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THE PROPULSION OF A MILLION TON TANKER

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The building and operating of a one million ton ship has, in recent years, become a distinct possibility and one that poses intriguing problems for the many who will be concerned in such an enterprise. The speed, power and possible proportions of such large vessels are of obvious interest, but when a paper on the subject was first proposed in December 1968 the title was to be "The Propulsion of a 500 000 Ton Deadweight Tanker". It is indicative of the speed of development of such vessels that before the end of 1969 the title had to be altered to cover the 1 000 000 ton vessel, thus preserving some element of adventure in the exercise. It was hoped that the launching of the megaton ship would be delayed at least until after 8th December, 1970, when this paper was presented.

It was the original intention to discuss primarily the propeller problems involved on such a ship, but at the suggestion of the Institute the scope was widened to include some consideration of the hull itself and possible suitable propulsion machinery to provide a technically acceptable overall propulsion system giving the vessel a reasonable service speed. The increased scope meant inevitable omissions in the detailed content of the paper and it is hoped therefore that in reducing it to readable proportions it will nevertheless stimulate discussion on what is assuredly an exciting engineering possibility.

INTRODUCTION

World oil consumption continues to increase, largely because of the ever increasing demand for power and the development of world transport of all types. When total transportation costs are evaluated, it is believed that it can be shown that oil is carried more cheaply as ship size increases and as a result, tankers have become the largest vessels afloat.

The increase in tanker size has been too rapid to allow the normal design procedure where progress is made in relatively small steps embodying at each stage minor advances in technology. As a result of this sensational rate of progress, there is now an accumulation of technical problems which require consideration if the increase in size continues at the present rate. In an endeavour to anticipate some of these problems the propulsion of ships of increasing size from 500 000 to 750 000 and 1 000 000 dwt has been examined from the propulsion point of view.

Although oil companies and other organizations are concerned with the whole transportation problem and may, as a result, produce a completely new ship type or a completely new oil transportation system, there has been no evidence of this available to the authors at this stage. In discussing the propulsion problem, therefore, the first part of the paper reviews the proportions of conventional ships and other design and operating factors, while the main body of the paper is concerned with possible propeller arrangements and the corresponding machinery installations. The physical and technical limitations in various areas of such a project are shown, and some indication is given of the ways in which these can be overcome. The short final section gives a summary of the various possibilities, although it is extremely difficult because of the number of variables involved to give any firm recommendations.

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PART I—SHIP DIMENSIONS AND HULL FORM

General Discussions Effecting the Hull and Propulsion

If the 1 000 000 ton tanker is to be built, technical facilities for servicing, loading and discharging must be created to anticipate this development. In this connexion a vessel must be accommodated having a length perhaps up to almost 1700 ft, a beam which may be around 300 ft and a draught that may be well over 100 ft. These dimensions would appear to be permissible utilising off-shore loading facilities and deep water routes. Loaded draughts over 100 ft, however, may be excessive for many purposes and the relatively narrow, deep ship may involve problems in launching and in docking because of its draught when light. It is therefore reasonable to suppose that for flexibility in service during the early years of operation, the trend will be towards shallower and wider ships.

The choice of 1 000 000 tons as the deadweight is, of course, an arbitrary one, but the arguments applying to this ship apply in modified form to bigger and smaller ships. For smaller vessels, down to 500 000 dwt, small changes in proportions may be critical because the smaller draught makes available a wider range of ports and docks. If the draught is restricted because of routes and facilities, the structural consideration of smaller tankers suggests that there is likely to be a rapid increase in steel weight and cost when the ratio of breadth to depth exceeds about 2:1, and the most expensive dimensional increase would be that of length. In selecting a typical hull, the extreme case of a tanker operating over a shallow water route has been avoided because for these wide, shallow forms strength and course keeping become the major problems.

For these very large ships there are few safe refuges available, and the towing problems in the event of casualty at sea are formidable. There is therefore more chance that an accident or failure will lead to total loss of ship or cargo than in the smaller vessel, and a greater need for the ship to be as far as possible independent of outside assistance when away from terminal ports. The question of safety, reliability and duplication of essential components will therefore be a matter of prime importance, as will be the question of control and manoeuvrability.

Up to the present, despite the increasing size of tanker, the ship speed has remained fairly constant at between 15 and 16 knots. Manoeuvrability and stopping ability are, in the main, a function of the power available and therefore, although there may be advantages arising from a reduction of speed and therefore power, it is considered prudent that a speed of at least 16 knots in service should be assumed.

At low speeds widely spaced wing propellers can provide an appreciable turning moment and controllable pitch propellers in this position would produce a significant improvement from this point of view as compared with fixed pitch propellers. Unfortunately twin screws, with the necessity for twin rudders and perhaps twin skegs with the added hull resistance, do not appear to be an attractive propulsion system, and we are therefore led to the consideration of triple screws which may offer certain advantages. A high percentage of the power could, in this case, be taken on the central propeller forward of a centrally disposed rudder, giving velocity dependent steering at full speed. Manoeuvrability at lower speeds could be conveniently achieved by the widely spaced wing propellers (preferably controllable pitch) without the added appendage resistance of additional rudders and skegs.

At low speeds there is an important safety requirement and manoeuvrability will involve the use of large powered tug boats. It is natural therefore to consider what further manoeuvring facility can be contributed by lateral thrust units. Although experience has shown that transverse propulsion units are not very effective in the normal way for speeds much above three knots, an important manoeuvring advantage could be achieved at this and higher speeds if the water is drawn from forward, perhaps with a uni-directional propulsion motor disposed on the fore and aft centre line of the ship. The thrust required to check or reverse a swing of the head of the vessel is large, and therefore instantaneous availability of power is a necessity for this purpose.

In short, safety requirements would appear to demand adequate power and a shafting configuration which will provide good stopping and steering ability preferably augmented by a bow steering device at a low and zero speeds.

Hull Form

The hull form used as a basis for propellers and machinery is shown as a body plan in Fig. 1. Figs. 2 and 3 show outline stern

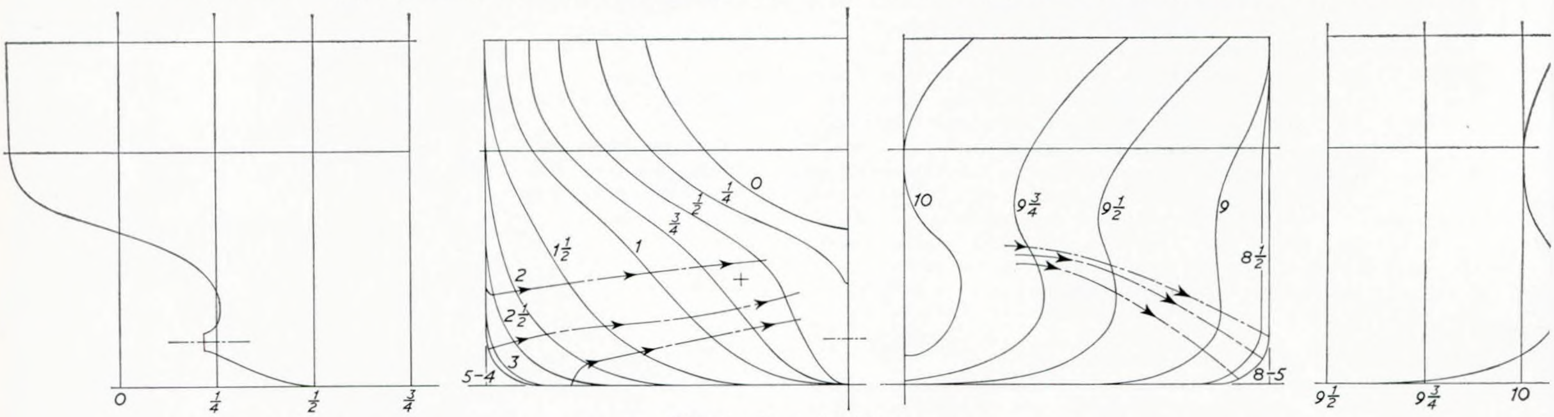


FIG. 1—*Basic hull form*

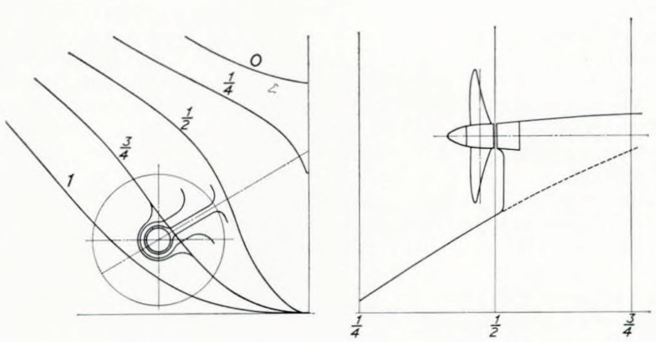


FIG. 2—*Stern arrangement—twin screw ship*

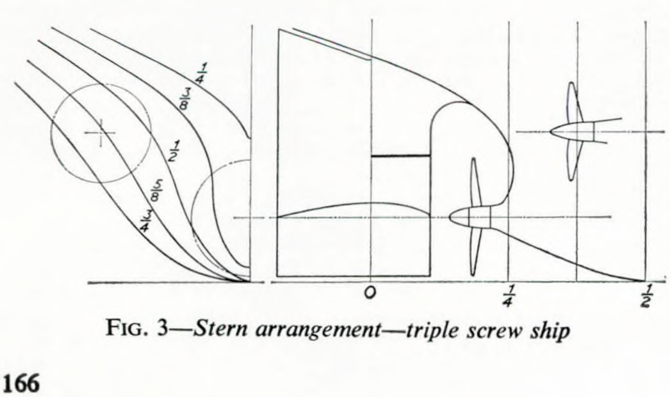


FIG. 3—*Stern arrangement—triple screw ship*

arrangements for particular twin and triple screw systems corresponding to specific model experiments now being carried out at St. Albans.

The 1 000 000 ton ship is 1600 ft length bp × 300 ft beam × 97·6 ft draught having a 0·85 block coefficient and a depth of about 140 ft. Similar ships for 750 000 dwt and 500 000 dwt would have dimensions 1460 ft × 274 ft × 89·4 ft and 1282 ft × 241 ft × 78·3 ft respectively. The smallest ship could use existing terminals and some of the present facilities could be dredged to take ships of 90 ft draught.

The service speed is 16 knots and as the wave making resistance is relatively unimportant up to 18 knots, the resistance penalty for increased fullness above 0·85 may be economically viable. For a length/breadth ratio of 5·3 course holding will require special study and it appears dangerous to increase the fullness coefficient up to the limit. Flow round the stern is rising slowly in way of the propellers and this fits in with orthodox bossing design for the wing screws and tends to rule out the use of vertical skegs for the twin screw design. The fore end must be shaped to avoid separation at the fore shoulder and a number of alternative forms would satisfy the requirements with some variation in resistance. For instance, the partially straight framed hull shown in Fig. 4 would ease construction although increasing the resistance by up to four per cent.

So far as the general arrangement of the ship is concerned it is preferable that the machinery should be concentrated into a short length leaving a clear block of cargo space. The triple screw ship has the wing shafts raised so that the wing engines do not need to be placed forward of the centre engine. The powers absorbed are large, and it is inadvisable to risk increased vibration and noise by making the stern fuller or reducing clearance.

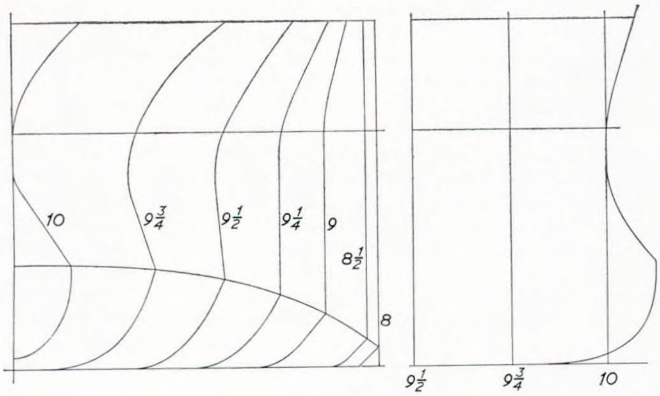


FIG. 4—*Basic hull form incorporating straight frames*

In considering 500 000 to 1 000 000 dwt ships, it was felt that for confirmatory purposes model tests should be carried out in the middle of the range and the 750 000 ton model was chosen. For this size of vessel a single screw arrangement was designed for 50 rev/min giving a diameter of 54 ft. The twin screw arrangement was designed for 70 rev/min giving a diameter of 35 ft and the triple screws were designed for 90 rev/min giving a centre propeller 31·5 ft in diameter and wing screws of 26·7 ft diameter.

It had been hoped to include these tank results in the body of the paper but this was found impossible and they will be given as a contribution to the discussion. As a result of this, instead of having available the ehp values from these tests on a 24 ft model, figures were supplied from tests on relatively small models run in the Newcastle University tank as follows:

500 000 dwt—47 500 ehp (naked)
750 000 dwt—61 000 ehp (naked)
1 000 000 dwt—72 600 ehp (naked)

There are two other points for discussion in this connexion, the correlation factor relating model estimate to the ship performance and the service allowance. Results up to 1965 showed that for single screw ships model predictions using the ITTC skin friction line required a correlation factor $(1 + X)$ varying from 1·16 for a 400 ft ship to 0·97 for an 800 ft ship to obtain the ship power with the best hull surface and best trial conditions. Applying this to tankers, if a form factor of about 20 per cent is added to the ITTC skin friction line and the ship prediction calculated on this basis, a constant addition to the resistance coefficient will fit the empirical correlation factor at 400 ft and 800 ft lengths. Extrapolating on this basis to greater lengths the correlation factor curve flattens out and may even be rising at 1600 ft. For the present estimates a $(1 + X)$ ITTC of 0·96, or 0·80 using the extended Froude coefficients is reasonable.

With increased length the tanker will pitch and heave less than the present day ships and the weather allowance could be reduced. But unless docking and cleaning arrangements are planned in advance, provision should be made for longer periods at sea with increased average resistance due to hull surface deterioration and fouling.

PART 2—POWERING AND PROPELLERS

The problem facing the propeller designer is to select practical propeller arrangements that will give good performance consistent with manufacturing feasibility and operating costs. Unfortunately the variables involved are so numerous that a general solution is almost impossible at this time. Certain basic assumptions must therefore be made, to reduce the number of variables to manageable proportions and provide a practical starting point from which the propulsion of these large vessels can be considered.

There are a number of propeller arrangements available which could possibly be applied to advantage, for example, contra-rotating, tandem, shrouded and overlapping propellers. However, the more conventional arrangements of single, twin and triple screws have much to commend them and before any major changes in propulsion device are considered it would seem desirable that every effort should be made to examine them in detail. Also, up to now, tankers have almost invariably been propelled by single screws and it is therefore natural to investigate first single screw propeller designs so that these may be used as comparisons with other arrangements and also with traditional practice. Detailed study has therefore been concentrated on single and multi screws fitted to the half, three-quarter and one million ton deadweight tankers discussed in the previous section.

Speed and Propulsion Factors

An examination of previous data has shown that throughout the range of ship size, the operating speed has remained almost constant at around 16 knots. Therefore, this speed has been selected to represent the average fair weather fully loaded service condition for these ships. This is, of course, a very low speed for the length of the vessels considered, but on the other hand, the total power, particularly when the million ton ship is considered, is extremely high and an advance on 16 knots seems most unlikely. A reduction in speed is the more possible alternative, and a speed reduction to 15 or 14 knots would exert a considerable influence on machinery requirements. As previously mentioned, however, 16 knots was chosen for this exercise as being a more reasonable speed for ships of this length and also because of the need for adequate manoeuvring and stopping power.

It is, of course, the speed of advance, rather than the ship speed, that is the important variable and this involves an estimate of the mean wake fraction. From a consideration of available data the following values have been derived, and have been assumed to apply to a propeller having a diameter of 41 per cent of the loaded draught aft.

TABLE I

	Deadweight 500 000	750 000	1 000 000
Single screw	0·437	0·425	0·420
Twin screw	0·254	0·251	0·250

As will be seen later, a series of diameters have been calculated for varying revolutions per minute, and in deriving these, the wake fraction was adjusted from the above values so that smaller mean wakes applied to the propellers having larger diameters and higher wakes for diameters less than the standard referred to above. Adjustments were made to the basic values assuming proportionate radial wake distributions similar to those given by Van Lammeren (Reference 1).

There is little data to assist in the selection of thrust deduction fraction for a particular single, twin or triple screw ship. We have therefore used the B.S.R.A. $0{\cdot}85C_B$ series as a guide (Reference 2), assuming constant thrust deduction factors of 0·19 and 0·17 for the single and twin screw ships respectively, although in practice these may well vary with different diameters and clearances. Similarly, relative rotative efficiencies of 1·01 and 0·99 for the single and twin screw ships were selected.

Correlation Factor and Service Allowances Applied to Effective Horsepower

The naked effective horsepowers as given in part one of this paper have first been multiplied by the recommended ship/model correlation factor of 0·8. This then represents the effective horsepower of a single screw hull on trial. An additional appendage allowance of five per cent is made in the case of the twin screw hulls, to account for the additional resistance of the twin bossings. Some attempt has been made to account for the increased bossing resistance on the triple screw ships, when allowances between zero and 5 per cent are added, depending upon the power absorbed by the centre screw. In every case, a further allowance of 18 per cent has been added to represent average sea conditions.

All of these allowances are obviously debatable, but they have been shown, in the case of the existing single and twin screw ships, to be reasonable in practice for power estimation.

Propeller Diameter and Rev/Min

Much has been written about the "apparent" fall-off in propulsive efficiency with increasing ship size (Reference 3). This can be shown to be due to the increasing power requirements of these very big hulls with constant speed and without commensurate change in rev/min, thus leading to propeller diameters small in relation to the beam and draught of the vessel.

Today tankers with deadweights over 300 000 tons are operating with almost the same service speed and rev/min as a 20 000 ton deadweight tanker of the 1950s, but now the engine output has increased over four-fold. The significance of this is obvious from an examination of any propeller $B_p - \delta$ diagram. Although changes in propeller efficiency are not exactly reflected in the propulsive coefficient, due to changes in wake fraction etc., there is still an important relationship. In order to retain the same propeller efficiency while, for example, doubling the power and keeping the same speed of advance, it would be necessary to reduce the rev/min by a factor of $\sqrt{2}$. At the same time the optimum advance coefficient δ would be the same and thus the propeller diameter would increase by this factor. This simple example does not account for such variables as blade-area-ratio or possible reductions in wake fraction etc., but the principle remains unchanged. It can therefore be said that the "ideal" propulsion arrangement is one in which the largest possible propeller diameter is used running at its optimum revolutions.

In the normal case of the dry cargo or container ship, the propeller diameter is limited by considerations of draught and immersion. It is fortunate that, under such conditions, rev/min of 110 to 140 are appropriate and involve conventional marine engines. The tanker is, on the other hand, a special case with a relatively unlimited draught where the diameter can, with advantage, be increased substantially. This could lead to the condition where the propeller diameter was only just covered in the ballast condition. Diameters in the region of 50 ft could then be expected. It may well be that the revolutions under such conditions are so low that the gearing required would be impracticable, in which case a compromise would have to be worked.

This principle was demonstrated on a proposed series of large Esso tankers some years ago when, following discussions with the propeller manufacturer, tests were run at the National Physical

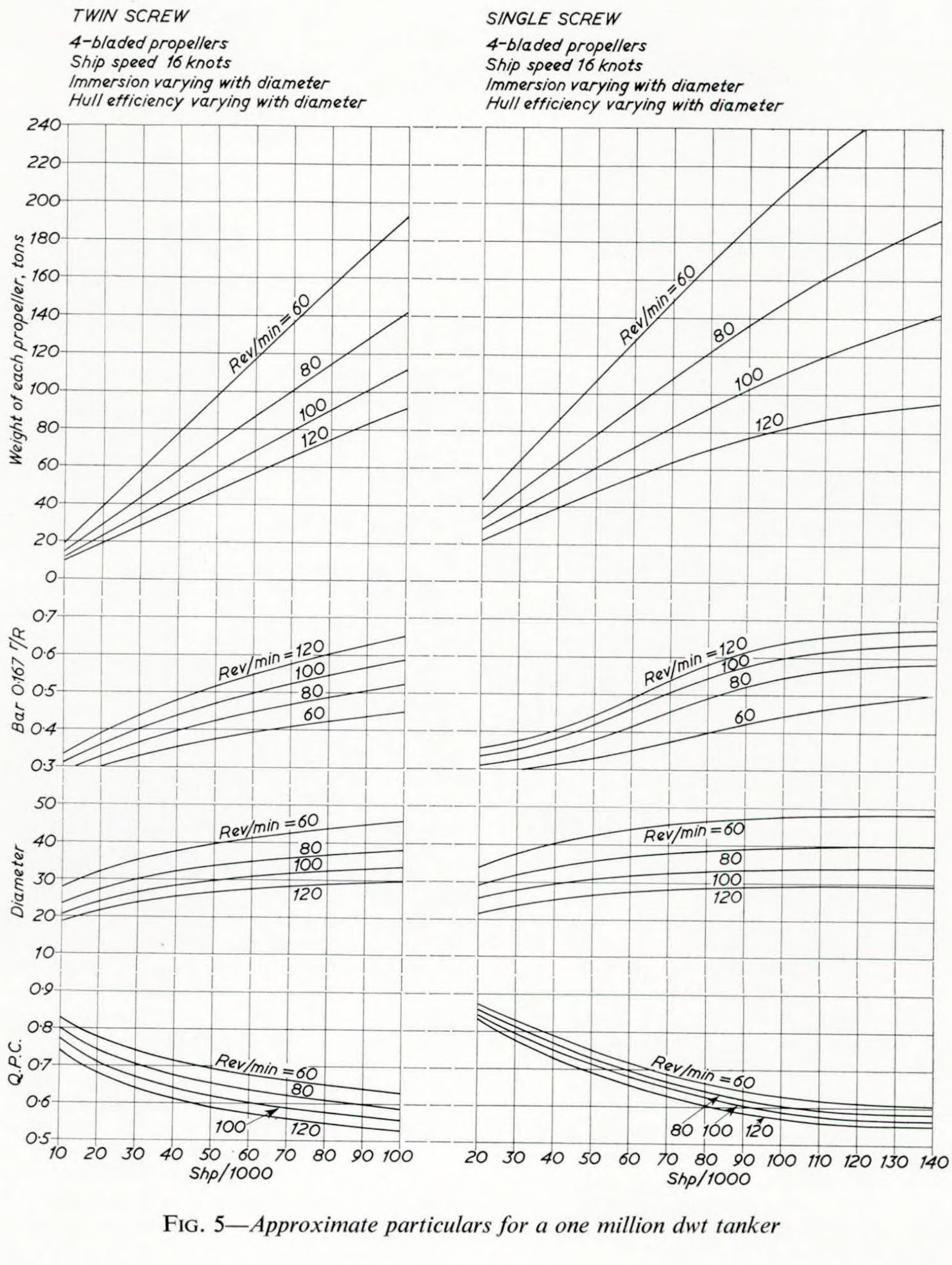


FIG. 5—*Approximate particulars for a one million dwt tanker*

Laboratory Ship Division which showed a marked gain in performance, when the revolutions were reduced from 120 to 80 per minute and the propeller dimensions increased to bring them more in keeping with the size of the particular ships involved.

Basic Propeller Curves

A computer programme was used to produce curves of q.p.c., propeller diameter, b.a.r. and weight, to a base of shp for various values of rev/min. Curves were produced for both centre and wing screws fitted to the three ships being considered. The resulting diagrams for the million ton ships are shown in Fig. 5.

This computer programme makes use of Troost systematic series model data for optimum diameters and open propeller efficiencies and the Burrill method for blade surface area estima-

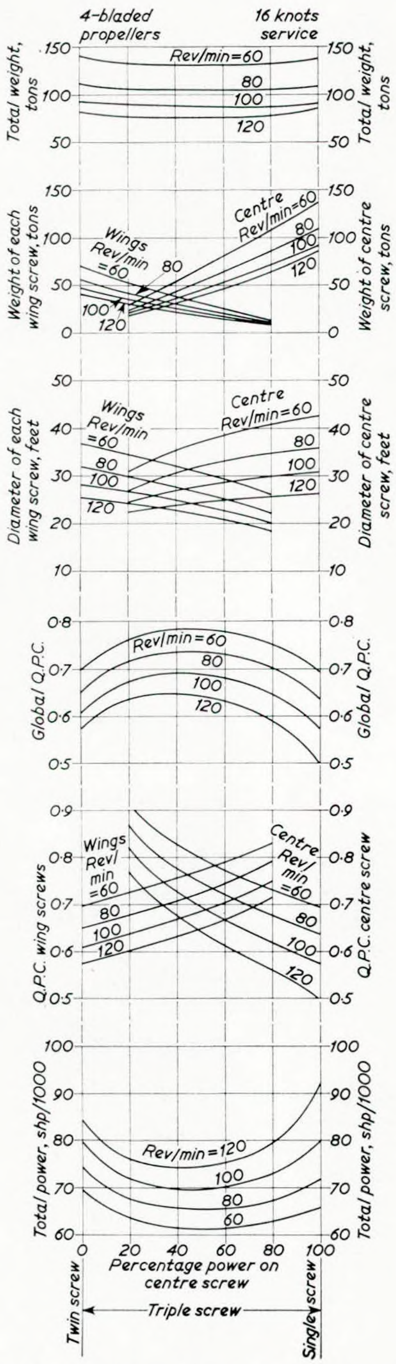


FIG. 6—*Approximate single, twin and triple screw particulars for a 500 000 dwt tanker*

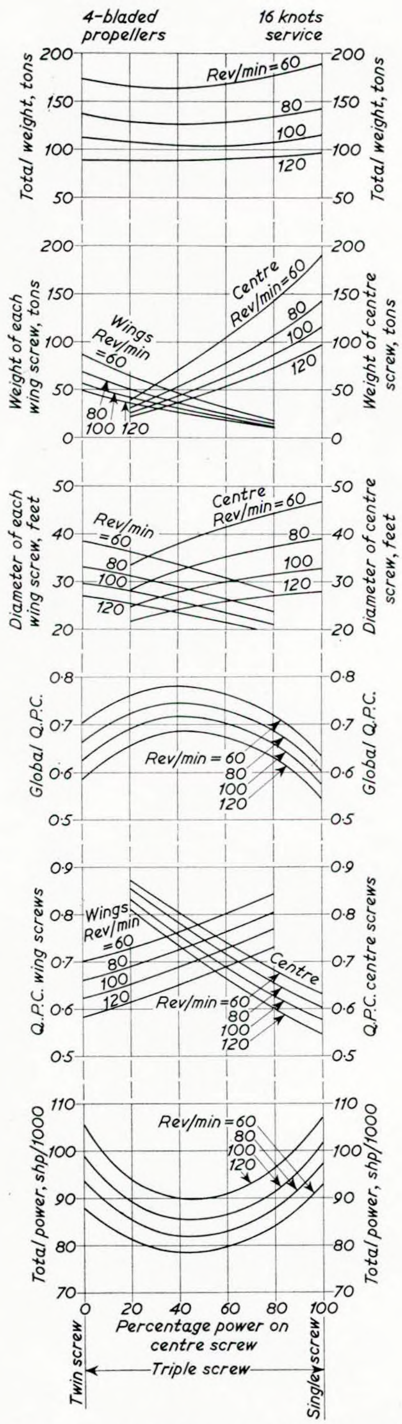


FIG. 7—*Approximate single, twin and triple screw particulars for a 750 000 dwt tanker*

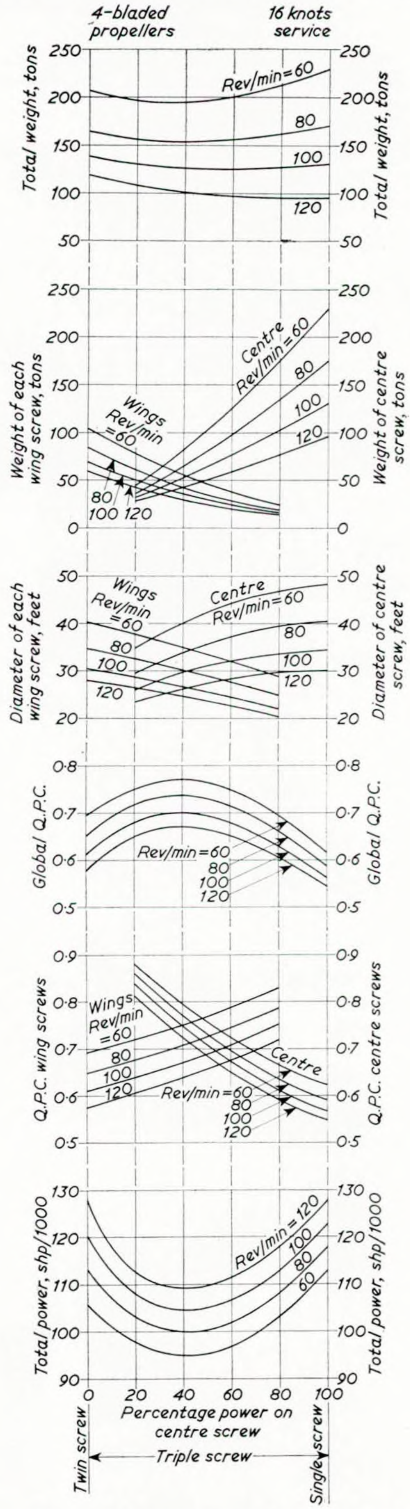


FIG. 8—*Approximate single, twin and triple screw particulars for a 1 million dwt tanker*

tion and blade thickness assessment. The programme gives the appropriate propeller weight based on boss diameters derived from the root blade widths and Lloyds' requirements concerning shaft diameter and bearing length.

