STEAM TURBINE MACHINERY

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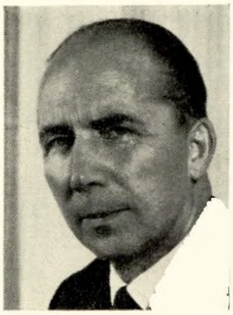

Despite the apparent decline, some years ago, of the steam turbine as marine main propulsion machinery, modern developments in ship size and design have provided marine turbine manufacturers with an incentive to develop their products in competition with the internal combustion engine, especially in the region of the high powers now demanded for large tankers and container ships.

The author states that these new ship designs must be powered by optimized propulsion systems and goes on to discuss propulsion problems peculiar to large tankers and those peculiar to container ships. The design of the steam propulsion system is then surveyed, the turbine, the gearing and the boiler. Finally, a section is devoted to discussing optimum fuel rates and design optimization.

INTRODUCTION

A few years ago the steam turbine seemed to disappear from the statistics for ships on order while the Diesel could show statistics which, when extrapolated, indicated that outside the U.S. the number of turbine ships launched would drop to zero in the period around 1970[1].

At an earlier stage, in the beginning of the sixties—when this trend of declining turbine orders became apparent—the turbine makers had only the choice of dying slowly or making an attempt to rejuvenate marine steam propulsion units design in order to reduce cost and improve efficiency while preserving the availability of orthodox simple units. In this situation the author's company resolved to invest in development work.

At a meeting before S.N.A.M.E. in New York, September 1963, the author told the boiler manufacturers present that if all elements including the boilers were not improved, marine steam engineers would all be out of business soon. Today the remaining few leading marine turbine manufacturers who survived the critical period cannot complain about orders received during the last two years. S.S. and T.S. are again prefixes for many great ships in the yards' order lists. The subject of this paper is to recapitulate developments over recent years seen from the author's point of view.

As designs of ships' propulsion are changing more rapidly than ever before, the propulsion units have to be adapted to follow the trends of the world's shipping demands. Thus marine turbine engineers still have the incentive to further develop turbine machinery—steam or gas—to withstand the competition, not only from the direct drive, slow speed Diesel engine, but also from the fast improving and fast growing geared medium speed Diesel.

NEW TYPES OF SHIPS MUST ADOPT OPTIMIZED PROPULSION SYSTEMS

During the last ten years there have been great changes in the design and size of ships (Fig. 1). The range of the transatlantic passenger ships has decreased, the numbers have dwindled, the powers have decreased compared with the time before the break-through of jet air transport. Some modern ships, such as *Queen Elizabeth* 2, the Swedish *Kungsholm* and some Italian and French liners have been built or are under construction. Most of these ships are turbine driven with powers between 30 000 to 55 000 shp on each of the twin shafts. The *United States* which is equipped with four Westinghouse turbines of 60 000 shp each is still the most powerful merchant marine vessel in the world. The situation for the fast passenger superships will be still more unfavourable in the near future when the supersonic airliners and the airbuses take over from the 100–200 seat passenger jets of today. Powers of the future cruising and transocean passenger service will probably stay in the region 30 000–60 000 shp where the turbine is definitely competitive, even if propeller revolutions are sufficiently high for competition from the slow speed Diesel. The fuel consumption will still be somewhat higher for the turbine drive but this is compensated by more acceptable noise and vibration levels and lower maintenance cost. Even so, this area of declining passenger transport at sea will not even keep one turbine manufacturer alive.

The areas of special interest for turbine engineers are mainly

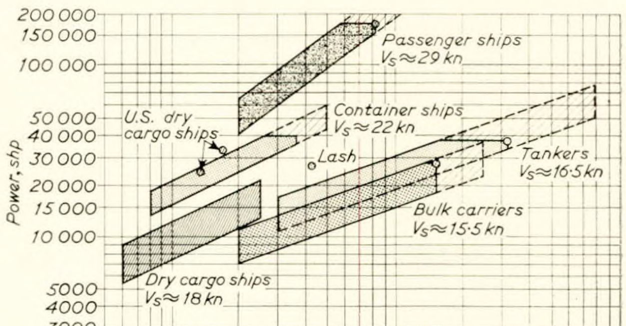

Fig. 1—Power requirements for different ship types

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those of the tankers and the container ships shown in Fig. 1, where power demands during the last years have increased up to or above the limits of the proven Diesel units and single screw propeller design. These rather high powers cause many complications with regard to the hydraulic propulsion system. Thus for these ships outside the well known and traditional regions it is essential to study and optimize the complete propulsion system of hydraulic propulsors, shafting as well as engines and auxiliary drives with respect to first cost, annual running cost and availability. For comparison of alternative propulsion systems it is essential to refer to an adequate base of reference. A low fuel cost is therefore of no significance unless related to the effective horsepower.

Even today's improved steam turbine machinery is hardly competitive below 18 000–20 000 shp and the normal dry cargo ship has not yet grown into this power range, whereas the bulk carriers are moving into power regions where the steam turbine plant becomes a favourable alternative. At present some fifteen bulk carriers on order are to be fitted with steam turbines.

The special dry cargo ships developed in the U.S.A. with high speeds and powers outside the normal range for this type of ship are mainly turbine driven. These ships and their machinery are, however, built to completely different economic optimization rules than are valid outside the U.S.

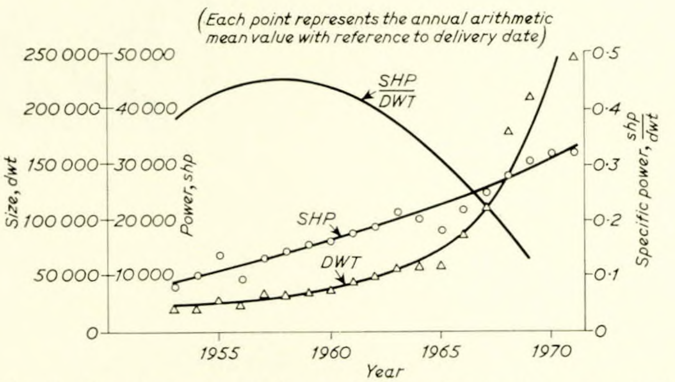


Fig. 2—Development of size and power for tankers propelled with Stal-Laval turbines

Fig. 1 also shows the trends for future growth especially where the next generation of container ships and tankers are of interest. Tanker design has changed rapidly (Fig. 2). Size has grown so fast that the load carrying capacity has doubled in 4–5 years and the trend is accelerating. Speeds, however, have stayed more or less constant. This is contrary to the laws of similar ships. A large ship should definitely have its economic speed higher than a small ship. Reasons for the more or less constant speed seem to be twofold. Smaller tankers built during the boom years were often overpowered with respect to minimum transport cost and the great supertankers of today are probably somewhat underpowered due to propeller and cavitation problems. The specific power has decreased further than the simple law of similarity:

$$\text{shp/dwt} \sim \text{dwt}^{-1/3}$$

because of better lines, bulbous bows and increased Reynolds numbers.

The revival of the turbine tanker in 1966 is very obvious from the statistics presented in Fig. 3. Since then, the numbers of turbine tankers as well as the total power of turbine machinery have increased rapidly, with the result that, in 1967, more turbine tanker shp was under construction than Diesel shp. The world order book for different drives for tankers, container ships and bulk carriers is presented in Fig. 4. In addition it should be mentioned that the majority of dry cargo ships and bulk carriers are propelled by Diesel engines. How very different the situation is in U.S. shipbuilding, particularly with regard to drive of general cargo ships, is seen from Fig. 5.

The largest and the fastest container ships and tankers have power demands approaching or exceeding the limits of what can be taken out of a normal single propeller with regard to dimensions, load, efficiency or risk of cavitation damage.

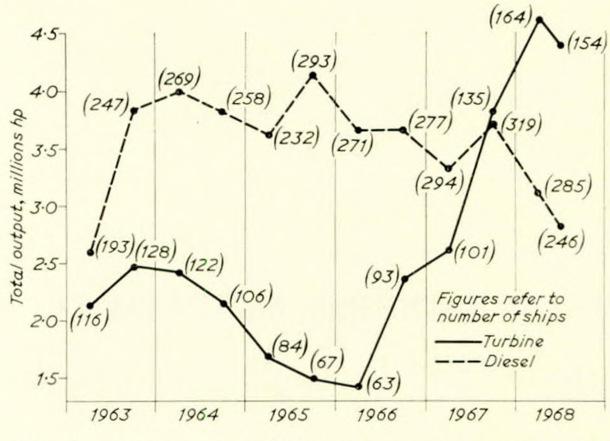


Fig. 3—Tankers above 2000 dwt on order 1963–1968

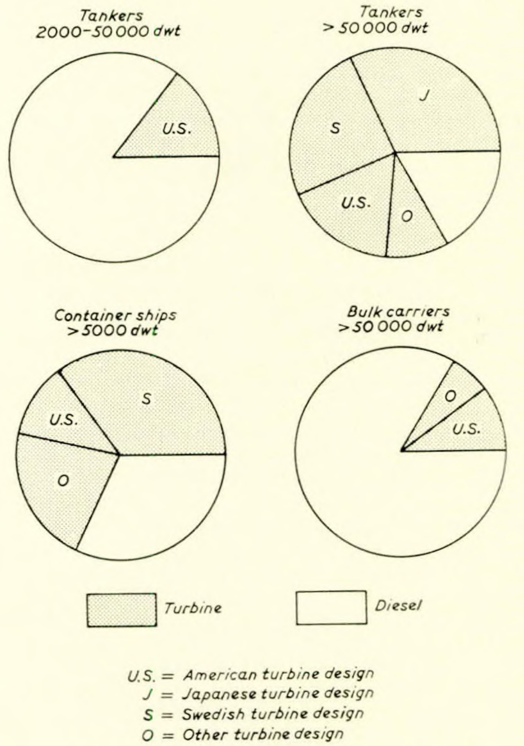


Fig. 4—World order book, 1.8.68

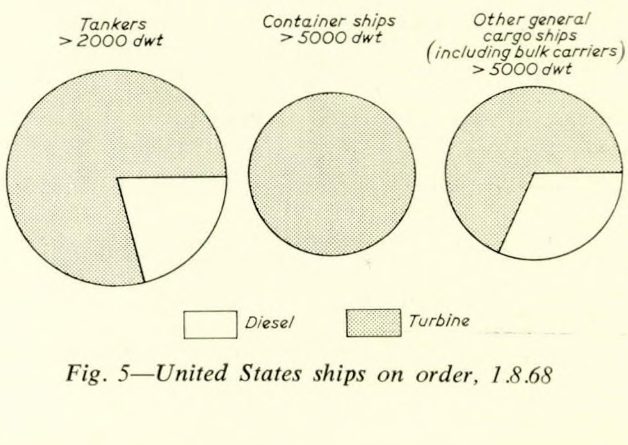


Fig. 5—United States ships on order, 1.8.68

Several alternative arrangements make it possible to exceed the power range of the single propeller or to improve the propulsive efficiency (Fig. 6). The twin-screw arrangement is traditional at high powers and may also give a higher propulsive efficiency for large tankers than the single propeller drive. Contra-rotating propellers have been much discussed since the Manhattan tests

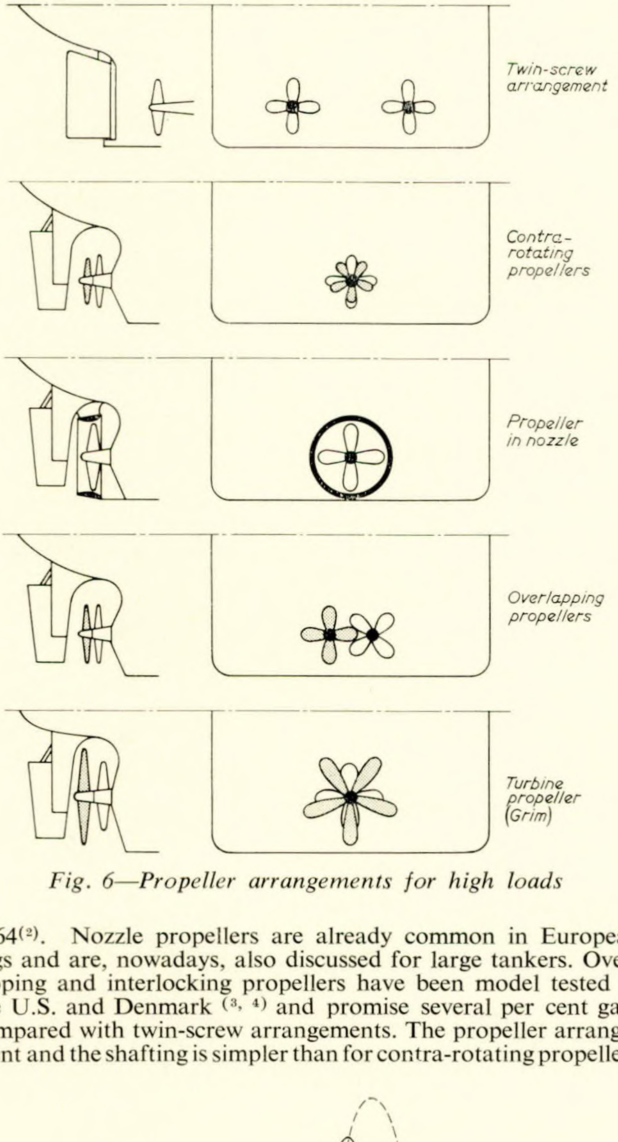


Fig. 6—Propeller arrangements for high loads

1964[2]. Nozzle propellers are already common in European tugs and are, nowadays, also discussed for large tankers. Overlapping and interlocking propellers have been model tested in the U.S. and Denmark [3, 4] and promise several per cent gain compared with twin-screw arrangements. The propeller arrangement and the shafting is simpler than for contra-rotating propellers while the possible gain in propulsive efficiency is of the same order. By the application of interlocking propellers the shafts cannot turn independently and should be geared for outward turning. Also by overlapping propellers there is need for some synchronizing device to secure that the aft propeller passes through undisturbed eddy-free water (Fig. 7). According to propeller designers this is important to avoid cavitation damage. With regard to reliability and emergency drive with only one shaft, the overlapping arrangement is preferable. Finally, the turbine propeller (Grim) should also be mentioned[5]. At high propeller load factors this propeller arrangement should give about the same gain as the nozzle propeller. Unfortunately full scale tests are not available—neither for this propeller arrangement or for the overlapping nor contra-rotating propellers.

