

The authors said that the propulsive coefficients used in the calculations were accurate and practical. On what evidence could they make that statement in view of the fact that very few ships of 200 000 dwt and above were afloat? Was it a reasonable assumption to expect the wake fraction to remain constant in the deadweight range 100 000-500 000 tons?

While the authors' calculations might provide a useful first guide for reviewing the size of propellers required in future, in his opinion they must be viewed with caution when deciding whether to adopt single or multi-screw propulsion because all tankers would not travel at 16 knots.

The authors had proceeded to review alternative methods of propulsion for tankers, and he expressed interest in the possibility of employing ducted propellers as a means of propropulsion. The National Physical Laboratory had had some experience of testing with these devices, and some interesting factors had emerged which indicated power reductions of the order of those mentioned by the authors, 6-7 per cent, and greater, might be obtained for the loaded condition, while even greater gains could be obtained in the ballast condition. Power reductions of 12-18 per cent had been measured in the ballast condition, clearly arising from the interaction between the device and the hull rather than from an improvement in the open-water efficiency of the device.

In an 18 knot ballast condition for a tanker of about 170 000 tons, the hull efficiency in the case of a conventional ship was 1·88; with ducted propulsion it was 2·29, an increase of 22 per cent. The propeller open-water efficiency was 0·396 for a conventional arrangement, and it reduced to 0·380, a reduction of 4 per cent, in the case of the ducted propeller. It was apparent that more work was required on this side of the subject.

Results coming from the N.S.R.D.C. indicated very similar trends in that the improvement came in the hull efficiency rather than the open-water efficiency. Had the authors experienced similar effects when using c.r. propellers on fast cargo vessels? Perhaps the question ought to be directed to Mr. Lyster, whom he would remind that A.R.L. had had c.r. propeller dynamometry for some time.

MR. W. MCCLIMONT, B.Sc. (Member) said that reference was made at a number of points to the problems connected with the increasing weight of propellers. In spite of the statement that "the propeller manufacturer has stated that he is prepared to produce the largest propeller required by the shipbuilder", the observation was made elsewhere that "presumably there is a limit to the size of propeller that can be manufactured, handled and transported". He had tried for some years to clear up the question of the maximum size but was still rather in the dark. What would determine such a limit—manufacturing problems such as maintenance of quality in casting? Was the limit likely to be dictated by road width restrictions or even the width of the doors of the foundry? Or was it the fitting of the propeller and the handling involved. He hoped he would be a little clearer after the discussion as to whether there was a limit and, if so, what set it.

If there was a limit due to any of the three reasons, the use of built-up propellers should enable it to be avoided, and propeller weights could go ever upwards. The comments of the authors would be welcomed.

It would be wise to look at the weight of the propeller that was desirable in operation. Looking at the data in the paper, it was not unreasonable to postulate that the ratio of the propeller weight to the rule diameter of the propeller shaft obeyed effectively the square law. Provided then that the bearing length was maintained proportional to the diameter there should not be an increase in stern bearing trouble. He found it difficult to be so optimistic and would be glad to have the views of the authors. Perhaps, on the other hand, the rather poor record of stern bearings with large propellers was due more to the adverse weight conditions in which large propellers normally operated rather than to sheer propeller weight.

The data so far available suggested that contra-rotating propellers were attractive in respect of efficiency, but the engineering problems should not be underestimated. He had looked in vain for a comparison of the combined weights of the two propellers with that of the single propeller they would replace. He suspected that in terms of the total weight at the stern bearing they were likely to be worse off.

To him the most interesting part of the paper began under the heading "Propeller Performance with Increasing Size". Perhaps the fact that there were 8½ pages on propulsive efficiency and 2½ pages on cavitation and vibration reflected the imbalance in the attention that had been given in the United Kingdom to the real problems of single screw vessels of large power. This comment was not aimed at the authors, who, he felt, believed much as he did. The important thing was that that state of affairs should be rectified, and no better beginning could be made than by pursuing work to determine the wake of the actual ship. There was abundant indirect evidence to support the authors' contention that it was likely that the model and ship wakes differed substantially. The sources of Figs. 15, 16 and 17 would be useful. Were they all calculated or did they derive from model tests?

The very severe cyclic forces which were induced in the propeller might be increased in light ship conditions and almost certainly multiplied in heavy seas. Work on actual ships to determine the severity of the problem was very necessary. Instrumentation presented problems, but it was believed that none of these was insuperable. A great deal of ground work was necessary to define more factually the present operating conditions, before one could give appropriate implementation to the authors' recommendation, with which he agreed, that serious attention must be paid to hull form, particularly on single-screw vessels, to give more reasonable flow conditions into the propeller.

MR. J. F. BUTLER, M.A. (Member of Council) said that he had done a little homework into the material about fast cargo ships, and the economics appeared quite startling.

In the case of the fast cargo or container ship, the fuel consumption for the three types of propeller—single, twin and triple-screw would be 158, 149 and 136 tons/day, representing a saving for the twin screw of £10 800 per year and for the triple screw of £26 500 per year. Capitalizing this, taking interest and amortization at 12½ per cent, one got the equivalent of capital savings of £86 000 and £212 000.

In addition, the lower horsepower with multiscrew installations would represent capital savings of £53 000 and £130 000 respectively, taking a rate of £23 per horsepower to cover the engine, installation and associated plant. Against this the extra cost of propellers and shafting would be of the order of £13 000 and £28 000, leaving net capital savings of £126 000 and £314 000 for the twin and triple screw arrangements compared with the single screw.

Obviously, the hull would cost more for the twin and triple screw arrangements, and possibly there would be rather more spent on propeller maintenance, but it appeared to him that there should be a very considerable financial gain on a large ship as a result of the multi-screw aspects, apart from the advantages of increased safety and reliability.

Taking the figures for the 300 000 ton tanker with twin screws, the horsepower per shaft would be only 27 000 and for the triple screw it would be only 16 000. In the case of a fast cargo ship the corresponding figures would be 19 000 and 11 600. Admittedly to get those efficiencies one might have to bring down shaft speeds below what was practical for direct drive engines. He would like the authors' views on whether the super large engines would become obsolescent within the next few years.

They had been shown a very nice picture of the ideal stern arrangement for a ship. Surely this was only one ideal. The ideal for the ducted arrangements would be to have a big hole in the hull below the water line where one pushed water through slowly. Was this not another possibility?

MR. A. STEEL said that the paper cast a highlight on problems which were as new as tomorrow because the powers talked about were those that they were just moving into. They had been asked to revise long-held ideas about the supremacy of single-screw propulsion because of the growth of power, and it was suggested that twin and triple screw installations should be considered.

He was putting a few simple points forward as a naval architect. The comparison in the paper where twin was shown as being advantageous was associated with a reduction of revolutions from 110 single screw to 70 twin screw. There had been quite a lobby in recent years for reduction of the revolutions to about 70 per minute in the interests of increased propeller efficiency but his impression was that the idea had not caught on much. However, Mr. Sinclair's remarks suggested that forthcoming events were going to disprove that. Mr. Sinclair had his ear closer to the ground so far as new work was concerned. Mr. Steel would like to hear the views of others particularly marine engineers, about such low revolutions. He suspected that reducing to 70 rev/min introduced possible difficulties or expense with gear design, and one speaker had referred to the adoption of triple reduction gear. He would like to know more about the practicality of that.

He agreed with Mr. Sinclair that Fig. 6 showed clearly, particularly at the higher powers and the deadweights up to 500 000 tons, that there was a distinct advantage even on a revolutions for revolutions basis with the triple screw compared with the single screw (both at 70 rev/min). He was struck by the very high power concentrated on the centre screw, about 75 to 80 per cent, for the most favourable triple screw distribution which at the 500 000 ton tanker level produced about 50 000 shp on the centre and a mere 8000 or 9000 on each wing.

From experience on only one triple screw ship, he was left with some suspicion about this mode of propulsion; Mr. Sinclair would probably be aware of the vessel. The propeller problems, so far as he could recall, had never really been satisfactorily resolved. It was difficult to tell whether the power was being adequately distributed on three screws or whether the centre screw was not just acting as a drag. But that had been a long time ago. In the light of this he would like some assurance that there were no inherent difficulties with triple screw arrangements.

In connexion with the very large tanker, he presumed that physical space existed within the hull for satisfactory triple screw or twin screw installations. The dimensions of the hulls were quite considerable when one went up to such high deadweights. Presumably the trend was definitely towards such large ships with some political help on the way such as the closure of the Suez Canal.

In Tables III and IV, columns giving the diameters of the relevant propellers would be useful. He would also like to have seen relevant diameters given for Fig. 6, though he did not see how that could readily be done because it was already a complex and ingenious diagram.

With regard to room inside the hull with multiple screw installations, while that might be available in the tankers he doubted whether the same situation obtained with the second class of ship, particularly the container ship which was attracting so much attention. He referred to the drop in container ships. With that type of vessel cargo cubic capacity of good rectangular form was at a premium. The draughts were low. There was talk about design draughts of 30 ft, but they would be lower than that in operation because besides a tendency to carry fresh air in the ship one carried fresh air in incompletely filled containers. Another feature was high powers and the demand for higher speeds, leading to higher power still.

Because of the large modular nature of the cargo in a container ship, the machinery needed to be right aft. Higher powers meant finer forms and larger machinery, and, therefore, the machinery went creeping forward. The great danger was designing a ship which carried mainly machinery. He and another member present at the meeting had been associated in an exercise to accommodate high power machinery in a twin screw container ship arrangement. Reduction gearing had been the worst offender. Accommodating this bulky item, if it was not to occupy a bulge sticking out from the ship, meant pushing the machinery space well forward.

He had looked at some of the engines-aft arrangements for passenger ships which were generally similar so far as form was concerned, although their speeds were less than those they had been looking into at that time. The forward machinery bulkheads in each case were located about a third of the length of the ships from aft. When one wanted to carry bulky containers, that posed a problem. He envisaged difficulty in attempting to put twin or multi screws on a normal fast container ship if speeds were going up. Therefore, he was pinning his hopes on the contra-rotating propeller and would welcome the release of some interim indications of what the designers would be presented with. For instance, what extra power could be considered as being practical to be transmitted by a c.r. propeller compared with a single screw? How did the optimum diameters compare?

Still thinking of the same type of vessel, at a branch meeting of the Institute during the past year reference was made to fully cavitating screws turning at very high revolutions. The suggestion was made that while they did not give particularly good open-water efficiencies, there was a remarkable and unexplained increase in the behind-hull efficiency. High revving propellers of that nature for large ships might at first sight appear impracticable and he felt that the reference made at the time was not intended to be taken too seriously. However, very high revving screws suggested compactness in both machinery and gearing with the possibility of these being more easily fitted into fine after ends. Perhaps the authors would care to comment on the idea?

DR. F. ØRBECK said that the general view had been held for some time that to minimize propeller excitation, and its effect, one ought to use generous clearances and, in addition, the tendency had been towards more blades on the propellers.

The vibration influenced by the propeller consisted of the torsional and axial vibration of the main engine shafting, the bending vibration of the tail shaft, the lateral vibration of the rudder, and the hull, and local vibration of the after end of the ship, often most troublesome in the superstructure. The torsional and axial vibration of the main engine shafting was excited by the engine as well as the propeller.

It had often been stated that one should avoid having the same number of propeller blades as there were cylinders on the engine. His company had ample evidence that this could be advantageous rather than disadvantageous. They had installed a number of four-cylinder engines with four-bladed propellers, with a net result of a reduction of the 4th order torsional vibration shaft stress of about 15 per cent.*

His second example was an installation with a six-cylinder engine for which he had calculated the effect of fitting a three-bladed propeller (see Fig. 23). Three-bladed propellers

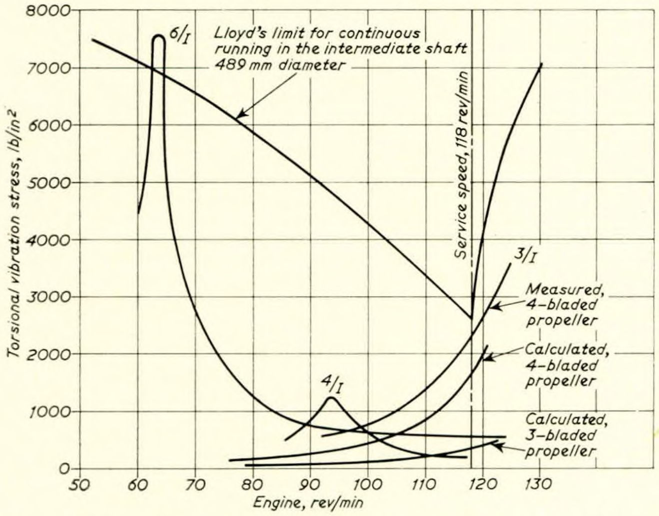


FIG. 23—*Torsional vibration stress versus engine rev/min—Records taken on sea trial*

were viewed with considerable mistrust, which he could well understand. As to the torsional vibration, it was possible with that arrangement to reduce the stress in the intermediate shaft at full speed from about 2500 lb/in² for a four-bladed propeller to about 500 lb/in² provided the propeller was correctly phased. They reckoned they would be able to do this with trial and error, or perhaps a little better than with trial and error. In the tail shaft, where they were really more concerned about the stress, there would be a reduction of about 700-1000 lb/in².

Figs. 17 and 18 were very interesting. The even blade numbers were preferable from the point of view of bending, whereas they gave rather high thrust and torque variations. Both variations could be an advantage rather than a disadvantage when they were used against the engine excitation; in certain circumstances one might even consider raising them.