The computer programme was arranged to automatically correct the wake fraction and propeller shaft immersion according to the calculated propeller diameter. To accommodate varying propeller diameters and maintain reasonable clearances from the keel to the lower tip of the propeller, the immersion was calculated from the ship draught minus the bottom clearance and propeller radius; the bottom clearance varying to comply with normal practice.

Triple Screws

With the basic single and twin screw diagrams it is possible to estimate the power required by triple screw ships having various combinations of power on the centre and wing screws. The most convenient way to do this is to carry out an iterative process at a particular rev/min, by first making an estimate at the possible shp, calculate the ehp using dhp and q.p.c.'s derived from the curves. If the calculated value is not the same, within reasonable limits, as that estimated for the ship, then the process is repeated with a revised estimate for shp. In these calculations transmission losses of two and three per cent are incorporated for the centre and wing screws respectively.

Using the derived shp's, propeller weights and diameters were obtained from the appropriate curves on the diagrams. The results of this work being presented in Figs. 6, 7 and 8 for the half, three-quarter and one million dwt ships respectively.

These diagrams show diameters, weights, q.p.c. and power requirement, over the range 60 to 120 rev/min for the three ships. Twin, triple and single screw information is given respectively, on the left, centre and right of each diagram. From the point of view of propulsive efficiency, triple screws are attractive as these extend the advantages of twin screws and take advantage of the wake gain near the ship's centre line, as in the case of single screws.

Single or Multi-Screw Ships

In the past it has been commonly assumed that it was always best from the point of view of propulsive efficiency to use single screws because of the high wake gain near the centre line of the ship. On ships such as those under consideration here, having exceptionally large beam and draught, this is not necessarily true providing the diameters and distribution of power between multi-screws are correctly chosen.

For optimum propulsive efficiency, the ideal approach is to design for the maximum propeller diameter that can be accommodated and by an iterative process calculate the optimum rev/min. This approach has been followed using extensions of Fig. 5 to lower rev/min and the results for the 1 000 000 ton ship are shown in Fig. 9.

In the course of these investigations it was found that very considerable gains in propulsive efficiency could be obtained

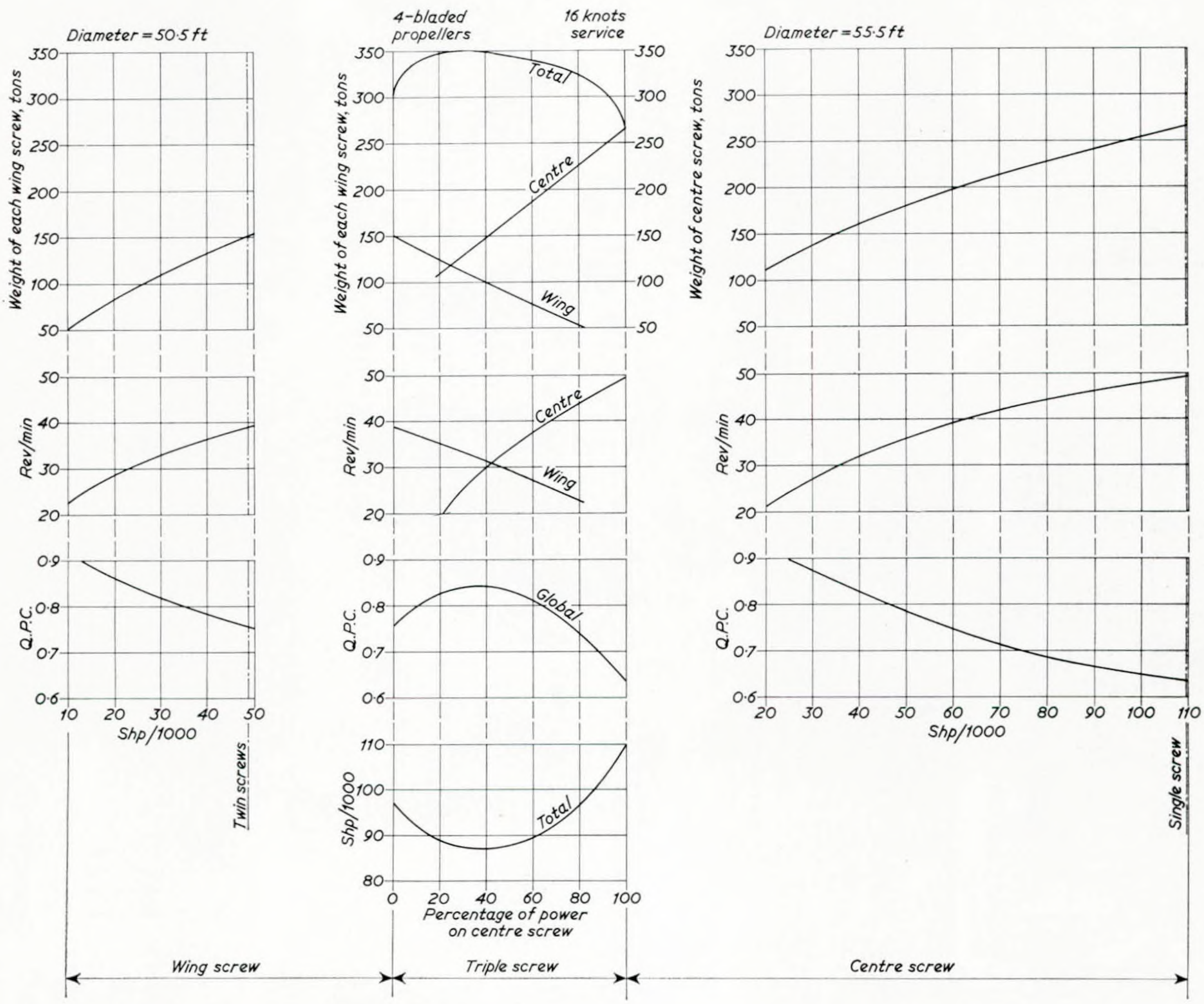


FIG. 9—*Approximate single, twin and triple screw particulars for a one million dwt tanker*

with triple screws; the optimum rev/min were found to be about 30 for 1 000 000 tons and 40 for the 500 000 tons deadweight ship. The propeller diameters are the largest consistent with the ballast draught and it is possible that these low rev/min may eventually be practicable. On the other hand for practical purposes at this stage, it was considered that 60 rev/min should be used as a lower limit. An examination of the diagrams shows that at these and at higher rev/min, there are still substantial propulsive advantages in using triple screws.

The comparative results for single, twin and triple screws for the three ships are discussed in more detail in the following sections.

TABLE II

Ship dwt	Prop. arrange-ment	Total shp	Global q.p.c.	% power on centre prop.	Rev/min		Diameter ft		Weight tons			Notes
					centre	wings	centre	wings	centre	wings	total	
	Limiting Diameter on all Propellers											
	Single	64100	0·713	100	65·0		44·0		142		142	
	Twin	66670	0·728			50·0		40·0		81	162	
	Triple	57360	0·838	45	46·0	40·0	44·0	40·0	86	50	186	optimum power
	Triple	57990	0·834	33⅓	35·0	43·0	44·0	40·0	72	58	188	equal power distribution
	60 Rev/min											
	Single	65790	0·695	100	60·0		42·5		137		137	
500 000	Twin	69550	0·697			60·0		36·5		71	142	
	Triple	61180	0·784	50	60·0	60·0	37·0	30·5	67	32	131	optimum power
	Triple	62040	0·779	33⅓	60·0	60·0	34·5	33·0	46	43	132	equal power distribution
	100 Rev/min											
	Single	79760	0·573	100	100·0		31·0		91		91	
	Twin	79950	0·607			100·0		28·0		47	94	
	Triple	69660	0·690	45	100·0	100·0	27·5	24·5	40	23	86	optimum power
	Triple	70310	0·688	33⅓	100·0	100·0	26·0	25·5	29	29	87	equal power distribution
	Limiting Diameter on all Propellers											
	Single	90800	0·647	100	53·0		50·5		214		214	
	Twin	82600	0·750			42·5		46·0		117	234	
	Triple	74000	0·830	41	34·5	34·5	50·5	46·0	116	78	272	optimum power
	Triple	74200	0·830	33⅓	31·0	36·0	50·5	46·0	105	85	275	equal power distribution
	60 Rev/min											
	Single	93000	0·632	100	60·0		47·0		191		191	
750 000	Twin	88000	0·703			60·0		38·5		87	174	
	Triple	78400	0·778	44	60·0	60·0	39·0	33·0	77	44	165	optimum power
	Triple	78900	0·778	33⅓	60·0	60·0	36·5	34·5	59	52	163	equal power distribution
	100 Rev/min											
	Single	102000	0·575	100	100·0		33·0		116		116	
	Twin	99000	0·625			100·0		30·0		56	112	
	Triple	85600	0·716	44	100·0	100·0	28·5	25·5	48	28	104	optimum power
	Triple	86300	0·713	33⅓	100·0	100·0	27·0	26·5	38	34	106	equal power distribution
	Limiting Diameter on all Propellers											
	Single	109800	0·637	100	49·5		55·5		266		266	
	Twin	97400	0·756			39·0		50·5		151	302	
	Triple	86900	0·843	41	31·0	31·0	55·5	50·5	152	98	348	optimum power
	Triple	87000	0·843	33⅓	27·0	32·5	55·5	50·5	135	108	351	equal power distribution
	60 Rev/min											
	Single	113000	0·621	100	60·0		48·5		229		229	
1 000 000	Twin	106000	0·693			60·0		40·5		104	208	
	Triple	95000	0·770	41	60·0	60·0	40·5	35·0	85	53	191	optimum power
	Triple	95500	0·768	33⅓	60·0	60·0	38·5	36·0	68	62	192	equal power distribution
	100 Rev/min											
	Single	122500	0·570	100	100·0		34·0		130		130	
	Twin	121000	0·609			100·0		31·0		69	138	
	Triple	104700	0·699	41	100·0	100·0	30·0	26·5	55	34	123	optimum power
	Triple	105100	0·698	33⅓	100·0	100·0	28·5	27·5	46	40	126	equal power distribution

(A)—500 000 DWT SHIP

A ship of about this size is already on order and is therefore no longer purely speculative. For 60 rev/min (probably powered by turbines), a single propeller of optimum diameter (42·5 ft) would weigh 137 tons and require about 66 000 shp. The largest diameter that could reasonably be accommodated on this ship is 44 ft for which the optimum rev/min are about 56 requiring about 64 000 shp with a propeller weight of 141·5 tons. These propellers are for the moment outside manufacturing capacity as solid propellers, but if the need arises prior to the requisite increase in capacity, the conditions could be met with a built-up propeller arrangement.

Twin screws at 60 rev/min would be 36·5 ft diameter, weigh 71 tons each and require 69 500 shp. The largest reasonable wing propellers are 40 ft diameter weighing 81 tons each and requiring 66 700 shp at the optimum rev/min of about 50.

With triple screws at 60 rev/min the power required is reduced to a minimum of 61 200 when 50 per cent is taken on the centre propeller which would be 37 ft diameter weighing 67 tons, while the wing propellers would be 30·5 ft diameter and 32 tons weight. Using the largest reasonable diameters, the power is reduced to 57 400, 45 per cent of this on the centre at 38 rev/min, the propeller weighing 86 tons, the wing screws turning at 40 rev/min would weigh 49·5 tons each.

A further alternative is to distribute the power equally on the three shafts, enabling all power units to be of the same type and only requiring about 1 per cent more power than the optimum distribution.

At 100 rev/min (a reasonable speed for Diesel engines) a single propeller of 31 ft diameter would weigh about 90 tons and require about 80 000 shp (20 per cent more than at 60 rev/min). Twin screws would require about the same power but triple screws would only require about 70 000 shp. Only the single screw is outside the present capacity for solid propellers and even 90 tons is possible with some modifications to existing manufacturing facilities.

(B)—750 000 TON, 1 000 000 DWT SHIPS

Single, twin and triple screws were examined for both these ships at 60 rev/min, as a possible speed for use with turbines and at 100 rev/min, for use with Diesels. The possibility of using the largest possible propeller diameters was also considered. For the 750 000 ton ship these would require shaft speeds of about 53 rev/min for a single screw, about 42 rev/min for twin screws and about 34 rev/min for the triple screws with 41 per cent of the absorbed power taken on the centre shaft. The corresponding figures for the 1 000 000 ton ship are 49 rev/min for the single, 39 rev/min for the twin and about 31 rev/min for triple screws with 41 per cent of the power on the centre screw.

In the course of the investigations to find the optimum revolutions required when using propellers of the maximum allowable diameter, it was found that, for the best distribution of power between the triple screws, the same rev/min were required on centre and wing shafts.

In the case of equal distribution of power between the shafts, the rev/min required on the centre shaft were lower than on the wings; 31 and 36 for 750 000 tons; 27 and 32·5 for the 1 000 000 ton ship.

The triple screws are within the present manufacturing capacity for a one piece casting, except for some of those with limiting diameters. "Built" propellers would be of considerably greater weight than the equivalent "solid", and the increase in power requirement would probably be at least 2 per cent.

A summary of the results obtained from the investigations carried out on the three ships is given in Table II.

General Remarks

As already referred to above, there is an appreciable advantage to be gained by reducing the rev/min to below the present limit of around 80 per minute. This will, of course, mean that very large propellers may be called for in the very big vessels now contemplated, involving propeller weights beyond present manufacturing capacity if made as fixed pitch solid screws. However, the advantages are so great that the usual disadvantages of built-up propellers are more than offset by the high efficiency achieved. The comparative possibilities have been investigated for the centre propeller of the triple screw ship of 1 000 000 dwt when 31 600 shp is to be transmitted at 60 rev/min. A conventional

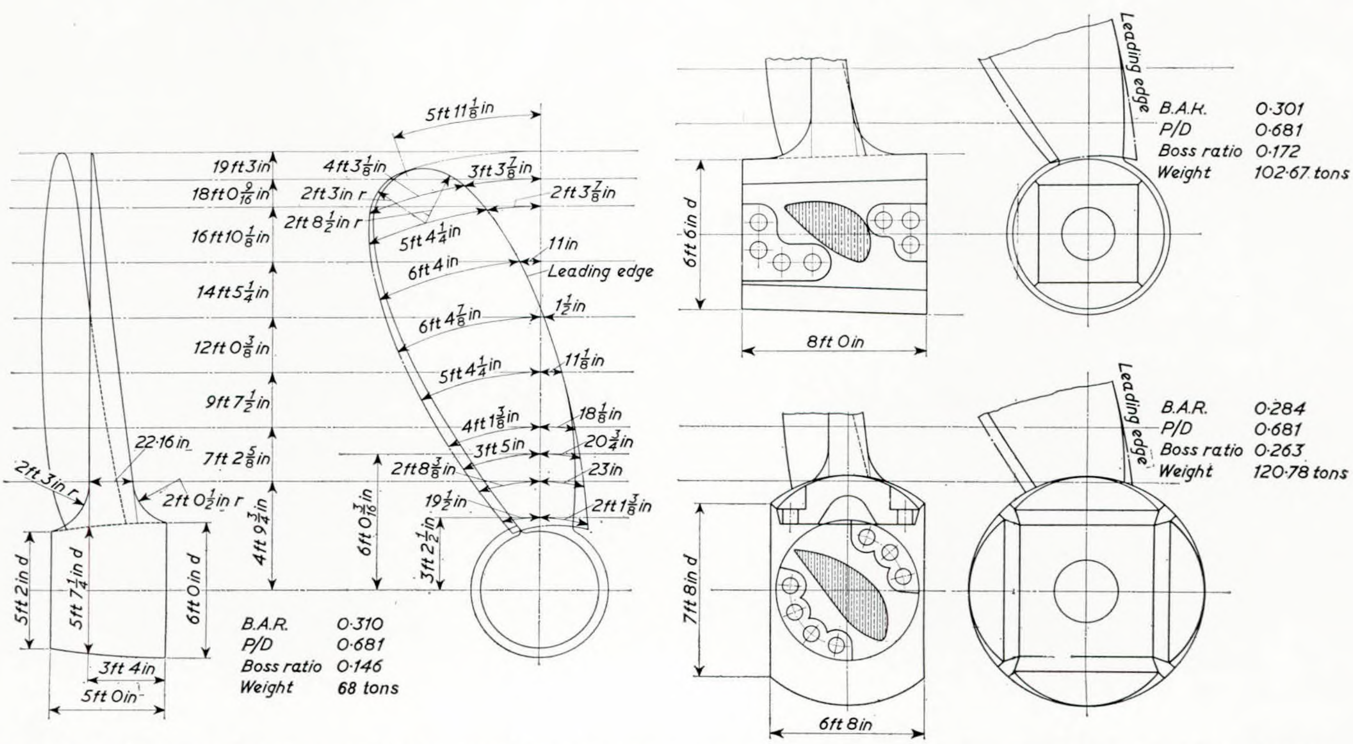


FIG. 10—*One million dwt tanker centre line propeller, triple screw arrangement. Solid and built-up propellers designed for* 31 600 *shp at* 60 *rev/min. Material—Nikalium. Scale—1:100*

built-up propeller and an improved design having a smaller boss/diameter ratio are shown in Fig. 10 alongside the normal "solid" design, which in this case is still within the present manufacturing capacity.

One of the advantages of the triple screw arrangement is the improved manoeuvrability gained thereby. It was emphasised in Part 1 of this paper that this will be a vital consideration in these mammoth ships. This advantage in steering can be further improved by fitting controllable pitch propellers on the wing shafts. These not only give instant change in propeller thrust by direct control from the bridge, but can also dramatically improve the stopping characteristics of the system (Reference 4). Contrary to what might have been expected, although these propellers are very large, due to the correct selection of rev/min, the blade-area-ratio is small, and the mechanical problems concerned with the pitch change mechanism are by no means insurmountable. A typical design for the wing screws of the triple arrangement for the 1 000 000 ton tanker is shown in Fig. 11 and an arrangement of the mechanism in Fig. 12. These propellers were also designed for 31 600 shp and 60 rev/min.

Transverse propulsion units in the bow undoubtedly give an additional aid to manoeuvrability although it is appreciated that the effectiveness of t.p.u.'s is reduced as the ship speed increases. Consideration should therefore be given to a fore and aft tunnel for the inflow, perhaps through the bulbous bow, to transversely disposed discharge tunnels. A uni-directional impeller drive could then be placed on the fore and aft centre line of the vessel abaft the athwart ship tunnel driving the impeller which would, of course, be situated in the inflow tunnel. This arrangement would make the unit effective at manoeuvring and higher speeds, the direction of thrust being controlled by butterfly valves controlled from the bridge.

This section of the paper covers a wide spectrum of different propeller arrangements and could be used as guidance for a variety of propulsion machinery. For the "Megaton" vessel which is the subject of this paper, some selection must be made for detailed consideration of suitable machinery and a practical and attractive triple screw machinery arrangement has been worked out in the next section of the paper. The power is equally divided between the three shafts, i.e. 31 600 shp at 60 rev/min throughout.

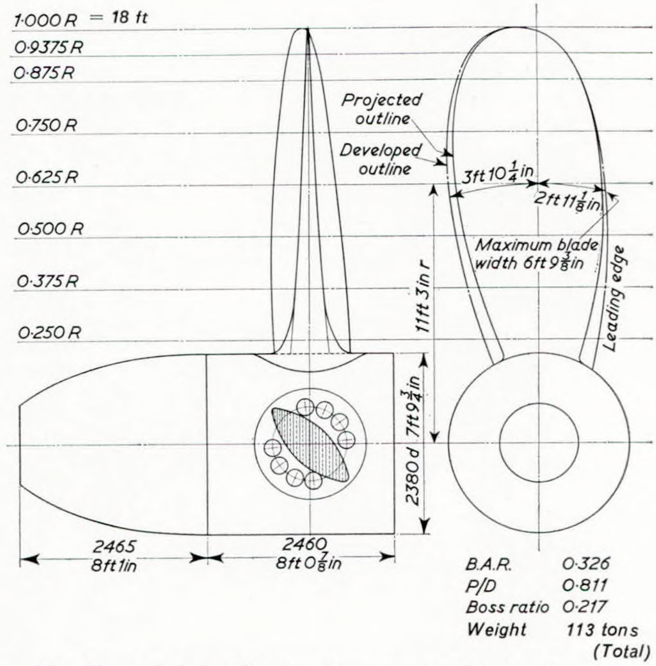


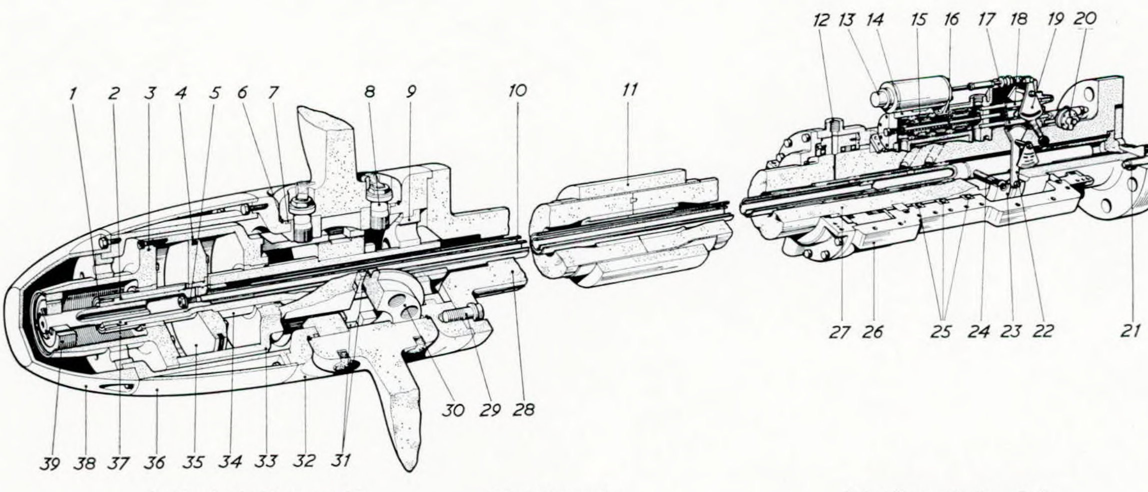
FIG. 11—*Typical design for the wing screws of the triple arrangement for the one million ton tanker*

PART 3—MACHINERY INSTALLATION

This part of the paper considers the selection of the propulsion machinery available for a million ton tanker. The increased size would take certain features of the installation close to their design limit and the more important of these are discussed.

Type of Machinery

Recent studies (Reference 5) have indicated that the optimum speed of a half million ton deadweight tanker is 16 knots, a figure which has also become established during the recent rapid



1) Vernier locking assembly
2) Thrust collar
3) Cylinder head
4) Piston ring
5) Circulation valve
6) Sealing ring
7) Bearing ring
8) Blate bolt
9) Piston rod bearing
10) Pressure tubes
11) Shaft coupling
12) Head tank connexion
13) Pneumatic actuator
14) Valve casing
15) Valve
16) Valve spindle
17) Pitch feed-back rack
18) Command rack
19) Operating lever
20) Feed-back transmitter
21) Pitch locking device
22) Feed-back lever
23) Feed-back sliding ring assembly
24) Carrier pin
25) High pressure seals
26) Oil transfer box casing
27) Oil transfer box shaft
28) Tail shaft
29) Tail shaft bolt
30) Crank pin ring
31) Sliding blocks
32) Hub body
33) Servo cylinder
34) Piston rod
35) Servo piston
36) Cone
37) Driving sleeve
38) Cone cap
39) Jacking screw

FIG. 12—*Controllable pitch propeller*

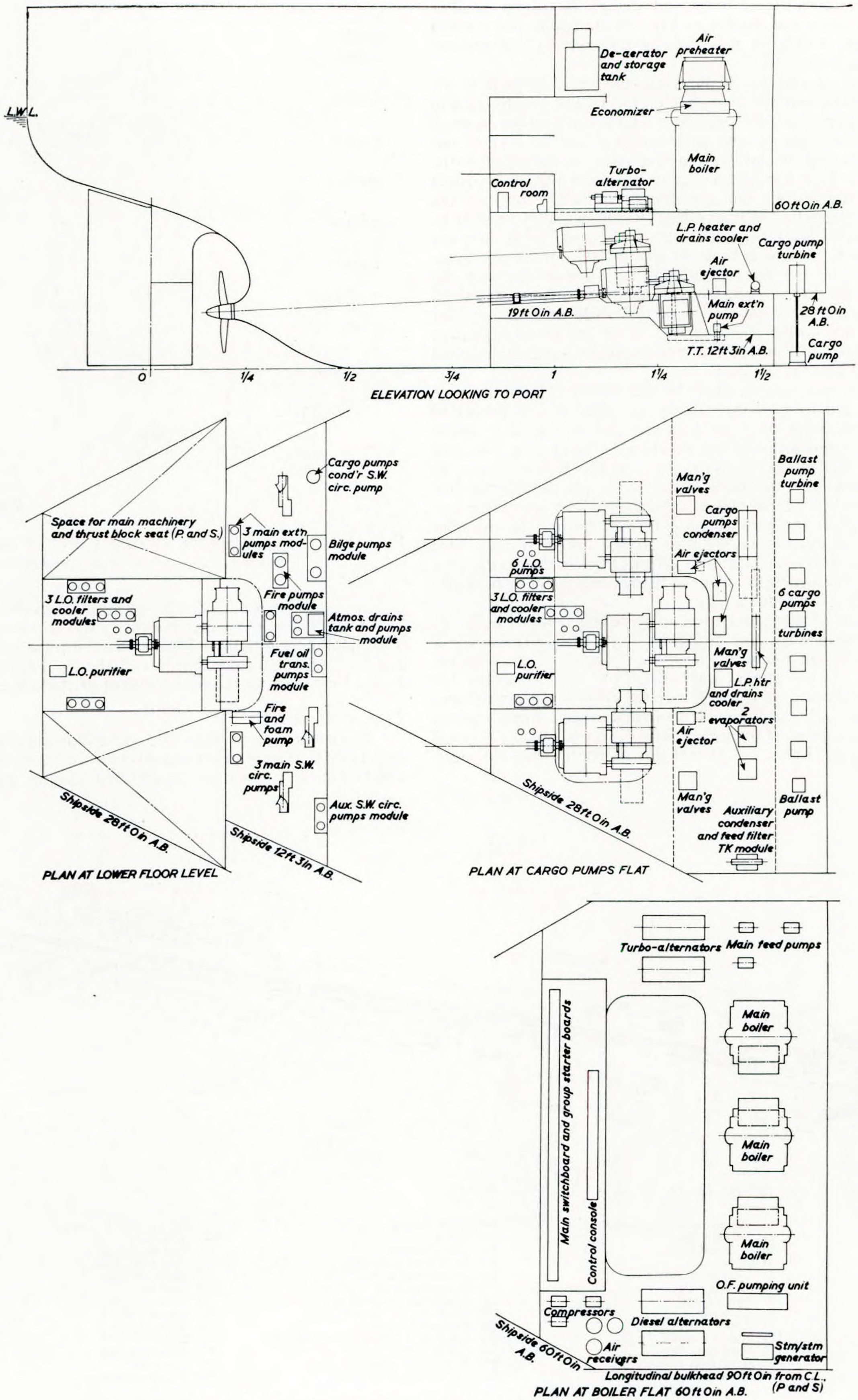


FIG. 13—*Machinery arrangement for triple screw installation*

increase in size up to, and beyond, a quarter million tons. Assuming this is still valid at a million tons then a delivered horsepower of between 95 000 and 130 000 is required which can only be met by either a triple screw or twin screw installations. (Fig. 8).

In this application reliability and safety must receive greater emphasis than usual so that equipment is chosen from designs which are already available and proven. When making this selection it has become common to compare different installations or consider additional equipment by making cost calculations. There are, however, factors which are not allowed for and are particularly important when the value of the ship as a whole, is compared with the cost of the machinery installation. Continued availability of the vessel and the possibility of influencing insurance premiums with designs which give safety a proper emphasis cannot be calculated. The relative stability of fuel prices and increasing capital cost of equipment has meant that the contribution to the running costs represented by fuel has been reduced and become less important. Maximum efficiency in terms of fuel consumption would, therefore, not be attempted because it is not always accompanied by maximum reliability. An equally important consideration is that this high powered plant should be designed to be within the capability of the staff who operate and maintain it. Unless this is achieved availability will be reduced due to maloperation and the design performance will not be obtained. In recent years there has been a tendency to ignore this and equipment has been installed which has given no benefit or actually reduced the standard of watchkeeping by diverting the attention of the engineers from the more conventional but important features.

For a twin screw installation only the steam turbine plant would be considered because the power at 100 rev/min is above the service rating of current slow speed Diesel designs. The saving in power and improved manoeuvrability of the vessel make a triple screw installation of interest. Referring to Fig. 8 the reduction in the range 60–90 rev/min is not less than 10 000 shaft horsepower. This also indicates that the optimum share in power between the screws is when 35 per cent to 40 per cent is delivered on the centre shaft and, as the curve is flat in this region, three identical sets could be used. Steam turbine plant is predominant in large tankers and has the additional advantage of providing boilers for the cargo pumping duty. It also allows a range of rev/min to be considered highlighting some of the more critical design features. Slow and medium speed Diesels have not been adopted because of the number of cylinders required. The drive for the cargo pumps is an additional disadvantage until proposals for connexions taken from the main engines themselves have been proved in service. The progressive improvement in the fuel economy of gas turbines suggest that they may provide an alternative propulsion plant in the future. The trend towards using better quality oils in main engines as residual fuels become scarcer reduces the cost differential arising from the different grades burned. Acceptability of these designs will be improved when methods of eliminating the entrained salts from the inlet air and treatment of the fuel oil before combustion have been developed.