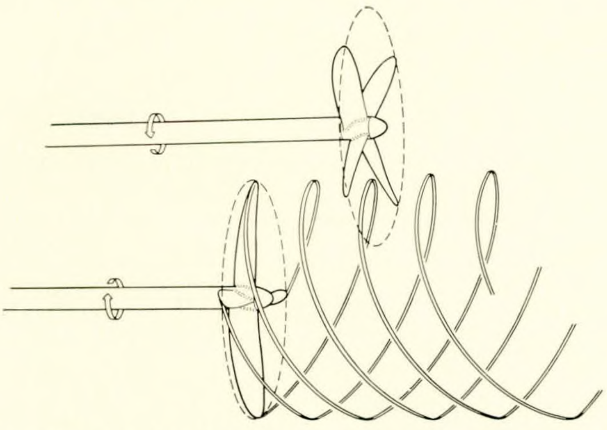


Fig. 7—Blade tip eddies from the fore propeller passing between the blades of the aft propeller in overlapping propeller arrangement

PROPULSION PROBLEMS FOR LARGE TANKERS

By enlarging the hull by a proportional increase of all linear dimensions, the simplified law of similarity shows that the propelling power demand is proportional to the square of the linear scale factor, α, at constant ship speed. The specific power demand, shp/dwt is then inversely proportional to α. Thus, the annual cost of transport work will decrease with ship size and as first cost increases less than proportional to the displacement, total cost of cargo transport will be more favourable the larger the ship. There are, however, hydrodynamic laws for propulsion which have negative effects on the efficiency as the ship size is increasing.

If at constant ship speed the propeller diameter increases linearly with α, the shaft speed should be reduced. Assuming then that the propeller speed is inversely proportional to α, i.e. keeping the tip speed constant, a series of similar ships may be obtained with characteristics according to Table I. The propeller load factor:

$$Bp = \frac{n\sqrt{\text{shp}}}{[V(1-w)]^{2\cdot5}} \sim \sqrt{\frac{\triangle V}{V_A}}$$

where

n = propeller speed (rev/min)
w = wake fraction
V = ship speed (knots)
V_A = $V(1-w)$ (knots)
$\triangle V$ = water speed increase in propeller stream tube (knots)

—a very defective characteristic number—would in such a series of ships be independent of size. In a dimensionless figure the Bp-value represents the square root of the water speed increase in the stream tube around the propeller tips divided by the speed of advance of the propeller. At constant load factor, Bp, the shaft speeds for the largest members of the family become unreasonably low with regard to dimensions of propellers, shaftings and engines. The influence of the load factor on the efficiency is presented by van Manen[6] in a very comprehensive diagram (Fig. 8). With increasing propeller load factor, i.e. increased impulse to the water at constant ship speed, the efficiency declines almost linearly.

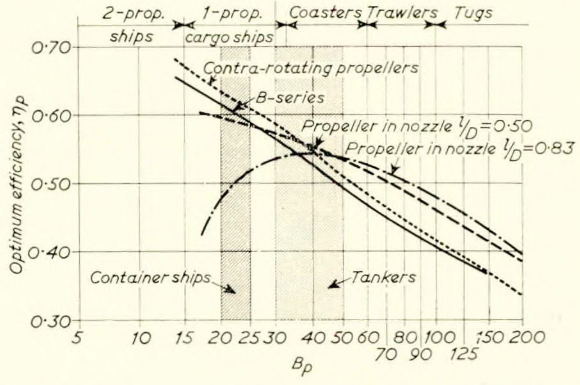


Fig. 8—Comparison of optimum efficiency values for different types of propulsors

At constant propeller revolutions, diameter and hence constant tip speed, the propeller load factor will increase proportionally to α. For the largest members this would imply very low efficiency values.

Neither of these solutions related are advisable. Propeller speeds inversely proportional to α as shown in Table I are unrealistic and constant propeller speeds lead to unacceptable values of the propulsive efficiency. The propeller speeds have therefore to be optimized between these two limits with regard to all economic factors.

Emerson [7, 8] long ago proposed that propeller speeds should decrease with $\alpha^{-0 \cdot 5}$. This relationship for similar ships, shown in Table II, should be of serious interest. The more favourable Reynolds numbers for the large vessels compensate for the loss due to the increased propeller load factors. At the same time the propeller speeds stay within reasonable values.

The van Manen diagram (Fig. 8) also indicates the gains to be achieved in propulsive efficiency by alternative propeller arrangements. Contra-rotating propellers are more favourable than the single screw B-series over a wide range of *B*p-values, whilst the nozzle propeller shows substantial gains at high propeller loads.

The Swedish Model Tank, S.S.P.A., has made tests with tanker models of 150 000 dwt [9, 10]. The results (Fig. 9) clearly show the importance of low propeller speed. A decrease in rev/min gives an improvement of approximately 2·6 per cent/10 rev/min in the actual region. The measured gain of contra-rotating propellers at constant rev/min is 6 per cent.

For overlapping and interlocking propellers very few model test results are published [3, 4]. According to the Manhattan tests the gain achieved compared with conventional twin-screw amounts to some per cent (Fig. 2)[2, 3].

Table III shows the possible improvements of propulsive efficiency for alternative propeller arrangements for supertankers as compared with conventional single-screw propulsion.

PROPULSION PROBLEMS FOR CONTAINER SHIPS

The container ship is a very new specimen of cargo ship and has not yet found its definitive form. Different trades, cargoes and types and sizes of container will influence the design of the future container ships. They will, however, have some features in common; they will be large and fast and require high propelling powers.

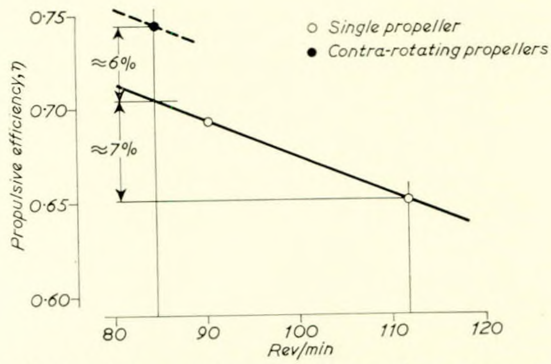


Fig. 9—Swedish model test results with contra-rotating propellers for 150 000 dwt tanker

The propulsion characteristics for some typical container-ships now under construction on leading yards around the world are shown in Table IV. None of these ships has more than 32 000 shp on one propeller shaft. For higher power requirements the twin-screw arrangement has been chosen. 32 000–35 000 shp is

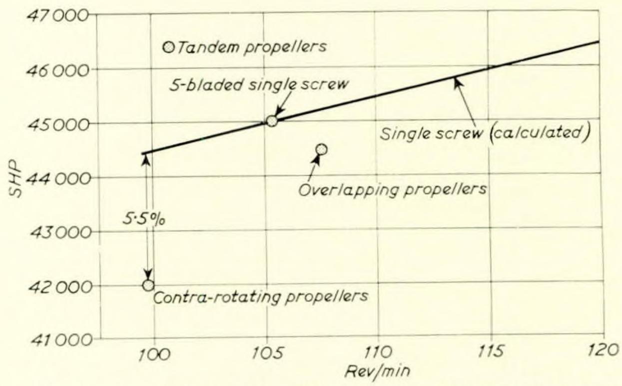


Fig. 10—Project Manhattan *19·25 knots 106 000 dwt tanker*

TABLE I—FAMILY OF SIMILAR TANKERS WITH CONSTANT SHIP SPEED, LINEAR PROPELLER DIAMETER, PROPELLER SPEED PROPORTIONAL TO α^{-1}, CONSTANT TIP SPEED AND CONSTANT PROPELLER LOAD FACTOR (Bp $\approx$ 20)

Linear factor, α	1·0	1·5	2·0	2·5	3·0
Displacement, tons	25 000	84 000	200 000	390 000	675 000
Shaft power, shp	8 000	18 000	32 000	50 000	72 000
Specific power, shp/tons	0·32	0·21	0·16	0·13	0·11
Propeller speed, rev/min	110	73	55	44	37
Propeller diameter, ft	18	27	36	45	54
Draught (linear), ft	29	44	58	73	87
Draught (actual or extrapolated), ft	29	42	57	71	84

TABLE II—FAMILY OF SIMILAR TANKERS WITH CONSTANT SHIP SPEED, LINEAR PROPELLER DIAMETER, PROPELLER SPEED PROPORTIONAL TO $\alpha^{-0 \cdot 5}$ AND TIP SPEED PROPORTIONAL TO $\alpha^{0 \cdot 5}$

Linear factor, α	1·0	1·5	2·0	2·5	3·0
Displacement, tons	25 000	84 000	200 000	390 000	675 000
Shaft power, shp	8 000	18 000	32 000	50 000	72 000
Specific power, shp/tons	0·32	0·21	0·16	0·13	0·11
Propeller speed, rev/min	110	90	78	70	64
Propeller diameter, ft	18	27	36	45	54
Draught (linear), ft	29	44	58	73	87
Draught (actual or extrapolated), ft	29	42	57	71	84
Propeller load factor, Bp	20	25	28	32	35

TABLE III—POSSIBLE IMPROVEMENTS OF PROPULSIVE EFFICIENCY FOR SUPERTANKERS

Comparison with single screw drive	Twin screw arrangement	Turbine propeller (Grim)	Overlapping propellers	Contra-rotating propellers	Propeller in nozzle
Propeller load factor	Reduced	Reduced	Reduced	Reduced	?
Rotational energy loss in race	Constant	Reduced	?	Reduced	Constant
Hull efficiency	Reduced	?	Increased	Constant	Constant
Propulsive efficiency at constant rev/min	Gain 0–3 per cent	Gain ≈ 5 per cent	Gain 1–7 per cent	Gain 5–7 per cent	Gain 4–7 per cent

today assumed to be the power limit for a single screw arrangement with regard to practical hull forms and propeller sizes. The risks of cavitation and hull vibrations when utilizing a high powered single screw have been considered so severe that the higher cost of the twin-screw arrangement is well justified.

The propeller load factors of container ships are favourable (Fig. 8) compared with those of the supertankers, whilst the propeller disc area loads and the cavitation numbers are extremely high. With the conventional twin-screw arrangement these limitations are eliminated but a loss of propulsive efficiency due to lower hull efficiency can hardly be avoided. The overlapping

TABLE IV—CONTAINER SHIPS ON ORDER 1968

Shipowner	Output shp	Ship speed knots	Propeller speed, rev/min	Propeller load factor
Overseas Container Ltd.	32 000	21·5	140	24
Associated Container Transportation Ltd.	30 000	22·0	137	22
Matson Line	32 000	23·0	110	16
Prudential Line (Lash)	32 000	23·0	105	15
Atlantic Container Line	2x17 500	24·0	120	8
Johnson Line	2x13 000	23·0	150	10
Project	2x50 000	27·0	180	15

Wake fraction: Single screw, W = 0·25
Twin screw, W = 0·13

TABLE V—POSSIBLE IMPROVEMENTS OF PROPULSIVE EFFICIENCY FOR HIGH POWERED CONTAINER SHIPS

Comparison with single screw drive	Twin screw arrangement	Overlapping propellers	Contra-rotating propellers
Propeller load factor	Reduced	Reduced	Reduced
Propeller disc load	Reduced	Reduced	Reduced
Rotational energy loss in race	Constant	?	Reduced
Hull efficiency	Reduced	Increased	Constant
Propulsive efficiency at limited propeller diameter	Loss 2–5 per cent	Gain 3–10 per cent	Gain 8–14 per cent
Power limit with regard to propeller cavitation (For single screw at present 32 000–35 000 shp)	> 60 000	> 60 000	> 60 000

and contra-rotating propeller arrangements offer, therefore, significant advantages for this type of ship. Apart from less risk of cavitation damage they both offer an increase of the propulsive efficiency due to higher propeller efficiency and better hull efficiency than the conventional twin-screw arrangement.