The odd blade numbers gave large bending moments of the tail shaft, e.g. the five-bladed propeller gave a bending moment without dynamic magnification of about + 30 per cent of the power torque. It was likely that a ship with the six-cylinder engine mentioned and normal shaft diameters would have a bending frequency of the order of 800 vibrations per minute, and at a running speed of 118 rev/min there would be a dynamic amplification of 2·24, giving a bending moment on the shaft ± 67 per cent of the power torque. From Fig. 23 one could assume that the bending moment variation was inversely in proportion to the number of blades for propellers with uneven blade numbers. The three-bladed propeller would have about ± 50 per cent of the power torque as a bending moment vibration. On the other hand, the frequency was far more favourable because the resonance peak would be much further away from the running range and the dynamic amplification would only be 1·22, with the result that the bending moment on the tail shaft was reduced to ± 61 per cent of the power torque.

In a comparison of fitting a three-bladed propeller with fitting a four-bladed propeller on the basis of resultant stress in the tail shaft due to torque as well as bending, his calculations showed that with a four-bladed propeller the maximum shear stress in the tail shaft would be 3955 lb/in², and with a three-bladed propeller is would be 3830 lb/in². As far as the shafting was concerned the three-bladed propeller was the better the two.

There were other factors which had to be considered, for instance, the stern tube bearing.

Some results† had recently been published stating that a three-bladed propeller scored by about 4·5 per cent on propeller efficiency compared with a six-bladed propeller. The present authors stated that the improvement in efficiency obtainable with fewer blades seemed to be negligible. Did the tests referred to extend to cover three-bladed propellers and were they based on the best and most modern design of the three-bladed propeller?

MR. R. NORRBY said that the choice of one or two propellers was based on safe manœuvring, propeller casting capacity, handling of the propellers and propulsive performance. Regarding propulsive efficiency, an investigation was made, at the Swedish State Shipbuilding Experimental Tank*, of single and twin screw arrangements for three vessels. It showed that the single screw solution was the most attractive for a 13 000 dwt cargo liner; for a 70 000 dwt tanker, approximately the same power was needed with single or twin screws; for a 300 000 dwt tanker, the twin screw arrangement offered higher propulsive efficiency, requiring 5-6 per cent lower power than the single screw arrangement.

The authors seemed to use a propeller weight criterion to decide on single, twin or triple screw propulsion. Propeller size could be a problem in the casting and handling of f.p. propellers and in the loading of the stern tube bearing. The casting and handling problems of large c.p. propellers, or f.p. propellers with bolted on blades, could be overcome. The blades were cast and manufactured separately and assembled to the hub on the propeller shaft.

By designing the propeller shaft with a large enough diameter shaft deflexion and loading could be kept within acceptable values to prevent excessive wear of the stern tube bearing. On a bulk carrier of 72 500 dwt with a 43-ton c.p. propeller, the largest bearing wear measured was less than 0·004 in after one and a half years service. The shaft diameter was 28 in and the calculated mean load for a bearing length of 84 in was 63 lb/in². The shaft boring was made horizontal.

The mechanical design problems which could arise with a c.r. propeller shaft system had not yet been solved for merchant ships. The c.r. propeller system had a greater total propeller weight and its centre of gravity was further aft, compared to a conventional arrangement. The two co-axial shafts had a speed of rotation, relative to each other, more than twice as high as for a conventional arrangement. This could give difficulties with seals as well as bearings.

In the U.S.A., the c.r. propeller principle had been used on a submarine with a somewhat simpler mechanical solution for the transmission between machinery and propeller shafts. However, according to his information the U.S. Navy at present considered the c.r. propeller system too complicated mechanically and did not intend using it for surface ships, although it offered high propulsive efficiency for some ship

* Orbeck, F. Recent advances in Computer Calculations of Torsional Vibrations wth examples of some Practical Consequences. North East Coast Institution of Engineers and Shipbuilders, Trans., 1963, Vol. 79.

† O'Brian, T. P. Contra-rotating Propellers for Large Tankers. International Marine Design and Equipment, 1967.

* Williams, A:"Jämförande undersökningar betr. en- och tvapropellerdrift för handelsfartyg" (Comparison between single and twin screw propulsion for merchant ships), SSPA Allm. Rapport, 1967, No. 21.

types. A shipowner in the U.S.A. was planning c.r. propeller propulsion on three cargo barge carriers with 36 000 hp each†. Experience from these vessels would show whether the c.r. propeller system was yet sufficiently developed.

The c.p. propeller was a logical component in the modern ship where the demands on automation and optimum utilization of the machinery were high‡.§ It was now quite common to use c.p. propellers for large powers; for ships with machinery above 1000 hp, built in 1967, an estimated 12-14 per cent were so equipped. His company and licensees had delivered 1361 c.p. propellers for a total power of around 5 200 000 hp, since 1937.

Among the advantages which a c.p. propeller system could offer were¶:

i) safe and fast manœuvring;
ii) optimum utilization of the machinery;
iii) automatic machinery load control;
iv) simple arrangement of shaft-driven generators and pumps.

The authors' statement that the relation between horsepower and rev/min for f.p. propeller on a tanker was independent of draught was too simple.

It was more frequent that rev/min (ballast) differed from rev/min (full load) by up to ± 3 rev/min for the same power.

It was obvious that when the ship's frictional resistance and frictional wake increased, then the advance ratio would decrease so that the propeller could utilize the full power. After some years in service, the diameter or the pitch of the f.p. propeller consequently had to be decreased. With a c.p. propeller, the desired power and propeller speed could always be developed, by introducing a machinery load control system with the c.p. propeller. The mean ship speed could be kept higher than with an f.p. propeller. With a load control system the machinery was protected against overloading by a decrease in propeller pitch.

The influence of hub size on propeller efficiency had been investigated at the Swedish State Shipbuilding Experimental Tank. Free-running tests and self-propulsion tests with a model of a dry cargo ship were carried out. The results were shown in Table VI. When estimating full-scale efficiencies from model tests it should be taken into account that the figures arrived at were slightly on the low side. The negative influence of a larger hub tested in model scale was greater than on the ship because of the lower Reynolds number.

TABLE VI—INFLUENCE OF HUB SIZE ON PROPELLER EFFICIENCY
Design speed 16·5 knots

Hub diameter ratio, per cent:	20	25	30	35	40
propulsive/ η propulsive at 20 per cent:	1·000	0·994	0·988	0·985	0·981

The maximum expanded blade area ratio his company had made for a fully reversible c.p. propeller was 82 per cent. For an f.p. propeller with the same margin against cavitation, the blade area ratio had to be increased by 10-15 per cent, depending on the hub diameters of the propellers compared.

Crash stop times for some ships with c.p. and f.p. propellers were given in Table VII‖. For the sister ships *Nuolja* and *Nikkala,* the stopping time was almost the same, although *Nuolja* was tested in ballast and *Nikkala* in full load. A recent investigation** into the stopping of a single screw 100 000 dwt tanker showed that the c.p. propeller offered the shortest stopping time and head reach both for Diesel and steam turbine machinery. The investigation also covered propellers of conventional f.p. ducted, and c.r. types.

TABLE VII—CRASH STOP TIMES

Ship	dwt	Power, bhp	Initial speed, knots	Stopping time, seconds	Comments
m.s. *Andorra*	12 000	12 000	19·8	183	c.p. propeller, single screw, ballast
m.s. *Azuma*	13 150	15 000	22·0	160	c.p. propeller, single screw, ballast
m.t. *Esso Fawley*	16 700	10 080	17·0	199	c.p. propeller, single screw, ballast
m.s. *Columbia-land*	24 850	11 400	15·8	240	c.p. propeller, single screw, ballast
m.s. *Holtefjell*	35 500	12 600	15·5	366	c.p. propeller, single screw, ballast
m.t. *Sinclair Venezuela*	51 300	2×8400	16·5	286	c.p. propeller, twin screw, ballast
m.s. *Nuolja*	72 500	17 600	17·0	420	c.p. propeller, single screw, ballast
m.s. *Nikkala*	72 500	17 600	16·3	426	c.p. propeller, single screw, full load
tanker	18 000	not published	15·0	534	f.p. propeller, single screw, full load
tanker	33 000	not published	15·0	558	f.p. propeller, single screw, full load
tanker	35 000	not published	16·3	582	f.p. propeller, single screw, full load
tanker	47 000	not published	16·6	560	f.p. propeller, single screw, full load
tanker	48 500	not published	15·8	630	f.p. propeller, single screw, full load
tanker	65 000	not published	17·0	690	f.p. propeller, single screw, full load

The nozzle propeller gave low vibration with good stopping and seagoing performance. With a rudder nozzle, the manœuvrability was good, especially at low speed. No separate rudder was needed. With a nozzle propeller on a tanker, the power needed was up to 7-15 per cent lower than with a conventional propeller, depending on the propeller loading.

A Kort nozzle rudder with his company's c.p. propeller had been fitted in a ship intended for service on the Great Lakes. Data were: 25 000 dwt, 9600 bhp, 119 rev/min, propeller diameter 17·2 ft, 14·7 knots. The nozzle weighed 35 tons. Recently they had received an order for a c.p. propeller and a fixed nozzle for an oceangoing bulk carrier of 25 000 dwt with 11 400 bhp, 120 rev/min, propeller diameter 18 ft, 15·5 knots. A c.p. propeller in the rudder nozzle was easier to handle than an f.p. propeller because the blades could be erected on the hub when in the nozzle. This could be important for high powered tankers and bulk carriers with large propellers.

At present shipowners and shipyards discussing nozzle propellers for large powers only considered fixed nozzles

† "Contra-rotating propellers and 36 000 bhp medium-speed engines proposed for unique U.S. cargo barge ships". *The Motor Ship,* September 1967.

‡ "Periodically unmanned engine room in medium-speed engined French cargo liners", *The Motor Ship,* September 1967.

§ "Controllable-pitch propeller ordered for 130 000 dwt tanker" *The Motor Ship,* March 1967.

¶ Yamashita, I. 1967. "Recent development of marine propulsion machinery", *Japan Shipbuilding and Marine Engineering.*

‖ Shell International Marine Ltd. (MRP/22), London: "Manövereigenschaften von Tankern", mansa, 1967, No. 23.

** Hooft, J. P., and Van Manen, J. D. 1967. "The effect of propeller type on the stopping abilities of large ships". Paper to Royal Institution of Naval Architects.

combined with an ordinary rudder. Soon it would probably not be unusual to equip large ships with nozzle rudders.

The full scale observations made on cavitation patterns confirmed that the model tests in a simulated wake field were fairly realistic. From Figs 12 and 14 it could be seen that the full scale propeller had been just slightly damaged by cavitation in approximately the same region as the model cavitated in the top position. The conclusion could be drawn that the model cavitation shown was not very aggressive. This could help the designer to judge if the next model propeller tested would have harmful cavitation or not. This empirical method was not fully satisfactory, but better means of judging from model tests were still lacking.

Almost every modern propeller cavitated. The cavitation could be more or less harmful to the propeller, depending mainly on the hydrodynamic design but also on the material chosen.

His company delivered c.p. propellers both in stainless steel and in bronze. In 1967 orders were received for 67 per cent of the propellers in stainless steel and the rest mainly in a high manganese copper alloy. Regarding resistance against cavitation damage both materials performed well. The authors wrote about cavitation erosion but it should be mentioned that aggressive cavitation started both corrosion and erosion of the materials. The magnetostriction tests were usually used for investigating resistance of materials against cavitation damage. Some of these tests showed that stainless steel was inferior to bronze. In practice no difference could be seen in these materials on propellers in service. The magnetostriction test did not tell the whole story of the performance in full scale. The reasons were mainly that the implosion forces were affected by scale effects and that the corrosion resistance of the material tested was not properly taken into account.

From propellers in service it was clear that stainless steel had a much better resistance to impingement corrosion than copper alloy. The impingement corrosion was located on the outer area of the propeller blade where the rotational speed exceeded a certain value, characteristic for the blade material. This type of corrosion resulted in loss of material and gave a rough blade surface and therefore a lower propeller efficiency.

MR. G. A. SKELTON, M.B.E. (Member) said there seemed to be a misprint in regard to displacement for vessels (a) and (b) in Table I. Were the block coefficients, a to e, 0·81, 0·82, 0·83, 0·84 and 0·85? If these were correct, the block coefficient could be written in the form:

$$Cb = \frac{Deadweight\ tons \times 17}{LBP \times B \times H} + 0{\cdot}491$$

the constant 0·491 could be varied to suit any class of vessel built. From this type of approximation hydrostatic curves could be arrived at closely in preliminary estimating.

From Table II, analysing the Q.P.C. of a 100 000 dwt ship and assuming:

$$Q.P.C. = \frac{e.h.p. \times 1{\cdot}30 \times 0{\cdot}95}{s.h.p. \times 0{\cdot}98}$$

correlation factor = 0·95
shaft efficiency = 0·98

the Q.P.C. equalled 0·8, 0·744 and 0·7 for 70, 90 and 110 rev/min respectively. This showed that the flexibility of the large Diesel engine could not be fully used because when using 70 rev/min against 110 rev/min there was a decrease in power of 14 per cent, the shipowner paying that much extra in fuel. The medium speed geared installation, with a loss of 2½ per cent in the couplings and 1½ per cent for gearing, would gain ten per cent in its own favour.

Unfortunately, the comparative table of the twin screws had only been shown at the presentation of the paper. This had brought out the limitation of the single screw, when one thought of the safety angle in navigation. The Esso tanker, in difficulties off Northern Ireland, showed that the twin-screw installation should be brought more to the fore. *Mitsitu,* a 400 000 dwt tanker, practically corresponding in dimensions to the vessel referred to in Table I, was designed as a twin-screw vessel, the only difference was in its speed of 15 knots, against 16 knots given by the authors.