Content and Layout of Plant

The layout of the three triple reduction steam turbines delivering 31 600 shp at 60 rev/min is shown in Fig. 13. This gives specific loads in the stern bearing slightly beyond existing values but this is considered acceptable with the designs proposed. Below this speed propeller weight increases more rapidly for a given reduction in revolutions (Fig. 14) and the associated shafting and reduction gearing also become increasingly expensive. Six cargo pumps are arranged in line in an area of flat of bottom as far aft as the ships lines allow. The boilers are placed forward and the centre main engine located under this flat. Shaft heights are chosen to give adequate propeller immersion in the ballast conditions and the wing engines raised to suit the lines of the ship. They are located at the aft end of the boiler flat so that overhauling access is available but far enough forward to allow shaft withdrawal inboard. Equipment has been arranged to use the space up to the main deck and give the shortest engine room. A control level has been created with a room looking out on to the boiler fronts and across an operational platform carrying the more critical machinery items such as generators and feed pumps.

Steam for the plant is provided by three equal sized front fired boilers with outlet conditions of 900 lb/in²g and 950°F at the superheater with 15 per cent CO_2 at the furnace outlet and a final undiluted uptake temperature beyond the rotary gas air heaters of 280°F. A steam/steam generator is used for contaminated services which are mainly for heating requirements, as hydraulic deck machinery is considered appropriate. The three main engines are cross compound units with astern elements in the L.P. casing.

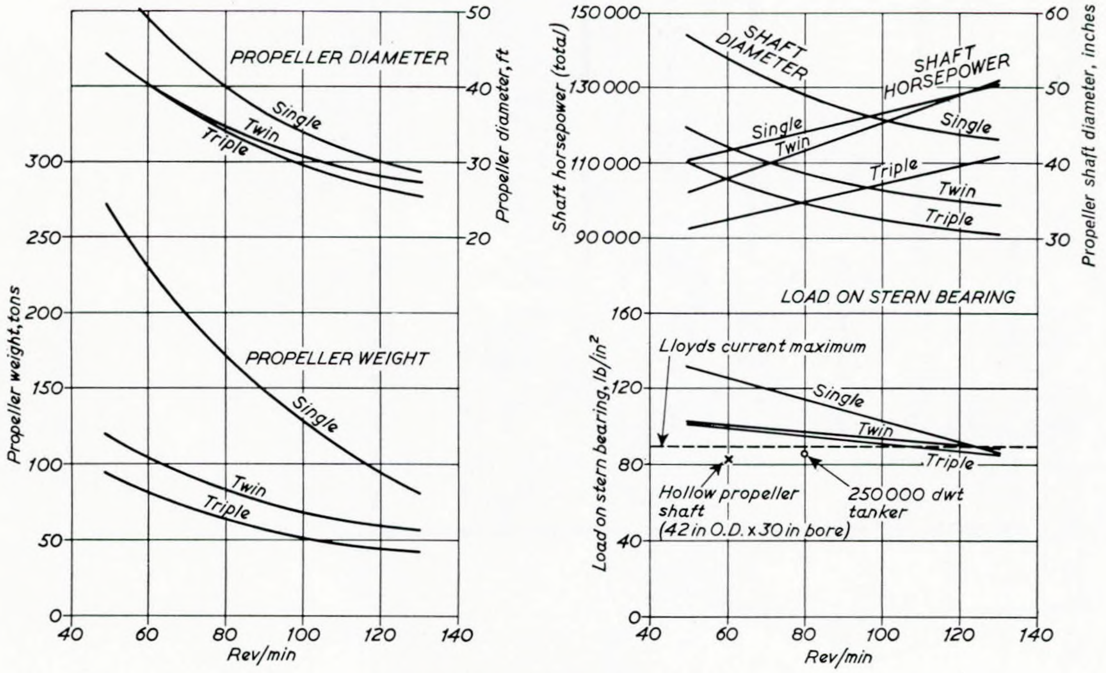
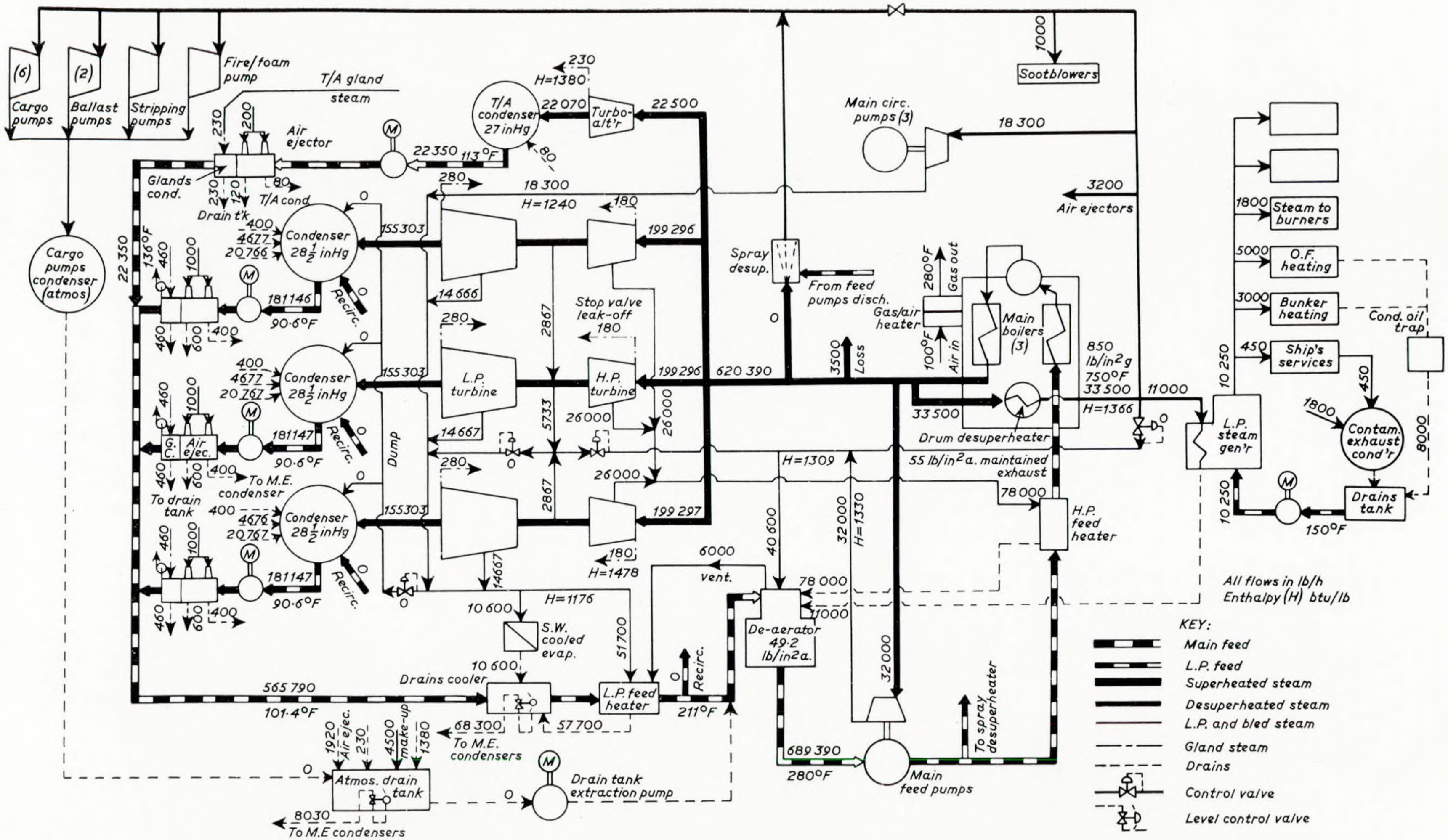


FIG. 14—*Stern gear characteristics*



STEAM CONDITIONS	
Superhtr. outlet	900 lb/in²g/950°F
Turbine inlet	850 lb/in²g/940°F
Main condensers vac.	28½ in Hg
Sea temperature	75°F
N.B.S.R.	5·41 lb/shp h
Boiler efficiency	89·75 per cent
Percentage CO_2	15
Uptake temperature	280°F
H.C.V. fuel	18 500 btu/lb

ELECTRICAL LOAD	
Hull	466 kW
E.R.	2034 kW
Total	2500 kW
No. altrs.	1

EVAPORATION LOAD	
Make up main system	4500 lb/h
Make up L.P. system	1800 lb/h
Total storage	1170 lb/h
Total	7470 lb/h
	80 ton/day

LB FUEL/LB STEAM	
Sup. steam	0·0658
Desup. steam	0·0590

FUEL RATES	
	45 080 lb/h
	484 ton/day

SPEC. FUEL CONSUMPTION

0·47 lb/shp h

FIG. 15—*Heat flow diagram*

Each has its own condenser circulated by a steam driven pump drawing from a sea tube with a faired entry which is not dependent upon scoop action because the characteristics of the boundary layer at this position on such a long ship would be unpredictable. The case for scoop circulation appears to be based primarily on the standby facility it provides as it is doubtful whether the economy in circulating power within the machinery space is greater than the increased propulsive power required for the scoop action. The feed cycle is shown in Fig. 15, consisting of air ejectors, gland steam condenser, one L.P. feed heater, a high level de-aerator and a single stage of H.P. feed heating. After the two extraction pumps and single steam air ejectors serving each condenser the systems become common at the L.P. heater. Three feed pumps are arranged so that two are running with the third standby. There are two condensing turbo alternators each capable of carrying the normal service sea load and together covering the manoeuvring and cargo pumping conditions. Standby capacity is provided by two Diesel generators with a further emergency generator capable of supplying emergency lighting, lubricating oil pumps for steering gear and turning gear. Both turbo generators will be run at sea, efficiency being maintained with an inlet nozzle control. In the event of one generator failing the other would accept the system load to avoid a blackout condition and a Diesel generator run up automatically as standby. A full size motor driven two speed F.D. fan is provided for each boiler and fuel oil is supplied from a common pumping and heating plant consisting of three heaters to give full capacity and three pumps which are arranged to have two working with the other standby.

Six steam driven cargo pumps of 5500 tons capacity are driven by multi-stage turbines receiving de-superheated steam from sprayers external to the boilers and exhausting to an atmospheric condenser. The discharge time is 48 hours and the steam requirements within the capability of the boilers, which are sized for normal service power plus cold tank cleaning. Inlet steam conditions of 800 lb/in^{2}g and 750°F are chosen to give an exhaust temperature which is acceptable to the materials in the condenser. The cycle is arranged so that condensate is returned to the main system through an atmospheric drains tank.

A fuel rate of 0·47-lbs per shaft hp hour at normal service has been obtained by using static elements such as heat exchangers and avoiding high initial steam conditions or very low uptake temperatures from the boiler. Advantage has been taken of these features up to the limit considered satisfactory from a reliability point of view. Engine driven auxiliaries have not been considered because of the increased mechanical and operational complexity. Salt water circulated evaporators are used to avoid the additional connexions required by condensate circulated units. The performance of the main engines and end conditions determine the efficiency of the cycle and small changes in the details of the system only make marginal improvements in the oil fuel rate. Realistic margins have been used in the calculations to obtain rates which could be maintained in service and not briefly achieved in a consumption trial.

Machinery units are chosen so that failure leads to either a reduction in power or a closing down of one of the main engines. For larger units such as F.D. fans and main circulating pumps where the provision of standby capability is expensive, one full sized unit is being provided on the basis that there are standby engines. On other occasions for feed pumps or fuel oil pumps where the duties tend to lead to a lower reliability, common services with standbys have been provided. In certain cases such as lubricating oil pumps or extraction pumps where the cost of failure is high or the provision or cross connexion standby arrangements comparable in cost to the unit, standbys have been provided for each working unit. Systems would be as simple as possible avoiding unnecessary cross connexions and other complications with a view to reducing the number of pipes from the unacceptably high levels found in current installations.

A centralized control room containing a console, the main switchboard and a group starter board is required to operate this large installation. The room is located in the centre of the installation so that it is possible to view the boiler front, the main engines and vital auxiliaries such as generators and feed pumps during manned operation. For the initial service of the vessel it is intended that there should be two watchkeepers, one senior and a junior but that the controls and instrumentation complex be designed so that at a future date periods of unmanned operation could be considered. During this time a duty engineer would be nominated to tour the installation once per watch reporting to the bridge before and after the inspection. Remote control of the main engines from the bridge would be provided to make this possible and to improve the response of the ship to bridge orders during manoeuvring. The exhaust steam from feed pumps, and the main circulating pump give a high enough standby load so that the turn-down during manoeuvring would be achieved within the range of the burners. The number of burners would be chosen to avoid flame impingement and turndowns beyond the standby requirement achieved by the remote removal of burners. Auto synchronizing would not be required and instrumentation limited to that required for manoeuvring and start up of the plant except for a number of chart recorders and an alarm panel display. The log taken once per watch would be limited to parameters which have a significant effect on safety, maintenance and performance and these would be used to review the operation of the plant.

Structure in Engine Room and Shafting System

The machinery installation is unlikely to be successful unless the structure at the aft end is carefully considered and exciting forces from the propeller minimised. The flow of water into the propeller disc should be studied to see if it can be improved by making local changes in the lines. A review should also be made of the general arrangement of stern gear to ensure that clearances are as large as possible to reduce inter-action between the propeller and adjacent parts of the ship. The propeller itself should have an even number of blades because the larger variations in torsional and axial excitation they cause are more readily accommodated than vertical forces transmitted through the stern gear. Structure in the space has to be arranged to avoid local vibration and provide a satisfactory platform for the propulsion machinery by distributing the loads. The principles are illustrated in Fig. 16, which shows arrangements suitable for a single screw vessel.

A major consideration is the continuity of longitudinal strength through from the main body of the vessel by bringing the longitudinal bulkheads from cargo tanks into the machinery space to form side tanks. Longitudinal sections on the upper deck and the side shell are also continued and tied together by heavy vertical web frames forming a ring structure in the side tanks and within the space itself. These frames carry the double bottom structure and the flats, including one flat which has been taken into the aft end to carry the steering gear. The double bottom is a box structure carrying heavy longitudinals near the centre line as well as the normal transverse floors. These run aft until they meet the shell and provide a platform for the shaft bearings meeting a similar box structure in the lower peak. The platform in way of the main machinery itself is made very stiff with transverse members tying into the web frames and longitudinals meeting double bottom members. The thrust block which is centre line mounted is located near the gearbox seat. There are no journal bearings in the block and the stiffness of this together with associated structure is checked to ensure that the first mode axial resonance is at least 25 per cent away from the maximum speed. Torsionals for the complete system are also checked to ensure acceptable stress and tooth load variations.

At the aft end the longitudinals on the shell run horizontally into brackets on the forward bulkhead of the after peak. Double spaced longitudinal wash bulkheads are cantilevered from this to carry the steering gear flat. The direction of the sections is also changed on the aft side of the peak bulkhead so that they run parallel to the stern outline until they meet the shell. This type of structure is used until the width is too narrow and the lower parts of the after body revert to the box structure found in the double bottom. Lines in this area need to be a compromise between hydrodynamic and practical considerations to ensure that the stern frame is tied into the structure so that loads from the aft bearing can be properly transmitted. Access to shaft couplings and oil seals must also be provided. In the case of a multiple screw installation the tunnels into the bossing are in the form of a cylindrical pod to satisfy these requirements.

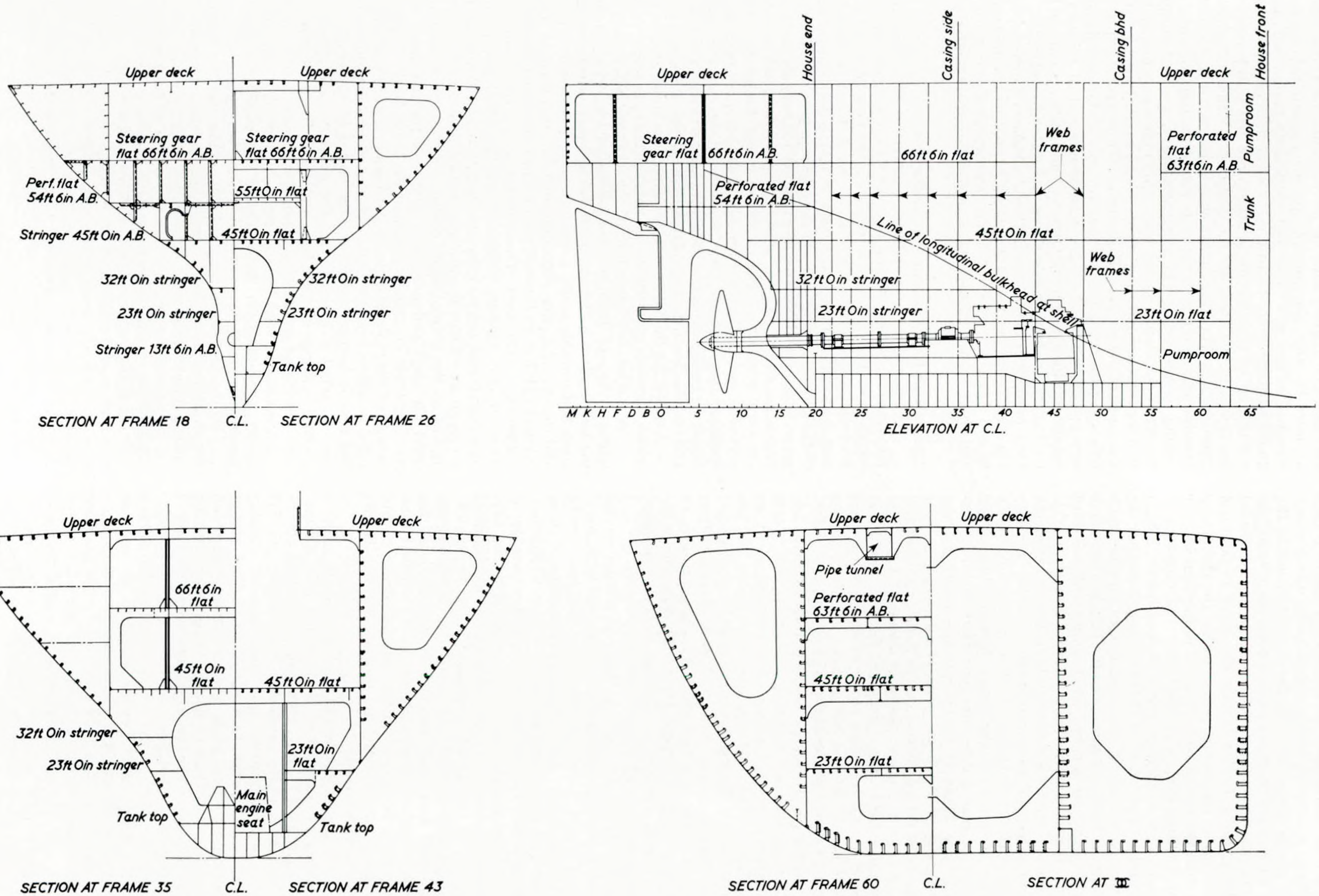


FIG. 16—*Machinery space structure for a large single screw tanker*

In the triple screw installation the machinery occupies only part of the space available in the length of the ship in which it is located. Cargo tanks can be brought around the sides of the pump room into the side tanks in the engine room and across the aft end above the shafting tunnels. Fuel oil can be located in the double bottom tanks. There are two funnels sited forward of the accommodation block which contains an emergency service centre. A raised control tower for the navigation of the ship is located near midships.

Spacing in the shafting system is arranged to give the maximum flexibility consistent with bearing loads and shaft whirl. A check is made to ensure that the fundamental whirling frequency is at least 50 per cent above the maximum running speed. A single white metal bush located in the stern frame and well spaced journal bearings form the basis of the system. This ensures that any deflexions in the hull can be accommodated except in the immediate area of the gearbox. The high influence numbers are at the main wheel bearings which are mounted on the stiffest part of the propulsion platform to isolate it from changes. Hardened gears are preferred because they provide a margin on tooth loading to accommodate any misalignment there may be in the secondary reduction. The calculated alignment is achieved using gap and sag methods and a final check made on the bearing loads using a hydraulic jack. (Reference 6). An allowance is made in the cold alignment for thermal rise in way of the gearbox up to the running condition. A correction is also made in the attitude of the thrust collar face to ensure an even distribution of load under power. Results taken at main wheel bearings indicate that calculated influence numbers are only valid for one condition and jacking the shaft changes the effective centre of support, altering the local characteristics of the system. Careful positioning of the gearbox is, therefore, vital as no final check on loads can be made.

Stern Bearings

One component near its current classification design limit is the stern bearing which can be demonstrated by considering a series of solid shaft designs based on Lloyds Rules. Fig. 14, shows the variation of specific load with revolutions on an L/D ratio of 2:1, for the four bladed fix pitch propellers on a triple screw 16 knot million ton tanker. Powers are based on an equal share between the shafts and a series of alignments calculated to establish the loads. Bearings of this type are at present limited to a design specific load of 90 lb/in²g, a figure reached at 105 rev/min, but, as the increase with reduction in rev/min is gradual, the 60 rev/min chosen is considered reasonable. Propeller weights for this installation are 80 tons with a diameter of 40 ft. These characteristics are within projected manufacturing facilities for a pit capable of over 80 tons and 35 ft diameter when the periphery is cut to accommodate the tips of the blades. A build up propeller could also be used consisting of four separate blades attached to the centre boss.

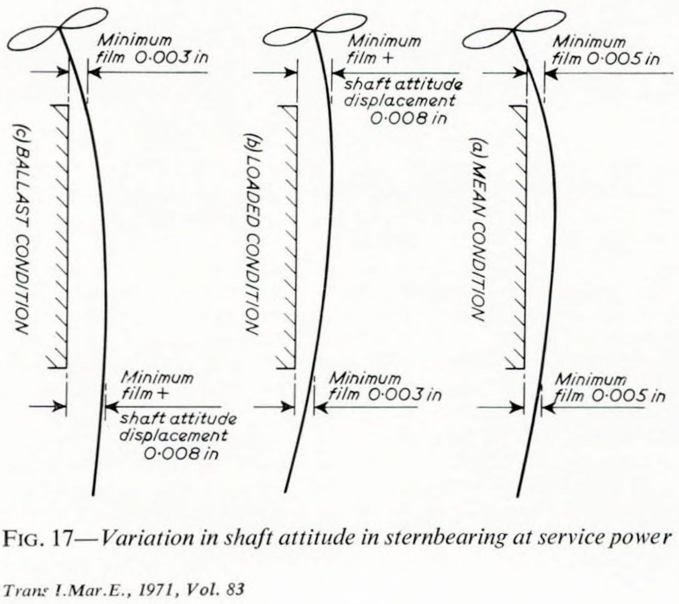


FIG. 17—*Variation in shaft attitude in sternbearing at service power*

A more detailed investigation is required on an actual installation because of the variation of attitude of the shaft in the bearing caused by the change in the centre of thrust of the propeller during operation. This is illustrated in Fig. 17, which is a calculated alignment for an average service condition (A) on a typical quarter million ton tanker. Variations about this mean take place in the loaded and ballast conditions shown in (B) and (C) respectively which are caused by eccentric thrust that is always above the centre line of the shaft but its position and value change. Each case shows the calculated variation of oil film thickness for an oil inlet temperature of 110°F. The first step in this analysis is to establish the line of the tail shaft through the bearing and then to calculate the height it will be above the whitemetal in the running condition. To do this the length of the bearing is split up into sections, each treated as a journal bearing with an oil film equal to the mean gap. It is assumed that 85 per cent of the length is load carrying to allow for the end leakage. The load carrying capacities for various heights above the whitemetal are calculated (using Reference 7) to meet the requirements of the load carried. This analysis is carried out for a number of oil temperatures and the minimum oil film thickness at the ends of the bearing obtained. A graph of minimum oil film thickness against temperature can be plotted and after allowing a 50 per cent margin between the degree of filtration and the minimum oil thickness, the maximum permissible oil temperature at the bearing inlet can be established. For the quarter million ton tanker being considered this indicated 50 micron filtration at a maximum inlet temperature of 110°F. Further calculations at half speed gave a figure of 25 microns. A forced lubrication system is necessary to obtain these conditions and includes filters, coolers and pumps as shown in Fig. 18. The method used in making this calculation is approximate but illustrates the factors involved, including the significant effect of change in attitude of the shaft during operation. It highlights the need for research into design methods for this type of journal bearing so that a more accurate assessment can be made of existing bearings and fuller investigations made into alternative approaches.

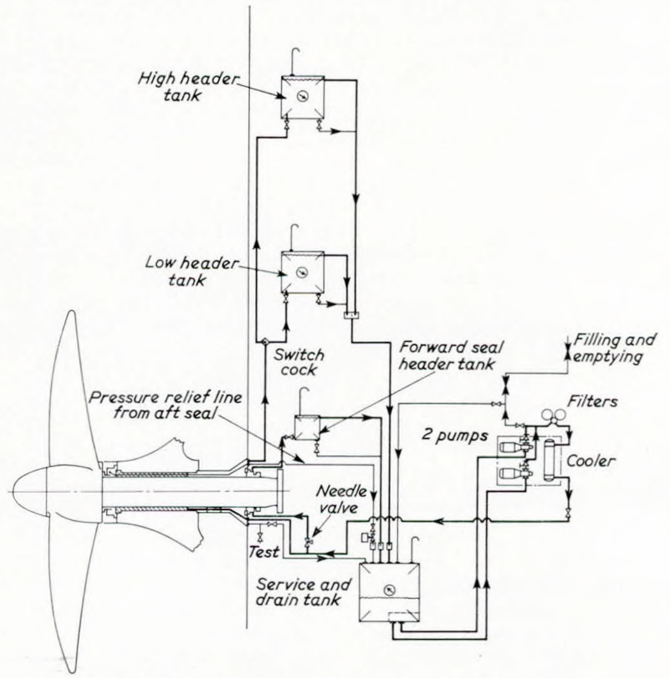


FIG. 18—*Sternbearing forced lubrication system*

The previous discussion has been confined to normal service speed but manoeuvring and turning gear operation where boundary lubrication conditions may exist are also important. An oil with viscosity of 0·6 poise at 140°F is used to increase the oil film thickness so that variations in load distribution can be accommodated. When higher specific loads are used oil could be injected into the bearing to float the shaft at low revolutions and avoid wear. The total system would, of course, be chemically treated and cleaned in the same way as the main engine supply to minimize the dirt entering the bearing.

The arrangement in Fig. 18 shows a solid tail shaft with the propeller attached by interference fit to avoid stress concentration at a keyway. A single steel bush is located in the frame with a design interference of the order of 0·003 in. Fitting loads of 100 tons are common on smaller ships but this allows a maximum in interference of under 0·002 in for the proposed installation which is too near the limit of existing machining techniques. A stepped stern frame bore with five fitting bands is used and a thin copper coating applied before installation.

Shaft seals require further development although the designs discussed in the paper do not exceed the rubbing speeds quoted by suppliers for cooled assemblies. A limit of 16·4 ft/s or a maximum diameter of 29·5 in is sometimes recommended when the seal is uncooled which goes up to a maximum of 32·8 ft/s when cooling is applied. New materials and improvements in surface treatment of the sealing element should raise the present limits and improve reliability. The forced lubrication system is arranged so that it does not pressurize the seals and as a further precaution two header tanks are used to allow for the differences in draught between the loaded and ballast conditions. Changeover is covered by an alarm in the control room console to ensure that the pressure difference across the seal is kept at an acceptable value.

The stern bearing design is based on the best features of existing installations including the introduction of forced lubrication. This area requires further development and new designs put to sea in smaller tonnage to establish performance. The first step is to consider shorter bearings approaching the optimum L/D ratio of 1:1 with a design specific load nearing 150 lb/in²g. This would reduce the variation of oil film thickness in the length of the bearing and improve its operating condition. Tilting pads might also be considered to distribute the load along the bearing face. Shell type bearings split at the half joint could be used and ways of obtaining access so that the bottom half can be turned or drawn out for inspection investigated. This would avoid some of the complex proposals available at present for the inspection of stern bearings without drydocking the vessel. A method of reducing bearing specific load is to use a hollow shaft which on the present installation gives a new point in Fig. 14 when the wall thickness is 6 in and the outer diameter 42 in. A flanged connexion is proposed and the propeller arranged in such a way that overhang is minimized and a muff coupling adopted at the forward end to allow tail shaft withdrawal aft.

CONCLUSION

To keep the paper to manageable proportions the problem has been tackled as though a design based on sound practice was required immediately. This means that the solution for the 1 000 000 tons vessel is a triple screw system powered by steam turbines with the same power on each shaft. If the bearing load on the wing shaft can be allowed to rise to 120 lb/in²g, then the use of controllable pitch propellers on these shafts would be an attractive aid to the overall safety and economy of the ship.

If there is to be a long period before the million ton ship is built, other systems may be developed to the stage where they may be used safely in such a vessel. For example, the steam raising plant may be atomic powered or the efficiency of twin screw systems may be improved by the use of ducted propellers. If the orthodox systems had encountered serious difficulties, these and other untried methods would have required closer study.