The Swedish Model Tank has carried out tests with container ships of 19 000 tons displacement, Fig. 11 [10]. It will appear from the diagram that, for propeller speeds down to about 140 rev/min, the efficiency of an optimized propeller will increase with decreasing rev/min. As the hull does not allow a further increase in propeller diameter a further reduction of rev/min means a less favourable screw with increasing pitch and blade area. Contra-rotating propellers may bring about 8–14 per cent gain at reduced propeller speed corresponding to maximum allowable propeller diameter.

Table V shows the possible improvements of propulsive efficiency for high powered container ships as compared with conventional single-screw propulsion.

STEAM TURBINE DESIGN

The steam turbine of today is, with a few unimportant exceptions, of the cross-compound impulse type with gashed solid rotors. The H.P. turbine is usually fast running with a service speed of about 6000 rev/min. The casing is very simple (Fig. 12). The rotor is slender with its first critical resonance speed at about $\frac{2}{3}$–$\frac{3}{4}$ of the maximum service speed. With good balancing and modern bearings with sufficient damping, the resonance speed can hardly be recognized without special instruments. The rotor is connected to the first reduction pinions by fine tooth couplings. Diaphragms are welded and radially supported and provided with spring-loaded split glands. Blade seals allow optimum reaction in the blades with small outside leakage losses.

The L.P. casing (Fig. 13), is built up of plate material and only some parts such as the astern casing are of cast material. The last stage has minimum reaction at the root and is designed for radial equilibrium and adapted essentially for constant mass flow distribution. The astern turbine should have axial exhaust to the condenser and the same direction of steam flow as the ahead L.P. turbine. With respect to thermal shock from steam

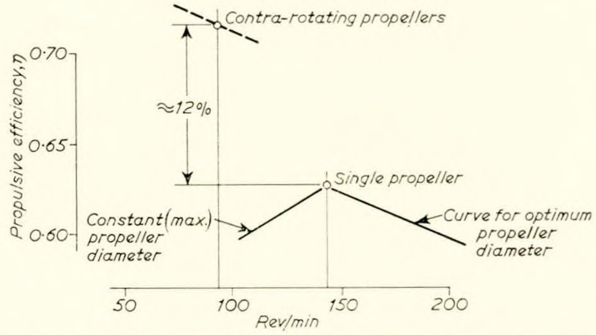


Fig. 11—Swedish model test results with contra-rotating propellers for container ship with 19 000 tons displacement

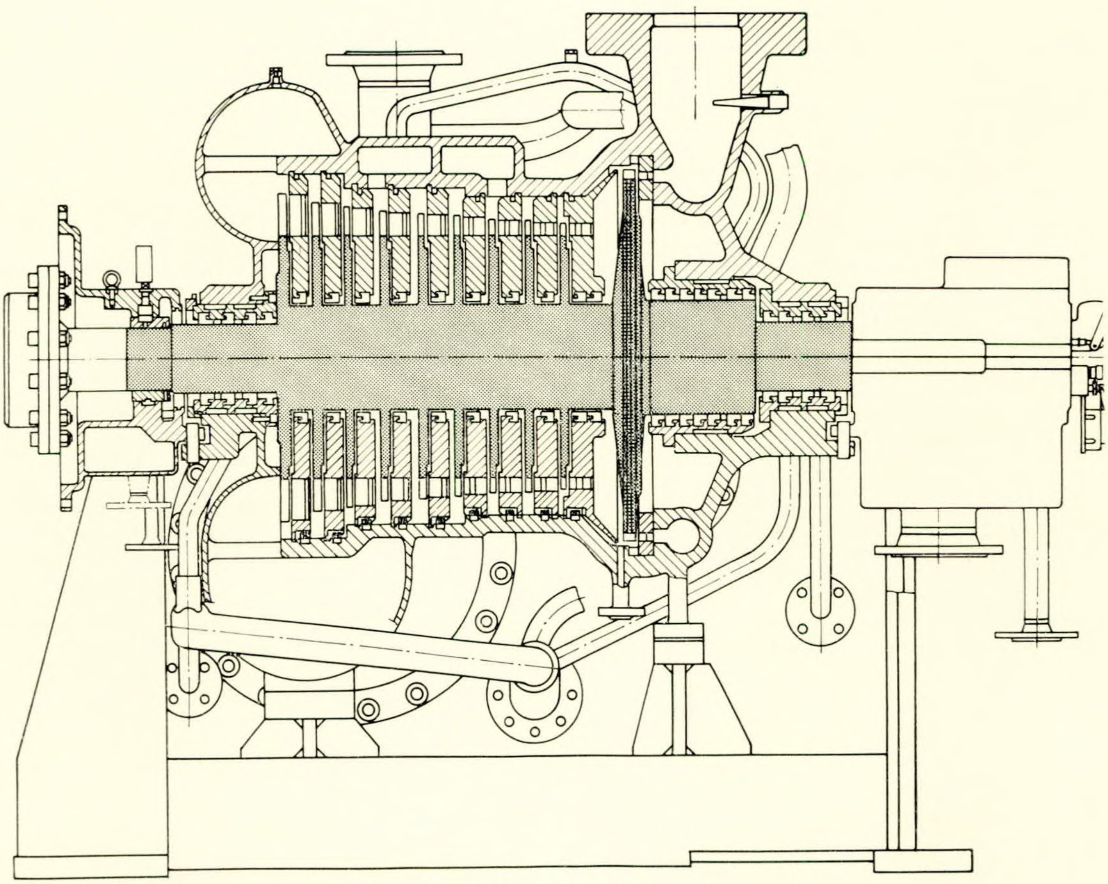


Fig. 12—H.P. turbine, APH 40, for 40 000 shp at 6000 rev/min

temperatures of more than 950°F (510°C) when manoeuvring, the astern casing and the astern wheels should be free to expand and, at the same time, be fixed radially.

Axial exhausts are used with single plane arrangements of turbines and gearing. This gives a lower but somewhat longer construction of the L.P. turbine-condenser unit than the underslung design with vertical exhaust. Only in cases where the engine length is of very great importance could there be reasons to leave the low cost standardized single-plane arrangement, since the exhaust loss is greater for the underslung design than for the axial exhaust design.

The blades have integral shrouds with rolled-in shroud wires in grooves combined with loose damping wires in the longer blades. L.P. blades without shroud or damping wires have been used in only a few cases by a German manufacturer[11]. Most turbine designs still have nozzle groups for load control even if most tanker turbines usually run with fully open governing valves.

In order to achieve the dimensioning criterion, 75 per cent ahead torque at 65 per cent ahead rev/min, the astern turbine needs only three or four blade rows. The double Curtis wheel design needs smaller diameters for the same torque and gives somewhat lower windage losses.

Single cylinder turbines (Fig. 14) have been topical for the last few years with regard to the simple design and lower cost [12, 13]. If one considers 6000 rev/min the maximum practical rotor speed with regard to the gearing, a single casing turbine is the natural solution at about 10 000 shp (Fig. 15), where 6000 rev/min is about ideal with regard to the tip speed and exhaust area of the last L.P. blade. By 20 000 shp the 6000 rev/min cross-compound H.P. turbine becomes smaller and more efficient than the H.P. section of the single rotor turbine. This difference becomes more accentuated at higher powers and lower speed of the single rotor. At 20 000 shp the cross-compound unit has already about 1·5 per cent better efficiency than the single casing turbine of equivalent design. The single casing turbine is less expensive only somewhere below 20 000 shp.

Reheat complicates the H.P. turbine design somewhat (Fig. 16) but the design will, in principle, be similar and the shaft speed about the same. More stages are needed because of larger heat drop. The longer rotor requires a more rigid shaft. The reheat gland has to be guided radially and the casing must be as symmetrical as possible in order to avoid distortion. Even at large steam flows corresponding to 100 000 shp a single H.P. + I.P. turbine will have rather small physical dimension. With reheat, the tip speed of the last L.P. turbine blade can be increased due to lower steam wetness and as the steam mass flow can be decreased, the L.P. turbine can be reduced in size.

A few years ago, marine L.P. turbines of the reaction type had double exhausts at powers higher than 20 000 shp. Today 32 000 and 40 000 shp single impulse type exhausts are used at sea and larger units of this type are under construction.

The last stage blades of these designs are somewhat conservative compared with stationary 3000 rev/min blades (Table VI). For stationary turbines operating at constant speed it is sufficient if the blades can be tuned to be vibration free inside a narrow region around 3000 rev/min. However, the marine blade must be strong enough and have sufficient damping to run continuously within the whole speed range even at the unavoidable resonance speeds. Free stationary blades up to 29 in length are in service today with the first resonance so low as to fall between the first and the second shaft speed multiple.

Until some years ago, marine blades were designed with all first resonance frequencies outside the fifth shaft speed multiple. The author recommends marine blades with continuous shrouds and/or damping wires in order to reduce the risk of dangerous first tangential blade resonance. By continuous shrouds and good damping this resonance can be allowed to come down even somewhat below the second shaft multiple. The axial nodal disc and blade resonance should, however, stay outside the fourth