In Fig. 4 why had the authors gone back to the old approximate Q.P.C. constant of 0·83? Mr. Emerson in his North East Coast paper had recommended 0·86, which was comparable to Ayre's Q.P.C. as follows:

$$Q.P.C. = x - \frac{\sqrt{D} \times N}{3333}$$

If one substituted $x = 0{\cdot}83$, making Ayre's formula equal to Emerson's, then the ratio of diameter to L.B.P. equal 0·0342 was constant. If one analysed Table I one found the ratio of D/L varied from 0·35 to 0·0238 which meant the 18 000 constant varied from 17 800 to 21 600. Would the authors comment?

Mr. Skelton was interested in power estimates; he liked quick methods sometimes. He had found out that using Doig's method of propeller diameter approximation, the constant was 53·3 throughout the series, therefore:

$$D = \frac{53{\cdot}3\ D.H.P.^{0.2}}{N^{0.6}}$$

Now if the combined Ayre's and Doig's formula:

$$Q.P.C. = 0{\cdot}83 - \frac{\sqrt{\dfrac{53{\cdot}3 \times D.H.P.^{0.2}}{N^{0.6}}} \times N}{3333}$$

one arrived finally at the following equation:

$$Q.P.C. = 0{\cdot}83 - \frac{D.H.P.^{0.1} \times N^{0.7}}{456}$$

It was regretted that the hull efficiency, thrust deduction factor and the relative rotative efficiency were missing from the paper. Would the authors care to state a figure of the grip pressure that was needed on the propeller shaft taper?

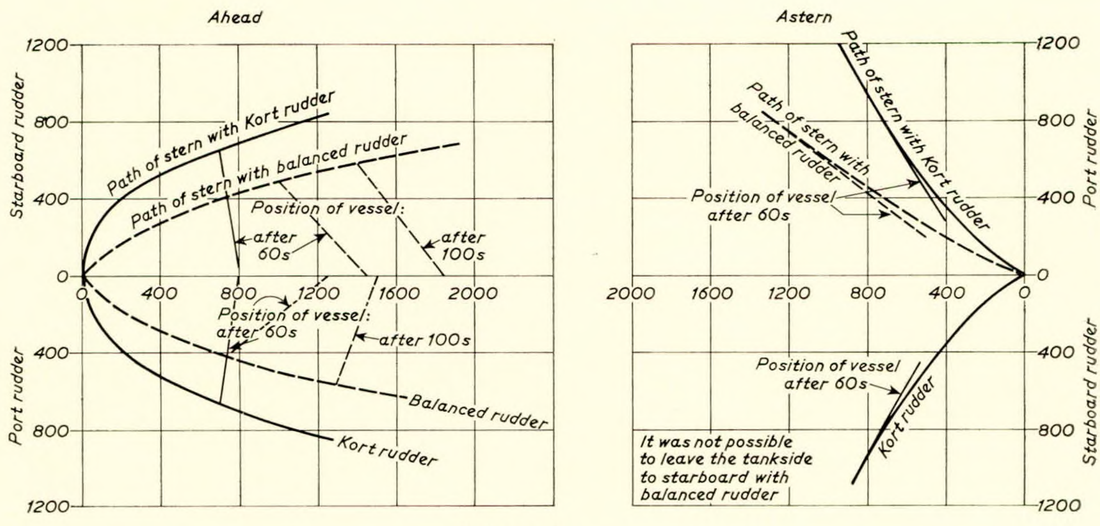
Correspondence

MR. J. N. WOOD (Associate) wrote that the authors had apparently passed over what was probably the most promising form of propulsion for this class of vessel—ducted screw, more popularly known as the Kort nozzle.

Mr. Wood had been associated with the design of this form of propulsion for some years and would offer the following comments.

Van Manen had shown that with propeller loadings of $Bp = 30$ or more, the use of nozzle offered worthwhile gains. These were dependent upon the Bp value and one experiment with a fixed type shroud showed a direct gain in efficiency of 11 per cent at a Bp of 53·2. It was probable that the use of the rudder type would show an even bigger improvement. It should be noted that while propeller loading was the major factor determining the probable efficiency increase, the effect of the hull factors could not be ignored. The effect of the hull factors, so far as the limited test data in his possession were concerned, was capricious to say the least. It would appear from this information that ordinary open water tests could not be used direct, but both propeller and nozzle portions must be modified for the behind condition.

The well known hydrodynamic phenomenon of an



Scales in ft
Helm angle 2 × 45°

FIG. 24—*Comparison between model fitted with Kort rudder and same model with balanced rudder*

accelerating flow "ironing out" irregularities had the effect in a nozzle application in that wake variations in the nozzle mouth of up to 80 per cent were reduced to less than five per cent at the plane of the propeller. Thus a screw designed to operate inside a nozzle could be designed for almost ideal wake variation, i.e. in a radial direction only. Providing that this was observed, the fitting of a shroud to a suitable ship could have the clear advantage of minimizing the problems of propeller excited vibration. The "large unbalanced hydrodynamic forces" mentioned in the opening paragraph cancelled each other and the smoother wake would also lessen the cyclic stress variations in the propeller blades.

The structural problems were obviously the most difficult in this application and the authors were wise to mention them. However, his own principals had recently supplied a nozzle to a bulk carrier in Canada. The vessel concerned was of 25 000 dwt and the propeller which was of 17 ft 3 in diameter absorbed 9600 bhp. The propeller was of the c.p. type. The nozzle weighed some 35 tons and yielded during a tank test an increase of efficiency of 12·7 per cent. It would be seen from this that these structural problems had, in fact, been largely overcome and he would have no fear in preparing designs based upon experience—for nozzles of over 20 ft bore. Fig. 24 showed a comparison between a Kort rudder propellered bulk carrier model and the same model without nozzle.

Authors' Reply

In reply to the discussion the authors said that it was originally intended that the paper should be one dealing with large single screw ships and in particular with slow running propellers for tankers. However, as power increased, it was necessary to consider alternatives to the single propeller and in its final form the paper represented an exercise carried out by the authors and their associates on the problems of high powers, the possible solutions and the particular investigations needed to meet requests for information relating to various alternative systems. With limited resources they had had to be selective, so that in this general paper most of the problems and solutions were merely mentioned and only one or two were dealt with in some detail.

Having reconciled themselves to the situation that if they wished to have experimental information on tandem and contra-rotating propellers in this country it would be necessary to provide suitable apparatus, a contract for this was, as mentioned in the paper, awarded to Vickers', St. Albans tank. The authors wished to express their pleasure at the close co-operation which ensued and the contribution of Mr. Lyster regarding the design of the equipment formed an essential part of the paper.

The ten per cent improvement over a very carefully designed single screw result was very significant. It was thought that there would be a number of further small gains when advantage was taken of the reduced cavitation and improved flow to reduce the blade area, particularly of the after propeller, by detailed change in design and perhaps by altering the after end lines to suit the greater length of a contra-pair.

The authors were happy to learn that Mr. Ayling was still interested in propeller excited disturbances and regretted that he was no longer fully occupied in answering questions instead of asking them. In a paper concerned with the use of increasing powers on full ships it was necessary to state that the pressure disturbances and unbalanced forces constituted a serious problem, but they had not attempted to provide a solution from the propeller end. Model wake data were relatively sparse and possibly insufficiently reliable and unrepresentative of full scale to permit anything but very broad conclusions to be drawn, even on the subject of blade number. In response to Mr. Ayling's specific question the authors would say that, bearing the foregoing in mind and the further limitations imposed by incomplete knowledge of the problem, computer programmes were available which gave reasonably qualitative values of the thrust and torque variations through 360° rotation of the propeller. The use of pronounced "skew-back" to give a smoother entry into what might be a sharply varying wake field was now common practice on both naval and merchant ship propellers. In applying this treatment, however, care must be exercised to avoid the risk of singing, a manifestation of blade vibration always associated with heavy "skew-back" or throw-round.

A built-up propeller had the initial disadvantages of greater weight and less efficiency. Apart from ease of transport there did not seem to be sufficient compensatory advantages because the individual blades on a large propeller were, in any case, quite heavy and awkward to handle. This was a matter for decision and investigation in any particular case.

Dr. English's practical approach, making the owners' requirements for deadweight and speed fit the particular engine available did not commend itself for the purpose of the paper. Although in practice a propeller must be designed to fulfil specified requirements of speed, power and rev/min it was much better for a study of this kind to use a less inhibited approach, thus attempting to meet the requirements of the future rather than the past.

With reference to Dr. English's remarks regarding the propulsive coefficient of very large ships, while it was agreed that little full scale data were available, information on vessels up to 200 000 tons dwt did not lead one to expect important changes in the elements of hull efficiency on larger vessels whether single, twin or triple screws. If, however, the reasonable estimates made for these were incorrect, this would not necessarily invalidate the qualitative nature of the published information. With regard to the question of speed, it was a fact that statistics showed that 16 knots had been used, with small deviation, with increasing ship size. Small changes in speed of up to one knot would not seriously affect any general conclusion which might be drawn from the paper.

In reply to Mr. McClimont, the limiting size of propellers was more a practical rather than a theoretical question. At present provision was being made for manufacturing propellers of 36 ft diameter and 80 tons finished weight. Foundry facilities and melting capacity could be increased and thus propeller size could continue to increase, providing adequate transport and handling arrangements could be made. The propeller manufacturers had been keenly interested in methods of attaching the propeller to its shaft and would agree that the conventional bearing had its limitations for these very large propellers. A number of methods of attaching propellers to shafts had been proposed and the authors included here a diagram (Fig. 25) showing what, in their opinion, was a good approach to the

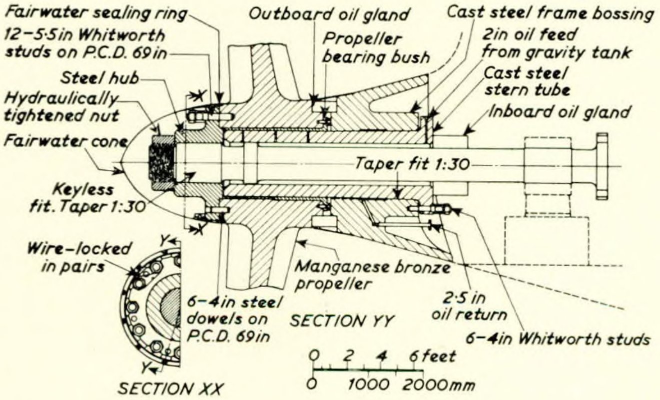


FIG. 25—*Proposed arrangement of Stone-Milton stern gear (Patent No. 34 984/65)—30 000 shp at 80 rev/min*

problem, which was based on the idea of separating the bending moment on the shaft from the torque. By efforts of this kind it was possible to ease the effect of the steadily increasing propeller weight.

The questions of built-up propellers and propeller disturbing forces had already been discussed in the reply to Mr. Ayling. There was little doubt that bad flow conditions did not always adversely affect model resistance and propulsion results and so in the crucial design stage the possibility of improving the vibration conditions was frequently missed.

The opportunity to speculate on alternative propeller arrangements was a refreshing change from the limitations of designing the best propeller to suit a fixed hull, stern arrangement and engine. The authors were grateful to Mr. Butler for translating their results in pounds, shillings and pence. There was some indication that the cross-over in efficiency from single to twin, the cross-over on ship costs and possibly the demands of manœuvring and safety occurred in the same general area. Up to this cross-over region it was possible to supply the large single engine and the large single propeller. Whether the demand would then be for two large engines in a very large, twin screw tanker the authors were probably less able to predict than Mr. Butler. A duct tended to smooth the flow into the propeller by accelerating the slower moving streams. There seemed to be a good case for a radical redesign of the stern, but there was rarely time to provide this sort of exercise when a particular ship was designed.

Mr. Steel discussed the propeller results in the frame-work of general ship design and this must add greatly to the value of the paper. There certainly appeared to have been a change in ideas on gearing so that very low revolutions with turbine and geared reduction on Diesel engines were now considered to be practical propositions. The propeller diameters corresponding to Tables III and IV and to Fig. 6 could be read off the curves in Fig. 5. Thus for the single screw at 110 rev/min in Table III the diameter of the propellers for 60, 70, 80, 90 and 100 tons were respectively 27·5, 28·5, 29·5, 30·25 and 31·0 feet.

The authors agreed entirely with the criticism that, in any particular case, it would be necessary to consider the complete single screw ship design and the complete twin screw design and then make an overall comparison. In the case of the tanker the changes were less extensive and so the simple comparison was reasonably correct. For the container ship, for the reasons given by Mr. Steel, there was not a simple comparison and the use of contra-propellers on a single shaft offered increasing advantages as the speed was increased and the power rose from 30 000 to 40 000 hp.

As a sighting shot, the contra-propeller diameter might be obtained from propeller $B_P - \delta$ charts using half the total power. The fully cavitating propeller does not seem to be a likely solution for either the tanker or the container ship as there would be difficulty in ensuring full cavitation, particularly at slow speeds and manœuvring speeds and a considerable loss in efficiency to off-set against the reduction in thrust deduction fraction.

In considering vibration and disturbances of the stern of the vessel, the authors had hitherto believed that only the propeller was normal and balanced and was driven by a fluctuating engine in the fluctuating flow behind the hull. Improved conditions could only be achieved, therefore, by a propeller with no inertia and instantaneous and automatic pitch adjustment to constant load. Dr. Ørbeck's comments suggested an alternative solution which in the limit, meant arranging the engine fluctuations to match the wake fluctuations. His figures showed that even a comparatively crude matching made a substantial improvement. Perhaps a limited freedom in twist for each propeller blade would allow even closer matching of characteristics. With smaller numbers of blades and increased diameter there would be an increase in efficiency at light loading. This increase in efficiency tended to disappear if there was a restriction on diameter and if the propeller was heavily loaded. For the large tanker propeller the difference was small enough to be outweighed by the importance of blade frequency vibrations and possible critical responses in the shaft or engine.