For smaller giants it is hoped that the information given in the paper will be useful in considering twin screw as well as triple screw propulsion, and the possible use of Diesels, gas turbines or a mixture of power units instead of steam turbines. When the many variations in size, speed and ship type are taken into account, it would be unrealistic to expect a unique solution to the problem of propelling such very large tankers.

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Discussion

MR. R. COATS, M.I.Mar.E., confined his remarks to the propulsion machinery aspect of the paper. He had no quarrel with the authors on the choice of steam turbine machinery, which was eminently suitable for the purpose, and with which no extrapolation beyond present knowledge and experience was required. He did, however, draw attention to another alternative which offered the same reliability together with a significant improvement in fuel consumption, which was the use of steam and gas turbines in a combined cycle arrangement. This scheme had been worked out for a triple screw installation of 90 000 shp, but this was not the limit of possibility with the arrangement.

The wing shafts would be driven by heavy duty gas turbines of the simple cycle, non-regenerative type, as described by White*, producing 26 000 shp each at an ambient temperature of 80°F. The centre shaft would be driven by compound steam turbine machinery of 38 000 shp, taking steam from two waste heat boilers accepting the gas from the gas turbines at about 950°F, plus additional steam from an oil-fired boiler rated at 100 000 lb/h, but operating at a lower rating at full gas turbine output. The arrangement was illustrated in Fig. 19.

This offered a fuel rate of 0·4 lb/shp/h when burning residual fuel in both gas turbines and fired boiler, at the maximum output of 90 000 shp. When operating without supplementary firing, at an output of 81 000 shp, the fuel rate improved to 0·38 lb/shp/h, and in fact the fuel rate curve was comparatively flat down to 55 000 shp (Fig. 20) and reached a minimum of 0·378 at 70 000 shp.

The steam turbine would have astern elements in the L.P. turbine only. The gas turbines could either be in association with c.p. propeller as suggested in the paper for the steam turbine proposal, or could use hydraulic reversing transmission within the gearbox.

* "Design and Development of a Marine Gas Turbine", A. O. White, IMAS proceedings.

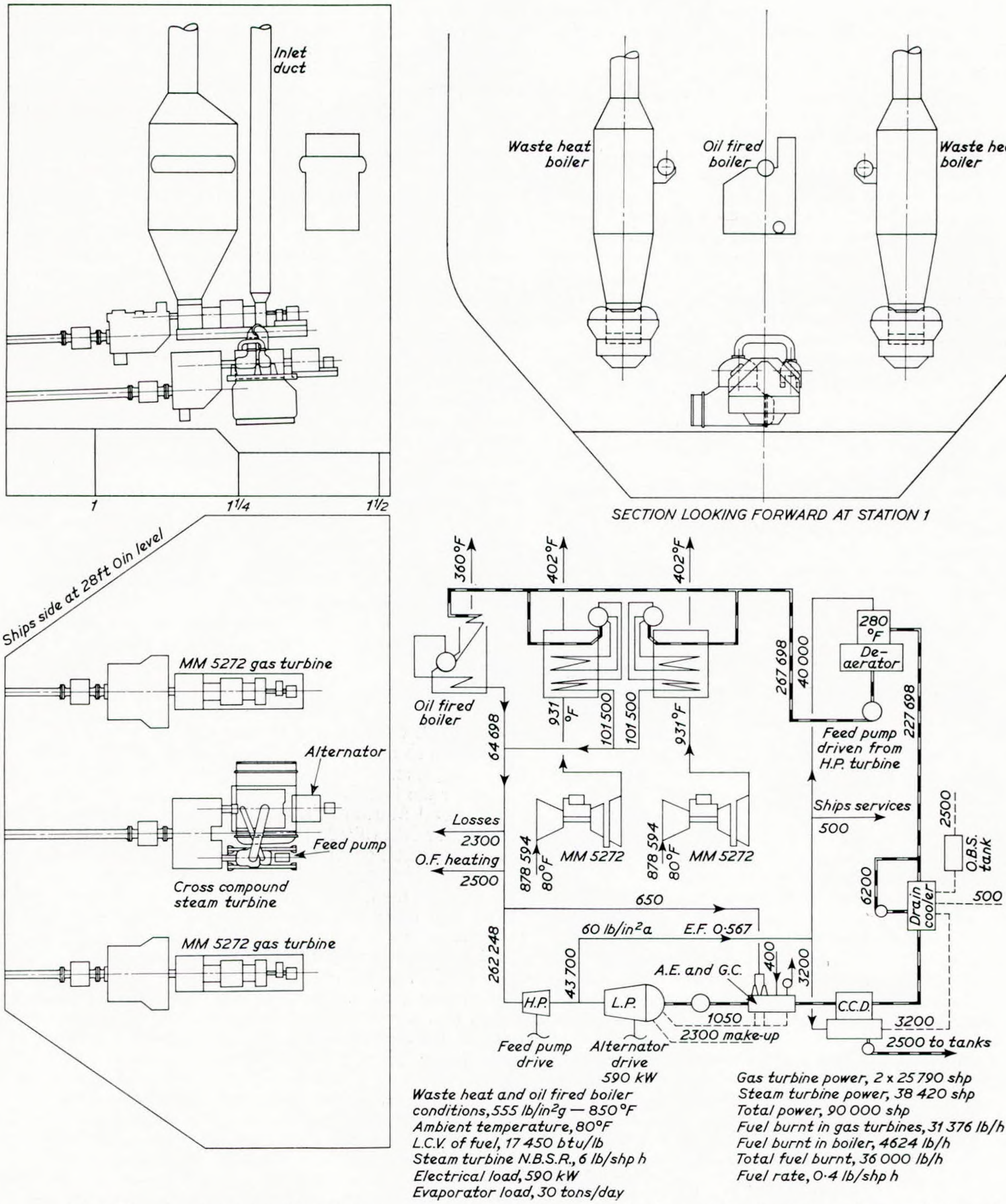


FIG. 19—*Arrangement of machinery and heat balance for 90 000 shp combined steam and gas turbine cycle*

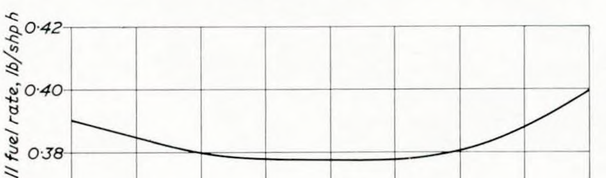


FIG. 20—*Variation of fuel rate against power for combined steam and gas turbine cycle*

The advantages were, firstly, that it was an efficient and simple system. The steam conditions for the steam turbine were moderate at 555 lb/in²g/850°F, and the feed cycle parts were kept to a minimum.

Secondly, continuing with the emphasis given to the safety aspects and availability of the main machinery in the body of the paper, the system proposed gave a very substantial back up of main body propulsion items against breakdown of any item, particularly if a supplementary firing facility were incorporated in the exhaust gas boiler design.

For instance, if one gas turbine were out of action the total power available with supplementary firing on one boiler to maintain the steam flow to the steam turbine would be 64 000 shp.

If both gas turbines were out of action and both exhaust gas boilers were supplementarily fired, the full steam turbine output of 38 000 shp would be available.

If the complete steam turbine set were out of action the power available would be 52 000 shp.

Thirdly, the arrangement gave adequate availability of steam for cargo pumping and cargo heating from the fired boilers.

Fourthly, either gas turbine could be used for cargo pumping by disconnecting from the propulsion gearing and connecting to a lay shaft arrangement giving mechanical drive to these pumps.

Fifthly, and most important, such an arrangement would be expected to be cheaper in the first cost and considerably cheaper in installation cost.

Finally, considering the saving in yearly fuel consumption that would accrue by using this combination instead of the installation described by the authors, it would be found to amount to no less than £170 000 per year; or, putting it in another way, in the first six years the savings would pay for both gas turbines.

Mr. A. Norris, M.I.Mar.E., was sure that several shipbuilding firms would welcome the challenge of building these enormous ships and it was known that the Japanese were already willing to build the 500 000 tonners. There were constraints other than technical ability, however, and there were formidable natural limitations on draught even for some of the quarter million ton ships which were already in service. For the lowest transportation cost it was important to get the largest ship possible to terminals which were close to large centres of population; Rotterdam was an example of a desirable terminal since it served so many European countries. When the ship was discharging there were other major problems of pipelines and storage tanks ashore which must be provided for and which could be costly.

With those and other restrictions it was probable that the enormous ships would be obliged to discharge at off-shore terminals or a few specially selected land terminals which might well have to be connected by pipeline to a complex of remote refineries.

The sheer number of ships which would be required to meet the ever-growing need for oil might force use of these enormous ships to ease congestion at approaches to large centres of population, or compel legislation to keep them remote from such places. In committing a large capital sum to the building of such units for trading between specified deep water ports care would have to be taken that in a few years time the trade did not disappear; for example, massive oil finds in conveniently located positions for transport to Northern Europe might change the optimum ship size required. This might increase the investment risk in building large ships.

Table II in the paper listed propeller weights and desirable shaft speeds for the large ships. Despite the technical desirability of advancing the art to the stage where the heavy weights and low speeds were proven, it was necessary to be realistic and accept that it would not be proper to take the inherent financial risks involved in doing the work on these valuable ships, also to accept that for a long time to come the factors mentioned would enforce the use of plant in the 80 to 100 rev/min range. The paper properly took the view that steam turbine propulsion would probably be used for these large ships, although it mentioned that Diesel was a possibility. Although it was probable that Mr. Norris's employer would take that view for use in ships of about 500 000 tons; use of the triple screw would, with some unorthodoxy, make a Diesel installation feasible. Mr. Norris's first approach to this would be to put a super large bore Diesel on the centre screw. Each wing screw would be driven by a medium speed engine of about 1000 hp for each cylinder with the engine arranged at a low level and the gearing train arranged for a vertical displacement to suit the wing propellers located at the kind of level shown in Fig. 3. The medium speed engines would be geared to drive cargo pumps, either through the gearing or electrically, and this would avoid the use of massive boilers in the ship and make available a greatly increased power for cargo pumping and improvement in discharge time, given in the paper as 48 hours. This could give a blend of the slow speed Diesel to suit the truly conservative and the medium speed engines for those who were slightly more daring. The installation would in some ways resemble the Diesel plant in a small ship since there would be one large and two higher speed Diesels corresponding to the main engine and Diesel generators commonly used. Such a configuration would allow overhauling and repair to be carried out at sea and there would be excellent prospects of obtaining a high effectiveness from the ship because of this provision.

A possible steam plant was shown in Figs 13 and 15 of the authors' paper. This could not be considered in detail at present as so many variables were involved and so many design studies had been publicized on plant of the same unit power level that there was a very wide choice of detail which would require a further paper to identify the optimum.

Looking at Fig. 13 of the authors' paper, the machinery arrangement for the triple screw installation, a cargo tank space is shown aft of the engine room. This would not be desirable from the safety aspect even if accepted for classification, since it would be a partial reversion to the outmoded centre castle in which living accommodation was surrounded by potentially dangerous cargo tanks.

Whatever type of machinery installation was adopted for these very large ships it might well be necessary to build in a substantial power and speed margin into the plants in order to allow the ships to be operated to a schedule similar to that which was applicable to passenger ships. The thought of having two tankers each of one million tons arriving on the same day at a terminal was rather disturbing in view of the tankage involved and the ships would have to be programmed to avoid such an occurrence—or even worse, the situation of neither arriving.

Mr. G. Crombie felt that in a paper of such scope it was impossible to discuss every improvement which had been made in recent years.

Over the last five years a great deal of time had been spent by Turnbull Marine Design Company in research and development of a range of split stern bearings. Mr. Crombie's company felt some years ago that the removal of tailshafts and propellers from their working positions for maintenance or examination was wasted effort, time and money. It was felt that it should be possible to carry out an examination of the stern bearing and tailshaft while the ship was at loaded draught.

The suggestion of the authors was that new designs should be proved in service in smaller vessels before being considered for these large ships. Three bearings of Mr. Crombie's company's design had now completed six years' service and a further five bearings were on order. These bearings were fitted in 16 000 shp controllable pitch propeller-driven installations classified by Lloyds for Class I* Ice Conditions. The use of these bearings enabled tailshaft and bearing maintenance to be carried out while the vessel was afloat without moving the propeller or shafting from their working position. In the Mark I and II designs the vessel must be drydocked to remove the bottom half bearing and outboard seal but, again, this work could be done without removing either the propeller or shafting. However, the accessibility of these bearings enabled weardown measurements to be taken regularly and this, together with a shaft and bearing top half examination, might indicate that it was not necessary to remove the bottom half bearing.

With the latest design, the Mark IV, the complete bearing assembly, including oil seals, could be moved into the vessel while it was at loaded draught. The Mark IV bearing could be angled to get the best alignment, again with the ship at loaded draught. Two bearings of this design were currently being manufactured for installation in two 225 000 ton single screw tankers with 36¾ in diameter tailshafts transmitting 50 000 shp.

The splitting of the stern bearing enabled the tailshaft to be flanged at each end, thus eliminating the expensive muff coupling used with c.p. propellers. It also allowed propeller manufacturers to produce flange mounted propellers which were bolted to the tailshaft. This not only produced a cheaper propeller—for example, a 45 ton taper mounted propeller would be reduced to about 38 tons if constructed as a flange mounted propeller—but also reduced bearing loads and shaft bending moments and meant that there was no need to disconnect the propeller from the tailshaft at the time of survey.

In Fig. 17 of the authors' paper reference was made to the movement of the shaft within the bearing for different powers and draughts. Tests carried out during trials on two container ships fitted with the Mark I bearing indicated similar results to those given by the authors. As the shaft came up to speed at zero pitch it moved from the bottom of the bearing and ran at about 4 o'clock. As the full pitch condition was approached the shaft moved round the bearing and finished up running at about 1 o'clock.

Mr. Crombie wondered why the authors considered that these split bearings were of complex design since he would consider that drydocking a ship to remove large propellers and shafts into and out of white metal bearings for the mere inspection of shafts and bearings, particularly with c.p. propellers, was a more complex problem than the removal of a top half bearing whilst the ship was at loaded draught.

Mr. Crombie would be pleased if the authors would comment on these points.

Mr. P. J. Adolph, A.M.I.Mar.E., offered some comments on the section headed "Type of Machinery", which was primarily on page 175 of the authors' paper. In particular the authors had made one statement, as follows: "The relative stability of fuel prices and increasing capital cost of equipment has meant that the contribution to the running costs represented by fuel has been reduced and become less important". It was of course appreciated that this statement might have been written some time ago and at that time may therefore have been substantially correct. However, a considerable escalation in fuel prices had taken place, particularly in the last nine months, which had resulted in fuel playing an important part in the cost of running a vessel.

The present high price situation had been brought about by a combination of factors and while one school of thought tended towards a possible drop-off in the present price levels once these factors returned to normal, another school of thought considered that the recent considerable upsurge in domestic oil consumption —well in excess of industry's own forecasted volumes—would tend to maintain the present price levels indefinitely.

Consequently it must be considered that the cost of fuel was likely to play an important role in the overall cost of running the projected vessel and it was therefore prudent to give close attention to the type of fuel to be used in the projected million ton tanker. The authors had come out in favour of steam turbine propulsion and in that event no doubt crude oil would be burned in such a vessel. Several turbine-powered mammoth tankers were operating successfully already on crude oil and the cost of installing the additional equipment to burn crude oil in a new vessel was relatively low.

However, it must be admitted that there was one economic argument against the use of crude oil and that was that when burning Bunker C only the heavy end of the barrel of crude was being used and the whole of the light end of the barrel was available to the refinery to produce the more expensive products; for instance: distillate Diesels, petrol, paraffin, etc. When burning crude oil the entire barrel was being burned in the boiler and it was considered by some that the burning under those circumstances of the light end of the barrel that could otherwise be used to produce the more sophisticated and higher priced product was a rather appalling waste.

The question of bunkering very large tankers had been given a lot of consideration by the major oil companies in recent years. To take a typical case of a tanker running from the Persian Gulf to North-West Europe, and burning Bunker C, if the vessel bunkered in the Persian Gulf and in North-West Europe then the bunkers taken at the European end of the run would have had to have been hauled as crude from the Persian Gulf in the first place. Conversely, if the vessel bunkered in the Persian Gulf for the round trip then you would be shutting out cargo on the voyage from the Persian Gulf to Europe, because of the loading restrictions. Both of these types of bunkering were used depending on a number of conditions which could vary according to the season and the size of the vessel involved. However, in the case of crude oil burning no refined bunkers were required and the bunkers could be taken, in effect, from the cargo. On the return voyage from Europe to the Persian Gulf when the vessel was in ballast it should be borne in mind that a quantity of crude oil was recovered during the tank cleaning process, and after a minimum of treatment, this crude oil could then contribute towards the return trip reserve bunkers.

Mr. Adolph also commented on the authors' statement regarding the possible use of gas turbines. Up to the present time one of the things that had held back gas turbines in the marine propulsion field had been the requirement to use a distillate quality fuel. The first speaker in the discussion (Mr. R. Coats) had proposed the use of gas turbines with residual fuel. Mr. Adolph was not aware that marine gas turbines were available for this use with proven reliability on residual fuels. Only now were we beginning to see the use of a blended fuel for marine gas turbines and it would appear that the day when gas turbines could successfully burn Bunker C was still some way off. However, when that day came it was safe to assume that the gas turbine would be able to burn crude oil with equal facility, and this may well make this form of propulsion more attractive to tanker operators. However, for the time being it seemed inevitable that the use of gas turbines for the main propulsion of mammoth tankers was still a long way off.

Mr. K. Brownlie, A.M.I.Mar.E., said that it was always difficult in a technical paper on a subject that covered a wide field of engineering to strike a balance in content that did justice to each critical area. The authors were to be congratulated in maintaining a discipline that did not detract in any way from the interest of the paper.

In the authors' precis they had talked of the "element of adventure" and "an exciting engineering possibility" and certainly the magnitude of this project evoked those sentiments. There were exciting possibilities in the design and layout of the machinery installation. The authors had quite rightly advocated a combination of machinery elements that were well proved, arranged as simply as possible and gave a high degree of machinery stand-by facility. This, coupled with sensible cross-connexions, gave an arrangement with maximum availability potential.

The advantages in propeller design and propulsion efficiency given by the triple screw concept allowed tremendous scope for exploitation of the inherent reliability that this arrangement could give to the machinery selection and layout. The machinery complex outlined in the paper provided what was possibly the present day ultimate in reliability provided that the basic engineering was carried out with diligence and that standard-frame machinery was used which had been proved by service in the present generation of large tankers.

Nevertheless, a study of possible alternative machinery was appropriate to the spirit of the paper and studies carried out in recent years by Mr. Brownlie's company had resulted in a triple screw, reheat plant giving a simplified turbine arrangement illustrated in Fig. 21.

It would be seen that the number of turbine rotors was reduced from six to three and the total number of ahead stages was reduced by about 50 per cent. This machinery probably had the lowest number of moving parts of any triple screw proposal.

The basic feature was the centre screw drive provided by a combined H.P./I.P. cylinder of only seven stages, and a triple reduction gearbox. The wing drives were conventional single flow L.P. turbines with double reduction gearboxes. Astern power was provided by conventional elements within the L.P. casing, plus a separate H.P. element at the forward end of the H.P./I.P. cylinder. The wing screws could be driven independently of the centre screw and it was proposed to manoeuvre with the H.P./I.P. ahead turbine isolated and at condenser vacuum, while the wing turbines were provided with steam throttled from full boiler

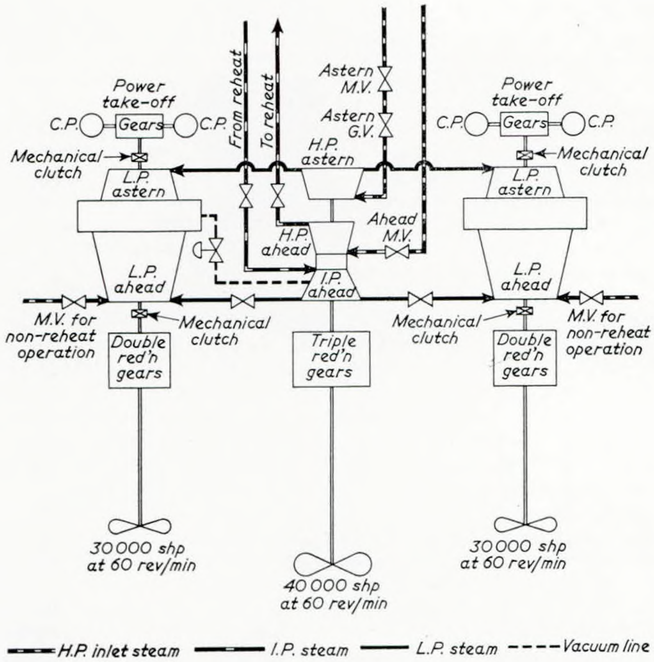


FIG. 21—*Reheat turbine power plant proposal for 100 000 shp, triple screw tanker machinery with astern elements*

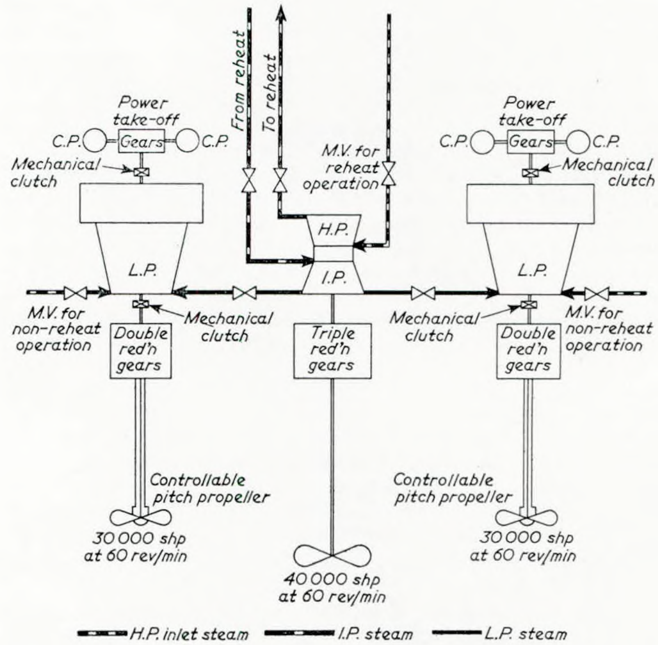


FIG. 22—*Reheat turbine power plant proposal for 100 000 shp, triple screw tanker machinery—with c.p. propellers*

superheat and pressure conditions. The throttled steam gave a temperature at inlet to the L.P. turbines only 45°F (25°C) higher than the normal design condition and eliminated the need for complex boiler control. Using this characteristic the wing turbines could utilize conventional standard frame elements. The high temperature primary, reheat and astern steam was contained in the centre screw drive. Detail design of the H.P./I.P. cylinder had the benefit of twelve years of actual company operating experience with reheat turbines using elevated steam pressures and temperatures.

Steam conditions for this study were selected at 1400 lb/in² g/ 950°F/950°F and were moderate compared with accepted and well proven land-based reheat turbine plant. For this arrangement the optimum power split between the screws was approaching 40 per cent power on the centre shaft and 30 per cent power on each wing shaft, each propeller speed had been taken as 60 rev/min. The feed heating train need not be complex or a detriment to reliability, bearing in mind that we were dealing with conservatively rated, non-rotating machinery, to produce a fuel rate of 0·4 lb/shp h. Taking current fuel oil prices the above improvement over a fuel rate of 0·47 lb/shp h, would reduce the annual fuel cost by approximately £140 000. Furthermore, the capital cost of the complete machinery installation for the ship when compared with the three cross-compound turbine non-reheat plant was reduced by approximately £250 000.

The auxiliary drives had not been fully optimized but work to date indicated promising returns on the use of combined units having a self-condensing turbine driving both a generator and a feed pump. To utilize the L.P. turbine facility of throttling full boiler steam conditions, the cargo pumps were shown driven through the forward end of the L.P. cylinder via simple manually operated clutches, two or three pumps could be driven from each L.P. cylinder. The drive to the pumps could be through right angle bevel gearing of the type described by Faber in a recent paper to this Institute as having been used in hydrofoil transmissions.* The ability to make such gears for the power involved did not at present exist in this country, but they represented the ideal drive for this application.

If c.p. propellers were considered the turbine could be simplified and Fig. 22 showed the arrangement using a fixed pitch centre propeller and c.p. wing propellers. In that case all the turbines were unidirectional and astern power was not provided on the centre shaft. If it was considered desirable to provide c.p. propellers on all three shafts to give improved manoeuvrability a comment from the authors regarding the feasibility of a centre propeller designed to absorb 40 000 shp would be appreciated.

A feasible boiler arrangement for this scheme would be the adoption of a two boiler concept. A comment on the reliability, maintenance and cost aspects of a one-and-a-half, two or three boiler arrangement would be useful.

* "Hydrofoil Craft and their Marine Engineering Aspects", E. Faber, *Transactions I.Mar.E.*, Vol. 82, No. 10, October 1970.

MR. A. N. S. BURNETT, M.I.Mar.E., quoted page 175 of the authors' paper: "An equally important consideration is that this high powered plant should be designed to be within the capability of the staff who operate and maintain it."

On page 172, Section B, the authors stated that all of the single screws and most of the twin screw propeller designs were beyond the present manufacturing capacity for a one-piece casting but all triple screw propeller designs, except for some of those with limiting diameters, could still be produced in one piece.

On page 171 the authors recommended that the weight of propellers for the triple screw designs would be 62 tons each for the 60 rev/min condition, falling to 40 tons for the 100 rev/min condition.

When such ships were maintained or surveyed in service, then the propellers would have to be examined and/or replaced from time to time. Sixty-two tons was a great weight to manage either in water or in a dock and certainly presented a problem so far as underwater maintenance was concerned.

The 1970s had been forecast as being a decade in which new materials would make a tremendous impact on the engineering scene. We should pay heed to the development of explosively bonded materials such as, say, titanium on to aluminium; processes were also available for the development and manufacture of hollow blades.

Would the authors like to comment on the feasibility of the reduction of propeller weights by advance manufacturing processes using new materials? The quoted figures could be altered considerably and so the maintenance load eased. Ease of maintenance was becoming of increasing importance with these vast ships, as time out of service could cost an owner anything between 100 000 U.S. dollars to 250 000 U.S. dollars per day. It was thought that the use of lightweight materials and hollow individual blades for propellers for such ships had tremendous potential.

MR. J. H. MILTON, M.I.Mar.E., understood from the authors' paper that the weight of the single screw capacity for these ships was in the neighbourhood of 100 tons. The use of such propellers introduced a number of practical and operational difficulties particularly as such weights were not easy to manipulate over a ship's counter. Fairly recently it was suggested that the propeller should revolve as a separate entity on an extension of the stern frame and be driven by a quill shaft which, passing through it, thus relieved the propeller shaft of all bending stresses and provided the propeller with a fairly uniformly loaded bearing within itself.

This arrangement had the following advantages:

1) Leaving the propeller *in situ* at tailshaft survey times.
2) It had more flexibility from the shafting alignment aspect.
3) Any heat generated in the bearing would be more easily dissipated.
4) If the propeller was driven by a cone attachment this could be a steel-to-steel connexion, thus avoiding differential expansion effects at the cone joint (see Fig. 23).
5) In the case of multiple screws where shaft withdrawing outboard was an advantage, the tailshaft could be flange coupled to the after face of the propeller (Fig. 23).

In conclusion, talks with roller bearing manufacturers had prompted them to state that they could, for a 45 ton propeller, supply two roller bearings, one to be put in at either end, which, provided they were efficiently lubricated, would last for the life of the vessel.

Mr. Milton thought the foregoing points worthy of consideration, when contemplating such large vessels, and the authors' comments would be much appreciated.

MR. T. ISHERWOOD, M.I.Mar.E., said that his company operated three 210 000 ton tankers fitted with 30 000 shp re-heat installations; the turbine inlet conditions being 1200 lb/in^2 950°F from one boiler. The emphasis, therefore, was on reliability and the use of proven designs for the machinery installations, which really endeared him to the authors in this respect.

A tremendous amount seemed to go wrong with the layout of engine installations. Mr. Isherwood agreed that maximum efficiency in terms of fuel consumption should not be striven for, and he was in agreement with the rest of the very pertinent remarks in that particular paragraph of the paper (second paragraph, page 175).

Most of us were aware of the difficulty of finding good steam engineers who, when found, must be given good acquaintance time with these complex high powered installations.

Mr. Isherwood's company's tankers had been in service since March 1969 and mechanical failures in all three, still waiting to be put right, had made a mockery of trial fuel consumption figures. Other failures were restricting the power to be used in two ships and it was amazing that so many failures could occur in equipment from both home and abroad.

It was interesting to note that the controls and instrumentation were designed so that unmanned operation could be considered at a future date for the one million ton tanker. His company did not have remote control of the main engines from the bridge but generally operated from the control room with one engineer on watch and back-up provided in shallow or confined waters. Perhaps the unmanned situation would be reached when all had a little more faith and we had got to the flat part of the bath tub curve for the failure of electronic units and other items.

Noting the statement that there was a need for more research into design methods for sterntube bearings, it might be worth recalling that in 1963 at the time of the Pametrada-BSRA merger, two colleagues of Mr. Isherwood at that time proposed and designed a large test rig to investigate the many parameters surrounding the sterntube requirements, and other problems in this area, but it was turned down.