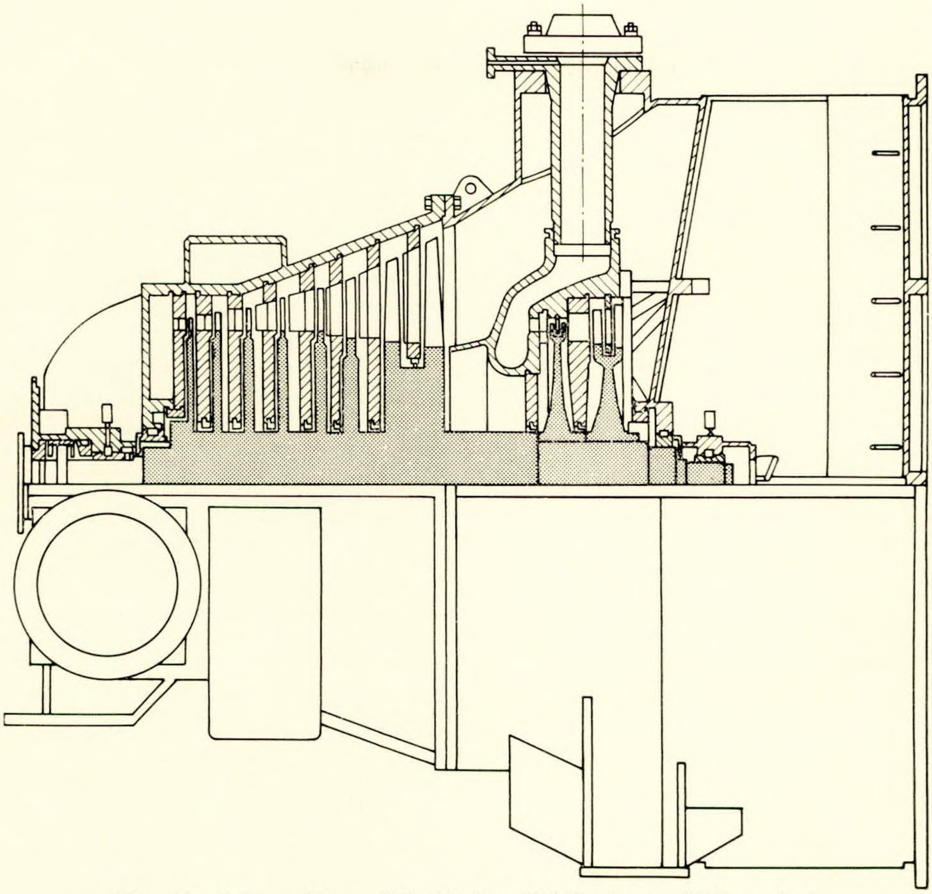


Fig. 13—L.P. turbine, APL 40, for 40 000 shp at 3100 rev/min

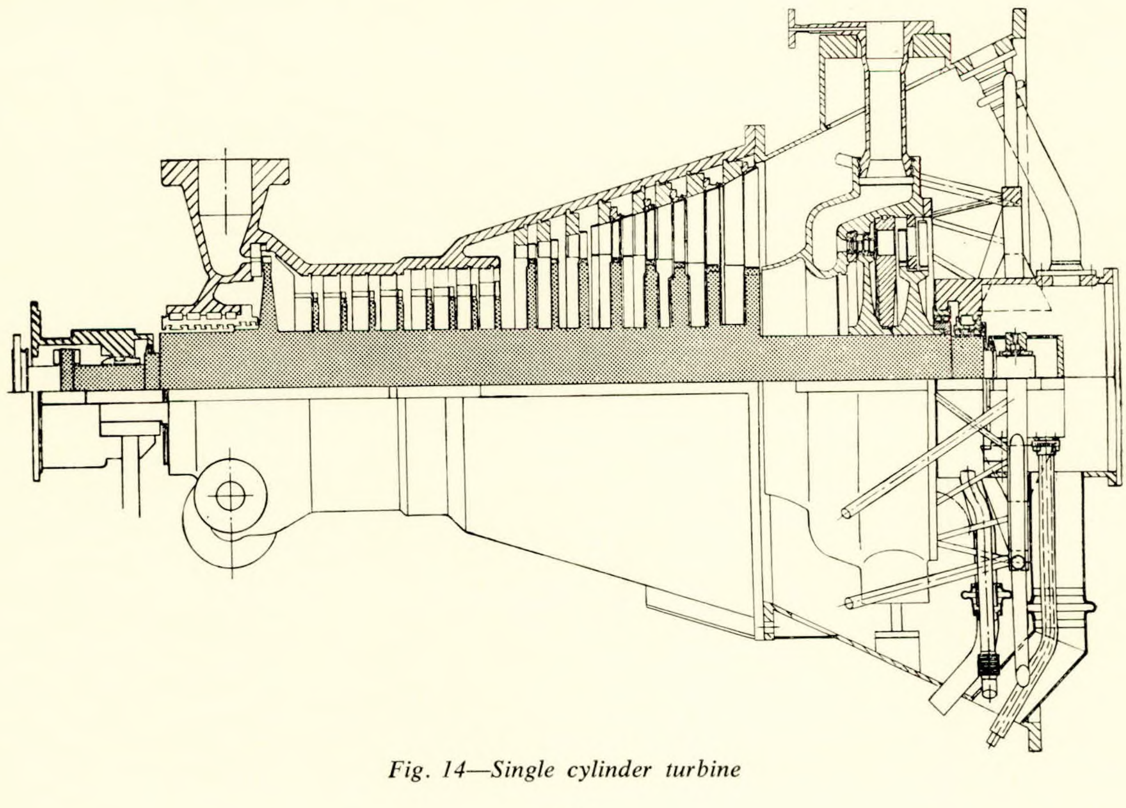


Fig. 14—Single cylinder turbine

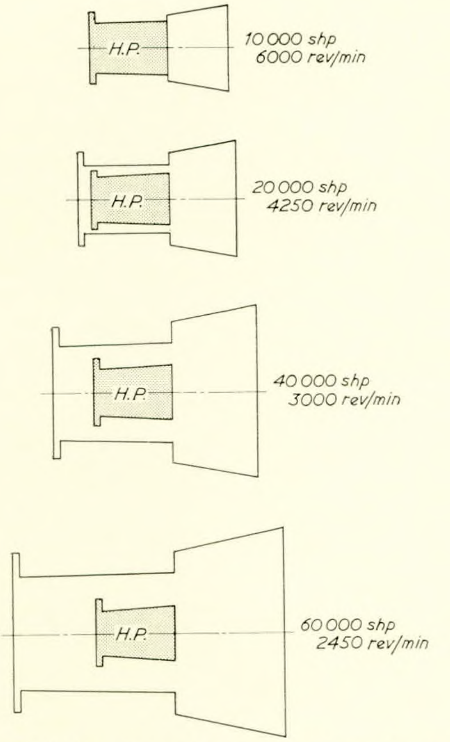


Fig. 15—Relative dimensions of single casing turbines and comparable cross-compound H.P. turbines

shaft speed multiple. As opposed to stationary turbines for generator drives where the shaft speed is fixed (3000 or 1500 rev/min), geared turbines for marine application can be designed for any rev/min, thus enabling tip speed and blade stresses to be kept at a desired level. The upper limit of exhaust area is therefore an economic question rather than a technical question. There is, therefore, no power limit for a single exhaust marine L.P. turbine if the speed is reduced sufficiently. With regard to the progress made during recent years in moisture extraction, blade erosion shields and blade vibration, less conservative tip speeds can be chosen in the future than the standard 1080 ft/s. This means (Table VI) that a shaft speed of 3000 rev/min and one exhaust, outputs of about 63 000 shp for non-reheat and 93 000 shp for reheat plants can be obtained.

GEARING DESIGN

The technical methods and facilities for gear making have now reached such a standard that there is no advantage in going further for normal marine gears (Table VII). Small deflexions in the ship are of more importance for line contact than more improved shaving in the climatized gear shop. Much higher K-factors than Lloyd's Register normal 95–125 limits are possible using standard materials if the laboratory accuracy from the gear shop should be maintained in the ship at all load and temperature conditions. With the AP design (Fig. 17) the author's company re-introduced the self-adjusting flexible pinion support invented by Dr de Laval and it has proved to be reliable. The design has simplified erection on board and ensures a good working tooth contact in spite of the large dimensions of the final stage, low speed, high power gears of today. The flexible gear casing reduces partial overloading and wear of the gear.

Increased hardness in pinions and rims will be introduced allowing higher K-factors by maintaining today's technique of cutting and shaving gear teeth. Surface hardened shaved or ground pinions will allow relatively smaller dimensions but tooth bending stresses will set a limit for larger pinion diameters before K-factors reach critical values. The hardened and ground gear will hardly bring additional advantages at large torques above those that can already be achieved by shaved gears with rims of medium hardness and surface hardened shaved or ground pinions.

The epicyclic gear introduced with the AP design has proved its reliability and its competitive ability. The triple reduction on the H.P. side for low rev/min was introduced three years ago with star-planetary gears before the final parallel stage and it is now dominant in tanker machinery above 25 000 shp, built in Europe (Fig. 18).

The single cylinder turbine may be combined with locked train parallel gearing or epicyclic gearing (Table VIII). The author has compared costs for these gear designs as well as gear

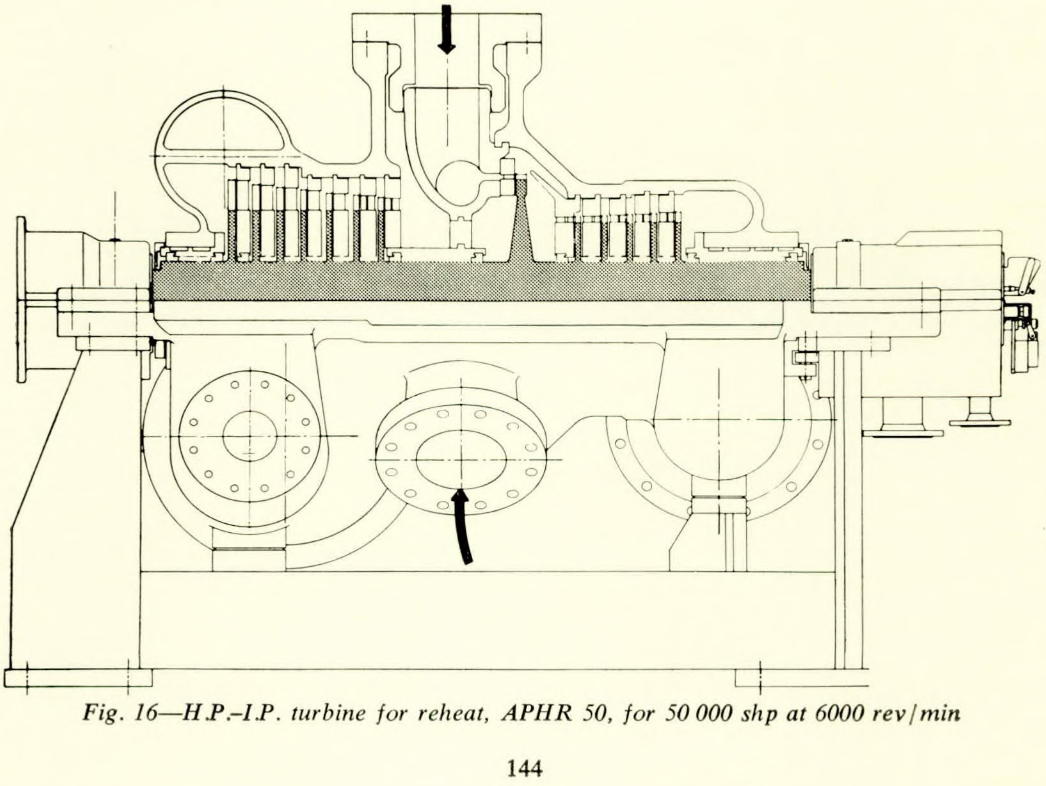


Fig. 16—H.P.-I.P. turbine for reheat, APHR 50, for 50 000 shp at 6000 rev/min

TABLE VI—MARINE AND STATIONARY EXHAUST BLADES

	Speed, rev/min	Blade length, in	Tip diameter, in	Tip speed, ft/s	Moisture per cent	Maximum power of one exhaust, shp
APL 32	3600	16	70	1080	12	32 000
APL 40	3100	18	81	1080	12	40 000
APL 63	2480	22·5	101	1080	12	63 000
Stationary I	3000	29·2	116	1510	12	93 000
Stationary II	3000	34·4	124	1610	8	107 000
Stationary III	3000	40	148	1910	8	134 000
Uprated limit	3000	22·5	101	1300	12	63 000
Marine blades	3000	26	115	1500	8	93 000

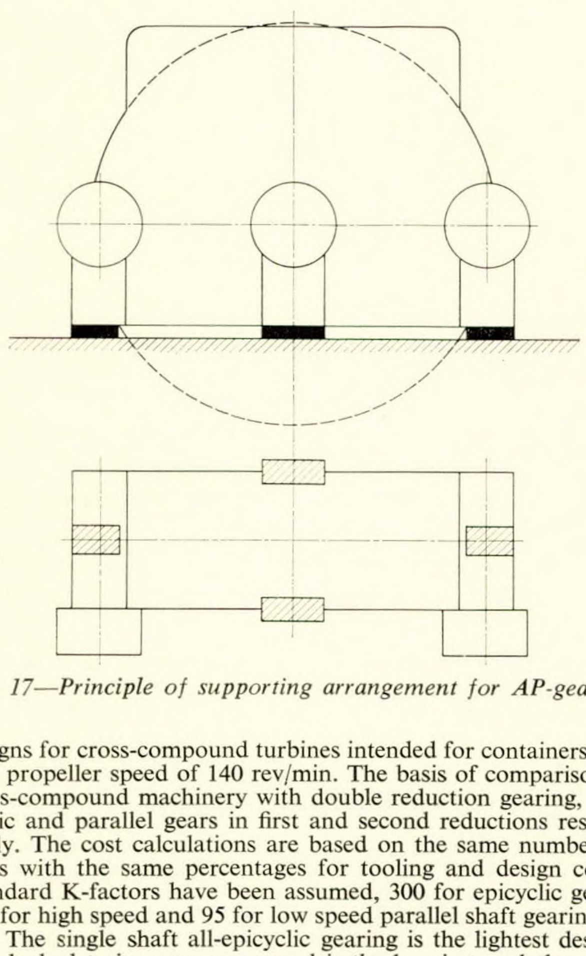


Fig. 17—Principle of supporting arrangement for AP-gearing

designs for cross-compound turbines intended for containerships at a propeller speed of 140 rev/min. The basis of comparison is cross-compound machinery with double reduction gearing, epicyclic and parallel gears in first and second reductions respectively. The cost calculations are based on the same number of units with the same percentages for tooling and design costs. Standard K-factors have been assumed, 300 for epicyclic gears, 125 for high speed and 95 for low speed parallel shaft gearing.

The single shaft all-epicyclic gearing is the lightest design. The locked train cross-compound is the heaviest and the most expensive.

Contra-rotation can be achieved simply by epicyclic gearing in the first stage (Fig. 19). If the epicyclics are combined with parallel gearing with adapted reduction ratios, equal torques can be transmitted to the propellers. If the epicyclics are placed in the final stage (Fig. 20), the aft propeller will have 25–40 per cent higher torque than the fore propeller.

Contra-rotating gears will hardly increase the degree of complication. Sufficient experience is at hand for star and planetary first stage gearing of similar size to ensure reliability. For final stage epicyclics larger sizes will be needed to handle the high torques required. Experience from high torque applications is lacking, but the future will prove the great possibilities of the high torque epicyclics.

For overlapping propellers, a gearing arrangement as in Fig. 21, with one H.P. turbine and two identical L.P. and astern turbines, gives several advantages. The single flow H.P. turbine with or without reheat I.P.-turbine means high efficiency. The two L.P. turbines provide sufficient exhaust area for 100 000 shp and above. The split power train from the H.P. turbine ensures tooth contacts independent of propeller torque variations. With three turbines, several emergency alternatives provide added safety. In such a case the propellers may also be driven independently if failure should occur in the gearing. The H.P. high speed gears are assembled in a separate casing as well as the direction changing gear. To allow deflexions and to simplify erection, long quill shafts with fine tooth couplings are used between the gear casings.