Mr. Norrby had given a full account of the merits of controllable pitch propellers. The authors also advocated the use of controllable pitch propellers in suitable cases and believed that for a number of very valid reasons the demand for these propellers would continue to grow. In the context of this paper,

TABLE VIII

Ship		a	b	c	d	e
Deadweight		100 000	200 000	300 000	400 000	500 000
70 rev/min	Shp	25 222	40 872	54 470	66 983	78 702
	diameter, ft	25·5	28·6	30·6	32·0	33·2
	weight, tons	22·5	37·5	50·5	62·0	72·0
90 rev/min	Shp	26 599	43 830	59 025	72 630	85 922
	diameter, ft	22·5	25·2	27·0	28·1	29·0
	weight, tons	18·5	31·0	41·5	51·0	60·0
110 rev/min	Shp	28 145	46 906	63 662	78 456	92 734
	diameter, ft	20·4	22·7	24·2	25·2	25·8
	weight, tons	15·5	26·5	36·0	41·5	52·0

however, the question of c.p. propellers only arose in connexion with the flexible use of a number of smaller engines. Commenting very briefly on Mr. Norrby's discussion, controllable pitch propellers were heavier, less efficient and very much more expensive than fixed pitch propellers. For the normal cargo ship a fixed pitch screw could be readily designed to perform very satisfactorily over the changes of slip which occurred due to draught changes and as the ship aged. This was particularly the case in the large tanker where the influence of draught had a negligible effect on the real slip and thus the optimum propeller for the full displacement was the same as that for the ballast condition. To compensate for its initial disadvantages, the controllable pitch propeller could apply greater stopping power in the first vital minute and if the ship operated in congested waterways this quality might well outweigh all the disadvantages.

The authors were grateful to Mr. Skelton for pointing out a misprint in Table I. Displacement (a) should have read 123250 and Displacement (b) should have read 243540; this had now been corrected. He was also correct in deducing the block coefficients used in the exercise. At the time this seemed a reasonable progression from current trends in hull dimensions but more recently block coefficients seemed to have been moderated for the larger vessels in the interests of improved flow at the aft end of the ship. It would, therefore, be unwise to base any general formula on this progression.

The comparison of the pros and cons of using lower rev/min was just the sort of use for which the authors hoped the paper would be employed. The twin screws table (Table VIII) comparable with Table II was included here as requested. The authors agree that twin screws offer additional advantages for better manœuvrability and increased safety.

Mr. Skelton's combined formula for Q.P.C. was interesting, this type of formula was chiefly useful for initial estimates and as a comparitor. The particular form of Emerson's formula had no special significance but happened to be the form used for comparison by the authors' firm for many years and in this case gave a simple measure of the effect of rev/min on Q.P.C.

The components of hull efficiency used in the paper were as follows:

TABLE IX

	Tankers		Fast cargo ships	
	Central propellers	Wing propellers	Central propellers	Wing propellers
Taylor wake	0·45	0·20	0·28	0·15
Thrust deduction fraction	0·18	0·18	0·18	0·14

The relative rotative efficiency was in all cases taken as 1·0.

It was difficult to quote a grip stress necessary when fitting the bore of the propeller to the shaft as this varied with the length of the boss and position of the lightening chamber. There were also differing opinions on what was desirable and what was possible. A general figure for axial pull-up was 0·005 in per inch of shaft diameter, provided that a key was fitted.

The authors were pleased to have Mr. Wood's contribution and agreed with much of what he said. There was a good case for the ducted propeller at high loading or high B_p values but in the authors' view, the first logical step in design was to strive for low loading by reducing rev/min and using the maximum propeller diameter consistent with the draught and beam of the vessel. In addition, at this stage, as Dr. English had pointed out, more work was needed to clarify the effect of the interaction of the propeller and nozzle with the hull.

The authors agreed that the ducted propeller offered a possible way of smoothing the fluctuation in flow due to the ship wake, with possible advantages in vibration and propeller design generally. However, the authors felt that the structural problems of fitting a nozzle round a very large propeller with minimal clearance on a vessel that was subjected to pitching, heaving and other hazards at sea, would require very careful consideration on the part of the designer and the ship operator.

Marine Engineering and Shipbuilding Abstracts

No. 5, May 1968

New Russian High Speed Craft

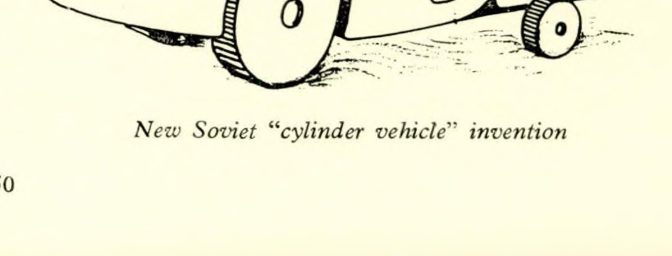
A Soviet engineer named Viktor Podorvanov has just developed a new kind of vehicle for amphibious use, applying the well-known Magnus effect which has been tried out for many years past in various media by numerous designers but hitherto without success. Viktor Podorvanov has achieved success in conditions as yet unexplored, in the boundary area between water and the atmosphere. In the course of a series of consecutive experiments with partly submerged revolving cylinders, he was able to determine the optimal parameters of the following for his purpose: the width and diameter of the cylinder, the circumferential speed at various speeds of the oncoming water flux and, what was the main unknown quantity, the change of lift.

When carrying out these experiments, he discovered, incidentally, that his unusual application of the Magnus effect, with cylinders partially submerged in the water flux, revealed a major advantage of the system: the cylindrical wheels not only do not create a bow-wave, which consumes energy by sucking water in beneath the hull, but they create both additional thrust and a vertical lift. He found, moreover, that this vertical lift eventually becomes sufficient to lift the hull completely clear of the water, thus reducing its hydrodynamic resistance to zero.

This effect, though outwardly similar to that of hydrofoils, proved to be very much more economical, for the energy required to revolve the cylinders is far less than that required for the propulsion of a comparable hydrofoil vessel and it seems probable that Viktor Podorvanov's "cylinder vehicles" will be the most economical and also the fastest.

Further interesting phenomena observed by the inventor were that at a speed exceeding 100 km/h (about 60 knots) the wheels barely touch the surface of the water and that, as the speed is increased beyond that, the aerodynamic lift, rather than the hydrodynamic lift, becomes the main force for the cylinders to provide, for the vessel then acquires a swift and smooth bouncing motion, as the result of which the cylinders themselves rise clear of the water for fairly long periods and the vessel becomes airborne. This phenomenon must be taken into account in the designing of the hull and superstructure which must tend towards an aircraft shaping and in the provision of a vertical stabilizer and an air rudder aft. Podorvanov has emphasized that he is not aiming at the development of a fundamentally new type of hydroplane or seaplane capable of taking off from water by means of its cylinders and then flying over land or water until it eventually comes down again on the water. His object is simply to develop an efficient high speed vehicle and he claims to have accomplished this with success. It reaches its maximum efficiency, from the point of view of economy, at a speed of 120 km/h (about 75 knots) but can very easily be made to achieve speeds twice, or even thrice as great as that, without final transition to flight by changing the cylinder speed within certain limits.

A very obvious major advantage of this "cylinder vehicle" is that it receives its thrust from cylinders working in the water, whereas the resistance to it is only that of the air. It has an advantage over a hydrofoil vessel, moreover, in that the foils of the latter are susceptible to damage by floating logs etc or submerged rocks, especially at high speeds, whereas



New Soviet "cylinder vehicle" invention the "cylinder vehicle" does not come into contact with such obstructions when it is moving at high speed and at low speed it simply rolls over them.—*Hovering Craft and Hydrofoil, July 1967, Vol. 6, pp. 6-8.*

Experience with Alclad Aluminium in Deep Sea Buoyancy Sphere

Alclad aluminium alloys are commonly used to minimize the likelihood of perforation by corrosion in aggressive environments. They consist of a relatively thick core of one alloy and relatively thin layers of a second alloy metallurgically bonded to one or both surfaces. A cladding alloy is selected to provide cathodic protection to the core alloy. As a result, any corrosion that takes place penetrates only to the cladding-core interface. It then spreads out laterally with substanially complete consumption of the cladding as it spreads. Occasionally, however, a peculiar type of corrosion has been observed in Alclad products consisting of an AlZnMgCu alloy core and AlZn alloy cladding where the lateral attack of the cladding is confined to a narrow region near the cladding core interface.

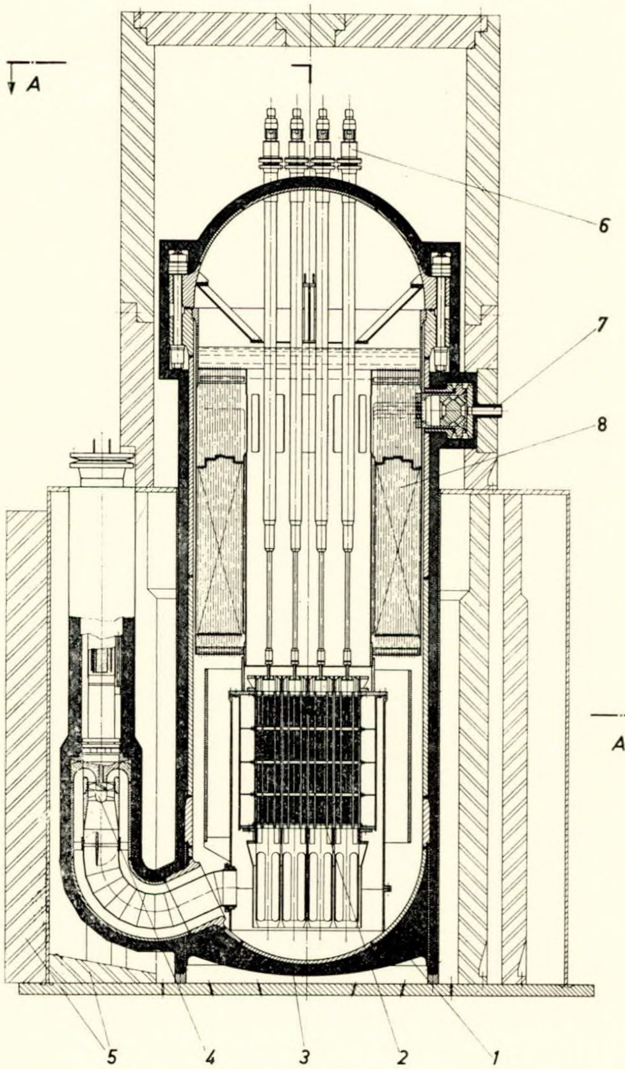
An exaggerated instance of this type of corrosion on an aluminium buoyancy sphere provided an excellent opportunity to study this phenomenon. In this instance, the corrosion was easily detected because it was severe enough to result in blistering.

The sphere, which was three feet in diameter and constructed from ¾ in thick Alclad (7072) 7178-T6 plate, was exposed in the Pacific Ocean for 6½ months at a depth of 300 feet off the coast at Port Hueneme, California, by the U.S. Naval Civil Engineering Laboratory as part of its hydrospace programme.

The selective attack suggests the presence of a narrow region more anodic than the metal on either side. The presence of such a zone was established several years ago in unpublished work on Alclad (7072) 7178 alloy sheet. The most plausible explanation is based on the fact that zinc in solid solution makes aluminium more anodic while copper makes it more cathodic; and on the assumption that diffusion could lead to a zone with a high proportion of zinc to copper.—*Wei, M. W., Corrosion, September 1967, Vol. 23, pp. 261-263.*

German Nuclear Powered Research Vessel

Otto Hahn is in her final stages of testing at the yard of her builders, Kieler Howaldtswerke and it is expected that this large (564-ft long) ore carrier will be at sea by summer 1968. Primarily designed as a research ship that will test the behaviour of the nuclear reactor, she will be carrying bulk ore primarily because of its non-hazardous characteristics and rapid handling. Speed of the ship and power of the plant are not critical. Thus she is designed to make 15¾ knots on 10 000 shp.



(1) Pressure vessel
(2) Core
(3) Insulation
(4) Primary transfer pump
(5) Shield
(6) Control rods
(7) Fresh water inlet
(8) Steam generator

Otto Hahn *reactor assembly*

The three compartments that contain machinery are located immediately aft of No. 4 hold. The reactor itself, in the forward compartment, will not only have the usual shielding but will be protected by a collision barrier. Safety assessment shows that only a very few ships at full speed would be able to penetrate the reinforced hull to impair the integrity of the high-tensile steel bulkheads to port and starboard of the reactor. Also, the height of the double bottom (8 ft 3 in) will protect the plant in the event of grounding.

In the compartment aft of the reactor are the conventional H.P. and L.P. steam turbines with double reduction gear. Although the H.P. turbine at full power will be running on steam of a slight degree of superheat (inlet conditions are 456 lb/in^2 abs and 523·4° F), a moisture separator is installed between the H.P. and L.P. units and the L.P. turbine casing is equipped with special drains to avoid erosion damage of the last stages blading due to excessive steam moisture. These turbines and the two auxiliary turbo-generators exhaust into a common condenser. Almost all auxiliaries are duplicated. Two conventional oil-fired boilers serve as main-steam generators in case of reactor shut-down. These are located in their own boiler room aft of the propulsion machinery compartment and will drive *Otto Hahn* at 8·5 knots. Two auxiliary condensers are provided for port operation of the generators.

As in the main turbines, the main condenser must have some special features in a nuclear-powered vessel. Special means are provided to detect any sea water that might leak through the tube sheets. An inner baffle is inserted between the tube sheet, forming two spaces completely separated from the centre section and hotwell. The condensate formed in the two spaces drains to the hotwell but not before being sampled by a salinity detector in each drain. The main condenser is also capable of directly condensing the full flow of live steam through on overflow system or steam dump. This allows it to cope with a very quick decrease in steam demand without any disturbance to the reactor.