He would like to ask the authors two questions which had featured in their paper. Which particular type of prime mover they might have considered for the bow thrust propulsion unit, whether electric motor or Diesel? And at what voltage could generation of the alternators be envisaged, at a load of 2500 kW? Could 440 be used, or was 3·3 k called for?

MR. L. HAWDON, M.I.Mar.E., said that one of the more important considerations in the operation of high powered ships was the avoidance of vibration, from the viewpoints of the comfort of the crew, the possibility of structural damage taking place, and the satisfactory operation of instruments and machinery. Such vibration was almost inevitably propeller excited, and this was often very difficult to avoid with a conventional hull form, due to the widely varying wake field in which the propeller must operate. Extreme wake variations were also the main cause of propeller cavitation, to control which it

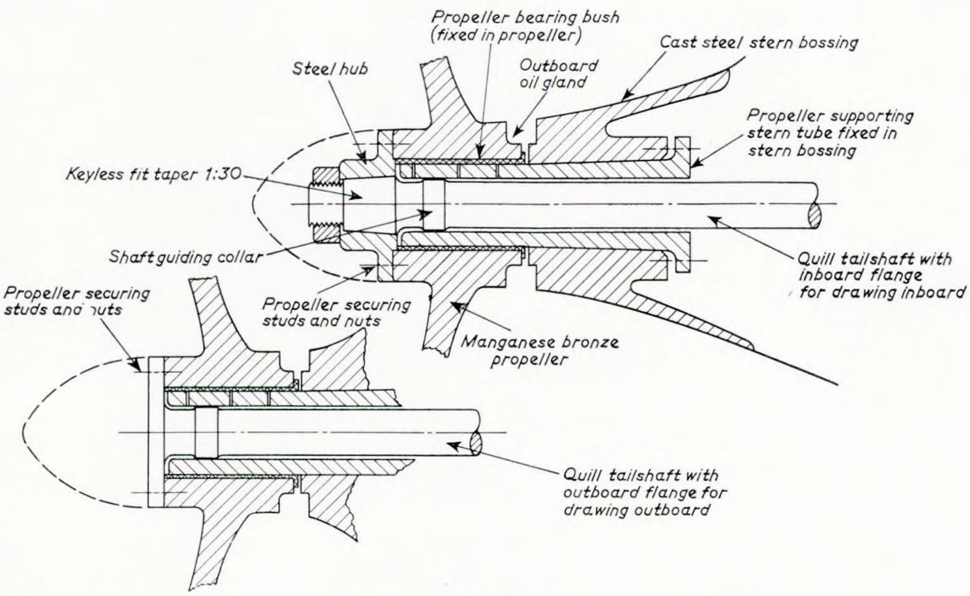


FIG. 23

was sometimes necessary to increase blade widths and so reduce propeller efficiency. With the high powers envisaged by the investigations in the authors' paper, particularly those on the centre shaft, it was fairly certain that the problems of vibration and cavitation presented on ships of the size presently in service would be increased considerably.

It was disappointing, therefore, that these subjects were not given more consideration by the authors, and that no apparent efforts were made in the hull design to move away from the conventional stern configuration in order to improve the wake distribution, particularly on the centre propeller.

The choice of 16 knots service speed followed general practice with large tankers building at present, but it may be worth considering some of the effects of adopting a lower service speed. For example: if 14 knots was to be considered, the tonnage required to maintain the same supply to refineries would need to be increased by about 12½ per cent, or two additional ships in fifteen. The capital outlay on these extra ships would obviously be considerable, but there were a number of advantages to be obtained in return.

The tangible gains lay firstly in the size of the machinery to be provided. The reduction of two knots in speed meant that the total power requirement would be reduced by about 33 per cent. For a million ton tanker the resultant power output would still be too great to be accommodated on a single shaft, but for a twin screw ship the necessary total power of about 73 000 shp could comfortably be carried on two shafts, while a very simple arrangement could be achieved with a triple screw installation with a required total power of about 65 000 shp.

The cost and size of turbines and steam generators would be reduced, and the triple screw alternative would be well within the range of presently available Diesel engines. It followed that the fuel consumption would be reduced by 33 per cent, and the gains in space made possible by the need for less bunkers, and smaller machinery, could be used to add to the cargo carrying capacity.

The intangible gains would lie in the simplification of the vibration and cavitation problems because of the reduction in power and thus in thrust and torque forces.

It may thus be interesting to see the results of a detailed economic study into the effects of a two knot reduction in service speed. The initial capital outlay for a given annual fleet delivery capacity would almost undoubtedly be increased, but would certainly be less than the 12½ per cent represented by the necessary deadweight increase. On the other hand the saving in fuel and maintenance costs over the life of the ship should be offset against this additional outlay, and the easier maintenance and possible reduction in vibration should allow owner, builder, superintendent, insurer, classification surveyor, propeller designer and not least the crew to sleep more soundly in their beds.

Correspondence

Mr. E. W. Bell, A.M.I.Mar.E., wrote that with the high capital investment incurred in the cargo pumping equipment, especially on the vessel described, equipment with a low usage factor (less than 5 per cent of the operating time of a vessel regularly plying the Arabian Gulf—U.K. route), he wondered whether we had arrived at a situation when serious consideration should be given to pumping at the discharge terminals with "off ship" pumping equipment?

Mr. Bell envisaged a multiplicity of self-propelled barges at the selected terminals used by such huge vessels. These barges would carry high capacity pumps, perhaps gas turbine driven, placed alongside the vessel, they in turn discharging to the refinery system. Such an arrangement would involve lower suction head over the "on barge" pumps, compared with that currently dealt with on ship board. Perhaps the pump designers would give some thought to this.

The design draught of the barge and consequently the position of its pumps suction had to satisfy a condition of acceptable suction head when the tanker was almost fully discharged.

Recent events have indicated that there is not necessarily an advantage in having full, self-discharging facilities on a vessel in a hazardous situation. In any event, pumps were always required on board for ballast and stripping purposes. Such pumps would possibly satisfy an emergency pumping condition.

Professor G. H. Chambers, M.I.Mar.E., in a written contribution, said that the triple screw arrangement brought back memories of the three shaft aircraft carriers of which the Navy has had a good deal of experience.

Reliability was emphasised to have been of quite exceptional importance in this very large tanker. Three truly indeoendent machinery installations would have made the chances of loss of all motive power very small. The proposed feed and steam circuit showed the three units cross-connected after the air ejectors to a single set of feed heaters and de-aerator, with a common steam system. Thus contamination in any of the three engines could immobilize the ship. Would the authors have considered it sound to have fitted completely separate feed and steam systems for each boiler-turbine unit, with no major steam or feed cross-connexions, so that they could have obtained the full benefits of three separate boiler-turbine units? In addition steam piping would probably have been reduced and control simplified.

A further reliability aspect was the possibility, with one large machinery space, that a fire or explosion in one of the boilers would have again immobilized the ship. This could have been countered by fireproof divisions, preferably watertight bulkheads, between the sets of machinery. In these days of unmanned engine rooms this would not increase the number of watchkeepers required. The single control room could still have a view of each set of machinery through the respective windows. Would the authors have expected such a measure to be practicable and economically justified bearing in mind the possibly extremely serious consequences of a major fire in the engine room with the ship immobile?

Mr. E. P. Crowdy, M.I.Mar.E., in a written contribution, noted that each of the three sections of the paper commented on the optmum speed of a megaton tanker. In the first section, at the start of page 166, was a statement "... it is considered prudent that a speed of at least 16 knots in service should be assumed". In the second section, second paragraph in the right column of page 167, was a statement containing " . . . and an advance on 16 knots seems most unlikely. A reduction in speed is the more possible alternative, and a speed reduction to 15 or 14 knots would exert a considerable influence on machinery requirements". In the third section, at the end of page 173, was a statement " . . . recent studies have indicated that the optimum speed of a half million ton deadweight tanker is 16 knots, a figure which has also become established during the recent rapid increase in size of up to, and beyond a quarter million tons".

Mr. Crowdy thought that further explanation was warranted on what was the most fundamental of all the parameters effecting the propulsion of a million ton tanker.

The graph shown in Fig. 24 and the information given in Table III has been prepared from the following data:

16 knot tanker overall purchase cost .. £30 per dwt
Diesel propulsion machinery cost, installed .. .. £40 per hp

Amortization charges taken as 15 per cent of capital costs, which was equivalent to raising the capital by mortgage at 8 per cent with repayment over 8 years or alternatively 12 per cent with repayment spread over 15 years.

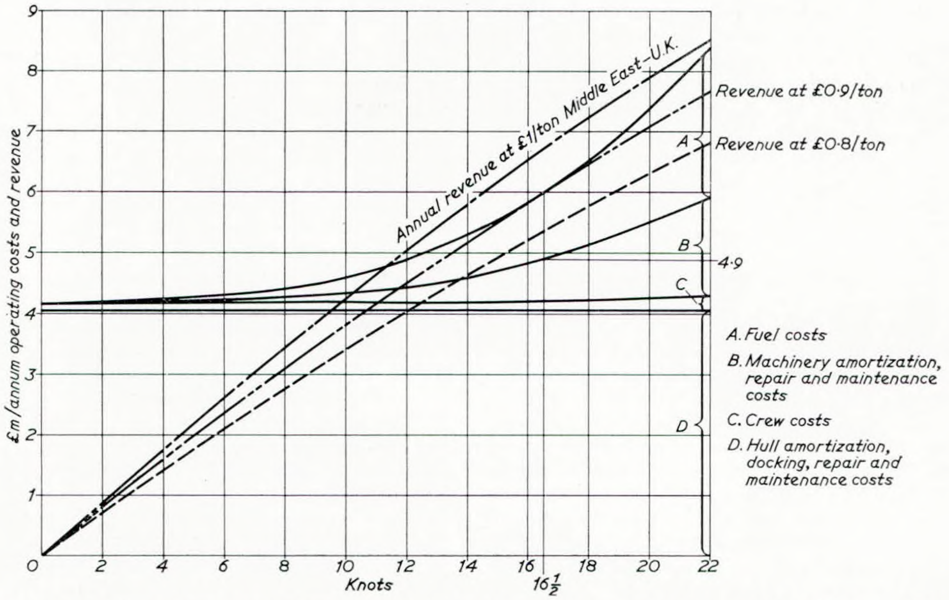


FIG. 24—*Operating costs and revenue return v speed for million ton Diesel tanker. Middle East—U.K. (Cape route)*

Fuel cost	£8 per ton
Overall fuel rate for Diesel propulsion ..	0·36 lb/shp h
A basic crew of 30 plus 1 per 10 000 hp at a charge of	£4000 per capita p.a.
Machinery repairs and maintenance ..	£2 per hp
Hull repairs and maintenance	£175 000 p.a.

Ten days off hire per annum.
Nineteen thousand mile round trip.
Four days in port per round trip.

In Fig. 24, which was plotted on a basis of ship's speed, the ordinates in area A indicate fuel costs, in area B machinery amortization and maintenance costs, in area C crew costs and in area D hull amortization and maintenance costs. The chain and dotted lines going through the origin represent the revenue arising from different freight rates per ton Middle East-U.K. and show how the "break-even" point with the lowest economical freight rate of £0·9/ton corresponds with a speed of about 16½ knots. Since freight rates would probably substantially exceed £1/ton (approximately 1d/gallon) the optimum speed would ıise and it would appear desirable to design a megaton tanker for at least 16½ knots thus requiring appıoximately 10 per cent more power than the figures given in the authors' Table II.

Mr. Crowdy was interested to note that the standing charges appropriate to a 16½ knot megaton tanker, including full internal finance and crew costs approximated to £4·9 × 10⁶ p.a. or some £13 500 per day. Demurrage at a higher figure than this would not have seemed to be financially justifiable.

It was suggested on page 175 that slow and medium speed Diesel engines were not suitable for a propulsive power of that magnitude because of the number of cylinders required. The triple screwed proposal with the propellers turning at 100 rev/min and power equally divided between all shafts required a service

TABLE III—1 000 000 TON TANKER

Tanker speed	knots	10	14	15	16	17	18	22
Propulsive power	bhp	24 414	66 992	82 397	100 000	119 916	142 383	259 961
Propulsion unit	—	Diesel	Diesel	Diesel	Diesel	Diesel	Diesel	Diesel
Length of round trip	miles	19 000	19 000	19 000	19 000	19 000	19 000	19 000
Duration of round trip	days	83	60	57	53	51	48	40
Time in port/round trip	days	4·0	4·0	4·0	4·0	4·0	4·0	4·0
No. of round trips p.a.		4·3	5·9	6·3	6·6	7·0	7·4	8·9
Capital cost of hull (amortization)	£10⁶	3·9	3·9	3·9	3·9	3·9	3·9	3·9
Repairs and maintenance of hull	£10⁶	0·15	0·15	0·15	0·15	0·15	0·15	0·15
Capital cost of machinery (amortization)	£10⁶	0·15	0·40	0·49	0·60	0·72	0·85	1·6
Repairs and maintenance of machinery	£10⁶	0·007	0·020	0·025	0·030	0·036	0·043	0·078
Fuel consumption/day	tons	91·3	250·0	307·9	374·5	449·7	534·7	975·4
Fuel consumption/round trip	tons	7248	14 206	16 308	18 555	20 947	23 483	35 080
Fuel consumption/p.a.	tons	30 949	83 247	101 923	123 203	147 044	173 776	311 159
Fuel cost/p.a. @ £8/ton	£10⁶	0·248	0·666	0·815	0·986	1·176	1·390	2·489
Number of crew	—	32	37	38	40	42	44	56
Cost of crew/p.a.	£10⁶	0·13	0·14	0·15	0·16	0·17	0·18	0·22
Total operating cost	£10⁶	4·585	5·276	5·53	5·826	6·152	6·513	8·437
Revenue @ £x/ton/round trip								
M. East/U.K.								
i) x = £0·8	£10⁶	3·39	4·65	4·92	5·21	5·5	5·78	6·85
ii) x = £0·9	£10⁶	3·82	5·23	5·53	5·87	6·19	6·5	7·7
iii) x = £1·0	£10⁶	4·24	5·82	6·15	6·52	6·87	7·23	8·56

power of 35 000 bhp per engine. With the super large bore slow speed Diesels now available this could be obtained from a ten cylinder engine resulting in 30 cylinders in total, a number substantially less than had been fitted in many smaller ships with medium speed engines. It could not be conceded that the number of cylinders per se was an absolute impediment against the adoption of Diesel machinery but merely one of the lesser of many factors to have been considered in balancing the advantages of different types of prime mover.

A direct drive Diesel did, however, incur a substantial propulsive efficiency penalty and a more attractive and flexible plant could have been achieved by adopting geared Diesels. The Doxford Seahorse engine might have been almost tailor-made for this application. Six seven-cylinder engines could have been geared in pairs to each of the three shafts with each of the four outer engines incorporating a ten megawatt alternator in the drive. Any three alternators would have provided, with a considerable margin, all the power required for cargo pumping, the appropriate engines having been disconnected from the main reduction gearing. If c.p. propellers had been adopted on the wing shafts, fining of the pitch would have provided a convenient source of power for a bow thrust unit of appropriate size. All auxiliary power would have been provided, both whilst cargo pumping and at sea, by a waste heat recovery scheme. Although such a scheme required operating expertise in Diesel, steam and electric plant, it was contended that the super abundance of exhaust gas would have allowed the steam plant to be extremely simple and trouble free. Electrical engineering competence was required for any marine installation. The overall fuel rate at sea on HVF fuel would thus have been 0·36 lb/shp h giving a fuel cost saving over the turbine machinery in excess of £300 000 p.a. and over £100 000 p.a. better than direct drive Diesels. Protagonists of turbine machinery were apt to claim that the availability of steam plant was substantially superior to that of Diesel machinery. Even if their claim could be substantiated, the Diesel plant would have to be off hire for an additional three weeks to absorb the fuel cost differential. Such machinery could have readily been accommodated within the space allocated by the authors for turbine machinery as could be seen from Figs 2, 3 and 4. The weight of such machinery would not have caused any embarrassment since the main engines and gearboxes complete would in total have been under 2000 tons, a mere 0·2 per cent of the cargo deadweight.

Although it was frequently contended that fuel costs were of decreasing relative importance, it was still an inescapable fact that this parameter alone had by far the greatest influence on the final profitability of a venture—always provided that the enhanced fuel economy was not offset by any loss of reliability or an excessive increase in maintenance costs. With the geared Diesel machinery suggested the flexibility and availability of the propulsion machinery would have been of an exceptionally high order and the chance of a total loss of propulsive power was minimal. It was appreciated that the authors were not in a position to put forward geared Diesels of the type now suggested

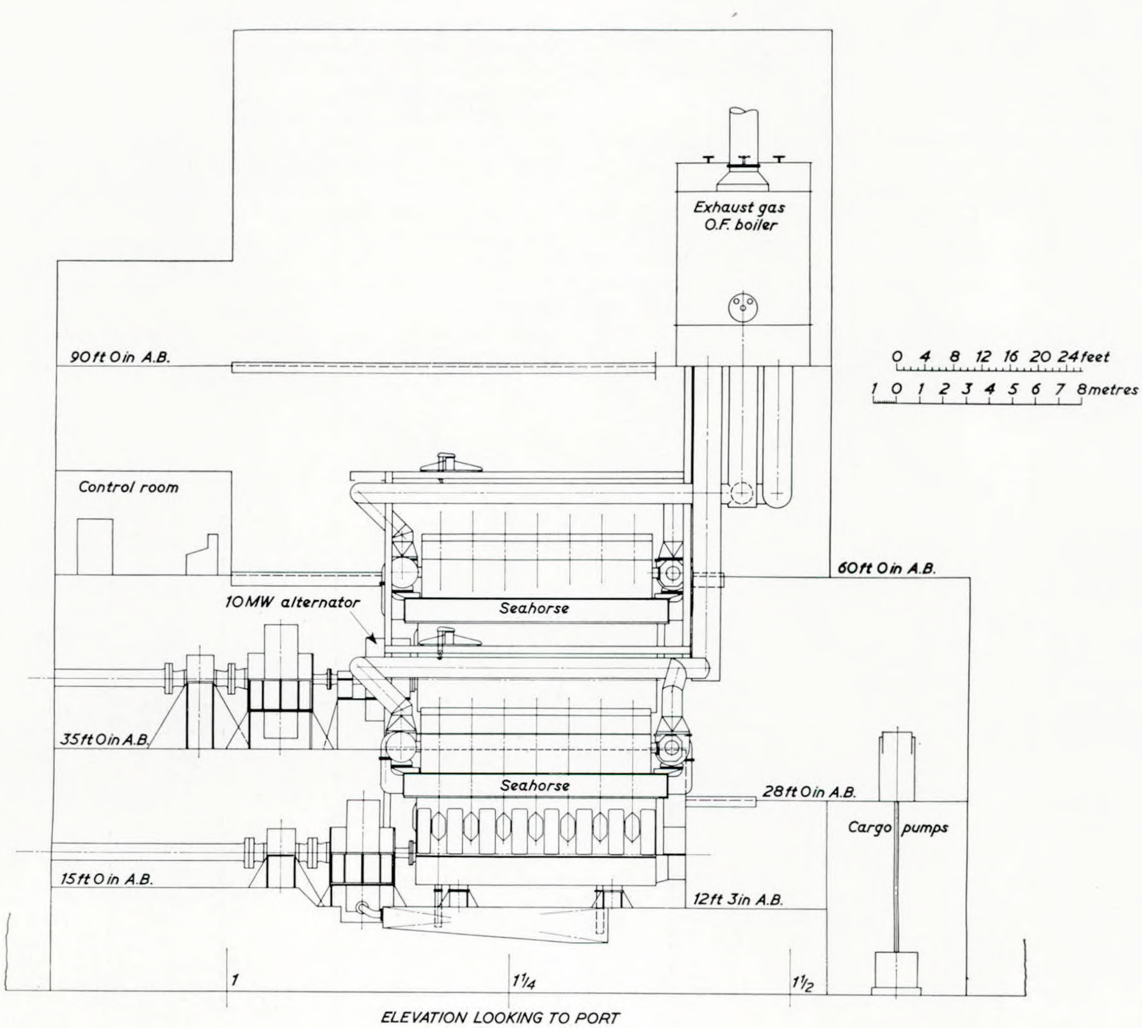


FIG. 25

since at that moment the Doxford Seahorse engine was not commercially available, but their reaction to this alternative proposal would be of great interest.

DR. J. W. ENGLISH thought that this paper was a very interesting extension of the authors' earlier paper on the same subject. He suggested that there was scope for at least one more paper on the subject in which the authors might perform economic calculations in an attempt to obtain a true comparison of costs for the various proposals. How the advantage of improved manoeuvrability and safety accruing from the use of a twin or triple screw ship might be quantified was difficult to see. Similarly the development costs that might be involved with supporting and lubricating the bearings of such large propellers and producing reliable c/s of the size proposed by the authors might be important factors. No doubt these calculations must be performed in detail before an owner would commit himself to using a triple screw arrangement with c.p. wing screws. Had the authors made any such comparative calculations?

He believed that the remarks in the paper on the advantages of using a bow thruster with an opening in the bow and side outlets for discharge could be misleading. Firstly they gave no indication of the size of thrust unit likely to be necessary to produce a worthwhile effect, even if it was possible to define the effect required. And secondly the advantages of the thrust unit configuration described by the authors would be mainly in the provision of a uni-directional propulsion motor, although this would tend to be counteracted by the need to provide flow divertors, and not so much from the intake of higher energy fluid when the ship was moving ahead. For example, at a ship speed of five knots the dynamic head of the free steam would only be about one fifth of the jet dynamic head while at ten knots it would be about 70 per cent. However, the effectiveness of a lateral thrust unit being used at forward ship speed was influenced mainly by the flow conditions at the jet outlets and the advantage of having a higher energy intake flow tended to be counterbalanced by an increased adverse suction effect in the lee of jet efflux.

Would the authors please explain the purpose of the knuckle line in the bow plating shown in Fig. 4 of the paper? Dr. English would have expected this to be a poor feature due to the flow crossing it at large angles and flow separation occurring in its lee. Could it be that a small separation here modified the flow in such a manner that the larger adverse pressure gradient encountered around the bilge could be withstood more easily? The comments of the authors regarding the very small scale model results used in the paper were noted, particularly in view of one of the author's insistence on running tests at high Reynolds numbers. The authors were making the tacit assumption in employing these small scale model results in the paper that the results from the large models compared closely with them. It would be interesting to see if this was the case.

Did the authors see the possibility of steel blades being used for large built up propellers?

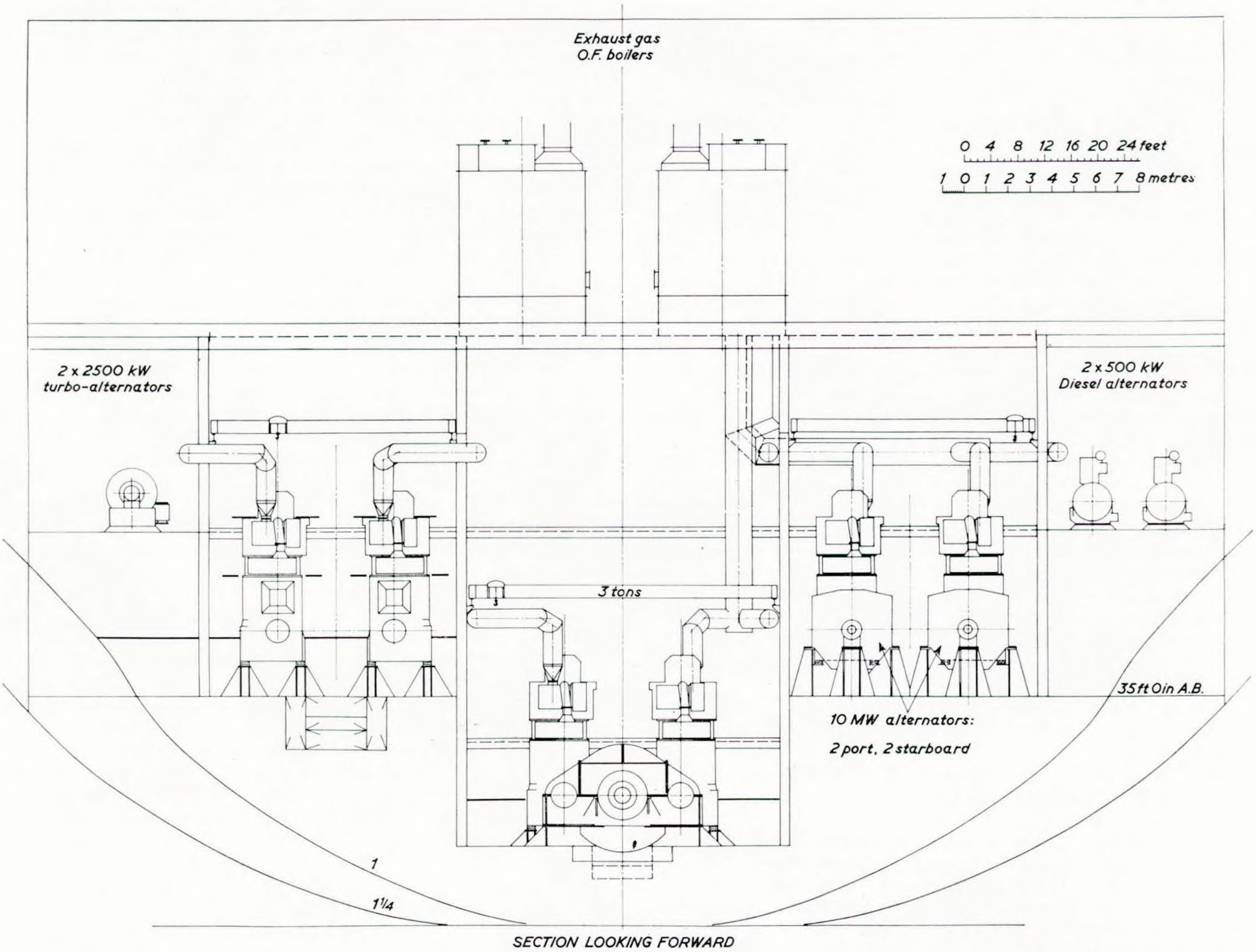


FIG. 26

Dr. English would support the authors in their proposals for twin or triple screw propulsion of these large tankers, but did not believe that contra-rotating, tandem or overlapping propellers, as options mentioned by the authors, would have any place in the propulsion of such large ships although ducted propellers of one form or another could have a very bright future.

MR. W. J. L. FOREMAN, M.I.Mar.E., wrote that in the last few paragraphs of the paper the authors concerned themselves with stern bearing design and lubrication and, on the latter point in particular, he would like to comment.

He thought that there was no doubt that the present permitted specific load of 90 lb/in^2 at 105 rev/min would have to be exceeded but, as the authors said, the increase would not be unreasonable.

What was of more concern, was the certainty that boundary lubricating conditions would occur in the bearing during lengthy operation of the turning gear.

The authors made a suggestion to use an "oil with viscosity of 0·6 poise at 140°F" to increase the oil film thickness. Mr. Foreman was compelled to make two observations here. In the first place, under "boundary conditions" viscosity was relatively unimportant. The oil film would be only a few molecules thick and likely to be broken by the larger surface asperities. When this happened damage to the shaft liner or bearing could result. To cater for this one had to look at those properties in the oil which might be termed "mild extreme pressure" or "polar". These determined the strength of adhesion between lubricant and bearing surface and in the simplest cases a suitable lubricant could be obtained by blending fatty material with mineral oil.

Secondly, the viscosity quoted by the authors, in units rather academic, suggested, in common or garden terms, an oil of about 280° secs Redwood 1 at 140°F or something between an SAE 40 and an SAE 50 grade. This oil would, at a temperature of 50°F, have a viscosity of over 3000 secs. Redwood 1 and this was making the most favourable assumptions on density and viscosity index.

It was not unreasonable to expect this temperature in the aft end of the ship where the forced lubricating system was situated. So, under these conditions, one would expect to see an oil heater as well as an oil cooler being employed.

This was all on the basis that hydrostatic assistance was not provided to lift the shaft as was done with some land based turbine rotors. Naturally, in this case boundary conditions were avoided and viscosity considerations applied.

Mr. Foreman recalled that the first suggestion of a 1 000 000 ton tanker had come in a public statement by a North-East coast shipbuilder some time ago and there was no reason to think the suggestion outrageous, at any rate from a technical point of view.

This was all that was being considered at the moment because the more delicate aspects of oil supply and transportation must be left to those able or willing to predict political and geographical situations.

Some twelve years ago, the first 100 000 ton plus tanker had been projected—a big step at that time but later, when it came into being, it did not meet with the success anticipated. Perhaps political motives had influenced design and commercial considerations, maybe it was just not economically viable in the market conditions then existing, perhaps there were lessons to be learned here.

But all those interested in the success of a 1 000 000 ton proposal would surely have in mind that such a ship could not be available for, at least, six years, would cost upwards of £30 000 000, carry a cargo worth £5 000 000 and have an earning capacity under moderate market conditions of up to £1 000 000 per month.

What was now being considered was some means with which to turn the propellers of a monstrous steel barge containing 1 000 000 tons of oil from one point to another with some degree of financial advantage to the operator. Presumably the naval architect would be satisfied that his part of the job could be effectively dealt with and from the paper one could be confident that Mr. Sinclair and his colleagues would be able to design and make available suitable propellers. It was then postulated that horsepower of the order of 90/110 000 would be required to give the ship a service speed of sixteen knots.