TABLE VII—DEVELOPMENT OF GEAR CUTTING MACHINES SINCE 1895

Machine of year	1895	1919	1957	1968
Index wheel diameter, mm	≈800	2195	4269	4250
Accumulated tooth error, μ	≈100	≈50	25	10
Accumulated tooth errors in seconds of arc	≈50	≈10	2	1
Number of teeth	180	360	540	700

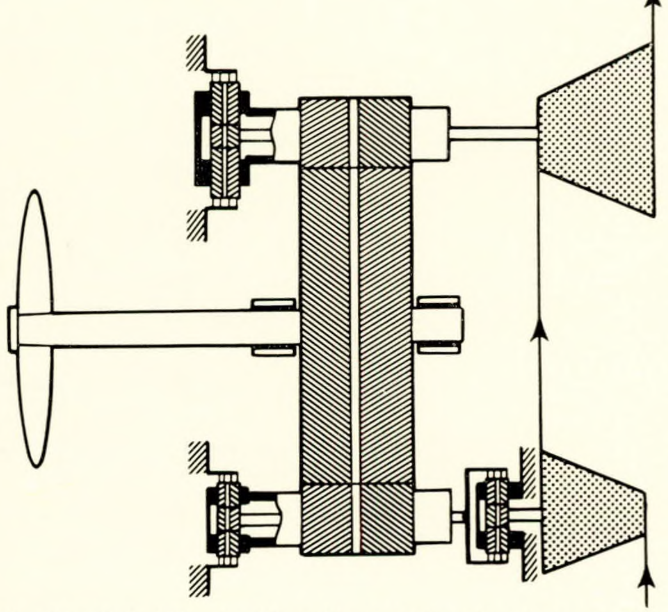
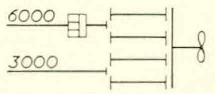
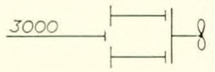
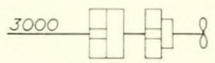
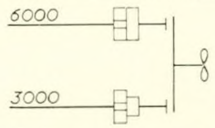
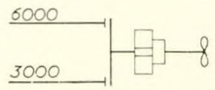


Fig. 18—Principle of AP-turbine gearing for low propeller speeds with triple reduction with star and planetary gears on the H.P. train and planetary gears on the L.P. train

TABLE VIII—DIMENSIONS, WEIGHTS AND APPROXIMATE RELATIVE PRICES OF DIFFERENT GEARING ARRANGEMENTS FOR 30 000 SHP AT 140 REV/MIN

Arrangements and speeds	Reduction	K	Diameter, in	Face width, in	Weight, ton	Approx. relative weights and prices
	1	300			101	1·03
	2	125				
	3	95	132	32		
	1	125			94	0·95
	2	95	128	45		
	1	300	58	18	71	0·75
	2	300	63	21		
	1	300			98	1·00
	2	95	169	43		
	1	125	122	23	73	0·75
	2	300	63	21		

BOILER DESIGN

Steam conditions have remained within the more or less standardized limits of 900 lb/in² 950°F (510°C) for several years. These, rather conservative admission data for stationary conditions, have proved as reliable as the 600 lb/in² 850°F (454°C) of around 1950. With water-cooled walls around the furnace, superheaters designed for low surface temperatures, reliable superheat temperature control, improved water treatment, improved combustion control and improved automatic soot blowers, the boiler designers are ready today to follow the trend towards the region normal for stationary boilers, i.e. 2500 lb/in²–1025°F (552°C). As long as manoeuvring is achieved by astern turbines, the steam temperatures should be kept 40–50°F (22–28°C) lower than in oil fired stationary boilers. When non-reheat boilers are used, the pressures should be kept below 1250 lb/in² with regard to erosion in the L.P. turbine blades. Due to established cost levels of piping, auxiliaries and valves good reasons must be put forward to justify steam data outside the standard condition of 900 lb/in²–950°F (510°C). However, with regard to technology and experience 1150 lb/in²–975°F (524°C) may be reasonable data today for non-reheat.

The modern non-reheat boiler shows great similarity to a stationary boiler (Fig. 22). It has a large radiating combustion region, two superheater banks and a steam cooler for temperature control before the secondary superheater. Rotating gas air heaters after economizer surfaces are becoming usual. Ships with reheat cycles of 1500 lb/in²–955°F/955°F (513°C/513°C) are in service and experience will soon be gained. One large oil company [14] is more conservative and would prefer less sophisticated admission data, 1150 lb/in²–932°F/932°F (500°C/500°C)

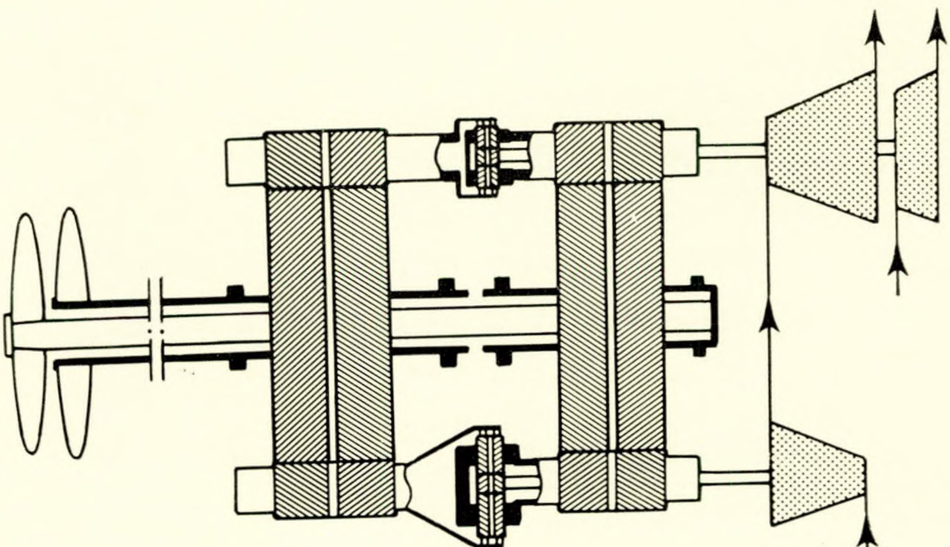


Fig. 19—Principle of contra-rotating gear with epicyclic gears in the first reduction

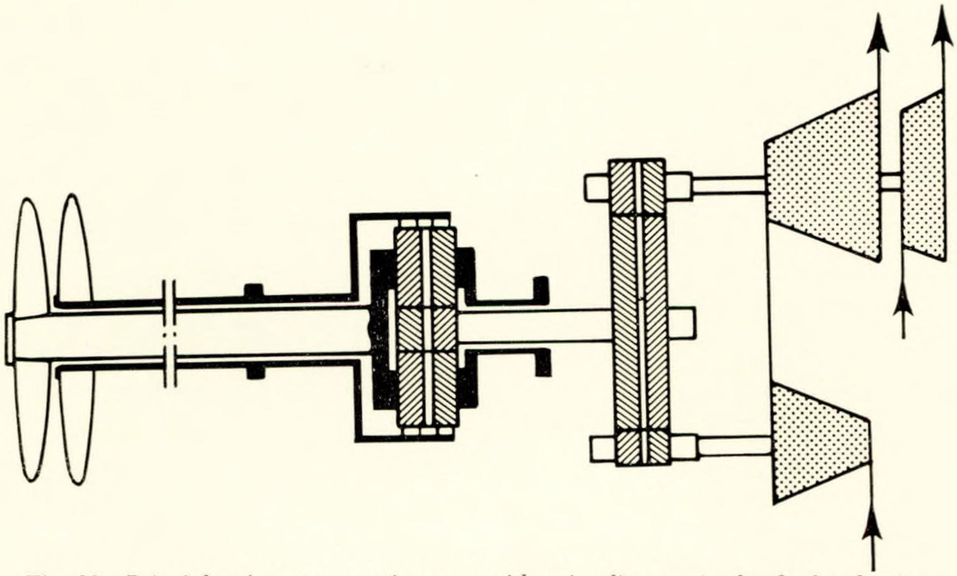


Fig. 20—Principle of contra-rotating gear with epicyclic gear in the final reduction

with regard to service problems. Since the author is no boiler expert he will not go into any second-hand detailed description of boiler designs.

The "one and a half" boiler arrangement has given improvements in total efficiency, comparable to what may be achieved by substantial increases in steam conditions or degree of complexity. The complete propulsion plant must be dimensioned for normal service load, an obvious design criterion proven for many years in the stationary one boiler/one turbine plants.

The author has made many studies during the last fifteen years in order to find out if reheat would be economically justified. Until recently the high extra costs for a reheat boiler compared with a non-reheat boiler for the same exergy* gave somewhat small, if any, economical gains as the fuel savings were counter-balanced by the higher capital costs. However, several reheat plants are now working at sea.

Some boilers are being built with a single furnace and two gas passages with dampers controlling the flow through the reheater (Fig. 23). Dampers in high temperature gas flows have often caused difficulties in service, but with today's knowledge of materials and with cooled bearings it should be possible to solve the damper control technique. Furthermore, with this design the

* The author defines exergy as meaning the energy transferable into mechanical energy in an ideal heat cycle.

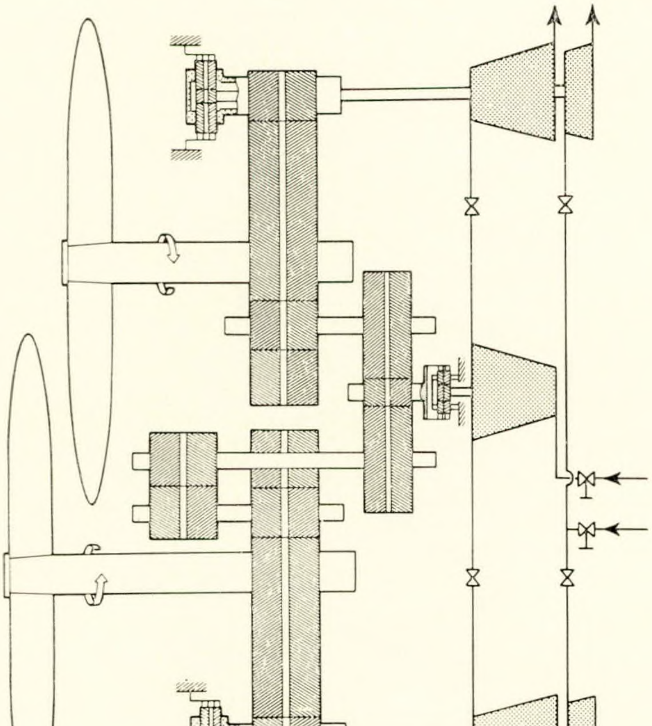


Fig. 21—Principle of gearing for turbine machinery with overlapping propellers

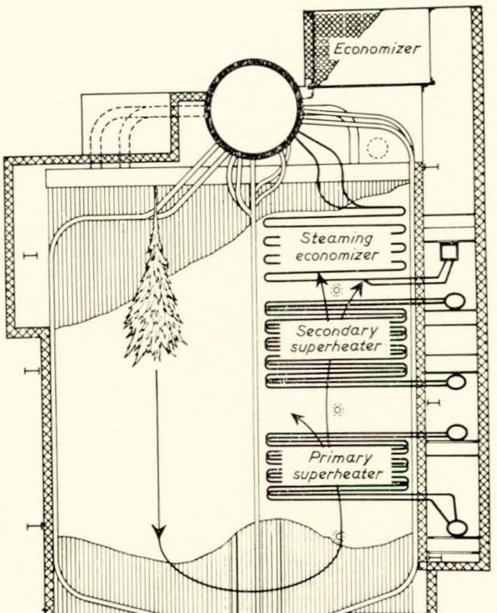
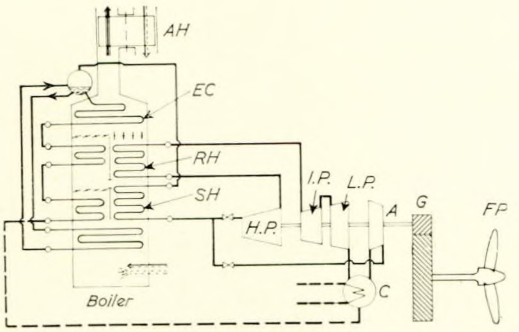


Fig. 22—Modern non-reheat radiant type boiler



AH = air heater
EC = economizer
RH = reheater
SH = superheater

Fig. 23—Principle of reheat boiler with damper controlled gas flows to superheater and reheater

convection surfaces must be dimensioned for maximum astern steam requirement when the superheater gas passage is fully utilized and the reheat passage is idle. The corresponding oversizing of the boiler implies unreasonably high costs with regard to the short astern operating periods.

The two-furnace design (Fig. 24) has also been built for several new ships. Even for this design the convection surfaces must be dimensioned for maximum astern operation and in

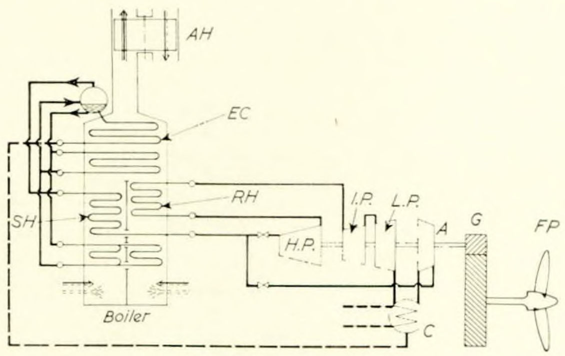


Fig. 24—Principle of two-furnace reheat boiler

addition the furnace for the non-reheat part of the boiler has to be dimensioned according to the same criterion.