The main difference between the *Otto Hahn* reactor and the *Savannah* reactor is that the former's core, steam generator and complete primary circuit are built in to the pressure vessel. This saves weight and space especially as high pressure piping with its large resistance to flow can be eliminated, the output of the reactor circulating pumps can be reduced to a fraction of that formerly required and the external pressurizing system is eliminated.

The core of the Deutsche Babcock and Wilcox-Interatom Reactor is designed for a thermal power of 38 mW with a primary circuit pressure of nearly 900 lb/in^2 abs with a coolant flow of 5·3 × 10^6 lb/h provided by three circulating pumps. The core is comprised of 12 normal square fuel elements and four corner triangular fuel elements. Each square fuel element has four guide paths for the control rod blades. Core diameter is 45·2 in and height 44 in (1150 by 1120 mm).

There is a total of 226 fuel rods in each square element. These are self-supporting, stainless-steel cans with an outside diameter of 0·429 in (11 mm) and are filled with uranium dioxide (UO_2) pellets, each with an outside diameter of 0·4 in (10·2 mm) and helium bonding. The cans are closed at each end with a welded cap and the sintered oxide pellets are held together by a spring under the cap.—*Marine Engineering/Log, October 1967, Vol. 72, pp. 71-72, 127-129.*

Diamond Electro-plating of Bearing Surfaces

The chief properties of diamond which make it ideal for use in bearings are its extreme hardness, high wear resistance, ability to support heavy loads, low coefficient of friction and good thermal conductivity, properties which are independent of size and apply equally whether the diamond is a large single crystal or in the form of tiny particles of diamond grit. Thus there is good reason to suppose that surfaces plated with a uniform layer of diamond particles whose tops have been smoothed and polished would possess excellent wear resistant and anti-friction characteristics.

The mass production of synthetic diamond has meant that this material is now generally available and research has therefore been initiated to investigate the possibility of providing parts with a diamond coated surface in order to extend their working life. This research has proceeded in two directions depending on the method of securing the diamond grit to the surface, the first method being to charge or impregnate and the second to diamond electro-plate.

At the Institute for Synthetic Superhard Materials in Kiev, investigations have begun on the second method and we present results of preliminary tests on the wear resistance of synthetic diamond electro-plated surfaces sliding on steel. Soviet-produced high strength synthetic diamond of 50-63 micron size, designated ASP5, was used in preparing the test specimens.

In preparing wear resistant surfaces, the degree to which the diamond particles are covered must be near complete or complete. It is easier but not always desirable to cover the particles completely, for although each particle must be securely held, total coverage can complicate the final process.

Since the diamond electro-plated part would act as an abrasive tool, further processing is necessary to blunt (facet) the sharp diamond points and provide a flat, even bearing surface. This was performed by sliding one diamond-plated surface against another until flat wear surfaces developed on the diamond particles and these were clearly visible under an MBS-1 stereomicroscope at × 87 magnification.

Testing the wear resistance of synthetic diamond electroplated surfaces in sliding friction with steel was carried out on an MI-1M friction device, according to the conventional roller-partial bearing system.—*Bakul, V. N., Sagarda, A. A., Orap, A. A. and Prudnikov, Y. L., Industrial Diamond Review, December 1967, Vol. 27, pp. 533-535.*

Stabilizer Fin Control System

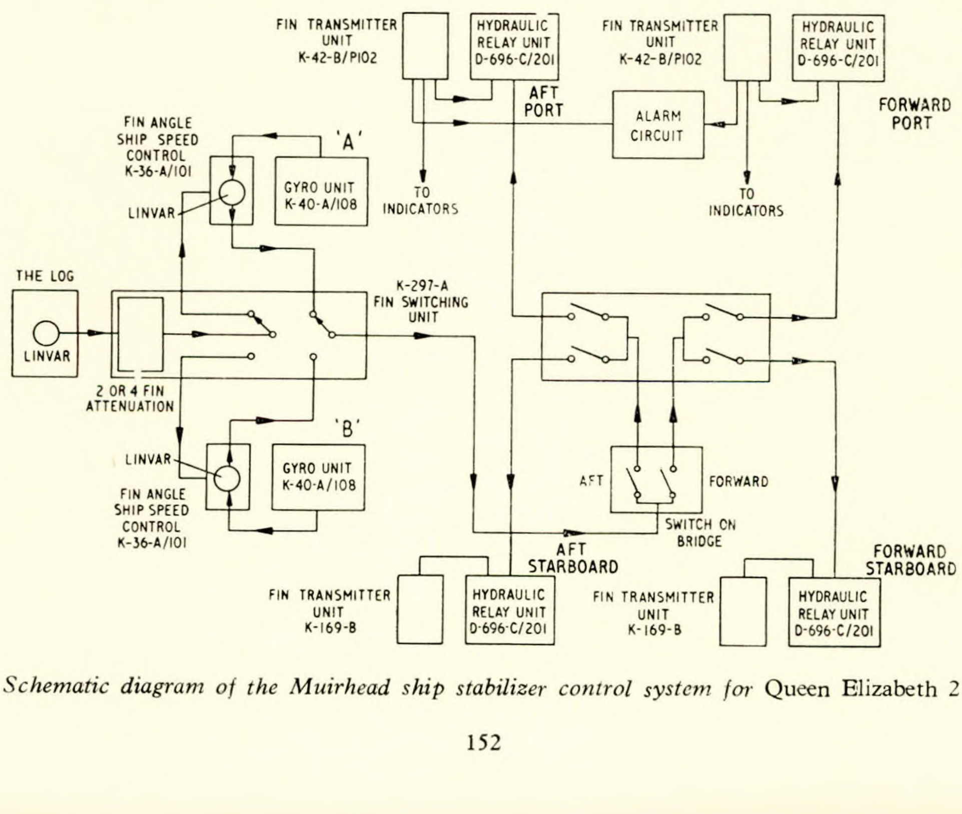
Four Muirhead-controlled Denny-Brown AEG stabilized fins are fitted to *Queen Elizabeth 2,* launched from John Brown's shipyard on Clydebank last September.

The latest features have been incorporated in this installation to ensure smooth travelling and to minimize any reduction in speed when the fins are in use.

Either two or four fins can be used depending on the prevailing conditions. The controls allow for any permanent list and for very slow, normal or rapid rolling of the ship. Folding fins have been incorporated as they take less space inboard than retractable fins when not in use.

The stabilizer fins have been designed and manufactured by Brown Brothers and Co. Ltd. and the electronic control and first-stage hydraulic amplification systems by Muirhead and Co. Ltd.

This is a four-fin installation intended primarily for two- or four-fin operation. Either 'A' or 'B' gyro unit (each with



Schematic diagram of the Muirhead ship stabilizer control system for Queen Elizabeth 2 its associated fin angle ship speed control) is selected on the fin-switching unit (see schematic diagram).

Following normal Multra practice, functions representing $S\phi$, ϕ, $\dot{\phi}$, $\ddot{\phi}$, are developed from the roll angle ϕ and added in proportions determined by the ship's characteristics. Briefly, $S\phi$, takes care of any permanent list in the ship; ϕ controls very slow rolling, $\dot{\phi}$ is the predominant term in normal rolling and $\ddot{\phi}$ becomes significant in more rapid movements.

In the fin angle ship speed control, the 60 Hz signal is passed to a Linvar and attenuated by an amount depending on the ship's speed so that the fins receive a reduced signal at high ship speeds. The Linvar is servo-controlled from a Linvar in the log and a cam with an appropriate law is arranged in the servo gearing so that the degree of reduction is correctly related to the ship speed requirements.

Depending on whether two or four fins are in operation, the log signal is itself attenuated since the fin angles must be smaller if four fins operate instead of two. Should the log signal disappear owing to a fault in the transmission from the log etc, the Linvar in the K-36-A/101 (fin angle ship speed control) automatically resets to the maximum fin angle at maximum speed.

From the K-36-A/101, the signal passes through the bridge control switch so that forward and/or aft fins may be selected and then through individual switches to each fin compartment.

Here, the signal is locked off by a signal from the fin transmitter unit for the individual fin and enters a phase sensitive detector in the D-696-C/201 hydraulic relay unit. The detector output operates a differential relay coupled to the pilot valve and so causes the output arm to follow the signal. The arm controls the hydraulic fin operating gear on the international type folding fin.—*Marine Systems, November/December 1967, Vol. 2, p. 115.*

Computer System on Board *Queen Elizabeth 2*

An advanced computer system, based on a Ferranti Argus 400 data processor is to be installed. It will be the most sophisticated computer system to be used on board a merchant ship.

The computer will have six main functions and is capable of fulfilling other requirements should the need arise.

The first main function is data logging. Automatic scanning of signals from transducers attached to the main engines and other machinery will be performed and a permanent record of engine speed, temperatures, oil pressures and other parameters will be produced on an automatic printer. Most large ships are fitted with data-logging equipment and the original intention for *Queen Elizabeth 2* was to install a data-logging system alone without the provision for other functions. However, after consultation with the National Research Development Corporation, Cunard realized that most of the available data logging systems capable of fulfilling the role required are computer-based anyway. It was therefore decided to purchase a computer which, for an additional £55 000 over and above the £100 000 price of a conventional data-logger, could be used to perform several tasks.

Alarm scanning is the second main function. A continuous check will be made on the main machinery, the computer scanning the signals from detectors and sensors and alarms will indicate when any temperature or pressure deviates from normal limits. A print-out of the alarm conditions will accompany the alarms.

Thirdly, monitoring and control of machinery will be performed. Transducers on the machinery will feed information to the computer which will provide data for display on a remote monitoring panel. This will enable the ship's engineers to adjust the machinery for maximum economy of fuel usage. Continuous automatic control of some of the machinery will also be provided by the computer.

Weather routing is a further task to be undertaken. Information on weather conditions will be fed into the computer on punched tape prepared by operators from received weather forecasts. The computer will then calculate the best speed and course to be taken for minimum fuel consumption without delay to the journey. A 1½ to 2 per cent saving in fuel consumption is expected and represents a saving of about 11 tons of fuel per day. The computed results will enable the captain to choose the best route for maximum economy and greatest comfort to passengers.

Fresh water requirements will be predicted by the computer. This will enable the evaporating plant which produces some 1 200 tons of fresh water from sea water per day to be operated with maximum efficiency and economy.

The Argus 400 computer system utilizes micro-miniature circuitry and the version on *Queen Elizabeth 2* has a 16 000-word memory store, features high-speed operation and can be extended in its use by a wide range of input/output and peripheral devices. If and when the situation requires expansion of the data-processing capabilities, a compatible but faster Argus 500 data processor can be easily substituted for the existing data processor.—*Industrial Electronics, November 1967, Vol. 6, pp. 490-492.*

Marine Radar Simulator

The techniques of flight simulation have now been applied to a marine radar simulator developed as a training aid for ships' officers, pilots and helmsmen. Designed in the U.S.A. by the Link Group and known as CART (Collision Avoidance Radar Trainer), the simulator provides radar interpretation training for all types of normal and emergency conditions from piloting in congested harbours to navigating in dense fog.

A digital computer is used to simulate radar presentations in river, harbour and coastal waters. The displays include movable ship targets which can be controlled by the instructor, navigation buoys which can be positioned by the instructor, selected land targets and 60 miles of coastline. Noise and sea-return signals are also simulated.

Ship manoeuvring controls are incorporated in the equipment as well as own-ship trainee stations so that realistic training conditions are experienced by the users of the system.

In the past, "radar-assisted collisions", arising from misinterpretations of radar displays, have accounted for several accidents at sea. This introduction of simulated training techniques is a step towards eliminating this unnecessary hazard. —*Industrial Electronics, January 1968, Vol. 6. p. 17.*

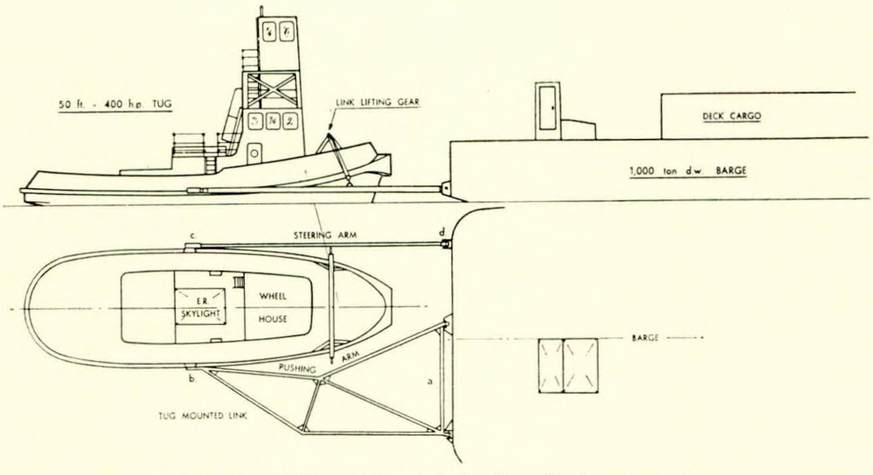
Sea-link

There has been a considerable growth, recently, of seagoing tug and barge operations, particularly on the high-cost United States and Canadian West Coast route and the open-sea San Francisco-Hawaii service.

Push-towing is now in general use on all the major waterways of the world and the advantages of the system, over line towing, in the way of propulsive economies and improvements in manœuvrability are accepted. Until now, however, there has been no satisfactory means of applying push-towing methods in wave heights in excess of 10 ft; after these conditions have been reached, it has normally been necessary to stream the tow to the conventional line position with consequent loss in propulsive efficiency and manœuvrability.