A single screw must surely be ruled out for many reasons, some of which were apparent in the paper. Of all the combinations of prime mover suggested by the authors, there should be strong support for a reasonably conventional, three boiler/two shaft arrangement. This should present no difficulty whatsoever to the turbine designer.

The success of the venture was most likely to depend on the attitude of those responsible for final selection and installation of all machinery.

These considerations had obviously been in the authors' minds, because in the first paragraph, page 175 of the paper, they gave very sound reasons for the adoption of robust, simple and well tried main and ancillary equipment, free of fanciful adaptations and appendages which might appear attractive on paper and even successful in other fields of transport and power generation, but which had proved so costly and embarrassing in marine practice.

Mr. Foreman did not advocate a temporary stand-still in technical development, but merely suggested that in the last decade too many extravagant ideas had found their way too quickly into marine machinery design and application.

More thought should be given to evolution not revolution.

MR. D. J. GIBBONS, M.I.Mar.E., wrote that the authors had rightly drawn attention to the need for reliable and maintainable machinery in this type of vessel but had failed within their paper to have stated the reasons for this requirement and the factors which would influence the degree of reliability which was called for. The investment and routine operating costs represented a high daily value even at normal rates but under the present inflated demand in the tanker charter market, the requirement for reliability became even more important. It would not have been unrealistic to have considered the cost of chartering in replacement tonnage, in the event of failure of a vessel of this size, at something over £100 000 per day. This high cost output for whatever reason, must have been considered at every phase of the ship's design and each factor must have been carefully considered against this cost, or an alternative figure specified by the prospective owner. To this end, it would have seemed essential, for the development of a successful design, that the owner, operators, designers and constructors must have collaborated from an early stage with realistic specifications and operational requirements.

In developing these specifications consideration should have been given to the manning which was to be adopted, both with regard to numbers, duties and the trade skills possessed by the crew, to the requirement for separate seagoing maintenance staff and to the range of repair and maintenance tasks that could reasonably be undertaken at sea without the need for shore support, or interruption of the vessel's trading routine. This should have been considered over the lifetime of the vessel, and consideration have been given to changes in manning level and maintenance procedures as the vessel aged. In all circumstances, consideration should have been given to the man hours required for each task, and the methods by which they would be available when required.

Considerations of this nature would lead to a rational choice in the number of units to be installed together with the degree of redundancy, and also to the excess capacity of individual units which might be installed to cover breakdown of another machine. An analysis of breakdown causes over a large number of vessels would also provide useful information on system reliability, and might well draw attention to the need for improved design of systems in order to take full advantage of the reliability which could be expected from the basic units of the system.

MR. J. P. GRAHAM, M.I.Mar.E. in a written contribution said that, when attending the delivery of this paper in the Picton Library, Liverpool, he had observed that many references had been made to propeller size during the discussions and felt that it was worth putting on record a few facts surrounding the two major dimensions of propellers—diameter and weight. At time of writing the largest propellers manufactured in one casting were the series for the 254 000 dwt single screw tankers for Esso. The propellers had a finished weight of 57 tons (cast weight 83 tons) and a diameter of just under 30 ft.

The capacity of the Stone Manganese Marine Ltd. factory at Birkenhead would enable a propeller of 40 ft diameter to be manufactured with a finished weight of 85 to 90 tons.

The rapid growth of tanker sizes and the possibility of large slow revving propellers had encouraged examination of manufacturing facilities for propellers of 50 ft diameter and heavy finished weights of 150 tons (190 tons cast weight). Already on the drawing board were plans for 200 ton foundries and finishing shops to accommodate these large propellers, and designers must not restrict their thinking, nor reduce the efficiency of a propulsion system, on the basis of present propeller manufacturing availability.

The problem of transporting these large propellers would be a major one were it not for the facilities made available by the Mersey Docks and Harbour Board. Road transport to any part of the U.K. was entirely ruled out, but Esparto Quay on the Wallasey side of the Wallasey Pool, only a quarter of a mile from the company's Birkenhead factory, was available, and floating cranes and coasters could deliver propellers to any part of the U.K., or overseas, if required.

The repair of large propellers was a major problem should damage occur remote from a dry-dock. Designers would, in the future, have to consider the provision of platforms (perhaps portable) and to tilt the vessel to permit repairs to be carried out on the shaft if the damage was not too severe. More severe damage might have to be repaired with the propeller removed from the shaft while in dry-dock and even this on site, since, again, road transport was ruled out and the prospect of returning the propeller to the factory was economically not on. Steps were being taken to ensure that any repair carried out on the propeller outside of the factory would be of a high quality and, again, would not prove to be an obstacle in the future planning of large propellers.

Mr. J. B. Hadler wrote that there was little that he could add to the authors' analysis of the propulsive performance of the single-, twin- and triple-screw configurations, but suggested that they should add to the twin-screw configuration the overlapping propeller arrangement. This configuration would have made the twin-screw arrangement more competitive by raising the propulsive efficiency. The smaller appendages associated with the propeller system tended to reduce the drag and the location of the propellers in the higher wake region resulted in a larger energy recovery.

The authors discussed the problem of propeller-induced vibration, but had not discussed the propeller blade vibration problem which became progressively more severe with increases in propeller size. Since the propeller blades were cantilevered beams the frequency of blade vibration decreased as blade length increased, thus, at the large diameters considered in this paper, the frequency was quite low. The exciting force arose from the harmonic components in the wake pattern in which the propeller operated. Although the rev/min were also reduced this change was not as great as the reduction in frequency due to increase in diameter. Hence, resonance frequency occurred at the lower wake harmonics which were significantly stronger. It was expected that greater root thickness (and greater propeller weight) might have been required to both raise the frequency and ensure low enough fatigue stresses so that propeller blade failure did not occur.

Mr. T. T. Hudson, G.I.Mar.E., in a written contribution said that in Dr. Milne's section of the paper fuel prices were quoted as being "relatively stable" and he would like to point out that, until the Suez crisis of 1967, this had been the case, but since then there had been a steady increase in the price of both residual and distillate fuel oils.

Prices given by *Shipbuilding and Shipping Record* were as follows:

	Distillate	Residual
September 1965	£9·77½ per ton	£5·62½ per ton
November 1970	£10·40 per ton	£7·35 per ton

Thus in 1965 when prices had been very stable the cost of distillate fuel was 1·74 times that of residual fuel. At the end of 1970 the cost of distillate fuel was only 1·42 times that of residual fuel. If this price differential continued to change at its present rate, the time could not be far away when the cost, of both types of fuel, would be practically the same.

Therefore, considering a fuel rate of 0·47 lb/shp/h, as quoted by Dr. Milne for his steam turbine installation, and a fuel rate of 0·48 lb/shp/h for the Olympus marine gas turbine type TM3B, as quoted by Captain R. D. Tatton Brown, R.N., in a paper entitled "Propulsion Gas Turbines in the Royal Navy", read before the North East Coast Institution of Engineers and Shipbuilders on 9 November 1969, it would have appeared that the economics of running a gas turbine plant might approach that of a steam turbine plant.

Considering also that the main cargo pumps could be propelled and controlled quite easily by electric motors, Mr. Hudson submitted that this type of main propulsion, gas turbine, ought to have been given a little more thought.

The authors did not seem to be aware of many of the recent innovations of the marine engineering industry, as they did not even consider generating electrical power at voltages above 440 V a.c. It would appear that Dr. Milne had chosen a steam turbine installation and then proceeded to justify this choice by giving one or two disadvantages inherent in other types of propulsion units.

Mr. M. Jourdain, M.I.Mar.E., wrote that there were many concepts of interest in this comprehensive paper, the most striking of which was the appreciation of the triple-screw arrangement. The authors were very convincing in their statement of the various advantages of this solution, which has been rather neglected for years. In addition, they could have mentioned that when it was combined with controllable-pitch propellers on the wing shafts, the astern turbines could have been dispensed with, as a ratio of two-thirds for the astern power to the ahead power looked largely sufficient. It would have been an appreciable simplification compensating for the inherent complication of a triple power plant.

In Figs 6 to 9 both solutions using single or twin screw appeared as the limits for the triple screw solution when the percentage power on the centre screw became 100 or 0. In fact, whatever this percentage might have been, the stern lines were different for each of the three arrangements and it was unlikely that the particular points pertaining to single and twin screw lay on the curve which was drawn for the triple screw.

The authors also mentioned that a built propeller was heavier and less efficient than the equivalent solid one. This was obvious if the blade lines were the same. The writer wondered whether the separate casting of the blades did not favour an improvement in the quality of the piece and perhaps also a reduction of the locked in stresses which allowed for a smaller thickness, more or less compensating for the boss overweight and increasing the efficiency.

Mr. J. M. Langham, M.I.Mar.E., in a written contribution, said that it had been his intention to include in this contribution the results of the model tests of single, twin and triple screw propulsive arrangements which were being run on behalf of Stone Manganese Marine by the Vickers Tank, at a cost of over £8000. Unfortunately these results were not yet available but it was hoped that the authors might be able to include them in their reply to the discussion.

This work, and in fact the whole of the very considerable amount of research and design effort that had gone into the preparation of this paper, had been undertaken to assist shipowners in the forward planning that was essential if ships of this size were to be built. It already appeared that this work was proving of use in this connexion.

Various references were made to the fact that manufacturing capacity limited the size of solid propellers which could be produced, and the authors then proceeded to develop proposals for built up propellers as an alternative although markedly less efficient solution.

This was all perfectly true as things were at the moment, but if the demand arose for such propellers in the future, however large they might be, the propeller manufacturers would increase their capacity accordingly. Mr. Langham said that his own company certainly would be prepared to do so as it had and always would be their policy to provide shipowners with the most efficient and economic propellers which could be designed. In this case there was no question that the very large single propeller was the best solution. There were no great technical or metallurgical problems to be solved in order to manufacture these enormous propellers. It was simply a matter of justifying the very substantial investment which would be involved.

Over the last decade the company had consistently implemented this policy. In 1960 the largest propellers then required had been about 24 ft in diameter and weighed 30 tons finished. Since then the capacity of their Birkenhead Works had been repeatedly increased and only a few months ago the largest coreless induction furnace for non-ferrous metals in the world had been commissioned. As a result, it was possible for them to manufacture propellers weighing up to 90 tons when finished, and with diameters of 40 ft.

Mr. J. A. H. Paffett, R.C.N.C., in a written contribution, said that the authors' approach to the hull and propeller design for a very large tanker had been studied with interest. A useful comparison could be made with some of the data from another similar but unrelated investigation into a million-ton design.

It was fairly generally known that the then Ministry of Technology had initiated design studies into two tankers of 400 000 and 1 000 000 tons respectively some two years ago. Two shipbuilders, together with Lloyd's Register, NCRE, BSRA and NPL had collaborated in these studies. The contribution from NPL had included the running of a fair number of hull models of various forms and scales.

From the hydrodynamic point of view the sheer size of the million-tonner, considered alone, presented no problem. The novelty arose from the low service speed; a ship speed of 16 knots at that size represented a Froude number of 0·12. This low speed meant that wavemaking contributed only a very small proportion of the hull resistance, and allowed the block coefficient to be pushed up towards unity. The Mintech million-ton study had dimensions 1650 ft by 250 ft by 105 ft loaded draught, with a block coefficient of 0·90. The block of 0·85 quoted in the paper was probably unnecessarily conservative at this low Froude number, as indeed the authors had suggested.

Resistance experiments had been carried out at NPL for several versions of the hull form. That which had been selected for detailed study had an effective power, scaled to full size, of 76 500 hp. The figure of 72 600 hp quoted in the paper was reasonably close, but since this was for a somewhat wider, shallower form it was perhaps just a little on the optimistic side.

It might have been as well to have sounded a warning about the interpretation of model results for those very full slow forms. Although only a small fraction of the resistance was due to wavemaking, the remainder cannot be described as skin friction. Viscous resistance became more complicated the more it was investigated, and there was little doubt that some forms with blocks above 0·8 suffered considerable drag due to trailing vortices and detached eddies. These phenomena were Reynolds number sensitive, and the largest possible model scale should have been used to make them as representative as possible.

There was another reason for using the largest possible scale; this was the need to avoid laminar-flow errors in the propellers when running self-propulsion tests for the interaction factors. A model of the proposed ship on a scale of 1/100 would have had a hull length of 16 ft, but a propeller which was 30 ft in diameter on the full scale would have been only 3½ in on the model, which was far too small for accuracy. Something like twice this diameter was desirable, and for this reason NPL had been running tanker models up to nearly 30 ft long, some displacing more than 6 tons in the tank. Ten such models had been run in the last two years in No. 3 tank at Feltham.

From model experiments NPL had carried out with forms of similar fullness and stern arrangement to those in the paper, they obtained wake fractions which were a little higher than those given; their values were about 0·1 higher than those quoted for single screw and 0·05 higher for twin shafts. They agreed generally with the R.R.E. and thrust deduction for single screw, but for twin screws got thrust deductions about 0·1 higher.

The authors' choice of correlation factors was reasonable and NPL would have used similar figures in the present state of knowledge. The authors' reasoning which led to the suggestion that the "correlation curve" had flattened out and might even have risen beyond 1600 ft was, however, far from clear. It should have been noted that the ITTC line was in fact a "correlation line" and not a skin friction line; if the Hughes' method had been used employing "form factors" the basic skin friction line would have been well below the ITTC line, indeed the latter could have been considered to have represented Hughes' basic line with a constant form factor already applied. An addition of 20 per cent to the ITTC line, did not amount to having added a "form factor" in the correct sense of the term. In any case, the complexity of the viscous drag on a bluff form was such that one could not have strictly assumed that the viscous pressure drag was a constant proportion of the viscous tangential drag. The nature and scaling of the viscous pressure resistance was the subject of current research at NPL.

Mr. Paffett echoed the authors' warning on page 166 about the dangers of separation of flow at the fore shoulder. NPL had observed separation of several kinds in the forebody, notably over hemispherical bulbs and at the turn of the bilge. After-body separation was well known, and very common; it might not have been so generally appreciated that the same danger lurked at the forward end. NPL found that flow visualization studies were very valuable for spotting separation while developing novel hull forms, particularly bluff ones. Here again it was important to have kept the scale up. Flow studies led them to expect that the flow lines in the fore body would be rather more steeply inclined than those shown in the authors' Fig. 1. The flow at the fore end of a typical full tanker plunged at an angle of as much as 45 degrees in places when viewed from the side, and they would have expected the knuckle shown in Fig. 4 to be distinctly "draggy".

The real meat of the paper was in its treatment of the propellers and machinery, which was as thorough as one would have expected. In their paper "The Design and Development of Propellers for High Powered Merchant Vessels", which was given before the Institute in January 1968, two of the present authors stated that when a limit of 60 tons had been put on the propeller weight, there was a small advantage in favour of the twin screw installation as compared to a single screw stern for ships of greater than 245 000 dwt. From Fig. 5 it would in fact be seen that the single screw stern was giving a better performance than the twin screw stern for the tanker of 500 000 dwt. This seemed to suggest that twin screws only gave a better performance, at least for ships up to 500 000 tons, if there was a limitation on the propeller weight and size. Was this the case?

The prospect of a single propeller 55 ft in diameter weighing 266 tons driving the million ton ship was an intriguing one. However, it seemed to the writer unlikely that single-shaft propulsion would ever be accepted for a ship of this size for safety reasons, quite apart from considerations of propulsive efficiency.

Regarding triple-shaft arrangements, NPL had not tried this combination in the 0·9 block million-tonner, but have made some twin- and triple-shaft comparisons in a slightly less full form of around half a million dwt. This work had suggested that the gain in propulsive efficiency in going from twin to triple was of the order of five per cent; this looked much less attractive than the 20-30 per cent suggested by the curves in Fig. 5.

Mr. A. Rolland, S.I.Mar.E., wrote that the problems of manoeuvring mammoth tankers were large, and in the paper consideration was given to the several screw arrangements possible. The triple screw arrangement with controllable pitch propellers on the wing shafts would, as was pointed out, provide the vessel with its best manoeuvring capabilities, using open water propellers.

Had the authors considered the possibilities of fitting shrouds to the two wing propellers, thus bringing about gains in both the propulsive efficiency and the manoeuvring capabilities of the ship?

MR. A. ROSE, A.M.I.Mar.E., wrote that the proposal to reduce the present L/D ratio for oil lubricated stern bearings from 2:1 was to be welcomed in that the shorter bearing could not only accommodate a greater degree of angular misalignment than the longer bearing but also would not suffer as great an absolute misalignment (end to end) relative to the shaft as a longer bearing on the same shaft.

However, because of the low speed, larger diameter shafting involving too high a unit loading could not be used. The reason lay in the relationship between oil temperature rise and bearing duty parameter: $(C/D)^2 (\mu N/P)$. A plot of $\Delta T/P$ v. duty parameter (Fig. 27) showed a distinct minimum point fairly close to the usual large stern gear operating conditions.

Any great increase in pressure would result in a movement of the operating point towards the left and should this go past the minimum point on the graph there would be an increase in operating temperature resulting in a decrease in operating viscosity and consequent reduction in film thickness. The graph of load carried against film thickness with this change of viscosity taken into account was shown in Fig. 28.

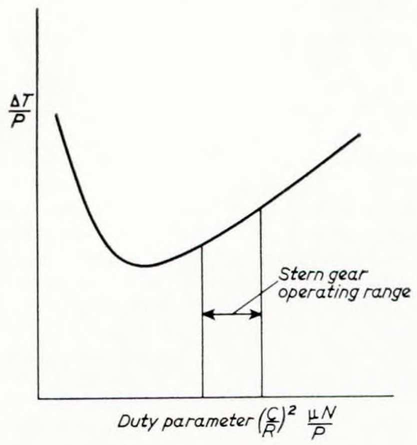


FIG. 27—*Temperature rise and duty parameter*

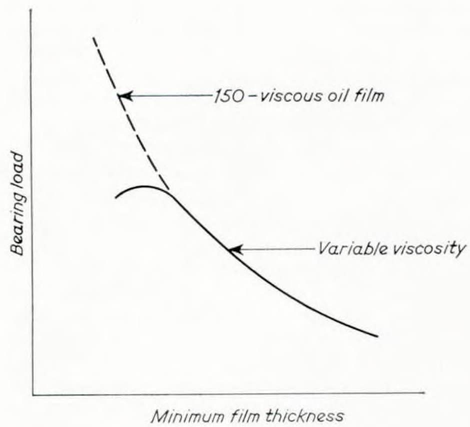


FIG. 28—*Effect of temperature rise on film thickness*

Fig. 29 had been prepared to show the relationship between minimum film thickness and load carried for various L/D ratios of a large diameter stern bearing. It had been assumed that the shaft had deflected in a parabolic shape over the length of the L/D = 2 bearing; the deflexion from the centre point of the bearing to the end of the bearing had been assumed to be 0·0002 L.

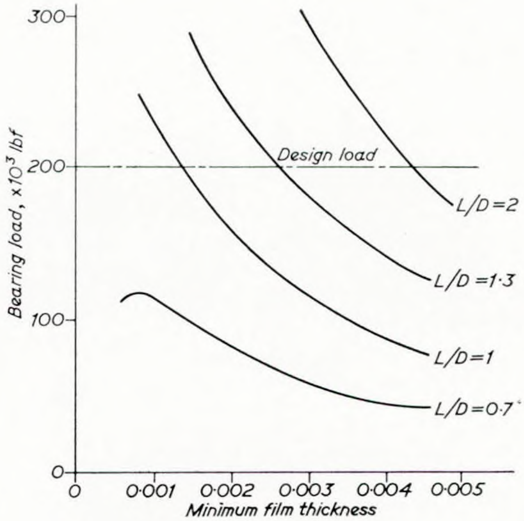


FIG. 29—*Load and oil film thickness for a deflected, large diameter, tail-shaft*

The method used had been to set the shaft to a given minimum film (i.e. at the ends of the bearing) and to summate the loads that would be carried by a series of strips of finite length along the bearing; side leakage had been taken into account but the assessment of side leakage was one of the least accurate factors in the calculation. In this particular case it would seem that L/D = 1·3 bearing would be a suitable choice; to go down to L/D = 1 could lead to metal to metal contact at manoeuvring speeds. A bearing L/D ratio of 0·7 suitable for high speed operation was shown to be incapable of supporting the design load.

The tilting pad journal-bearing suffered from the fact that under the loads and speeds encountered in stern bearings it would have a smaller minimum oil film than a plain bush.

To some extent this could be overcome by machining the pivot strips to allow a fair degree of self-alignment of individual pads.

The main advantages of the tilting pads, however, seemed to be the relative ease with which they could be withdrawn inboard and their dynamic properties. Because of the stiffer oil film the use of tilting pad bearings would result in higher transverse resonant frequencies and would also allow greater wear down before resonance became a problem. Also, seal damage ought to be reduced because of the reduced shaft movement.

Fig. 30 had been prepared to show the effect of wear-down upon resonant frequency for both a plain bush and a tilting pad unit.

The cures given had been based upon existing design methods which, although very satisfactory for smaller bearings, were not too reliable for larger ones. One reason for this might be the effect of tolerances upon the larger bearings invalidating some of the basic assumptions made in laying down the method. Not only was there a requirement for research into the overall design of the stern bearing but a requirement existed for a study of the operation of large diameter slow speed bearings.

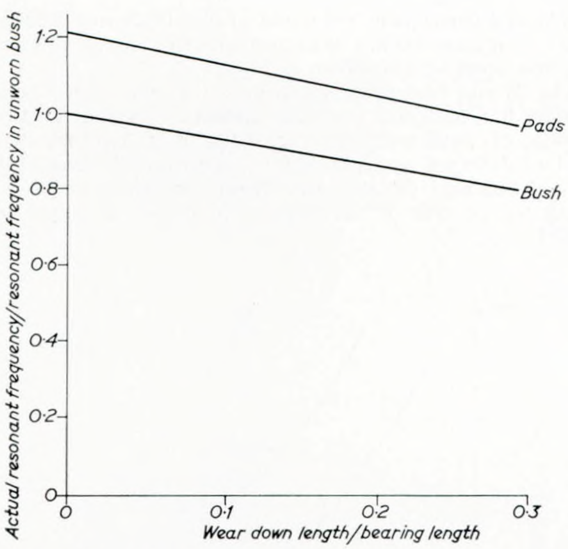


FIG. 30—*Effect of wear down on resonant frequency*

NOMENCLATURE

ΔT	oil temperature rise
P	pressure on bearing projected area
C	radial clearance
R	shaft radius
μ	absolute viscosity
N	shaft speed

DR. J. F. SHANNON, in a written contribution, thought that the power required for a million ton tanker could easily be met by the steam turbine and that in fact there was no limit to the power from this prime mover, for ships.

The practical limits of propellers were reached at the high power and low rev/min they required, and as the authors pointed out, the gearing might be impracticable under these conditions, in which case a compromise would have to be worked.

Since the authors had not dealt specifically with gearing it might have been useful to have given the limits so that proper adjustments to the overall arrangements could be made in this and in further surveys. The background to the gear limits was reviewed in the writer's joint paper, "Marine Turbine Propulsion Gearing", I.Mech.E. Gearing Conference, September 1970.

Extending this to suit the authors' paper, it was clear that with such low propeller rev/min, triple reduction from the turbine speed was required. With such large powers, the final reduction was where the limits might be reached with load factors allowed by the Classification Societies.

The limiting torque factor could be expressed as hp/N where N was the propeller rev/min.

Three distinct modern gear systems were possible with cross-compound turbines, viz:

1) Dual tandem parallel shaft gears with four pinions on the main wheel and with the first reductions either parallel shaft or epicyclic gears. Making adjustments in the gear ratios to suit the first and second reductions, the torque factor for the final reductions were:

   a) 126 in diameter wheel, nitrided, 250K, 2dp, $\frac{hp}{N} = 900$

   b) 180 in diameter wheel, through-hardened, 170K, $2\frac{1}{2}$dp, $\frac{hp}{N} = 1000$

   c) 192 in diameter wheel, through-hardened, 170K, $2\frac{1}{2}$dp, $\frac{hp}{N} = 1140$

2) First and second reduction epicyclic gears with two pinions on the main wheel:

   a) 126 in diameter wheel, nitrided, 250K, 2dp, $\frac{hp}{N} = 500$

   b) 192 in diameter wheel, through-hardened, 170K, $2\frac{1}{2}$dp, $\frac{hp}{N} = 570$

3) Final reduction epicyclic gear with parallel shaft second reduction gear:

   a) 100 in diameter annulus, $2 \times 12\frac{1}{2}$ in face, double helical, 5 planets, 250K, 2dp, $\frac{hp}{N} = 540$

   b) 160 in diameter annulus, 1×20 in face, spur, 4 planets, 250K, 1·5dp, $\frac{hp}{N} = 660$

The corresponding values of $\frac{hp}{N}$ for the million ton tanker were from Table II.

Single	Twin	Triple	Triple	
2220	1250	1150	1070	for the very low rev/min
1880	885	650	532	for 60 rev/min

Thus all the schemes except (2a) could meet the triple screw equal power arrangement at 60 rev/min. The dual tandem arrangements in group (1) with four pinions on the final reduction wheel had about double the capacity of the other schemes. They could be designed to give their maximum torque ratio or a reduced value giving an even greater margin of safety. The overall widths of the gearboxes were well within the centre distance between the shafts.

MR. A. STEEL, in a written contribution, said that it was indeed strange to hear at least one of the authors advocating the use of built propellers when his recollections were of a one time zeal in hunting down the possessors of such devices.

The authors had, however, readily acknowledged the usual disadvantages of built propellers and suggested that the advantages of their use in the circumstances considered have tipped the scales in their favour. And indeed when one had included in the range considered, propellers of up to 50·5 ft in diameter and weights of 150 tons, what else could be done?

Mr. Steel admitted to having been very impressed with the alleged achievement of the design of a built propeller with a boss/diameter ratio of only 0·172 as compared with the more usual 0·26 or thereabouts (Fig. 10). Were the authors sure that the design was workable on these small proportions? If it was, it was surprising that no one had at least approached these smaller proportions in the past.

He was struck by the peculiarly squat blade root section shown on the proposed new built boss. Was this a feature of the design? He appreciated that the t in the wt^2 function was the important feature for blade strength but would have thought that an adequate w was desirable to ensure against any possible torsional flexure of the blade about its own axis.

On the broader aspects of the paper, Mr. Steel was surprised at the concept of 97·6 ft draught and 300 ft beam ships. The former would surely increase the hazards of navigating and increase the chances of a *Torrey Canyon* type of disaster, with even more dreadful results.

Without knowing if the economies of scale continued above the present size of ship being built he had had the impression that the "oil men" considered that a plateau had been reached in ship size. Certainly while the operator enjoyed such economies arising from the large ships, it would have appeared from recent announcements that the shipbuilder did not share these advantages and that the sheer physical size of the structures had had an unpleasant and unforeseen effect on construction costs.

The manoeuvring of such large ships no doubt presented equally large problems as did the question of stopping them. He would have liked to have seen a diagram of the proposed fore and aft tunnel for inflow leading to transversely disposed discharge tunnels for the purpose of producing a transverse thrust at speeds above that at which a normal bow thruster became ineffective. That, without the impeller and with unrestricted discharge port and starboard, was a promising steady course stopping device. If the discharge could have been selected port or starboard with butterfly valves as suggested, then with the ship's forward motion would not an impeller in the system have been an impediment?

MR. J. STEFENSON wrote that the authors had given a very valuable background for those shipbuilders who were designing, projecting or developing ships requiring high powers. The paper would probably be referred to several times in connexion with large tankers and bulk carriers and fast and large container ships.

The authors had presented their proposals based on steam turbine machinery and had only briefly taken into consideration other types of prime mover. Figures were, however, given in Table II for the power requirements at 100 rev/min—these being for large bore or super bore Diesel engines. Mr. Stefenson therefore hoped that it would be of general interest to describe some features of a triple screw machinery for a large container ship which was being built at Oresundsvarvet; similar ships were under construction at Burmeister and Wain and Mitsui Shipyards. The machinery consisted of three Götaverken large bore Diesel engines, type DM 850/1700, one 12 cylinder engine for the middle propeller and two 10 cylinder engines at the wings. The total continuous service power was 75 000 shp.

When these ships were being discussed, different types of prime mover and machinery arrangements were discussed. The reason for choosing three large bore engines had been, according to the shipowners' statement, that they had wanted a proven type of engine from which the owners had had considerable experience since there were special requirements on the vessel that it should operate in a system of several ships on a timetable basis. The owners had insisted on Diesel engines for personnel reasons. The increasing costs of bunker oil were taken into consideration.

The operating system for the three engines had been described as follows:
The middle engine driving a KaMeWa variable pitch propeller would be operated in and out of harbour. In the open sea all three engines would be operated at full speed. Generally the three engines were provided with their own auxiliary systems so that they could operate independently of each other.

The maintenance of the total of 32 cylinders was planned as follows:
The middle engine would always be overhauled and maintained in harbour. For the two wing engines the maintenance and overhaul of cylinders could either be done at sea or in harbour. When one of the wing engines was stopped at sea, the propeller would be disconnected from the engine by a Renk tooth type coupling. This coupling was arranged between two thrust blocks, one for the propeller side and one for the engine side. The ship would continue the voyage on two engines with the third propeller free-wheeling. The propeller shafts on the wings were also provided with a break and locking device, capable of keeping the propeller locked when the clutch was operated for engagement or disengagement. A considerable amount of design work had been done for the wing shafting systems to provide the possibility of overhauling wing engines in the open sea.