Apart from the cost for reheat boiler surfaces and more complicated piping, the additional costs for reheat is to a considerable extent due to the oversizing mentioned. There are two main ways to overcome these undue costs for astern operation.

One is to utilize c.p. propellers in which case the astern

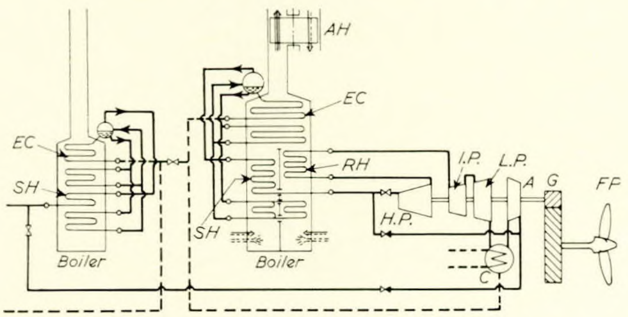


Fig. 25—Principle of two-furnace reheat boiler with supplementary astern steam from the "half" boiler

turbine can be omitted. c.p. propellers are well proven up to 20 000 shp but are still expensive. It is therefore possible that the additional cost for the c.p. propellers will exceed the cost reductions gained from the simplified boiler and turbine designs.

Another way is to utilize the capacity of the auxiliary boiler in a "one and a half" boiler arrangement to compensate for the reduced capacity of the main boiler during astern operation (Fig. 25). The steam from the "half" boiler should be taken through a separate control valve to the astern turbine. With this arrangement the main boiler can be dimensioned solely for ahead steaming.

In spite of the extra cost for astern steaming, the reheat boiler can today be built with low extra cost for the same amount of exergy produced for steam quantities corresponding to propulsive powers in excess of 40 000 shp. With supplementary steam from the "half" boiler for astern power, lower capital costs should be possible and thus improve the reheat economy.

The author would not be surprised if future high power container ships were to be equipped with reheat boilers and C.P. propellers, and the next generation of tankers with reheat and astern steam supplement from the "half" boiler.

OPTIMUM FUEL RATES AND DESIGN OPTIMIZATION

The theoretical calculated cycle efficiency of the marine steam plant varies within wide limits depending on the admission data and the degree of complexity for the cycles. In addition to this, the service fuel rate may deviate somewhat from the calculated or measured trial values. Experienced shipowners usually specify uncomplicated steam cycles with conservative steam data knowing that operating costs in practice can be lower for the simple plant than for the more sophisticated installations even if the estimated fuel rates are considerably lower. In this paper only the steam cycles characterized in Table IX will be considered, all of which are conservative compared with stationary practice. The three steam cycles in Table IX will have varying fuel rates depending on the propelling power and the main engine design. Fig. 26 shows the calculated fuel rates for the various cycles. All twin-engine arrangements have higher fuel rates than the comparable single engine plant because of lower steam flow per unit and consequently lower turbine efficiencies.

TABLE IX—MAIN DATA AND DESIGNATION CODES

Steam cycle designation	T4B	T5B	T5BR
Designations for alternative main turbines	AP APS	AP	APR APRCR APROL
Pressure before strainer, lb/in²	900	1150	1450/320
Temperature before strainer, °F	950	950	950/950
Condenser pressure, ″Hg	1·5	1·5	1·5
Number of HP feed heaters	1	2	2
Number of LP feed heaters	3	3	3
Boiler efficiency, per cent	90	90	90

Designation codes:

Steam cycles:

T	Tanker
4, 5	Number of feed heaters
B	Back pressure type auxiliary turbines
R	Reheat

Main turbines:

AP	Cross-compound, non-reheat
APS	Single casing
APR	Cross-compound, reheat
APRCR	Cross-compound, reheat, "contra-rotation"
APROL	Cross-compound, reheat, "overlapping"
x1	Single engine installation
x2	Twin engine installation

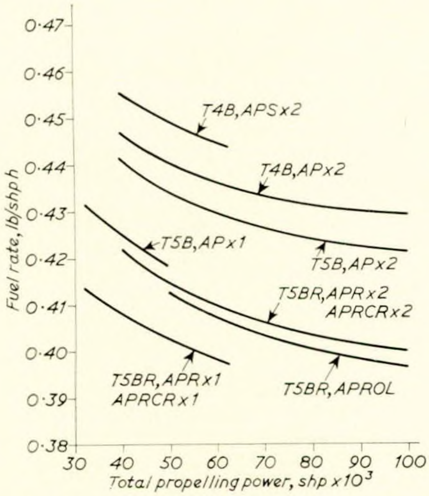


Fig. 26—Fuel rates of different steam cycles and turbine designs

The designation code appears in Table IX and the turbine and gear arrangements in general correspond to the following figures.

APS	Fig. 14
AP	Figs 12, 13 and 18
APR	Figs 16, 13 and 18
APRCR	Figs 16, 13 and 19
APROL	Figs 16, 13 and 21.

The APS turbine is coupled to a triple reduction epicyclic gearing. The steam generating plant considered is of the "one and a half" boiler type, and the main boiler is furnished with rotary gas/air heater.

When comparing the various propulsion plant alternatives with regard to fuel rates, the propulsive efficiency must be taken into consideration, i.e. the comparison is to be carried out at the same effective horsepower. A simple method is to calculate the equivalent fuel rates as follows:

$$g_{eq} \approx g_o\left(\frac{\eta_o}{\eta}\right) = g_o\left(\frac{1}{1+P}\right) \qquad \text{lb/shp h}$$

g = Specific fuel rate lb/shp h
η = Propulsive efficiency
P = Gain in propulsive efficiency per cent in decimal form.

The approximate gain in propulsive efficiencies (P) for the different alternatives can be seen from Table III.

For large tankers the propulsive efficiency improves, as already mentioned, by 2·5–3·5 per cent per 10 rev/min decrease of propeller speed depending on size and block coefficient. The actual saving in fuel rate at constant ship speed is somewhat less due to the corresponding reduction of engine power. The cost of propeller shafting and gear will increase with the torque. At constant ship speed the cost of boiler, turbine and auxiliaries will vary with the change of propeller speed. All these factors have been calculated for a single-screw tanker of 200 000 dwt with data as in Table X.

TABLE X—ASSUMPTIONS FOR PROPELLER SPEED OPTIMIZATION

Ship size, dwt	200 000
Ship speed, knots	16
Propeller speed, rev/min	50–110
Propelling power, shp	27 300–33 500
Steam cycle	T4B
Turbine type	AP32
Fuel rate, lb/shp–h	0·436
Operating time, days/year	330
Fuel price, £/ton	4·9
Freight rate, £/ton	1·6
Bunker capacity, round trip	0·5
Pay-off time, year	3·7

The relationship between capital investment and fuel consumption for propeller speeds of 50–110 rev/min is shown in Fig. 27. A pay-off time of 3·7 years corresponds to an optimum economic propeller speed of 76 rev/min. The saving in bunkers will increase the transport capacity. If this is considered, the annual gains will increase by about 16 per cent, reducing the optimum propeller speed to 66 rev/min.

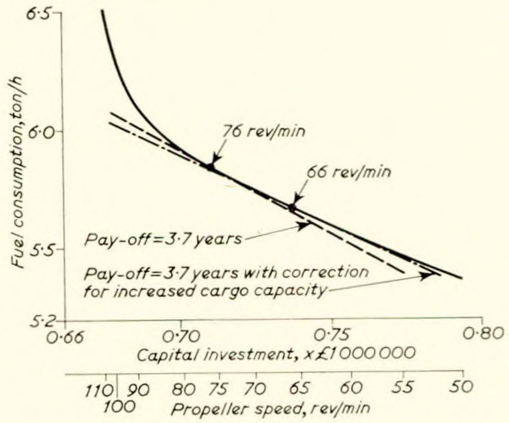


Fig. 27—*Optimum propeller speed 200 000 dwt tankers*

The author's organization has made an attempt to compare weights and first costs for the steam plants mentioned for super-tankers up to the largest sizes considered today.

The weights and first costs are based on the following:

1) steam generating plant;
2) propelling machinery;
3) electrical generating plant;
4) condensate and feed water system;
5) propulsion piping system;
6) propeller(s) and shafting;
7) controls and instrumentation;
8) cargo pumping system;
9) engine room outfit;
10) installation costs;
11) dock and sea trial.

The main boilers are dimensioned for full astern operation without additional steam from the auxiliary boiler. The propeller speeds are 80 rev/min both for single and twin-engine arrangements.

Figs. 28 and 29 show the total weights and first costs respectively for the plants investigated. All data are based on components of standard designs with the consequence that the curves have small steps. However, in this case where the curves are meant only as basis for general discussions, the steps have been regarded as unimportant and therefore eliminated.

When judging the results one should bear in mind the differences in thermal and propulsive efficiencies for the different alternatives. Taking this into consideration one comes to the following conclusions.

For propulsion powers beyond the upper limit for efficient single screws, say 40 000–50 000 shp, the single engine installation, AP $\times$ 1 and APR $\times$ 1 (not shown in the curves), are the most favourable alternatives. At higher powers up to the limit for efficient single contra-rotating propeller arrangement the reheat installation combined with contra-rotating propellers, APRCR $\times$ 1, is definitely the most attractive alternative from a technical/economic point of view. For the highest power range the reheat plant and overlapping propellers are superior. Due considerations should of course also be paid to the technical complexity of the

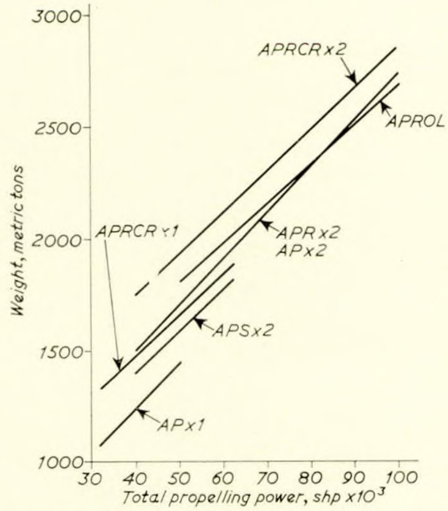


Fig. 28—Weight of steam turbine propulsion plant

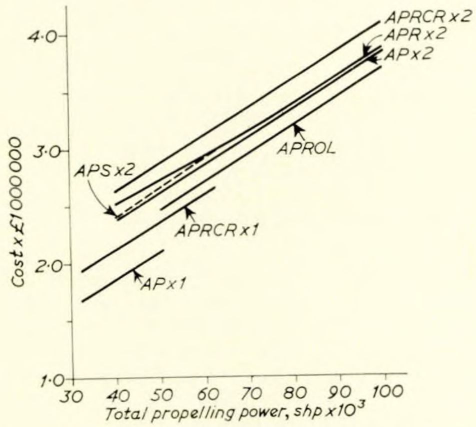


Fig. 29—Cost of steam turbine propulsion plant

different arrangements. The author believes, however, that all suggested arrangements are fully realistic from a technical point of view and deserve to be further discussed.

CONCLUSIONS

Steam turbine machinery has successfully proved its competitive potential as propulsion plant for large tankers and container ships. With regard to propulsion powers, supertankers may require up to 80 000 shp and for future containerships, 100 000 shp is being discussed.

Several propeller arrangements are possible at these high powers and, furthermore, give improved propulsive efficiencies compared with the conventional twin-screw arrangement. Contra-rotating and overlapping propellers will be of major interest. The first application of contra-rotating propellers which is technically economically justified, seems to be for the high powered containership as the propulsion power for the ships under construction has already touched the limit for efficient single screws.

Epicyclic gearing has proved competitive and reliable. In future, epicyclics will also be used for the final reduction. Three stage reduction is commonly used today on the H.P. side of gears for tankers at powers above 25 000 shp and at propeller speeds in the region of 80–90 rev/min. Gear dimensions will be reduced as harder materials and higher K-factors are used foɹ parallel gearing. With regard to costs, epicyclic gearing in the final reduction seems to give the best result.

The single-casing turbine is only technically and economically justified for powers below 20 000 shp per shaft and where weight and size are of special importance. Single exhaust L.P. turbines can be built for powers up to 63 000 shp with non-reheat and up to 93 000 shp with reheat retaining the normal speed of 3000 rev/min. A three-cylinder arrangement with two parallel L.P. turbines seems to be an interesting solution in an overlapping propeller installation.

The additional cost for reheat boilers as compared with non-reheat boilers will decrease with increasing size when referred to the exergy consumption, i.e. the propulsion power. When utilizing the auxiliary boiler as a complement in astern operation, the cost for reheat plants may be somewhat reduced. For ships where C.P. propellers are justified partly by other reasons, improved manoeuvring characteristics, its application becomes more attractive when combined with a reheat steam plant.

ACKNOWLEDGEMENTS

The author wants to express his gratitude to STAL-LAVAL Turbin AB, members of this company's marine section and especially Mr. Gunnar A. Larsen for valuable assistance in the preparation of this paper.