To remedy this situation Sea-link was developed by L. R. Glosten and Associates, naval architects of Seattle, who subsequently entered into partnership with Burness, Corlett and Partners to design Sea Linkages and their associated vessels.

It is the object of Sea-link to allow unrestrained rolling, pitching and heaving on the part of the tug relative to the barge although the barge influence on the tug is beneficial in that it reduces these movements. At the same time if thrust is to be transmitted and control effected, it is clear that the tug and barge must always point in virtually the same direction; they



Diagramatic illustration of the Sea Link arrangement

cannot separate from each other longitudinally and they must move laterally together. Hence relative yaw, surge and sway must be prevented. Briefly, vertical relative motions are allowed but horizontal relative motions are restrained.

The linkage consists of two parts: the main or pushing arm which is attached to the barge by two hinges in line and therefore has transverse rigidity, and the steering arm which is in effect universally jointed to the barge and hence can alter its angle relative to the barge both horizontally and vertically. Both of these arms are joined by universal linkages to the tug at approximately its longitudinal centre of gravity. Hence it will be appreciated that the four junctions—steering arm/barge (a), steering arm/tug (b), pushing arm/tug (c), pushing arm/barge (d)—all allow unrestricted rotation vertically. Three of them unrestricted rotation transversely but one, the pushing arm/barge connexion, only allows vertical movement.

The purpose of the steering arm is to prevent yaw between the tug and the barge and to transmit thrust; the purpose of the pushing arm is to prevent surge and hence to allow thrust to be applied to the barge and to apply transverse or sway forces to the tug. When the sway force is applied to the tug the steering arm, of course, prevents its developing into a rotation. The linkage is arranged, generally speaking, to be carried on the barge in a flotilla which may use more than one tug or on the tug where a flotilla may use more than one barge per tug. The latter is the normal arrangement. The tug anchorage points are on reinforced structure and aproximately at deck level. — *Shipbuilding and Shipping Record, 17th August 1967, Vol. 110, pp. 227-229.*

Semi-submersible Freighter for Container Cargo

K. W. Baldock, of Freighters Ltd, Moorabbin, Australia, has proposed a scheme for the transport of pre-loaded container-carrying barges in a ship incorporating a new type of tubular hull.

In this C.L.A.S.S. (Container-Lighters Aboard Ship System) the ship is designed somewhat on the floating dry dock principle in that it can be partly submerged to provide sufficient depth of water for vessels, in this case barges already loaded, to enter the reception space through a large bow opening. The barges are either pushed directly into their stowage positions or hauled in by means of a "towveyor" system under the carrier-ship's main deck. When all the loaded barges are in place, the bow opening is closed and the water is pumped out to allow the barges to rest in the hold where they are secured by nesting devices. All the barges can be open topped because the C.L.A.S.S. ship's cargo space is completely enclosed and has no hatchways.

The hull of the carrier ship resembles an oil tanker in external shape, but is of tubular structural design with portal or hoop frames and longitudinally-stressed shell plating. The lower half of the hull is of double-wall construction to allow the trim and draught to be adjusted as the loaded barges are taken aboard or discharged. Fuel, water, and supplies will be taken aboard, in the same way as the payload, in special barges. The upper bow portion hinges upwards.

The C.L.A.S.S. ship will load and discharge offshore and widespread use of the system would reduce capital expenditure on port equipment, in addition to reducing idle time of the carrier ship itself.—*Cargo Handling Quarterly, February 1967, Vol. 6, pp. 77-78; Jnl B.S.R.A., July 1967, Vol. 22, Abstract No. 25 487.*

Underkeel Clearance

It is of considerable commercial importance to limit the underkeel clearance required for a vessel operating in depths of water of the same order as the draught to the minimum compatible with safety, so that the allowable draught is as great as possible. For example an extra foot of draught allowed for a 200 000-ton tanker increases its earning capacity by £25 000 per annum.

In still water, when the ship's motion is unaffected by waves, the underkeel clearance required for a given tanker can be calculated with an accuracy sufficient for practical purposes. The factors that have to be taken into account are:

1) the accuracy with which the depth of a given sea area can be established;
2) the accuracy with which tidal height can be predicted for any given time and place;
3) the change in underkeel clearance due to "squat", i.e. changes of trim and water level due to the ship's forward motion;
4) the manœuvres which the ship has to perform;
5) the nature of the bottom.

Squat, which depends on a number of factors, including the degree of confinement of the waters in which the ship is operating, has been the subject of much research, and methods are available for calculating it with satisfactory accuracy. The author gives graphs which show the effects of squat for various tankers in three different types of channel. As a rule ships in confined channels are much less likely to ground owing to squat than ships with small underkeel clearances in relatively unconfined waterways. Large tankers have a tendency to trim by the head when under way and it might therefore be thought that a small stern trim when at rest might counteract this effect of squat but the author considers that even-keel trim is best in practice for the navigation of shoal waters because if the ship does ground, it is better that this should occur forward than aft. In very soft bottom conditions a ship can be navigated even though her forefoot is in the mud.

When all the above considerations have been taken into account, it is the custom of the author's company. Shell International Marine Ltd, to allow a final underkeel clearance of two foot for manœuvring reasons.

Knowledge of the effect of wave action on ship movements in shallow waters is at present insufficient to enable underkeel clearance in waves to be estimated with an accuracy approaching that possible for still-water conditions. Until recently the clearance was fixed largely by rule-of-thumb. The fact that very few cases of grounding due to wave motion have occurred suggests not so much that the allowances are reasonably accurate but that they are over generous for safety reasons.

The amount of the additional underkeel clearance required due to roll, pitch and heave is not equal to the sum of the clearances calculated for each motion separately because the rounding of the hull at the ends makes it unnecessary to add the effects of roll to pitch until a certain value of roll has been reached. The author gives graphs showing the combined effects of pitch and roll on tankers of 50 000 and 115 000 dwt. Moreover, the effect of shallow water on waves and swell reduces ship motion below that experienced in deep water. There is also evidence that in shallow water the ship motion is damped by an amount that increases with decrease in clearance.—*Dickson, A. F. Institute of Navigation, paper read 19th April 1967; Jnl B.S.R.A., November 1967, Vol. 22, Abstract No. 26 878.*

Girodin Axial-piston Drive Mechanism

Disposition of the cylinders of a reciprocating machine around and parallel to the driven or driving shaft has inherent advantages, such as compactness, convenient cylindrical shape of the machine, weight reduction and elimination of the conventional crankshaft. It is only recently that a satisfactory mechanism for efficient transmission of high powers between the pistons and the shaft has been developed. The construction and operation of this mechanism are described in detail. The pistons are linked by dumb-bell-shaped connecting rods, working in ball sockets at each end, to points near the periphery of a "wobbler" element. The inner part of the wobbler is formed as a truncated cone, whose small end carries a bearing bush. The journal of this bearing is the pin of a single oblique crank throw in the shaft. The intersection point of the projected crankpin axis with the shaft axis is the centre of curvature of fixed spherical-segment supporting surfaces which bear against corresponding concave surfaces attached to the inside of the cone. The reaction torque which tends to turn the wobbler is transferred to the engine frame through a shallow-bevel toothed ring on the large end of the wobbler and a corresponding ring on the frame; the rings do not rotate but their point of contact does. The cylinders can work on any of the usual systems (petrol or Diesel, two-stroke or four-stroke etc); an opposed-piston arrangement with a Girodin mechanism at each end has been used in test engines.

It is argued, with supporting considerations and data for specific designs, that this scheme is superior to conventional engine layouts in almost every respect and is attractive when compared with other developments such as gas turbines, Wankel and Stirling engines etc. Size and weight reductions in relation to conventional engines are striking, factors of 2–2·5 being involved. First cost is also considerably reduced, largely because all components are of quite simple form. The number of bearings is reduced and their lubrication conditions are greatly improved; this results in higher mechanical efficiency and less wear and maintenance. Girodin machines run more smoothly and quietly.

At the time of writing, over 100 Girodin air compressors were in service with the French Navy and many more were being manufactured. Several test engines have performed satisfactorily, the largest of these is a turbocharged two-stroke Diesel with a continuous rating of 2000 hp at 1500 rev/min. There seems to be no serious obstacle to the construction of larger, slower-running engines, in fact, the advantages increase with size. Particulars of several projected designs for marine propulsion are tabulated; the largest of these develops 15 000 hp at 265 rev/min. — *Bastide, M. P. Nouveautés Techniques Maritimes, 1966, pp. 147-161; Jnl B.S.R.A., November 1967, Vol. 22, Abstract No. 25 926.*

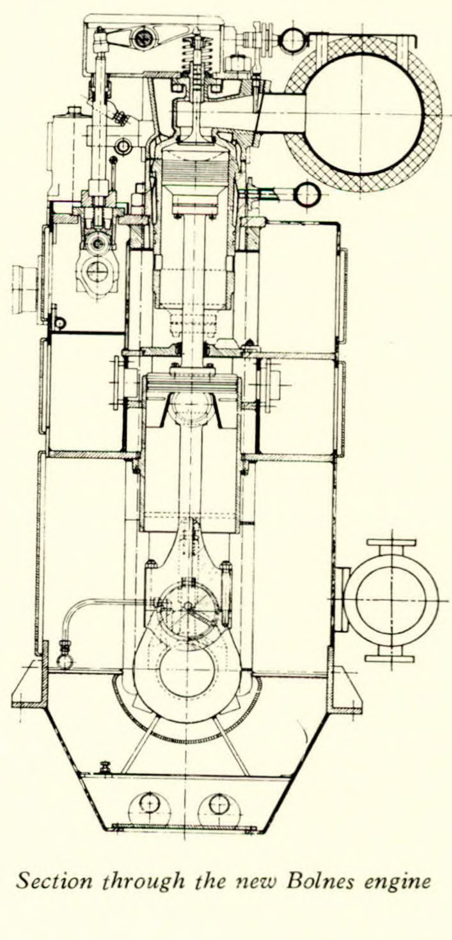
New Bolnes Engine

More than 17 years have elapsed since N.V. Machinefabriek Bolnes, introduced their L type engine, a medium speed two-stroke of crosshead design. Developing 50 bhp/cylinder, these early engines were constructed with up to 10 cylinders in-line but the range was soon to be augmented by vee-form versions up to 20 cylinders. The application of turbocharging followed later.

On display at the recent Europoort Exhibition was a newer, more powerful version of the Bolnes engine, a natural successor to the L type. The characteristic Bolnes features, uniflow scavenging with the crosshead acting as a scavenge pump piston, have of course been retained in this new design. In fact, the same cylinder dimensions, a bore of 190mm and a 350mm stroke, have been adopted but the speed is now 500 rev/min giving a mean piston speed of just under 6m/sec.

The m.e.p., moreover, is now upgraded to about seven and ten kg/cm^2 respectively for normally aspirated and turbocharged versions, increasing the output/cylinder for the two types to 70 and 100 bhp.

The new model is to be constructed in versions having from three to ten cylinders in-line, covering an output bracket from 210 up to 1000 bhp. Turbocharging is on the constant pressure system with the compressor working in series with the scavenge pumps. Besides the advantage of simplified construction, such a system offers an improved efficiency at higher m.e.p, resulting in a lower fuel consumption. With the pumps by themselves being capable of meeting scavenge air requirements at loads of up to 70 per cent, the system also affords a



Section through the new Bolnes engine

degree of security against turbocharger failure. Moreover, flexible operation over the entire range of loads and speeds is assured. Blowers and intercoolers are of Brown, Boveri manufacture.

Effective segregation of combustion chamber and crankcase is ensured by the crosshead/scavenge piston and combustion products are exhausted through a single, centrally arranged valve.—*Shipbuilding and Shipping Record, 14th December 1967, Vol. 110, pp. 838-847.*

Laser Modulation for Communications

The application of the laser to communications is an attractive proposal for several reasons. First, the extreme narrowness and low beam divergence of the laser provides a directional system subject to little spreading and diffraction losses. The same quality affords security and privacy in communications. Due to its extremely high frequency, the laser opens up new areas of the frequency spectrum to our use, which will relieve the already crowded transmission bands. Because of its frequency, the laser would allow many channels of intelligence to be carried on one beam. Theoretically, allowing a 100 kc bandwidth for each transmission, 10^9 channels could be impressed on one laser beam of frequency 10^{14} c/s. This fantastic capability is of course limited by the modulator techniques available, to the extent that the maximum modulation frequencies would be on the order of 20-30 Gc. Still, this would allow 30 000 channels 100 kc wide on one beam.

The effectiveness of the laser system is lessened somewhat by the fact that atmospheric interference diffuses and absorbs the laser beam over long distances, and destroys its coherence. Therefore, long range communications links would be limited to outer space or evacuated tubes. Because of its extremely directional properties, the laser will never replace micro-wave systems for broad coverage radio or television transmission.

The pulsed operated ruby laser is generally inapplicable to communications systems. Thus, this paper is concerned with the gas laser and GaAs semi-conductor laser.—*Amos, D. H. Naval Engineers Jnl, August 1967; Vol. 79, pp. 573-580.*

Multi-purpose Reefer with Maierform SV Bow

Schlichting-werft of Lübeck-Travemünde recently handed over the last in a series of three multi-purpose reefers constructed to the order of Norddeutsche Kühlschiffreederei Harmstorf and Company, Hamburg. The lead ship, *Pagensand,* completed towards the end of last year was soon to be joined by *Fährmannsand. Lühesand,* last of the trio, entered service at the end of October 1967.

In almost every respect identical, the three ships have been designed for economic world-wide trading, loading perishable commodities of any class. Indeed, the design has bestowed upon the vessels a cruising range of 10 000 miles at a speed of 16·5 knots.