The maximum continuous power available for a similar system as the one described above, but consisting of three 12 cylinder engines, would today be 95 400 shp which would almost cover the requirements set up by the authors for "The Propulsion of a Million Ton Tanker".

MR. E. A. STOKOE, M.I.Mar.E., wrote that the manoeuvring and stopping of such large units would indeed pose tremendous problems and he wondered whether the proposal of a ducted bow thrust unit would solve the problem of turning the ship. Would the authors care to speculate on the relative efficiencies of a bow thrust unit, an activated rudder and a bow rudder fitted above the ram bow?

There was considerable controversy regarding the stopping of ships in an emergency and it appeared that the ship's officers were given little conclusive data which would be upheld in a Court of Enquiry. The fitting of controllable pitch propellers provided an excellent opportunity to carry out research into the stopping of ships. With a fixed pitch propeller it was necessary to stop the engine and then build up to full speed astern, a time interval of between two minutes and five minutes depending upon the type of machinery and the circumstances. It had long been assumed that this time lag was an advantage in taking the way off the ship before running astern and giving reduced cavitation. The optimum time interval, if any, would be obtained by means of a c.p. propeller. The stopping distance of a triple-screw vessel might, perhaps, be reduced by the use of the rudder in addition to the propellers, the turning effect of the rudder being compensated by one of the wing screws.

The proposed propeller speed appeared low and Mr. Stokoe wondered whether sufficient control of ship speed might be obtained at low revs when manoeuvring.

Fig. 3 indicated the proposed disposition of the propellers. It was estimated that the draught to the wing propeller tips was in the order of 60 ft for a 1600 ft vessel. Would the ballast draught be sufficient to immerse the blade tips?

The proposed structural arrangements at the after end were excellent, providing complete continuity of longitudinal strength together with improved watertight subdivision. This was in line with recommendations made some 20 years ago and carried out to a smaller extent in a vessel built by S. H. & W. R. Ltd. at about that time. The subdivision could be further improved by separating the three engines by watertight bulkheads.

The support of the massive propellers caused some concern. If the optimum power arrangement was used for the one million ton vessel the resulting propeller weighed 85 tons. Mr. Stokoe suggested that this could well be supported on a shaft which passed through the propeller, with a bearing aft of the boss. This bearing could be integral with the rudder post.

This paper would no doubt be used as a basis for discussion for many years. It seemed a pity therefore, that the data were not expressed in S.I. units.

MR. D. G. YOKUM wrote that the authors of this paper had provided an intimation that some significant departures from the current trend of ship design were required to make their proposed million ton tanker a practicable reality. The emphasis on triple-screw propulsion, twin rudders, and controllable pitch propellers as essential or, at least, highly desirable factors in making this concept a reality constituted a rather radical change in thinking from current ideas in large marine propulsion plants. Their decisions on propulsion plant configuration seemed to involve only a cursory evaluation and a rather curt dismissal of the type of plant which powered most larger vessels today; that is, the high pressure-low pressure turbine combination driving a single fixed-pitch propeller through a reduction gear.

The authors predicated their equipment selection on such factors as take-home ability, manoeuvrability, propulsion efficiency, and technological incapability to produce a single unit propulsion plant of sufficient size to power such a vessel. The intention of this contribution was to point out that such a vessel could, and practically speaking should, be a direct outgrowth of current technological trends in merchant ship propulsion.

The triple-screw concept provided a virtually negligible contribution to manoeuvrability, and the maximum available manoeuvring thrust was that provided by the propeller wash acting against the rudder surface. This meant that a single screw design actually provided greater manoeuvring thrust than the triple screw, single rudder design due to the fact that a greater percentage of the total propeller wash acted upon the rudder surface. The wing screws of the triple-screw design contributed little towards manoeuvring power other than stopping capability, which was just as well provided by a single screw.

Propulsion efficiency was one of the major criteria used by the authors in propounding the need for the triple-screw arrangement. Though this was admirable from a technical standpoint, it must be considered that even a reduction of 20 per cent in installed horsepower did not fully compensate for the added cost of the more complex triple screw arrangement, with the necessary auxiliaries, support, and control equipment. A reduced fuel rate due to greater propulsion efficiency was attractive, but could be offset by higher costs for maintenance of the extra equipment and the possible need for added operating personnel.

Take-home ability was increased by utilization of triple screws, but how much was needed? Cross-compound turbines used for marine propulsion did, of course, provide emergency arrangements for supplying about 40 per cent of normal shp with either turbine disabled. If this was considered adequate for other major vessels, there should be no good reason to change the rules for this size of ship.

Propeller size was cited as a problem in a single screw arrangement, because facilities did not exist for manufacturing such a large wheel in one piece, the method the authors preferred. It was admitted, though, that the size posed no problem in a built-up design. The built-up propeller would seem to be particularly advantageous in a propeller of this size from a standpoint of repair cost reduction. Damage of a single blade on a one-piece propeller required considerably more time and effort to repair than on a built-up screw; one blade could be replaced with a spare, rather than replacing the entire propeller.

Reduction gearing posed perhaps the most significant problem. Limitations in available facilities did restrict gear size to the point that a conventional articulated double-reduction gear design could not presently be built to provide the low output rev/min required. Current designs were handling 32 500 shp with an output speed of about 85 rev/min. It was difficult to envisage this type of gearing growing to accommodate over 100 000 shp with a 60 rev/min output within the few years allotted before being faced with the practical problem. It was possible, however, to envisage this problem as resolved by a departure from current commercial marine gear train designs. For example, locked-train gearing offered obvious advantages in its ability to handle greater torques without resorting to extremes in face width. These units, which offered maximum space and weight savings, had found wide application aboard naval vessels, and their technology was well established. A practical approach to the gearing problem might have been to have utilized locked-train gearing with an output speed of, say, 150 rev/min to reduce gear size; the output shaft could then have been coupled to a planetary type, in-line gear to give the final speed reduction. This triple-reduction arrangement provided the most feasible approach to obtaining the required single-shaft speed and torque while presenting the gear industry with the minimum impact on its facilities and technology.

A 100 000 hp cross-compound marine turbine and condensing unit was something new. Mr. Yokum could not imagine for a second, however, any major suppliers of steam turbines turning down a request to quote on such a plant. Turbine manufacturers regularly produced even larger, in-line units for power generation, and would find few hurdles to jump in producing a marine propulsion unit of this power.

Current large vessel propulsion systems were configured the way they were for a number of good reasons, among them economy, reliability, maintainability, and simplicity of operation. On page 175 of the paper, the authors stated that the power requirements " . . . can only be met by either a triple screw or twin screw installations . . . ". Mr Yokum did not believe this to be the case; he thought it more probable that the super-size tanker of the future would be the result of an orderly, logical growth pattern, with few deviations from what had been found to be reliable and economically effective.

Authors' Reply

To reduce repetition the reply has been based on subjects dealt with in the order in which they appeared in the paper. It is hoped that contributors who have made comments on different sections will tolerate the rather disjointed replies to their separate points.

At the end of his discussion Mr. Norris outlined a system under which the very large crude oil carriers might work. This would mean a fleet of similar ships operating a scheduled liner service between terminals able to load or discharge the cargo in the interval between successive ships. When considering the risk involved with very big ships it was at least possible that a few ships on a scheduled service might be safer, as well as cheaper, than many smaller ships. The recent legislation on tank size might well cause the cost per ton to flatten out before reaching the million ton size.

Mr. Hawdon drew attention to the lack of discussion on propeller induced vibration. The hull used had been kept down to 0.85 block coefficient, care had been taken to provide adequate clearance and the power was divided between three propellers. These were the main precautions taken to control the problem. There could be little doubt that for a particular ship the ordinary resistance and propulsion tests would have been quite inadequate and that the hull shape should have been modified in special experiments with measurement of surface pressures and transmitted forces. The paper had been intended to explore the problems of large power requirements and the use of low speeds avoided the difficulties. For any particular service and set of costs there would be an optimum speed because the earning capacity decreased with speed whereas the hull capital charges were unaffected.

This point was brought out by Mr. Crowdy who filled a gap in the paper by making an economic analysis. When the paper was written it had been felt that there was insufficient data to determine the best speed for a ship operating in five years time. As tanker size increased the low resistance/speed^2 range extended to higher speeds but this had not been reflected in any change in average speed, which had remained at about 15 knots. For the reasons given in the paper a rather higher speed had been used.

The restriction in tank size and consequent increases in capital cost brought down the optimum size. An increased insurance premium would have removed very big ships from consideration. The answer to the first question asked by Dr. English was that adequate manoeuvrability and control might prevent the application of excess premiums. The transverse thrust units would be discussed later. The two basic questions which could not have been resolved were:

a) Was it worth while installing a 15 000 hp transverse unit which would only be used when the ship was almost at rest at the terminal ports?

b) Was it feasible to have a unit available at short notice to help in an emergency, which would work when the ship had appreciable ahead speed?

The chosen hull form was not of primary importance to the purpose of the paper; it represented a resistance to be overcome and provided real physical limitations to the siting of machinery and propellers. Nevertheless, Dr. English's intuitions were quite irrelevant. The streamlines shown on the normal hull drawing had been measured on the model. The knuckle lines followed these measured streamlines. The measured resistance with the straight frame model had shown that there was not a big increase due to crossflow. If, in spite of the published evidence, the eight foot model resistance results were not accepted, it was easy to check them against the 0.85 block coefficient results from a well known series, believed to be tested at N.P.L.

Mr. Jourdain was correct in his suggestion that the use of the same form was an over-simplification. For the large slow running single screw, cutting back the after end lines would have incurred an appreciable increase in resistance.

As Mr. Langham was aware, the original intention had been to make model experiments first and to use the results as the basis for the paper. The decision to start the model programme had been delayed and the two departures from orthodoxy – the use of a broad shallow hull or the raising of the wing screws would each have required extensive model testing to obtain optimum results. It would have been unreasonable to have expected the propeller company to pay for such a large scale investigation. It was expected that the sample results would give sufficient indication to allow ideas to develop and it was regretted that they were not yet available. After the last paper to the Institute by two of the authors, experiments had been carried out on the triple screw alternative to a single screw tanker and had verified the 10 per cent reduction in power predicted in that paper for the particular conditions and revolutions. In long term propulsion investigations it was not easy to time an up to date progress report for publication with full supporting information. It was hoped that the virtue of freshness outweighed the disadvantage of incompleteness in a paper of this type.

It was very useful to have had comparative information, particularly on propulsion, from the Ministry of Technology's study. As Mr. Paffett said, the block coefficient could have been increased to 0.90 without very heavy penalties in power. The real question was how much allowance should have been made for other considerations; the reduction in fullness to ensure directional stability; the reduction in fullness so that the ship could be controlled with very little water below the keel; the possible fining of the after body to reduce propeller induced vibration. For the present ship with a length/breadth ratio of 5.3, 0.85 was not regarded as a conservative choice of coefficient.

It would be interesting to see the evidence of idiosyncracies in the model results with full ships. One would have imagined that the nearer one approached a box shape, the less variation there could have been of separation position with Reynolds Number; model scale was hardly an answer. But in the present context there was sufficient evidence of consistent model results up to CB=0.85 and Mr. Paffett's own value of 76 500 hp for a 0.90 form against 72 600 hp for a 0.85 form was a very normal effect of increased fullness.

Mr. Paffett was, of course, correct in saying that:

$$Cp = \frac{0.075}{(\mathrm{Log}_{10} R_n - 2)^2}$$

had been defined as a correlation line, but in fact, it did not correlate model results with anything. If a comparison of model prediction with ship trials was required, the B.T.T.P. comparison in the range of 400 ft to 1000 ft ships, supplied an empirical correlation factor $(1+x)$ which decreased in that range. In the model range, correlation, that is, consistent results, were obtained by using a much higher proportion of the resistance as Reynolds Number dependent. For tankers models a numerator of 0.090 instead of 0.075 had this effect. Applying this form factor correction to the ship size left a fairly consistent percentage difference between a model prediction and ship trial result and this provided a basis for extrapolation to greater lengths. It would be of interest to have suitable form factors applied to the B.T.T.P data to see if this could be used as more than a rough guide.

There was a mistake in the published diagram for the 500 000 ton ship (Fig. 6) and this had exaggerated the apparent gain with triple screws. A corrected diagram has been produced for this final publication. As mentioned in the reply to Mr. Langham, experiments with a smaller and rather finer tanker substantiated the previous triple screw prediction and it was believed that the potential advantages in efficiency from lighter propeller loading, from frictional wake gain, and from the absence of twin rudders was much greater than the five per cent Mr. Paffett had so far achieved.

Mr. Burnett raised the question of manufacturing capacity for propellers. Propellers of nearly 60 tons had already been in service for some time and were now frequently required for large tankers. Problems associated with handling them had, therefore, been faced and overcome. If these much larger ships were to be built, it would not only be propellers that might need to be larger but nearly everything would be on a grander scale and it was reasonable to suppose that handling problems would continue to be soluble. Nevertheless, new materials and new manufacturer's processes must always be sought and considered in order to improve and progress. The cost of time out of service due to maintenance was a vital consideration and any methods of reducing this deserved examination.

New processes such as fabricated (hollow) propellers and new materials such as carbon fibre reinforcement have already been considered in some detail. While these might be technically possible, they were not economically viable, but it was possible that this might change. There were also technical disadvantages such as increased thickness in the case of hollow propellers and serious cavitation erosion problems in the case of carbon fibre reinforcement.

Flange mounted propellers advocated by Mr. Crombie were used in the case of controllable pitch propellers and there was no serious practical difficulty involved. It would appear, however, that the average marine engineer had been so far disinclined to change from the practice of the fitting of a bored propeller to a shaft taper which, after all, had been attended by considerable success over the years.

If the flanged principle was used, the necessary p.c.d. of the fitted bolts in the flange for adequate strength would be high and therefore would give rise to higher boss diameters than used hitherto on normal fixed pitch propellers. The reduction in propeller weight and bearing loads referred to was, therefore, a little optimistic, but none-the-less worthy of investigation. The consideration of these relatively minor details in such an installation could not have been effectively dealt with within the text of the paper, and therefore Mr. Crombie's remarks provided an extremely useful contribution.

Dr. English asked about the supporting of large slow-turning propellers and the lubrication problems but these did not seem to present insurmountable difficulties. It should not have been too difficult to persuade an owner to accept controllable pitch propellers which were now well established in the merchant ship field and which, from the exercise referred to in the paper, did not in this special case introduce new problems.

With regard to the remarks regarding bow thrusters, experience had shown that the conventional arrangement became increasingly inefficient as the speed of the vessel increased and this was largely related to the difficulty of ensuring an adequate flow of water to the tunnel when the ship had forward movement, even with the use of scoops or special tunnel endings. The fore and aft tunnel with the aperture in the fore end of the bulbous bow would have ensured adequate water flow whatever the ships speed and has the added advantages that result from the use of uni-directional engine and propeller. This could have simplified and cheapened the installation and would have also made it easier to have provided a prime mover. The magnitude of thrust required to provide adequate movement of the vessel's head was difficult to assess and there would have appeared to be scope here for experimental work not only in the case of the million ton ship but for ships down to the more normal size (if a quarter of a million tons could be considered normal).

There would have seemed to be no virtue in using steel for the large built-up or controllable pitch propeller blades as the well developed non-ferrous alloys would prove very adequate for this purpose. Blades of a weight as high as 20 tons each might be called for and the well developed foundry techniques and the high corrosion/erosion resistance of the conventional propeller bronzes, together with their relatively high fatigue limits made them extremely useful for this purpose. The cast steels were not as good from this point of view and the stainless steels which might be available giving comparable performance were difficult to obtain in castings of the required size.

Finally, the authors agreed with the remarks of Dr. English regarding the use of twin, triple, contra-rotating, tandem and ducted propellers, but would reserve their judgement of overlapping screw propulsion. They said that they had insufficient knowledge or experience of this innovation but could see that there might be advantages under certain conditions.

It was satisfying to be re-assured by Mr. Langham and Mr. Graham that whatever ship designers might require, the propeller manufacturer in this country would be able to fill the demand, and also arrange transport facilities for the largest propellers forseeable. The possibility of propellers being made weighing 200 tons might at this stage seem unlikely, but ten years ago the same could have been said of the propellers currently being made weighing around 60 tons. There was no doubt that from the point of view of efficiency and loading the trend towards lower rev/min and higher propeller diameters could usefully continue in the special case of the tanker where the draught was continuously increasing with ship size. The comments made regarding the repair and servicing of propellers were appropriate, particularly bearing in mind the enormous cost involved if such vessels were delayed or immobilized.

Mr. Hadler's remarks regarding single, twin and triple screw configurations were appreciated and it was agreed that the possibility of overlapping twin screws might be investigated in any final consideration of the propulsion of such a vessel. This should improve the competitive position of the twin screw arrangement, but it would be appreciated by Mr. Hadler that in an exercise of this kind the number of variables had to be limited to some extent and it had therefore been decided to restrict the investigation to relatively conventional shafting arrangements.

It was agreed that the possibility of blade vibration was a further factor to be taken into account on large propellers and experimental verification of the fundamental blade frequency had been carried out in the U.K. on a number of propellers which gave close agreement with an acceptable method of calculation. There was no doubt that if reasonable account was taken of the depression of the fundamental frequency on immersion in sea water that the blades of some large propellers had a fundamental frequency significantly close to rev/min x 10.

Hydrodynamic excitation might therefore be an important contributory factor in the premature failure of certain five bladed propellers. Full scale experiments were called for to investigate vessels where the possibility existed of the fundamental blade frequency being excited at resonance.

The remarks of Mr. Jourdain regarding the application of triple screw propulsion to such a vessel were appreciated, as well as his reminder of the economies to be achieved by the omission of a reversing gear on the wing shafts.

It was agreed that in the case of triple screws, as the power on the centre propeller reached the limiting values of 0 per cent and 100 per cent, there should have strictly been a discontinuity in the curves. In each case there was a change in the hull geometry due to the disappearance of the centre shaft in one case and the wing shafts in the other. It had been, however, desirable to make use of the relatively simple single screw and twin screw cases as end points on the curves in order to reduce the complexity of the diagrams. A compromise had therefore been made whereby the resistance with triple screws was assumed to be the same as that with twin screws while the power on the centre screws was small and then assumed to reduce gradually to that of the single screw as the power on the wing screws approached zero.

Provided satisfactory manufacturing techniques were employed there was no great difference between the soundness of separately cast blades and that of the face root fillets of solid propellers. Because of the faster rate of solidification the mechanical properties at the roots of the separately cast blades would be slightly better than those of the solid propeller, but in the case of very large propellers the difference was probably very small.

There was not, as yet, a great deal of information about residual stresses in propeller castings and, particularly in separately cast blades, it would have probably been unwise to make any assumptions in this respect until some more reliable data was obtained. Changes of thickness or stress would not therefore have been desirable and any that might have been justifiable would have only had a minimal effect on efficiency. Built-up propellers were relatively inefficient compared with solid propellers, primarily because of their large boss-diameter and restricted blade root design, and slight changes of blade thickness would not have significantly effected this situation.

It seemed likely that, as Mr. Rolland suggested, benefit could have been derived under certain circumstances from the fitting of shrouds to the outer most screws of the triple screw arrangement. Unfortunately it had been impossible, within the scope of the paper, to cover such a wide field, and although ducts and shrouds would have an increasing part to play in the years ahead, it had been agreed to leave out as far as possible, relatively untried devices in the consideration of the propulsion of such a vessel at this stage.

Mr. Steel, with tongue in cheek, referred to the day when this company had influenced a number of ship operators to replace built-up propellers with fixed pitch solid propellers on the basis of substantial savings in fuel consumption. Ships such as the Shaw Savill *Bays* and the Cunard *Franconia* class had been dealt with in this way shortly after the war and had shown a clear nine to ten per cent improvement in fuel consumption with a considerable reduction in total propeller weight. These ships had had propellers weighing between 10 and 15 tons, each absorbing 4000 to 6000 shp, and it could not be conceded that there had been any change in attitude to this type of propeller.

In the paper, however, the investigations had included a million ton tanker with a single screw, and if such a vessel were to have a service speed of 16 knots, the optimum propeller would have been 55 ft in diameter and 255 tons in weight. Such an installation might not be beyond the bounds of possibility, but all would agree that a single piece casting of this size was not a solution that would have found favour among ship owners and builders. Alternatives must therefore have been considered in such a paper and the loose bladed propeller had therefore been introduced, it having been understood that the motives for this had been very different to the earlier cases to which Mr. Steel referred.

Using these low rev/min and high propeller diameters, it was found that blade-area-ratios were surprisingly small, which was the reason why both controllable pitch propellers and loose bladed screws were not necessarily a difficult engineering possibility. The boss diameter in both cases was related to flange size, which in turn was related to blade root width, and Mr. Steel could be re-assured that for 32 000 shp at 62 rev/min the boss-diameter-ratio shown was a practical one. The high thickness/chord ratio shown in the drawing in the paper was neither desirable nor necessary for such a propeller and the authors apologized for an apparent mistake in the drawing.

In the case of such an enormous vessel the problems of manoeuvrability which had been raised by Mr. Stokoe, could not be over-emphasized, and one would immediately agree that any discussion on the relative merits of active and passive steering aids would be purely speculative. Experimental work, and the consideration of the full scale performance of large ships currently in service was needed to provide back-up information. Nevertheless, it appeared to the authors that the pushing over of the head of the vessel by a suitably powered and located bow thrust unit would have been the most effective way of assisting steering, and if this was situated in the ram bow as proposed, certain practical difficulties would be overcome. There was little information available on bow or activated rudders in vessels approaching this size.

Mr. Stokoe's remarks regarding the stopping and reversal of the main machinery were useful, as were those regarding the use of rudder to assist braking. It was not obvious to the authors why the relatively low rev/min favoured in the paper should have in any way reduced control at lower ship speeds. No deleterious effects from this point of view had been reported in the case of the latest vessels, where the rev/min had been successfully reduced from 100 to 110 rev/min down to about 80 rev/min for shaft horsepowers exceeding 30 000.

In making a general reply to Mr. Yokum, it should be pointed out that the choice of triple screws was not based purely on propulsive efficiency and if it was accepted that a speed of 16 knots was to be achieved on a vessel of one million dwt, a power in the region of 100 000 shp would be required. In these circumstances in the view of the authors it would have been hazardous and unrealistic to have assumed that one propeller would have been adequate.

Safety, reliability, propeller size and immersion as well as manoeuvrability must be taken into account and this had come out very strongly in the discussion on the paper. If one propeller was not favoured, all the authors were saying was that three rather than two shafts were perhaps the better alternative, giving the best solution while still using engines of a size commonly used at that moment. It would have been expected that manoeuvring in terms of turning would have been improved by the moment of thrust from the wing shafts about the centre line of the vessel.

The machinery installation had been chosen in the context of the vessel it had been designed to propel and equipment considered only if it had already been proved in service. Statistics for ships over 175 000 dwt under construction and on order during the three years up to 1970, had shown that about 90 per cent of these vessels were propelled by steam turbines. A single main engine of high power as had been suggested by Mr. Yokum was unrealistic and not part of a logical growth pattern. The decision on the number of boilers and main engines to be used had been influenced by the fact that units of this size were already in service, because advances in unit output or the introduction of new equipment had usually led to at least a temporary loss in reliability. This trend was confirmed by statistics contained in a survey published by Fearnley and Eger's Chartering Co. Ltd., showing a rise in the off hire periods for large tankers, the figures increasing with size of ship. Steam turbine propelled ships had shown an advantage in this respect of some two to three days, figures had been confirmed by a recent study carried out by the Norwegian Ship Research Institute using information from their shipowner and shipbuilding members. The report showed a steady financial gain for a turbine propelled vessel compared with the slow speed Diesel as the horsepower of the installation increased before any allowance was made for the improvement in availability. When it was considered that the cost of chartering replacement tonnage could be £100 000 per day for a million ton tanker the operational advantages of a steam turbine were further emphasised. It was significant that the contributions from Mr. Gibbons and Mr. Isherwood who were both concerned with the operation of large tankers gave most emphasis to the question of reliability. This and the associated problem of maintenance must now be given more consideration at the design stage, following the approach already common in the process industry. In addition to defining planned maintenance schedules for equipment and ensuring that removal routes were available the major tasks were studied to establish their content. The model for the installation was used to make this study and to determine the maintenance policy on both equipment and manning. A reference was also made to reliability engineering as an aid to the rational choice of equipment in systems. The mathematics of this technique were understood but it would be some time before information collected during the service of vessels would be sufficiently comprehensive to make this possible. The approach adopted in designing the installation was summarized by Mr. Foreman's comments asking that there be "evolution not revolution". In recent years installations had contained too much equipment unproved in the marine application and, in the case of controls and instrumentation, fashion had often been a greater consideration than function.

Professor Chambers and Mr. Stokoe suggested that the machinery be located in separate compartments so that an accident in the engine room would not immobilize the ship. A common machinery space which was a feature of many passenger ships had been adopted because it had not been considered to be a high risk. Subdivision of this type was more common on M.O.D. (N) ships where damage resulting from action was a factor. It was not intended that the machinery spaces be unmanned and comprehensive fire detection and fire fighting equipment would be provided. A development of this approach had been to make the feed system common after the air ejectors because an L.P. steam generator had been provided to supply the contaminated services to maintain the required level of purity in the main system. Mr. Norris had been anxious that the space aft of the machinery compartments nominated for the carriage of cargo be eliminated. This would have been possible if the lines could have been changed so that the flat of bottom region came further into the aft end. The cargo pumps had been arranged athwartships at the first station wide enough to accommodate them. Providing this change could be made without affecting the flow into the propellers and increasing aft end excitation, the pump room and the machinery spaces could be moved aft. In the present location, however, the volume aft of the engine room was too large to ignore and this had led to the proposal of tanks aft of these spaces isolated by a cofferdam.

Mr. Adolph and Mr. Hudson drew attention to the fact that fuel prices had risen rapidly during the last year. This was true of all operating costs so that the relative importance of fuel for any particular size and type of ship had not changed significantly. The proportion of the total operating cost represented by fuel for a given type of ship decreased with size so that reductions in fuel consumption became less significant. In addition to this, machinery with lower fuel consumption's were generally less reliable, thus reducing the availability and income of the ship. The reduction in the differential between distillate and residual fuel was still significant although it had been reported that certain owners had operated slow speed Diesels on higher grade fuel to reduce maintenance. Mr. Adolph's comments on the burning of crude oil were very interesting but this had not been considered in the paper because of the safety aspects.

A twin-screw installation, as suggested by Mr. Foreman, had not been adopted because the triple-screw installation, which gave a saving of 10 000 hp, kept the turbine frame sizes within existing ranges, improved the safety of the vessel by having three shafts and gave a marginal improvement in the stern-bearing load. The revolutions had been reduced to 60 rev/min. because the resulting propeller characteristics were within the capability of projected manufacturing facilities. The resulting stern bearing loads and propeller weight had been considered to be within acceptable limits. Ducted propellers, as suggested by Mr. Rolland, or an overlapping arrangement proposed by Mr. Hadler, had not been adopted because of the increased complexity of the installation. The overlapping arrangement would have certainly led to difficulties in arranging and aligning the propeller drives. It should be noted that the triple screw arrangement proposed in the paper was based on a fixed centre propeller and controllable pitch on the wing shafts was only suggested as a possibility. Mr. Stokoe asked if the immersion of the propeller would be adequate in the ballast condition. A mean ballast draught of 56 ft had been considered reasonable with a trim of 15 ft by the stern giving a draught aft of about 64 ft. This would have left a tip immersion of 7 ft which was equivalent to the figure for the current quarter million ton tankers.

Mr. Hudson advocated that the aero-engines used as gas generators by M.O.D.(N) for the propulsion of warships be considered. It should be noted, however, that the use of the Olympus gas turbine had not been proved and that shore tests up to 10 000 running hours were still in progress to assess its reliability. Previous experience indicated that salt in the air and fuel led to deposits on the compressor blades and corrosion in the turbine. This had been partly overcome by development in materials and water-washing techniques introduced to remove the salt, oil and dirt deposits from the compressor. These difficulties had, however, been experienced in installations using Diesel oil and had occurred together with other problems such as noise and failure of rotating components due to vibration. Mr. Coats suggested that the heavy duty industrial gas turbine already widely used in land base applications be used in a combined cycle with a steam turbine. This led to complications in operating two types of machinery and a need to introduce supplementary firing in the waste heat units to achieve a standby capability and also to provide cargo pumping capacity. One of the principal advantages of the gas turbine was its simplicity and the possibility that this might ultimately lead to significant reductions in maintenance. A combined cycle with the steam conditions suggested sacrificed these features and it was possible that an arrangement would lower steam conditions with a helper turbine driving the compressors of three separate gas turbines, one on each shaft would be a more attractive installation. There was no doubt that the gas turbine once it had been marinized would have considerable advantages, being easy to remotely control and basically a simple machine. The contract placed by the Maritime Administration in the United States for an £8-million, five year research effort gave some measure of the development still required. This programme was aimed at adapting the industrial heavy duty gas turbine for ship use. Among the subjects studied would be the large exhaust gas regenerators needed to improve the economy of the cycle, the unidirectional features of the turbine and the question of fuel treatment when using low cost bunkers.