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Discussion

MR. W. H. FALCONER (Associate Member of Council) said that Professor Jung had reminded them of the importance of using the same basis when comparing alternative systems; neither should trends nor their long term effect, both technically and sociologically, on total economics be overlooked. The relative costs of energy and labour had a significant effect on the complexity of the cycle chosen. If the cost of fuel continued to decrease then it was reasonable to expect that cycles would become simpler.

Whilst agreeing with the author that there was little to be gained by improving the accuracy of gear cutting beyond the figures quoted, it should not, however, be assumed that all manufacturers had reached such a high standard, but it did demonstrate that high quality gearing could be produced competitively today.

The value of Table VIII would be enhanced if the costs of the complete engine were included on the same basis, bearing in mind the limitations of the single cylinder design.

Mr. Falconer expressed the view that Professor Jung's comment on boiler design had passed unheeded by the majority of boiler manufacturers. It was a matter of regret that owners still found themselves in the position where an otherwise satisfactory design was modified, often with disastrous results, for short term economic advantages. Boiler designers should be given the opportunity to accept the responsibility for co-ordinating the overall design of the steam generating plant. Ideally, a standard boiler design should be produced for each turbine frame size.

The paper also reflected the advantages which could accrue when technology and management are integrated. It was not perhaps generally appreciated that over 30 per cent of all marine turbines at present under construction were of the types described. This was truly a remarkable achievement.

Snow and Levis never did quite agree about the "Two Cultures and the Scientific Revolution" but that was a long time ago. Ten years later we found ourselves in a similar position—society was divided into two camps—there were those who attributed our shortcomings to what was euphemistically called the "technological gap" and those who attributed our failures to the "management gap". To overcome these gaps we must learn not to separate technology from management—as Professor Jung had so ably demonstrated, they were complementary to each other. In the final analysis these gaps could only be closed by education. This Institute should concern itself with this aspect of management education.

If the older industries, such as marine engineering and shipbuilding, were to survive they must attract their fair share of the brighter young graduates. To attract such people these industries must change their outlook and introduce some of the resourcefulness and imagination which was now commonplace among the newer industries.

Professor Jung had shown that this was possible although it still remained a major problem for most of those present at the meeting. It must be appreciated that massive capital investment was also necessary.

Mr. Falconer commended Professor Jung's paper to a very much wider audience than was able to be present at the meeting.

MR. W. M. BROWN, B.Sc. (Associate Member) said he was particularly intrigued with the section of the paper devoted to gear design, because during the past eighteen months his company had been examining different gear configurations for various types of prime mover and propulsor. So far as the marine steam turbine was concerned, their conclusions were very similar to those arrived at by Professor Jung, although much higher gear loadings were used. That suggested that the weight of the parallel shaft arrangements presented in the paper could be reduced by at least 60 per cent with the use of K factors of 500, and it was emphasized that such loadings could now be considered to be somewhat conservative.

However, even at that level of loading, the use of epicyclic gears in the final reduction showed considerable savings in weight. Similar conclusions were also reached by Pametrada in 1965, with the result that the Paraplan design was produced for which it was claimed that with conventional gear units employing through hardened materials, a reduction in weight of 35 per cent could be expected.

Such savings in weight resulted because with an epicyclic arrangement the load could be carried through more than one mesh point, thus enabling smaller gear assemblies to be made for the transmission of the same power. However, the effectiveness of the arrangement depended upon equal load sharing between all the planet wheels, and to ensure that it was traditional for the plant spindles to be firmly supported at each end, and the carrier straddled the sun gear. The complexity of the carrier structure effectively limited the number of mesh points which could be achieved, and that, together with the complexity of introducing the required degree of flexibility into the system, considerably negated the basic advantages of the epicyclic arrangement.

Those disadvantages were overcome by Mr. R. J. Hicks who, two years previously, described to the Institute during the discussion to Mr. T. P. Jones' paper, a novel method of introducing flexibility into epicyclic gears. In that solution, the rights of which had been acquired by Mr. Brown's company, the planet wheels were mounted on flexible pins cantilever mounted from the carrier. Thus, the need for straddle mounting was eliminated, and consequently a much simpler planet carrier resulted; that allowed the maximum possible number of planets to be used for a particular gear ratio, so that the system became even more compact and less expensive.

Applying that system to the double planetary arrangement shown in Table VIII with loadings of 300 K in the primaries and secondaries, the result was:

First Reduction:
Diameter 52 in
Face 13·5 in
Second Reduction:
Diameter 54 in
Face 19·5 in
Total weight 46 tons.

That represented a saving in weight of 35 per cent of the double reduction epicyclic arrangement presented by Professor Jung, or a 53 per cent reduction in the weight of the standard AP design.

So far as Professor Jung's remarks about hardened and ground gears were concerned, it appeared relevant to indicate that 500 K and equivalent bending stresses were now considered to be quite conservative in naval circles, and if those were applied to the design he had just mentioned the total weight would be reduced to about 30 tons.

MR. I. T. YOUNG (Member) said that the AP series of turbine assemblies, incorporating single-plane gearing with epicyclic primary gears, had earned a well deserved success. On the other hand, he wondered if Professor Jung was not mistaken in implying that gearing arrangements embodying epicyclic gears were always the best. Epicyclic gears had their limitations, in many cases not shared by parallel shaft gears.

Mr. Young wondered whether the author was serious in implying that 6000 rev/min was the maximum practicable speed for input to an epicyclic gear. With parallel shaft gears, there was no such limit. Again, in Table VIII, 3000 rev/min was chosen for the single cylinder turbine. Was that to suit the epicyclic gear?

Also in Table VIII, 140 rev/min of the propeller was taken as the basis of comparison, which might be the speed at which the epicyclic arrangement was at its best. With parallel shaft gearing, on the other hand, designs with powers well over 30 000 shp at output speeds of 50 rev/min, or even lower, were perfectly practicable without going beyond well-proven techniques, and without resulting in an unduly cumbersome arrangement.

The serious defect in the epicyclic was its inability to split the drive, and thus for each turbine input only one pinion could operate on the final reduction wheel.

As an example of the adaptability of the parallel shaft gear in that regard, Fig. 30 showed a turbine/gearing arrangement for transmission of 18 000 shp at 50 rev/min of the

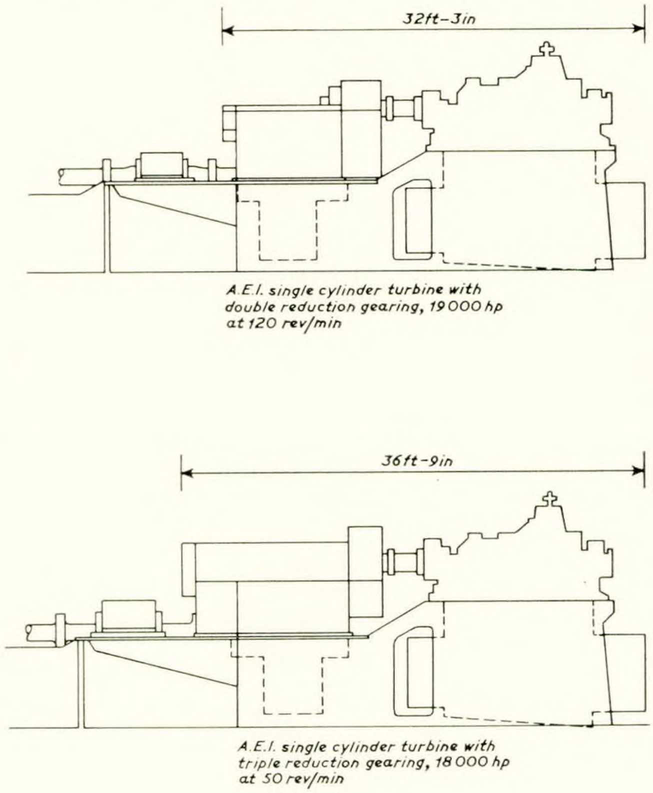

FIG. 30

propeller, contrasted against the same arrangement for an output speed of 120 rev/min. The compact gear-box arrangement had been obtained by three reductions, resulting in four pinions on the main wheel. The first reduction was forward, and the second reduction aft of the output wheel. The second reduction gears and all pinions were surface hardened by nitriding, while the first and final reduction wheels were of 50 tons UTS steel. K values were 160, 280, and 130 for the first, second and third reductions, respectively.

The design of gear case shown in Fig. 30 (top) was the speaker's firm's new single input type, designed to incorporate all rotating parts in a single fabrication, supported just below the level of the main shaft.

Fig. 31 showed this more graphically, and it could be seen how the turbine gearing and thrust block seatings were combined in a single rigid structure.

Professor Jung threw doubt on the use of surface hardened pinions and wheels for large power parallel shaft gears, and suggested that pitch limitation might not allow full advantage to be taken of the increased surface load capacity.

As he was aware, K values in the region of 500 were common with naval gears, without excessive pitches, but admittedly with much smaller diameter wheels than would be required for the low propeller speed now required for tankers and bulk carriers. Investigation showed, however, that with an output of 35 000 shp at 50 rev/min stresses could be kept within Lloyd's limits for nitrided pinion and wheel with

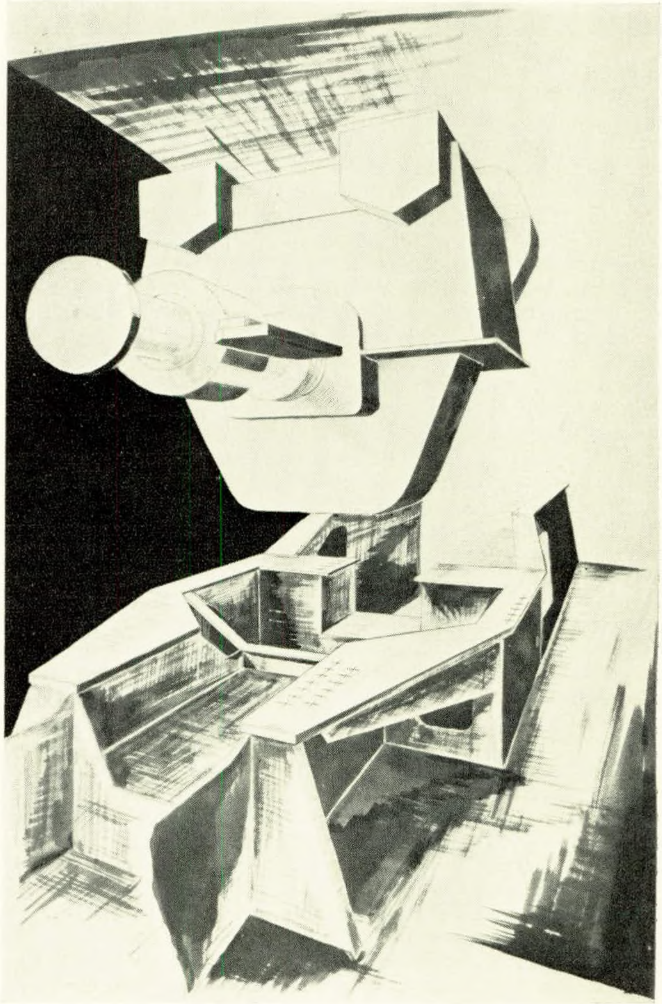

FIG. 31—*New A.E.I. single-input gear-box and proposed integral main thrust and gear-box seating*

a pitch little in excess of three Dp at a K value of 250. Admittedly if the design was limited to two pinions on the main wheel, the pitch would be greater, but even then it would be less than two Dp. Mr. Young said he would close by making reference to the most ingenious arrangement for overlapping propellers shown in Fig. 21. Three gear-box layouts for twin propeller drive with three independent power sources were, of course, no novelty, and his firm was at present producing a series of those sets for the U.S. Navy. But Professor Jung was advocating that system for very large powers, without apparently being able to take advantage of the flexibility provided by the use of controllable pitch propellers.

Could Professor Jung explain how overloading of one side of the H.P. train could be avoided at high rudder angles, when the distribution of torque between the two propellers would be far from uniform?

MR. T. B. HUTCHISON (Member) said that at the time when Professor Jung presented his paper before S.N.A.M.E. in September, 1963, he and his colleagues had already solved the boiler problems which had largely contributed to the decline of the steam turbine as a viable marine propulsion unit. At the same time, the need to optimize the total propelling system was dealt with. Professor Jung had made a continuous effort to provide elegant solutions in the optimization, and powering of large ships.

In Fig. 3 the dramatic recovery of the steam turbine was clearly shown, but if tankers of 150 000 tons or more alone were considered, the split between steam turbines and Diesel engines would probably show a 90/10 ratio in favour of the former.

In the development of large tankers, it had been observed that ship's speed had not increased to any marked extent, but in the opinion of the speaker, that had nothing to do with propulsion problems. Ship's speed was derived from an economic assessment of all factors in the total transportation system. That would include loading and discharge facilities, duration of voyage, tankage and refinery throughput, and while the increased hydrodynamic efficiency of large ships affected the final outcome, other factors had to be taken into account. In developing what was reasonable and acceptable in regard to propeller dimensions, and revolutions, what today appeared unreasonable might be quite realistic in the future development of large tankers, and particularly in those with unusual length/breadth dimensions.