With machinery and accommodation arranged all aft, the three-hatch vessels have been constructed in accordance with the rules prescribed by Germanischer Lloyd for the classification ✠ 100 A4E ✠ MC ✠ KAZ. The refrigerating installation was classed by Lloyd's Register.

The hull is transversely framed and features a cruiser stern and bulbous bow. It was stated above that the three vessels are substantially similar. The forefoot of *Lühesand* in fact, marks the only significant variation—it being the first Maierform SV bow to be incorporated into a German-built ship. It is claimed of the Maierform SV bow, first fitted to a cargo vessel built for Robert Bornhofen Reederei, Hamburg, at the Caledon yard, that performance may be enhanced by as much as 0·5-1·2 knots, depending, upon ship type, with propulsive power maintained. In this instance, *Lühesand* registered an improvement in speed over her sister ships of 0·35 knots, machinery output maintained.

Speeds, as open shelter-deckers, are given as 16·85 knots for *Lühesand* and 16·5 knots for the two earlier vessels, the machinery in either case developing 2750 bhp at 275 rev/min. Details of the bulb fitted to the first two vessels have not been disclosed.

The underhung semi-balanced rudder, fitted with an end shield for enhanced manœuvring qualities is actuated by an AEG rotary vane steering gear with two pumping units. The steering engine can be controlled for either follow-up or non-follow-up steering and is also linked with the gyro for automatic control.

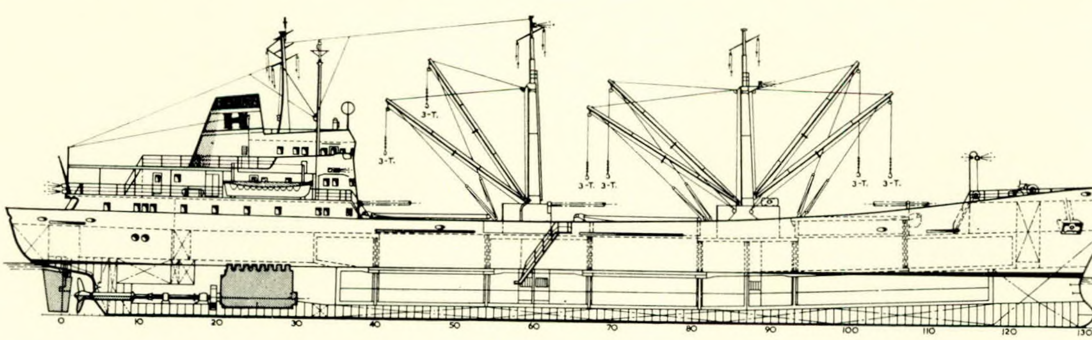
For main propulsion, the owners, standardized on an eight cylinder Mu551Ak engine developing 2750 bhp at 275 rev/min. The four-stroke engine is intended to be run from "pier to pier" on heavy oil (IFO 200), consuming about ten tons/day.

As the vessels will mainly be employed in tropical and sub-tropical waters, the entire accommodation is served by a Hi-Press air conditioning system. This guarantees an inside temperature of 27°C (80°F) with 60 per cent relative humidity when outside ambient conditions are 32°C (90°F) 87 per cent humidity. The heating capacity of the plant is such as to maintain a 20°C (68°F) room temperature when the outside temperature is −15°C (5°F).

The electric installation is powered by four English Electric 12 CSVM Diesel engines rated at 750 rev/min, 1350 bhp engines each driving a 745kW constant current d.c. generator and in tandem a 220kW-220 volt d.c. auxiliary generator.—*Shipbuilding and Shipping Record, 21st and 28th December 1967, Vol. 110, pp. 869-874.*

Metalliding

A technique of diffusing atoms of one material into the surface of another, thereby obtaining surface properties radically different from those of the bulk material, has been developed by the General Electric Co. The technique has been given the name "metalliding." The company foresees the process as a means of meeting some of the demands of space and nuclear technology by shielding various metals and alloys against corrosion and erosion. Other potential applications include the



Lühesand

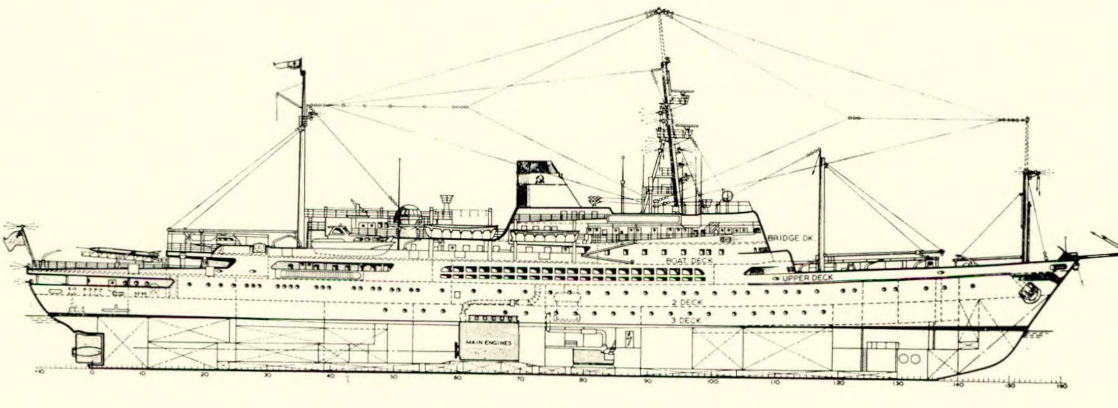
formation of extremely hard surfaces for bearings and dies, the solving of difficult problems of lubricating certain metals, and the creation of decorative finishes on base metals.

Metalliding is a high-temperature electrolytic technique. The bulk material, serving as a cathode, and the diffusing material, serving as an anode, are suspended in a bath of molten fluoride salts. When a direct current is passed from the anode to the cathode, the anode material diffuses into the surface of the cathode, producing a coating that is an alloy.—*Scientific American, September 1967, Vol. 217, p. 106.*

Russian Research Ships

At the VEB Mathias-Thesen-Werft, Wismar, a contract for five oceanographic and meteorological research ships is advancing towards completion with three units already in service, another launched and the final vessel under construction.

The design was prepared by the builders in collaboration with the Soviet Academy of Science and was for a full-scantling ship with three continuous decks. Construction to the assigned class was under the supervision of the Deutsche Schiffs-Revision und Klassifikation (DSRK).



General arrangement of research ships

As one-compartment ships for water-tight integrity, the hull below the second deck is divided by nine transverse bulkheads into the following main watertight compartments:

1) fore peak and chain locker;
2) magazine, bow thrust unit etc;
3) hold;
4) fuel tanks, accommodation etc;
5) forward stabilizing tanks, gyro compass compartment, accommodation etc;
6) auxiliary engine room and accommodation;
7) main engine room, workshops, service tanks etc;
8) after stabilizing tanks, pump room, fuel tanks, refrigerated store rooms etc;
9) fuel tanks, rocket store etc;
10) after peak and steering gear compartment.

Above the second deck the hull space is principally devoted to accommodation and laboratory and research quarters and the same applies to the superstructure decks.

An essential feature for research vessels is an ability to proceed and remain manœuvrable at slow speeds. This has been achieved by the provision of an active rudder powered by a 300 hp motor and two bow thrust units each powered by a 190 hp motor. The rudder can be put over to 90 deg each way in view of the attached active unit which, with main engines stopped, gives a speed of about four knots. Wind drift can be checked from an observation station on the monkey island using a smoke signal. The electric steering gear has a torque of 16 tons/m.

Main propulsion is by two six-cylinder M.A.N. type K6Z. 57/80 Diesel engines, built under licence by VEB Halberstadt, each rated at 4000 bhp at 225 rev/min. These units have exhaust gas superchargers which are F.W. cooled, as are the cylinder heads and fuel injectors, while the pistons are oil cooled, and are directly coupled to five-bladed chrome steel propellers with a diameter of 2950mm (9ft 8¾in) and a pitch ratio of 1·02.

Electric power is supplied by five 460kW (p.f. 0·8) type SSED 914/12 alternators each powered by an eight-cylinder type NVD36A Diesel engine developing 560 bhp at 500 rev/min.—*Shipbuilding and Shipping Record, 4th January, 1968, Vol. 111, pp. 13-16.*

Study of the Law of Crack Propagation

A general form of the mathematical function in the crack-propagation law for specimens of both infinite and finite width was determined by means of dimensional analysis. The exact function of the latter type was then determined by experiments. It was found that the general equation of the power law holds for specimens of finite width for three metals, cold-rolled types 301 and 310 stainless-steel sheet and 2014-T6 aluminium alloy sheet, tested at 78° F and low cyclic load rates.—*Yang, C. T., Trans. A.S.M.E., Jnl of Basic Engineering, September 1967, Vol. 89, pp. 487-495.*

Deceleration of an Unbalanced Rotor through a Critical Speed

The problem of a single-degree-of-freedom rotor decelerating slowly through its critical speed is solved by an energy approach; a closed solution is obtained. A small discontinuous downward jump of rotor speed across the critical speed is shown to be required, either with or without damping in the system. The maximum increment of deflexion, hence bending stress, in the rotor shaft is shown to be small, provided the rotor is carefully balanced and/or the system is sufficiently damped.—*Bodger, W. K. Trans. A.S.M.E., Jnl of Engineering for Industry, November 1967, Vol. 89, pp. 582-586.*

Magnetic Method of Measuring the Wall Thickness of Steam Generator Water Tubes

The reliability and accuracy of different non-destructive methods for measuring the wall thickness of boiler tubes during overhauls are compared and the magnetic method is recommended as the most reliable, convenient and least time-consuming. The instruments and method are described briefly. —*Schamberger, J., Strojirenstvi, February 1967, Vol. 17, pp. 138-139; Fuel Abstracts and Current Titles, July 1967, Vol. 8, p. 80.*

Torsional Vibration Analysis of a Hydrodynamic Split Torque Transmission

The steady-state torsional vibration for one mode of an engine transmission system was analysed and the analysis was verified by experimental data. The engine transmission system included a Diesel engine, torque divider which consisted of a fixed housing, single-stage torque converter and a planetary gear set and a dynamometer. The equations of motion are derived by an energy method (LaGrange's equation) and a numerical solution of these equations is obtained with the aid of a digital computer.—*Klokkenga, D. A. Trans. A.S.M.E., Jnl of Engineering for Industry, November 1967, Vol. 89, pp. 605-610.*

Dynamic Programming for Computing Optimal Plate Dimensions in Some Ship Structures

Research on automated optimization methods, using non-linear mathematical programming applied to minimum cost ship structures, is extended to discrete and integer variables in the design of plate sizes for decks and longitudinal and transverse bulkheads. This problem arose from the shipyard's desire to design plate layouts to minimize total costs including welding, extra charges for steel plates differing from a certain value and excess material used in meeting classification thickness requirements.—*Moses F. and Toennessen, A., European Shipbuilding, 1967, Vol. 16, No. 4, pp. 70-76.*

Admiralty Pattern Dissolved Oxygen Meter

Tests carried out on a dissolved oxygen meter developed by the Admiralty Materials Section to determine the instrument's suitability for measuring low concentrations of dissolved oxygen in boiler feedwater are described. The instrument consists of a perspex cell, a magnetic stirrer and measuring and calibrating electrodes. The oxygen concentration is determined by measuring the electrical conductivity of the sample after a drop of KOH has been added to increase conductivity.—*James, W. G. and Fisher, A. H., Chemistry and Industry, 18th February 1967, No. 7, pp. 272-274; Fuel Abstracts and Current Titles, July 1967, Vol. 8, p. 81.*

Heat Transfer and Pressure Loss in Spiral Tubes

The influence of secondary flow on heat transfer and pressure loss in helical pipe sections under conditions of induced convection was investigated. Spiral tubes show a higher heat flow than straight tubes.—*Schmidt, E. F., Chemie-Ingr-Tech., July 1967, Vol. 39, pp. 781-789; Fuel Abstracts and Current Titles, November 1967, Vol. 8, p. 104.*

Casting of Marine Propellers in Nickel-Aluminium Bronze

After reviewing the requirements of modern ships' propellers, the author describes in detail the procedures employed at the Theodor Zeise Foundry, Hamburg-Altona, to cast a six-bladed propeller, 8900 mm in diameter, for installation in a 190 000-ton tanker.

The propeller was produced in nickel aluminium bronze to German specification *DIN* 1714 (copper 70-80, iron 2-5, aluminium 6-11, nickel 2-5, manganese 5-8, per cent) and involved the casting of 74 tonnes of metal to give a finished weight of 54·9 tonnes. The author describes the preparation of the pattern and discusses moulding techniques, furnace control during melting of the nickel aluminium bronze, casting procedures and the grinding and machining of the casting.