Mr. Brownlie suggested that a re-heat steam cycle be used with the two L.P. turbines driving the outboard shafts, the remaining cylinders being coupled to the centre propeller. This made the three main machinery units inter-dependent, a situation which had been avoided in the proposed installation. It was noted that the number of rotors had been reduced but usually these were reliable elements whereas couplings for power take-offs and the operational difficulties they introduced led to a reduction in availability. The contribution from Mr. Isherwood supported the authors' view that design performance and the desired availability were seldom achieved in such advanced cycles. One of the main difficulties was in the operation of the plant, not only in a marine environment but with staff who generally had not got the specialized experience of the personnel working in land power stations. The number of boilers had been chosen so that the units were in the size range currently in operation at sea so that some operational experience existed. The information supplied by Dr. Shannon was very useful because it was always difficult to obtain data on the limits of performance of the various gearing proposals available. The dual tandem arrangement with four pinions on the final reduction wheel had been preferred and it had been noted that this gave double the capacity of the other schemes. A through hardened main wheel had been preferred and this would have been arranged to give a margin on tooth loadings to accommodate any misalignment that might occur.

The reference made in the paper to the number of cylinders in either a slow-speed or medium-speed Diesel installation was concerned with the maintenance load and reliability aspects. It was reasonable to assume that the reliability would in some way be related to the number of moving parts and in the case of the slow speed Diesel, the load referred not only to the manpower content but to the difficulties of working with the large component weights. It was difficult to believe that the large rotating masses and their associated unbalance were the correct engineering solution to providing the powers being considered. Mr. Stefenson described the use of large low-speed Diesel engines in a triple screw installation using both controllable and fixed pitch propellers. The revolutions of these engines were ideally suited to the container ship which had a restricted draught. In the case of larger tankers with a continuously increasing draught, the best results from the propeller point of view were achieved by taking maximum advantage of the greater immersion available by using the largest permissible propeller diameter running at the optimum revolutions per minute. This would lead to propellers running well below 100 rev/min on a one million ton vessel. The maintenance programme proposed for the container ships took advantage of the number of units available by proposing that the wing shaft engines be overhauled either at sea or in harbour. The provision of three separate units could have reduced the off hire time of the Diesel installation because failure would have at least initially resulted only in a reduction in ships speed. In general, however, with the tendency towards reduced manning and an emphasis on minimizing overhauling, this would not have been acceptable to every operator. Mr. Crowdy and Mr. Norris both proposed the use of medium speed Diesel engines in the installation; in the latter case these were confined to the wing shafts. Mr. Crowdy's contribution was particularly interesting as it gave more information about an engine that was currently being developed. This installation was preferable to the slow-speed alternative and had the additional advantage of being geared down to lower propeller revolutions. The analysis of operating costs suggested that the ships speed should have been 16½ knots but this was dependant upon the assumptions used in the calculations. For example, it was doubtful whether the machinery would have given a fuel consumption of better than 0·38 lb/shp/h based on net calorific value and some allowance should have been made for the consumption of the more expensive lubricating oil. The overall purchase cost assumed was probably at least 30 per cent too low and the off hire figures quoted were less than those currently established for quarter million ton tankers. A claim that the installation could have remained unavailable for an additional three weeks to absorb the fuel cost differentials was therefore wrong by a factor approaching 10. Current experience on slow speed Diesel maintenance costs for powers in the region of 30 000 indicated that they were three times those of a turbine installation. The differential increased with hp in a way that suggested that the large bore Diesels were proving expensive to maintain and a medium speed installation could have had advantages in this power range.

A number of contributors mentioned the problem of cargo pump drives which, in this case, would require approximately 30 000 hp employing two of the main boilers at their full output. For the inlet conditions of 800 lb/in^{2}g and 750°F exhausting to an atmospheric condenser, the cargo pumping load was of the order of 530 000 lb/h. This gave some indication of the additional firing required in the waste heat boilers proposed by Mr. Coats for the combined steam and gas turbine cycle. Electrically driven units have also been suggested but as the high voltage systems required to keep equipment and fault levels down to acceptable values have not proved reliable in service, they have not been adopted. The voltage of 440 would have probably been reasonable for the remaining equipment on the ship. A further alternative was to arrange the output from the main engines so that it was available for driving cargo pumps but as such arrangements required design and development they had not been considered. There would need to be at least a cold tank cleaning system and as reductions in speed were unlikely to be accepted for such vessels some means of driving two cargo pumps had to be provided. In the steam installation additional capacity had been added to the boilers so that an output beyond normal service evaporation was possible.

A discharge time of 48 hours had been selected because it had appeared to be a reasonable development from approximately 24 hours used on existing quarter million ton tankers. The discharge rates attainable towards the end of the operation would have been limited by the piping arrangements within the tanks and the flow around structural members. It might also have been uneconomic to provide shore facilities capable of accepting higher transfer rates. Mr. Bell suggested that an independent barge carrying cargo pumps be considered as an alternative to providing units on the ship. There were, however, occasions during the voyage such as ballasting, for the outward journey and deballasting which required such large quantities of water that additional pumps would have had to have been fitted if cargo pumps had not been available. The quantities required for tank cleaning also involved using the cargo oil pumps and the stripping lines on existing tonnage. Pipework for the cargo oil ballast systems would have still been necessary for working cargo and for trimming the ship, so that the reduction in first cost by saving equipment installed in the vessel might not be as significant as expected. Suction lift and flow around structure to the pipework was already a difficulty and was unlikely to be assisted by providing portable connextions from a barge to the tanker at, or around, the water line. In addition to this, control of the pumps was normally carried out from the tanker at a central position to ensure that optimum pumping conditions and acceptable load pattern were maintained. A barge itself could have presented additional problems when unloading in poor weather conditions and would have probably required its own propulsive power for manoeuvring into position. The question of prime mover for the bow thrust unit had been mentioned by a number of contributors and although this had not been considered in any detail cargo pumps could have been used to provide thrust by discharging through one side of the ship.

Mr. Crombie and Mr. Milton offered designs which made in the first proposal the bearing surfaces more accessible and in both cases enabled the tail shaft to be inspected for survey purposes without drydocking the vessel. Approaches of this type had particular relevance to very large tankers with their special docking problems. Both would have reduced or made unnecessary the drydocking of vessels purely for the survey of stern gear and helped to have eliminated time out of service. Both approaches introduced a degree of complexity and it could be expected that simpler solutions would become available after some experience of these designs at lower powers. Mr. Milton's arrangement had the advantage of placing the propeller bearing vertically below the load, eliminating the bending moment from overhang and in addition to this, separated the torque from the bending moment effects on the drive shaft. The advantage of being able to withdraw the tail shaft for survey was partly lost because it was not possible to inspect the bearing surfaces without removing the propeller. It was true that the flexibility of the shafting was improved by removing the propeller load from the tail shaft, but the drive would have had to be have been arranged to give sufficient load in the first plumber block to avoid whirling. Mr. Stokoe's suggestion that outrigger bearings should have been used located in the rudder post, could have only been applied to two of the shafts if they had been aligned with the twin rudders installed. The problem of sealing and providing lubrication to the outrigger bearing and the flexibility of the shafting system with particular reference to the distributuion of load between the bearings would have to be considered carefully. The bearing loads encountered in the installation were only marginally above those already in service in existing designs avoiding the need for special arrangements. The information given by Mr. Rose had illustrated the lack of an established design method for stern bearings. The plot of temperature rise against duty parameter showed a minimum value beyond which increase in load and the resulting increase in temperature rise caused a rapid decrease in the film thickness. If a bearing was designed to carry a given load and a series of length diameter ratios considered, it was found that for a given diameter the shorter length bearings were incapable of meeting the requirements. In addition to reviewing the theoretical basis for designing such bearings, results from a test rig or measurements taken during service were required to provide data so that design procedures could be developed. Mr. Foreman pointed out that the viscosity of the oil used in the system was high at the sea water temperatures encountered during operation. This characteristic was necessary to sustain hydrodynamic lubrication of the bearing in the greatest possible range of revolutions during manoeuvring. It was probable that on the turning gear and at similar low revolutions for the main engine boundary lubrication conditions existed. When the ship was started from 'cold' or was manoeuvring for a port at the end of a voyage, the coolers were kept in circuit to keep the oil temperatures down during low speed conditions.

The results of the model self-propulsion tests for the 750 000 ton ship have only just come to hand and therefore only a simple analysis has been possible in the time available.

For ease of comparison, the results for the single screw, designed for 50 rev/min, and those for the triple screws, designed for 90 rev/min, have been adjusted to give equivalent powers at 70 rev/min, the speed for which the twin screws were designed. These are presented in Fig. 31.

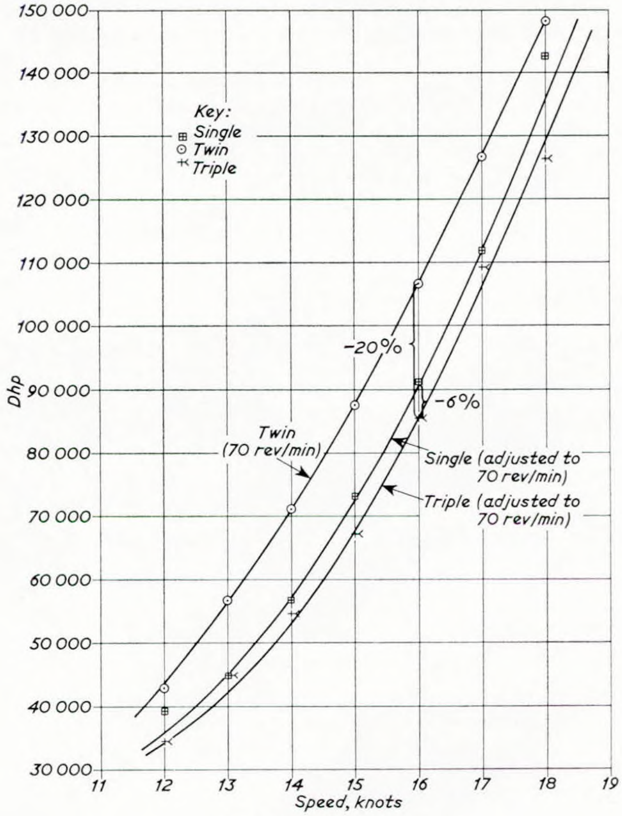


FIG. 31

The powers did not correspond exactly with those estimated in the paper but this was largely a matter of estimating correlation factors, wake fractions, thrust deductions and relative rotative efficiencies, all of which were largely unknown for ships of this size, particularly for twin and triple screws. One of the objects of these tests was to obtain guidance on these parameters. The other object was to ascertain if the advantages of triple screws, estimated from theoretical considerations, would also be indicated by model testing. At 16 knots, the power prediction for the triple screw arrangement was 20 per cent less than with twin screws and 6 per cent less than with a single screw. However, the single screw was hardly a viable proposition from consideration of ship safety; it would necessitate nearly 100 000 shp being carried on one shaft, also the propeller would need to be about 43 ft diameter weighing about 165 tons. The practicable comparison was therefore between twin and triple screws. The model results were considered to give reasonably satisfactory confirmation of the propulsive advantage that may be obtained by adopting a triple screw arrangement.

Related Abstracts

Advanced Steam Turbine Power Plant Designs

A symposium on marine steam power plants was held in London during late September by the General Electric Company (U.S.) and Babcock and Wilcox. The article consists of abstracts from some of the papers presented.

Mr. Prohl, Turbine Engineering Manager of the Marine Turbine and Gear Department, General Electric Co. (U.S.), stated in the opening paper, entitled "Marine Steam Plants", that due to the rapidly expanding requirements of the marine industry, it became apparent that the previous maximum of 45 000 bhp from a steam turbine would not be sufficient and a new range, the MST-16, was designed. Two basic sizes of HP turbines cover the power range, the smaller unit from 45 000 bhp to 70 000 bhp and the larger unit from 70 000 to 120 000 bhp. As reheat or non-reheat versions can be specified, four basic HP turbines are available as shown in Fig. 1.

The topmost combination in Fig. 1 represents the design as supplied to the 21 000 dwt, 33 knot Sealand container vessels at present building in Germany. Steam at the initial condition of 850 lb/in^2 and 950°F is admitted through sequentially-opening, hydraulically-operated valves located in the upper half of the casing at the forward end, and flows straight to the cross-over. Steam from the HP turbine enters symmetrically from the top inlet and flows in the forward

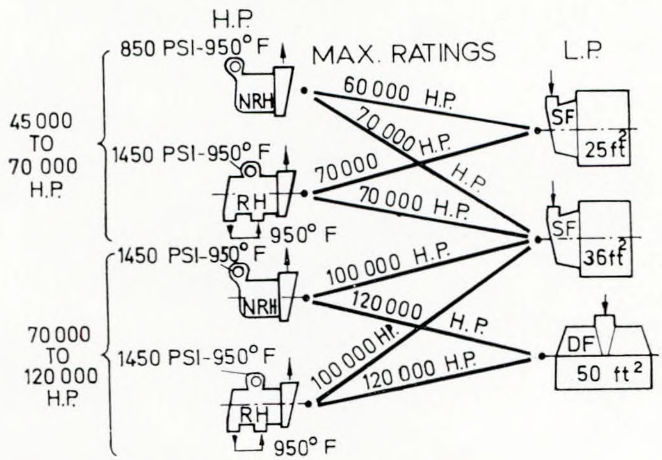


FIG. 1—*The MST-16 range*

direction to the downward exhaust and condenser. At this higher power level, the use of a top exhaust and cross-over for the entrance to the LP turbine in place of the conventional cross-under, provides the optimum arrangement. Included in the exhaust end of the LP turbine is the integral, standard two stage astern turbine. This is arranged with a two-row first stage and a single-row second stage. Since the astern turbine spins in a space where the full condenser vacuum prevails, the rotational losses are extremely low, approximately 0·33 per cent of the full ahead power. An axial flow exhaust arrangement is available for this particular LP turbine frame size for applications where space requirements and considerations dictate the use of this type of exhaust.

Estimates indicate that 50 000 bhp per shaft is the next step for propulsion requirements and several high-powered plants are already in service. Fig. 2 shows a triple reduction,

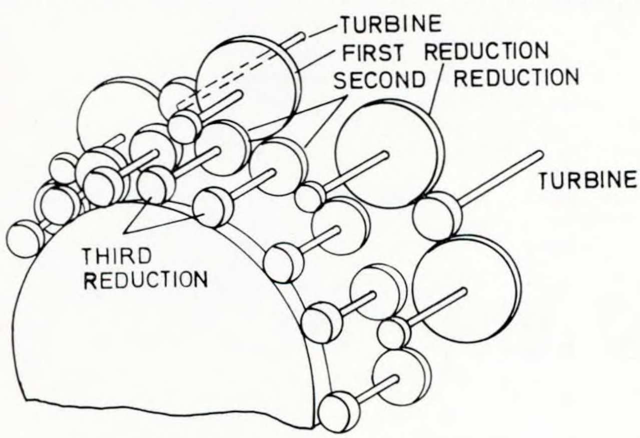


FIG. 2.—*100 000 bhp gearing arrangement*

parallel torque path gear driven by a cross-compounded turbine. This unit is rated at 100 000 bhp at 80 rev/min and the tooth loadings are within current marine practice. The eight pinions driving the main gear wheel are needed to control the tooth loadings due to the very high torque. This type of design would be suitable for large ice-breaking vessels where the requirements dictate that the capability to transmit the rated power at very low propeller speeds means that the machinery must continuously transmit very high torque levels.

A number of recent serious enquiries have considered power outputs of up to 200 000 bhp for very high speed container ships and ice breaking ships of various types. Design work has proceeded to the level of 120 000 bhp on a single shaft with 200 000 bhp being considered for two or three shafts. A 120 000 bhp single shaft steam plant would have two boilers generating steam at 1450 lb/in^2 and 950°F. At these high power levels, motor-driven forced draught fans are impracticable for economic reasons, so turbine-driven units are used. The steam pressure of 1450 lb/in^2 was chosen as the turbine sizes are smaller than a 850 lb/in^2 plant, permitting better thermal matching and good manoeuvrability.—*Shipping World and Shipbuilder, December 1970, Vol. 163, pp. 1673–1674; 1677.*

Experience with Controllable Pitch Propellers

Results from six cargo liners of the author's company, the first one built in 1964, have been evaluated in order to clarify the pros and cons for the c.p. propeller. It seems to be difficult to prove that the use of c.p. propellers is economically justified as such advantages as improved manoeuvring capability and reliability cannot be estimated in exact amounts. The effect on the service speed of the ships by using c.p. propellers instead of fixed propellers seems to be small and the cylinder liner wear is showing no marked difference. The time and cost in connexion with general overhaul and classification is higher for the c.p. propeller. Taken as a whole it is the author's opinion that the c.p. propeller is generally to be preferred in this type of ship.—*Bille, T., Transactions of the Institute of Marine Engineers, August 1970, Vol. 82, pp. 289–302.*

Optimum Propellers with a Duct of Finite Length

Numerical results are given for the quality coefficient of optimum ducted propellers. The influence of the number of blades, the advance ratio, the clearance between blade tips and shroud, and the hub diameter on this coefficient is shown.—*Slijper, C. A. and Sparenberg, J. A., Journal of Ship Research, December 1970, Vol. 14, pp. 296–299.*

Japanese Multi-Diesel 288 000 HP Proposal

The Japanese Ministry of Transport is reported to be supporting a design project for a remarkable vessel which would be able to carry some 3000 20 ft containers at a speed of 35 knots. The triple-screw 288 000 hp machinery proposed would consist of eight 24-cylinder medium-speed engines, each of 36 000 bhp, four of them coupled to the centre shaft —at 144 000 shp and two to each of the wing shafts. This infers engines of 1500 bhp per cylinder, an output 50 per cent higher than anything demonstrated and offered today by any established builder.

Mitsui has ben chosen to develop this engine which is to be of four-cycle type.—*Marine Engineer and Naval Architect, November 1970, Vol. 93, pp. 493–494.*

Marine Propulsion Turbines

The rapidly increasing size and consequent power demand of tankers and high speed containerships have resulted in significant changes in propulsion machinery. The propulsion power requirement for these large ships has increased from around 30 000 shp to the 60 000 shp range on one shaft.

Mitsubishi are now manufacturing four units of 80 000 shp twin-screw marine steam plants for high speed container-ships ordered by Japanese owners.

The special features of the MS and MR plants, developed on the basis of service experience obtained from the MTP and MWL accompanied by the extensive investigation made to improve the performance and economy are as follows:

MR plant is designed for steam conditions of 89·5 kg/cm^{2}g, 515°C/515°C (1270 lb/in^2, 960°F/960°F) or 103·5 kg/cm^{2}g, 543°C/543°C (1470 lb/in^2, 1010°F/1010°F) at superheater and reheater outlet. The MS plant uses standard steam conditions of 61·5 kg/cm^{2}g, 515°C (875 lb/in^2, 960°F).

A particularly interesting recent installation is that of the MS plant for the 128 000 dwt ore/oil carrier, built at Mitsubishi's Yokohama Shipyard for San Juan Carrier Co. This ship, *San Juan Venturer,* is the first steam turbine-powered vessel of this size to have a controllable-pitch propeller driven by a non-reversible turbine. The c.p. propeller is used continuously at the maximum rating and bridge control is provided. A single boiler with welded wall construction is installed and the main alternator and feed pump are directly driven by the high pressure turbine.

The principal particulars of this plant are:

Main turbine	
Type	1—Mitsubishi marine steam turbine, Type MS32
Maximum continuous rating	23 500 shp at 85 rev/min
HP turbine speed	6615 rev/min
LP turbine speed	4046 rev/min
Main boiler	
Type	1—Mitsubishi CE V2M-8W Boiler
Evaporation	110 t/h
Steam condition	61·2 kg/cm^{2}g, 515·6°C
Auxiliary boiler	
Type	1—Mitsubishi CE V2M-8 Boiler
Evaporation	35 t/h
Steam condition	22 kg/cm^{2}g, 218°C
Propeller	
Type	1—Mitsubishi KaMeWa four-bladed c.p. propeller. Type 220S/r
Diameter	8·2 m
Main alternator	
Type	1—direct driven 1—independent steam turbine driven
Rating	1250 kW at 1800 rev/min
Main feed pump	
Type	1—direct driven 1—independent steam turbine driven
Capacity	145 m^3/h × 87 kg/cm^{2}g
Back-up turbine	
Type	1—single stage steam turbine
Rating	2440 bhp at 6615 rev/min with steam conditions of 56 kg/cm^{2}g, 320°C

—Matsuoka, H., Marine Engineer and Naval Architect, November 1970, Vol. 93, pp. 559–565.

Steam Turbine Tanker of 255 374 dwt

The first in a series of 17 steam turbine powered tankers building at Kockums Mekaniska Verkstads AB, Sweden, has now been delivered to her owners, Cie Francais des Pétroles. This vessel, *Jade,* 255 374 dwt, is the largest ship built in Sweden so far, and is also the largest in the French merchant marine.

The longitudinally framed hull has been the subject of weight and cost saving exercises in several areas. The deep longitudinal bottom and deck stringers for example have been eliminated, with the exception of a so-called docking stringer on the centreline. These omissions have been compensated for by strengthened transverse frames and also by the adoption of high strength, yield point 36 kp/mm^2, steel in the centre tank bottom transverses. Previously such steel was used principally for the longitudinal strength members of the deck and bottom. For additional strength, and to prevent hull vibration, all transverse tank bulkheads are stiffened by horizontal stringers and secondary webs.

This particular hull design has been examined by Kockums on a computer using several up-to-date strength programmes and it has also been checked and approved by four major classification societies, namely Bureau Veritas, Lloyd's Register of Shipping, Det norske Veritas and the American Bureau of Shipping. Experts from Texaco and Chevron, two American oil companies who have ships of this type on order, have satisfied themselves regarding the soundness of the design.

To ensure that the findings of these theoretical investigations were confirmed in practice, during her sea trials *Jade* was fitted with 150 strain gauges which were combined into 360 circuits to measure static strain conditions on the ship.

Principal particulars are:

Length, o.a.	340·51 m
Length, b.p.	329·18 m
Breadth, moulded	51·82 m
Depth, moulded	25·60 m
Draught, summer	20·06 m
Deadweight	255 374 tons
Gross tonnage	126 370·21
Net tonnage	110 098·26
Cargo capacity	338 750 m^3
Lightship	33 676 tons
Speed on trial, fully loaded ...	16·24 knots
Service speed	15·70 knots

Jade's propelling machinery comprises a cross-compound, triple-reduction geared type AP32 Stal Laval steam turbine set which develops 32 000 shp (metric) at 85 rev/min.

The five-bladed propeller, in nickel aluminium bronze, has a diameter of 8600 mm, pitch of 6510 mm and blade area of 36 m^3. Suppled by Kobe Steel, it weighs 48·4 tons.

The main machinery can be controlled from the bridge through a Kockums/ASEA automation system. In the control room, which is situated on the forward part of the boiler platform, is a large console which carries most of the control and monitoring instruments.

Steam is supplied by two top-fired Kockum/Combustion Engineering type V2 M8 boilers, each normally generating 46·6 tons of superheated steam per hour at a pressure of 60·8 kp/cm^2 and temperature of 510°C. Maximum output is 71 ton/h. The boilers are provided with Ljungström type rotary air preheaters of Svenska Maskinverken's manufacture.—*Shipping World and Shipbuilder, April 1971, Vol. 164, pp. 445–448.*

World's Largest Tankers

The construction of what will for a while be the world's largest ship, the 372 400 dwt *Nisseki Maru* for the Tokyo Tanker Co. Ltd., a member of the Nisseki Group, commenced recently at the 400 000 dwt building dock of the Kure Shipyard of IHI, Japan. She will be powered by a 40 000 shp IHI-turbine and have a service speed of 14·5 knots.

Completion is scheduled for November, 1971.

The world's largest ships now in service are six 326 000 dwt tankers of the Universe Ireland class, which were delivered to National Bulk Carriers Inc. by IHI's Yokohama

Comparison of mammoth tankers:

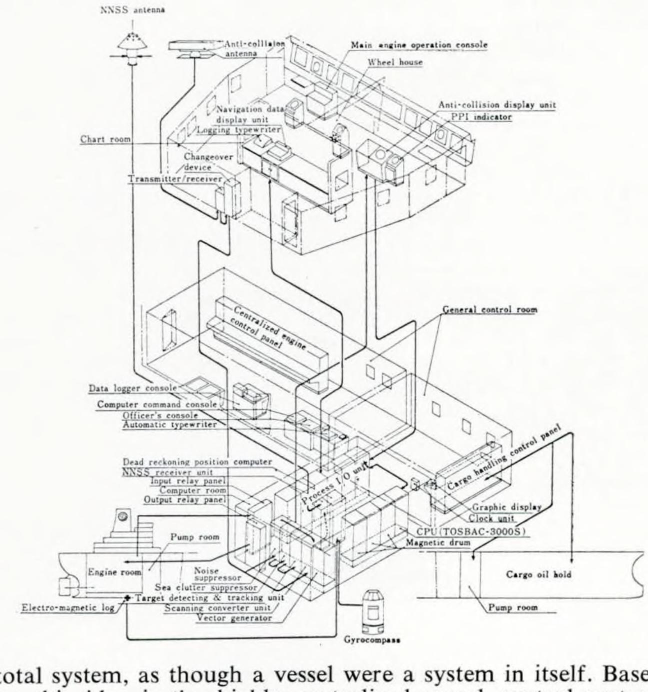
	Nisseki Maru	*Universe Ireland*	*Idemitsu Maru*	*Tokyo Maru*	*477 000 tonner*
	metres	metres	metres	metres	metres
Length, o.a. ...	347·0	346·0	342·0	306·5	379·0
Breadth, moulded ...	54·5	53·3	49·8	47·5	62·0
Depth, moulded ...	35·0	32·0	23·2	24·0	36·0
Draught ...	27·0	24·78	17·65	16·03	28·0
Gross tonnage ...	186 500	149 608	107 957	94 630	235 000
Deadweight tons ...	372 400	326 585	206 005	153 685	477 000
Power (shp) ...	40 000	2 × 18 700	33 000	30 000	45 000
Service speed, knots ...	14·5	14·6	16·5	16·0	15·0
Cargo capacity ...	470 000 m^3	399 600 m^3	245 058 m^3	192 000 m^3	581 000 m^3
Complement ...	—	51	32	29	35
Completion ...	Nov. 1971	Sep. 1968	Dec. 1966	Jan. 1966	Feb. 1973
Builder, IHI ...	Kure	Yokohama	Yokohama	Yokohama	Kure

Shipyard and Mitsubishi's Nagasaki shipyard between September 1968 and July 1969.

IHI also has two 477 000 dwt tankers, considerably larger than any yet built on order for Globtik Tankers Ltd., London. These vessels will each be powered by a 45 000-shp IHI turbine and have a service speed of 15 knots. Completion is scheduled for February 1973, and early 1974. —*The Motor Ship, March 1971, Vol. 51, p. 550.*

Computer Controlled System on the 138 000 dwt Oil Tanker "Seiko Maru"

The purpose of equipping vessels with a computer, though it differs with types of vessel, their routes, etc., is commonly threefold, 1) to overcome the shortage of labour, 2) to improve the safety of navigation, and 3) to contribute to economy in operation. To achieve this purpose it is necessary not only to automate highly each function of a vessel, but also to organize individual automation systems into a



total system, as though a vessel were a system in itself. Based on this idea is the highly centralized vessel control system, termed Super-Automation System.

The 138 000 dwt tanker *Seiko Maru* built at the Aioi Shipyard of IHI for the Sanko Steamship Co., is the first vessel to be equipped with this system. It was installed on an experimental basis to gather operational data. Its importance with respect to the scope of computer control deserves worldwide attention.

The structure of the whole system can be divided functionally into four systems for, respectively : i) navigation; ii) hull; iii) engine; iv) computer.

For this vessel, the centralized computer system was adopted to realize the aim of highly centralized control. The system is provided with a so-called on-line processing function which enables one computer to process data of differing contents and nature almost simultaneously in the order from higher priority to lower priority.

The arrangement on board of this system is shown in the diagram. With this layout the operation of various calculations and controls by the computer can be performed in the wheelhouse as regards navigation, and in the general control room (GCR) located at the forward end of the boat deck as regards the machinery and hull. The control room in the engine room, as seen in conventional vessels, has been abolished. Both main engine and cargo handling are operated from the GCR.—*Sakano, N., Holland Shipbuilding, January 1971, Vol. 19, pp. 48–51.*

212 350 dwt Tanker

The 212 350 dwt tanker *Eugenie S. Niarchos,* built by Kockums Mekaniska Verkstad, Malmo, Sweden, was delivered recently to the Bethel Shipping Company, a subsidiary of the Niarchos group. It was built under special survey of the American Bureau of Shipping, and safety arrangements comply with the SOLAS 60 recommendations.

Eugenie S. Niarchos has an overall length of 1037 ft, a beam of 160 ft, a depth of 80 ft $4\frac{1}{2}$ in and a draught of 62 ft $4\frac{3}{4}$ in. She has a cargo capacity of 9 161 775 ft^3 and a ballast capacity of 34 092 tons. The hull is longitudinally framed and is provided with a cylindrical bow raked at the upper end. High-tensile steel was used in the longitudinal strength members in the deck and bottom.

The main propulsion machinery consists of a triple-reduction geared Kockum-Stal-Laval advanced propulsion type turbine rated at 32 000 shp at 85 rev/min. Steam is supplied by two Kockum-Combustion Engineering type boilers, each having a maximum capacity of 66 tons of steam per hour at 865 lb/in^2g and 950°F. The boilers are regulated by a Kockum Combustion Control Mk 3 T. The turbine drives a 28 ft propeller. This machinery provides a service speed of 16 knots.

The engine room alarm system is based on continuous one-man operation from an open control station.—*Maritime Reporter/Engineering News, 1 October 1970, Vol. 32, p. 43.*