In conclusion the author gives a general review of the Salient characteristics of materials commonly used for service in ships' propellers which, in addition to nickel-bearing and nickel-free brasses, comprise alloy steels (stainless chromium or chromium nickel grades) used for special applications and cast iron or cast carbon steel, used for stand-by propellers. — *Rininsland, H. S. Giesserei, 28th September 1967, Vol. 54, pp. 513-518; Nickel Bulletin, 1967, Vol. 40, No. 11, p. 288.*

Ultrasonic Pressure Gauge

The transit time of short ultrasonic pulses in solid rods is used to measure pressures throughout the fluid range. Self-excited regenerating circuits are used. Measurements to 3500 bar with an uncertainty of 0·3 bar and to 20 kb with an uncertainty of two bar are reported. Further improvements, temperature compensation and stability are discussed.—*Heydemann, P. L. M., Trans. A.S.M.E. Jnl of Basic Engineering, September 1967, Vol. 89, pp. 551-553.*

Systems for the Control of Ship Motions

There are now a number of different types of system for the control of the rolling motions of ships, from the simple bilge keels to the sophisticated fin stabilizers. The choice of an individual system depends to a large extent upon the ship operating conditions, the initial cost of design and installation, the maintenance costs in service and upon the loss of deadweight due to space loss.—*Goodrich, G. J., Technical Symposium of Ships Gear International, 1966, Paper No. 9/1.*

Influence of Mode of Loading on Fatigue Behaviour of 18-10 Stainless Steel

Study of changes occurring in the mechanical hysteresis loop of a specimen during fatigue testing offers an extremely sensitive means of detecting variations in mechanical properties resulting from fatigue-induced structural transformations. Alternating-torsion tests at fixed strain amplitude are usually employed for this purpose and as the type of loading may be expected to influence the results, the authors compared the changes occurring in the mechanical-hysteresis loop of identical specimens subjected to alternating-torsion tests at fixed strain amplitude or at fixed maximum torque or stress.—*Levasseur, M. and de Fouquet, J., Memoires Scientifiques Revue de Metallurgie, February 1967, Vol. 64, pp. 133-139; Nickel Bulletin, 1967, Vol. 40, No. 11, p. 298.*

Mechanism of Inception of Hydraulic Cavitation

For the purpose of clarifying the physical nature of the inception of hydraulic cavitation, distilled water saturated with air at atmospheric pressure was injected as a free jet through small nozzles into the same kind of liquid under various states of thermodynamic meta-stability. The condition of the critical states for the inception of cavitation was observed. The results of this experiment seem to show that, contrary to the generally acceped hypothesis of the existence of "nuclear bubbles", some kind of direct coupling between the state of flow of the liquid and the inception of cavitation therein may be blamed for the phenomenon.—*Wakeshima, H. and Nishigaki, K., Japanese Jnl Applied Physics, February 1967, Vol. 6, pp. 263-266; Applied Mechanics Reviews, December 1967, Vol. 20, p. 1194.*

Proposed New Stern Arrangement

The trend of ship designs towards high speed and power has led to increasing propulsive difficulties due to the problems of cavitation and propeller-induced vibration. Consequently, in the design of stern arrangement, considerations must be focused on these problems in addition to propulsive efficiency. The report deals with the factors that influence the hull-propeller interaction problem. It discusses the advantages and disadvantages of single-screw and twin-screw arrangements presently used. A new twin-screw arrangement is described.—*Pien, P. C. and Strom-Tejsen, F., May 1967, Naval Ship Research and Development Centre Rep. No. 2410; Applied Mechanics Reviews, December 1967, Vol. 20, pp. 1236-1237.*

Reciprocating Compressors with Non-Lubricated Cylinders and Labyrinth Pistons or Plastic Piston Rings

The authors describe the development of the labyrinth piston compressor, tests on reciprocating compressors with plastic piston rings, test results, comparative measurements, design features and successful employment of non-lubricated piston rings.—*Zürcher, A. and Meier, H., Sulzer Technical Review, 1967, Vol. 49, No. 1, pp. 25-29.*

Static and Dynamic Tests of Speed Governors for Diesel Engines

Static and dynamic tests dealt with in this paper have been carried out on 14 speed governors for Diesel engines. The test stand described permitted a constant main speed to be adjusted between 500 and 1000 rev/min (static tests) and allowed the superimposition of speed oscillations of variable frequency and amplitude (dynamic tests) on the main speed. The test results are presented and discussed in the form of coefficients and frequency-response curves. — *Hutarew, G., 1967, A.S.M.E. Diesel and Gas Engine Power Conference and Exhibition Paper No. 67-DGP-3.*

Mechanisms of Wear in Misaligned Splines

Experimental results are presented on the wear behaviour of misaligned splines operating without lubrication and with either grease or oil lubrication, in various environmental atmospheres. On the basis of the observed results, it appears that protection from rapid wear stems from conditions which inhibit oxidation of the stressed metal surfaces. This inhibition process can be obtained by both physical and chemical means. The physical means include exclusion of oxygen from the contact region and the chemical means include the use of lubricant additives.—*Weatherford, W. D., Valtierra, M. L. and Ku, P. M. October 1967, A.S.L.E.-A.S.M.E. Lubrication Conference Paper No. 67-Lub.-1.*

Evaluating Mechanical and Corrosion Suitability of Materials

Fracture toughness, corrosion response and low cycle fatigue characteristics of advanced high strength metals are of critical importance in structural design and materials selection for present and future high performance structures. The methods for evaluating these materials and for application of the results to predicting structural performance are not widely understood. Recently developed concepts and methods are presented for characterizing these important properties for the new high strength metals in full thicknesses and for translating the results of predictions of structural performance.—*Brown, B. F. and Goode, R. J., A.S.M.E. Design Engineering Conference and Show, May 1967, Paper No. 67-DE-7.*

Aspects of the Stork Large Marine Diesel Engine

Several types of large marine Diesel engine are now available. Steady development resulting in an increase in mean effective pressures and dimensions is going on. The paper outlines the basic principles and characteristics of a special engine type in connexion with the demands for its turbocharging. Experimental results promising future development are mentioned and special design features favouring easy maintenance are described. — *van der Wal, H., 1967, A.S.M.E. Diesel and Gas Engine Power Conference and Exhibition Paper No. 67-DGP-4.*

Patent Specifications

Cargo Ship for Carrying Loaded Barges

Fig. 1 shows the forward end of a cargo ship having a hull (10) with the wheel house (11) at the forward end and in the aft part of the ship, that is after the wheel house, the vessel is divided into cargo holds (12), (13), (14), (15), (16) and (17). At both ends and between each of these holds are watertight bulkheads (18), (19), (20), (21), (22) and (23). Each is uninterrupted, as seen in Fig. 3 from the inner wall

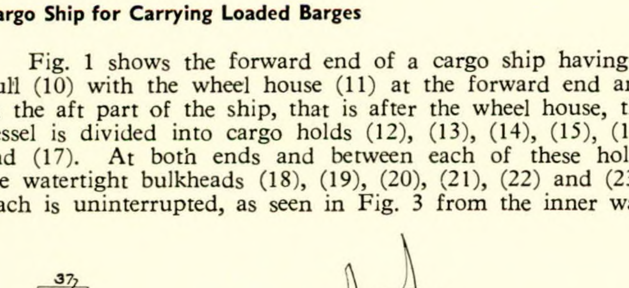
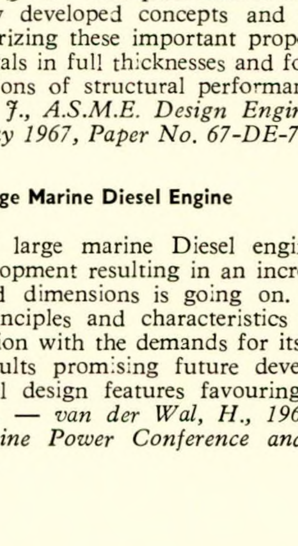


FIG. 1.

FIG. 3.



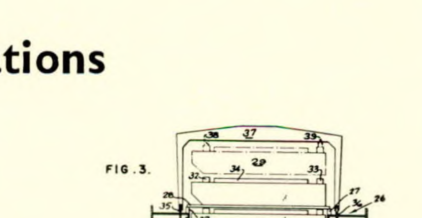


FIG. 2.

of the wing tank (24) to the inner wall of wing tanks (25). The vessel is provided with a main deck (26) having coaming (27) about a hatch for receiving a hatch cover (28) over each hold.

The cargo units or barges (29) have a bow (30) and stern (31). Each barge has a pair of pillars spaced apart (32) near to the bow and a similar pair (33) at its stern. The cargo carried by the barges (29) is shown at (34). It will be noted in Fig. 3 that the bottom of the barges above the first one stored in the vessel rests upon the pillars of the barge immediately beneath it to prevent crushing the cargo stacked in the other loaded barges. As shown in Figs 1 and 2, the barges (29) are positioned in the cargo-receiving hold with their major axis at right angles to the longitudinal axis of the main cargo ship (10). This orientation of the barge units with respect to the main vessel hold permits larger capacity holding of larger barges for quicker off-loading than has been suggested previously.—*British Patent No. 1 090 344 issued to J. Lee Goldman. Complete specification published 8th November 1967.*

Passive Stabilizers for Tankers

It is generally known that the sizes of newly designed liquid cargo tankers are now over 100 000 dwt and because of this increased cargo carrying capability, the structural requirements for vessels of this type are becoming more exacting. Consequently the present invention is drawn to improvements in the incorporation of passive stabilizers in vessels of this type. Referring to Figs 1 and 3, a number of transverse oil-tight bulkheads (12) are spaced along the longitudinal axis of

FIG.1.

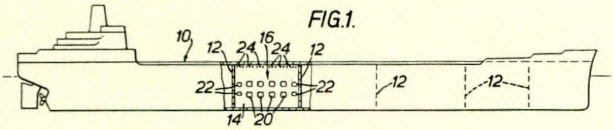
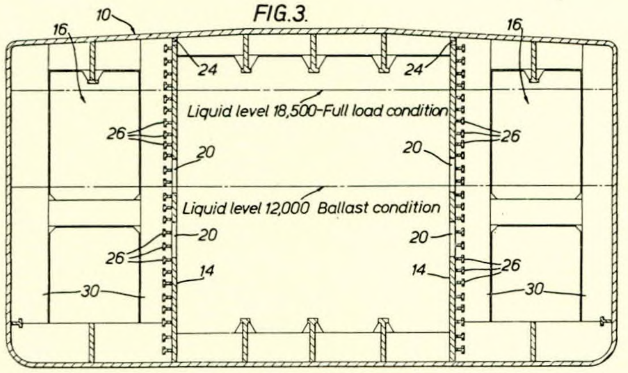


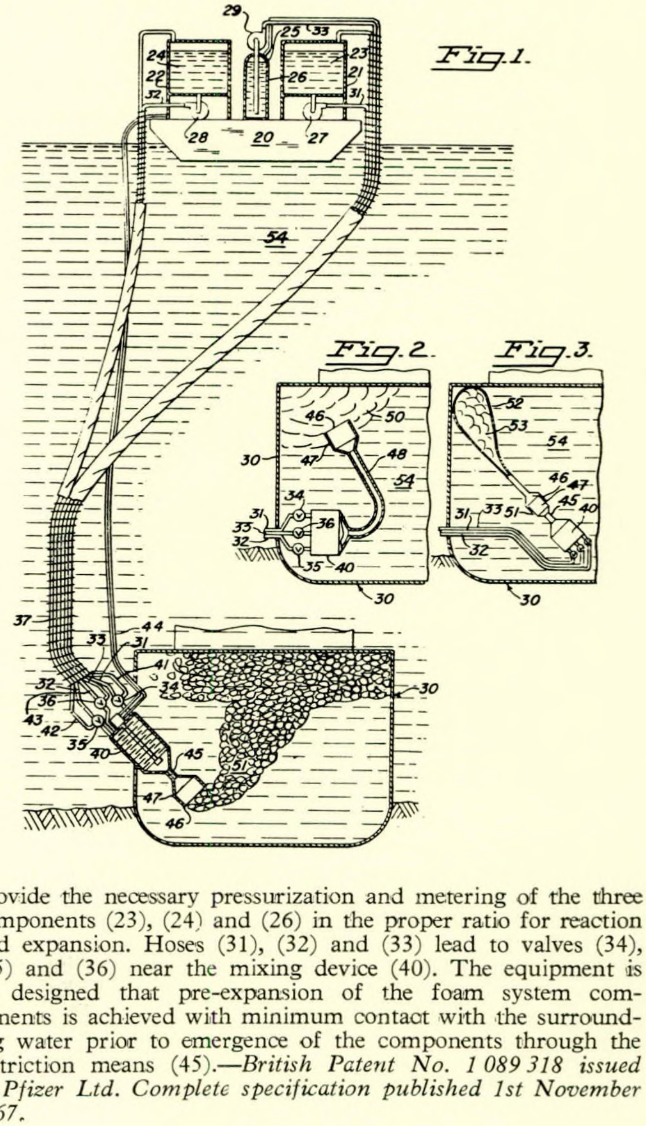
FIG.3.



the vessel (10). There is also one or more longitudinal bulkhead (12) which is oil-tight except as shown. The vessel shown has two such longitudinal bulkheads. At least one of the compartments (16) functions as a passive stabilization system. Bulkheads (14) divide compartment (16) into two outer wing tanks and a centre connecting tank. The portions of the bulk heads (14) within compartment (16) are provided with a number of large openings (20) and (22) to enable communication between each wing tank and the centre tank. Smaller openings (24) are also provided to enable unimpeded transfer of air in response to the movement of liquid through the larger openings (20) and (22). A number of horizontal T stiffeners (26) provide lateral support and stability in anticipation of large lateral forces extend by the liquid in the tanks.—*British Patent No. 1 093 972 issued to J. J. McMullen Associates, Inc. Complete specification published 6th December 1967.*

Method of Raising Submerged Objects

This invention relates to a method of raising submerged objects such as sunken vessels by the use of closed cell plastics foam. Referring to Fig. 1, a salvage vessel (20) may be equipped with a pair of tanks (21) and (22) each containing one or two mutually reactive foam-forming resinous ingredients (23) and (24) and a third tank (25) containing a compressed or liquefied gaseous expanding agent (26). Three pumps (27), (28) and (29)



provide the necessary pressurization and metering of the three components (23), (24) and (26) in the proper ratio for reaction and expansion. Hoses (31), (32) and (33) lead to valves (34), (35) and (36) near the mixing device (40). The equipment is so designed that pre-expansion of the foam system components is achieved with minimum contact with the surrounding water prior to emergence of the components through the restriction means (45).—*British Patent No. 1 089 318 issued to Pfizer Ltd. Complete specification published 1st November 1967.*