In a twin screw system it was possible to supply quite considerable power in each shaft, with propellers of relatively large diameter and low rotational speeds, even when considering the 30 ft diameter propellers running at 80 rev/min generally accepted today. When comparing twin screw arrangements with overlapping propellors, or contra-rotating systems, one should keep in mind that the former are not affected by any interactions.

The contra-rotating system had been discussed, and its problem areas recognized. An overlapping system was relatively new, having been dealt with by Pien and Strøm-Tejsen, and recently by Munk and Prohaska. Professor Jung, in his conclusions, considered that those areas would still be of major interest.

The speaker referred to the layout given in Fig. 21 and said that one must seriously question the engineering which would be involved to make the system possible. Perhaps Professor Jung could comment on the shipbuilders' capability to line up and pitch the stern tubes to give the degree of vertical and horizontal parellelism demanded by the spacing and tolerance of the machinery installation.

In a single screw arrangement, the machinery was aligned to the shaft coupling and chocked accordingly. In an overlapping system it could be seen that this would present some difficulties. As the speaker saw it, both couplings in correct position should have identical degree of deflexion throughout their length. With a large relatively stiff system, the shipbuilder might find difficulty in attaining the degree of accuracy required. The system was of some torsional complexity, particularly when account was taken of the propeller characteristics operating in different wake conditions. To deal with some of those problems, Professor Jung had drawn attention to the in-built flexibility of quill shafts, flexible couplings, etc.

In addition, there was the flexibility of the ship, which might impinge on the reliability of the installation. In the single screw or contra-rotating system longitudinal bending and torsional flexing of the hull structure would not be unduly upsetting. That would also apply to a twin screw arrangement, but with an overlapping system which was connected back to one single point, twisting of the hull structure or displacement of seating would have to be taken into account.

Emergency conditions also presented problems. Professor Jung's observations on how the system would cope with one lost propeller blade or what happened when one L.P. turbine failed, and load-sharing took place through the H.P. pinion at reduced output. Was there a possibility of excessive torque and tooth loading occurring? Additionally, in a seaway, wake conditions would vary when turning at speed or manoeuvring at large helm angles. Could Professor Jung see any problems which would give rise to unduly high vibratory stress in the gearing elements?

When considering highly efficient marine steam turbine installations the complexity of the system had been criticized. That mainly referred to the use of feed heaters and, in the speaker's opinion, the word "complex" was mis-used. Static pieces of equipment, if properly designed, added to the economic efficiency of the plant, and were desirable equipment. However, in the case of the two furnace proposal (see Fig. 25), surely that introduced the type of complexity which could lead to trouble? Was it possible that the boiler manufacturers would agree that a re-heat boiler had to be oversized to withstand astern load? The use of auxiliary boilers would surely lead to a more expensive and complex arrangement. The auxiliary boiler would need proper output characteristics, and additional piping, valves, etc. If there were a need to keep a re-heat boiler down in size, and cost, then it would be much simpler to add one stage to the astern turbine, raise the efficiency, and reduce the steam demand. The speaker felt he need hardly mention that the two furnace boiler concept was difficult to control at optimum level of efficiency.

He closed by agreeing with Professor Jung that the future of high powered installations lay in the use of re-heat with possibly CP propellors.

Mr. K. Brownlie (Associate Member) said the author's conclusions that the single cylinder turbine was less expensive than the cross-compound machine only for powers somewhere below 20 000 shp followed from his assumptions for the design of the single cylinder turbines. The figure of 6000 rev/min had been taken as the maximum speed for the rotor of a 10 000 shp turbine, and from Fig. 15 it could be seen that the author had assumed that for larger powers, the running speed had to be reduced in accordance with the relationship:

$$\text{speed} \propto \frac{1}{\sqrt{\text{shp.}}}$$

That relationship was correct if it was assumed that the last blade tip speed must remain constant, and that the exhaust stages were all similar, so that the tip diameter was proportional to the square root of the shaft horsepower.

However, although 6000 rev/min was a reasonable rotor speed for a 10 000 shp turbine, it did not necessarily represent a limiting value of either the centrifugal stresses, or the last blade tip speed. It was possible to design larger single cylinder turbines in such a way that the tip diameter of the exhaust blades increased much less rapidly with the power than the assumption that the author had made.

Therefore, large single cylinder turbines could be designed, having higher rotor speeds than those given in Fig. 15, without exceeding proven stress levels, or blade tip speeds.

Fig. 32 showed a view of turbine machinery for 17 500 shp normal; 19 250 shp maximum. Six of those machines were being fitted in twin screw container ships. The turbine was

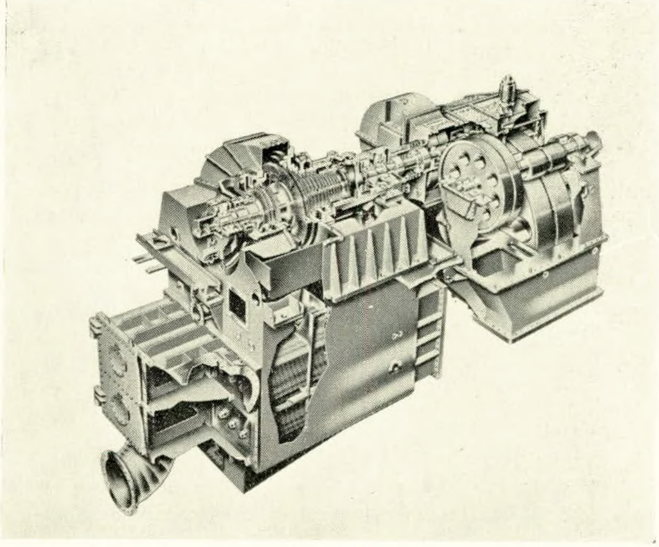

Fig. 32

suitable for 20 000 shp and the rotor speed at maximum power was 5360 rev/min.

The last blade tip speed was about 1300 ft/s, which corresponded with the author's proposed uprated limit for the 22·5 inch long blade operating at 3000 rev/min.

With skill it was possible to design a 40 000 shp single cylinder turbine for a running speed in the range of 4750 to 5000 rev/min without exceeding normal stress or tip speed limits.

Since the number of turbine stages required for a given diameter was proportional to the reciprocal of the speed squared, or alternatively, since for a given diameter the number of stages was proportional to the reciprocal of the speed, it was clear that by increasing the speed the number and the diameter of stages could be reduced, and the size and cost of the single cylinder turbine reduced.

The work carried out by Mr. Brownlie's company had shown that large single cylinder turbines and gears for the power range from 20 000 to 40 000 shp were approximately 15 per cent cheaper than the equivalent cross-compound machinery.

Economic studies based on that difference showed the single cylinder turbine to be very competitive. For example, for machinery for a 40 000 shp machine at 80 rev/min output speed assuming 300 days/annum at sea, £5 per ton fuel price, interest on capital of 5 per cent, fuel consumptions of 0·44 lb/sh p h for the cross-compound and 0·449 lb/sh p h for the single cylinder turbine machinery then the additional cost of the cross-compound machinery would only be recovered in fuel savings in approximately 14 years. If interest and insurance charges on capital of more than 5 per cent were taken, then the time required to recover the extra cost of the cross-compound machinery would be increased. Thus, for 7 per cent charges on capital the period would be approximately 19 years and for 9 per cent charges nearly 32 years. This did not include any allowances for the reduced maintenance, easier installation, etc., which was to be expected with the single cylinder turbine machinery.

The author's remarks concerning L.P. blading designs summarized the position very clearly, but even with continuous shrouding or lacing, Mr. Brownlie would prefer to keep the frequency of the fundamental tangential mode of vibration above the second shaft speed multiple.

MR. T. P. JONES (Member) thought that the value of the present paper would be increased if reference was made to the paper* the author and Mr. Lundstrom had presented before S.N.A.M.E. in 1963.

The author had referred to a number of factors which had led to the resurgence of the use of steam turbine machinery, but he had, of course, omitted any reference to possibly the biggest single factor which had led to the present position in Europe, i.e. his leadership and inspiration, both inside and outside his organization.

Up to 1963 marine engineers had all watched the swings in Europe between Diesel and steam turbine main propulsion. At that time Mr. Jones thought it was true to say no one could possibly have foreseen that the AP machinery then introduced in New York would become so successful so quickly.

Among its innovations was the use of epicyclic primary reduction gears and, as slower propeller speeds became required, so did the use of epicyclic primary and secondary reduction gears. Some of the development of those gears was described in a paper to the Institute two years ago. Service experience began in 1965 and, up to the present time, no design modifications as a result of service experience had been carried out or were contemplated.

Sixteen ships' sets of gears had been examined after twelve months or longer in service, one of them after two and a half years' service.

There was now a very considerable number of running hours experience engendering every confidence in the further use of epicycle gears in single and multi-propeller installations. Apart from their advantages, necessity almost, in contra-rotating installations, he agreed with the author that they would undoubtedly prove advantageous as final trains.

Professor Jung had stated that the reliability and competitive ability of epicyclic gears had now been proved. As a maker of the parallel shaft type, and a user of the epicyclic type of main propulsion gear, he was, of course, in an unique position to be able to make such a statement. He also stated that he would not be surprised if future high power container ships were to be equipped with re-heat boilers and controllable pitch propellers. The speaker said he would go a little further on another aspect and say he would be more than a little surprised if final train epicyclic gears were not used in future high power steam turbine installations, and go further still and suggest that possibly such installations might even be at sea in 1971/72.

Such final trains would be very similar to large epicyclic gears which would also come into use with the increasing power of medium speed Diesel engines. In fact there was likely to be a great similarity between final train gears used for steam or gas turbine main propulsion and single input/single output gears for high power medium speed Diesel main propulsion.

The author had illustrated various gearing configurations. He would have been the first to agree that in such cases there were nearly always a number of alternatives which could be used to achieve the same end. Mr. Jones was sure the author would not be in the least surprised to know that he favoured a solution for the arrangement given in Fig. 21, which would use rather more epicyclic gears, eliminate the use of the idler gear, and considerably reduce the size of the parallel shaft gearing, while still maintaining the same propeller centres.

Fig. 3 indicated a tremendous upsurge in tankers with steam turbine main propulsion machinery from the first quarter of 1966 to the middle of 1968, when around four and a half million horsepower were on order. Of those, about half had been, or would be, fitted with epicyclic primary and/or secondary reduction gears. It seemed possible that the recent enormous advance in the use of such gears had, at long last, really established the foundation for their continued and increasing use in large ships' main propulsion.

MR. J. F. PRESTON (Associate Member) felt that the paper constituted a valuable addition to the Institute's proceedings, since it contained a careful balance of established facts, and inspired forward thinking. The gathered facts had been clarified, helping to prepare one's mind for the look into the future.

On the subject of propeller arrangements, an impressive array of alternatives to the conventional single screw were examined. Before comparing those, however, one should be sure that the conventional had been optimized. They had formed the opinion that for large tankers, with very full block coefficients, much could be gained by further study on flow characteristics at the after end. No doubt that would raise the limit of the single screw. Eventually, of course, one must look at the alternatives.

From the practical aspect, one could never afford to ignore that, stern tube seals constituted a very real problem, even with a single screw, and, therefore, the application of contra-rotating propellers might have mechanical limitations. It could well be that they would see a progression from single screw conventional to the ducted, and then to twin ducted if the hoped for gains were established. Although not the subject of the paper, the ducted propeller would also seem to offer the improved steering characteristics which were necessary for the larger ships.

Whether the choice was twin-ducted or overlapping, the

* Jung, I. K. E., and Lundstrom, T. 1963. "Recent Developments in Propulsion Gears for Steam Turbines." S.N.A.M.E., New York Metropolitan Section, 24th September.

configuration shown in Fig. 21 was most interesting. One could imagine that excellent fuel rates, optimum application within the ship, and adequate safeguards could be had from such a plant.

The pros and cons of re-heat were well described in a condensed form, with the limitations of astern operation suitably highlighted. To some extent, the boiler load to meet cargo pumping requirements could become a limiting factor in boiler sizing.

The idea of using the half boiler as an additional steam source required parallel operation of two boilers with entirely different characteristics. That would certainly prove to be a mind-stretching exercise for the control engineers, and one had grave doubts on the ability to find a good solution.

In the section on Boiler Design, the author left one with the feeling that the marine industry was poised and ready to follow the trend towards stationary practice with steam conditions as high as 2500 lb/in^2.

Mr. Preston felt that that must be questioned. To date, the improvement in boiler water treatment was only relative, and fell short of the standard needed for operation at such high conditions.

During the commissioning of a plant operating at 900 lb/in^2, 950°F, in December 1968, silica recordings showed figures as high as 30 p.p.m. on the first trial. Emptying the boiler and flushing out carefully prior to the second trial did not prevent the steady increase, up to 70 p.p.m., on the next trip out.

Since no measurable trace of silica was recorded in the condensate, one had to assume that the origin lay in construction debris such as refractory carried into the boiler by personnel. Since that was really the first time one had tried to quantify silica, and the boiler had been prepared for service with particular care, applied to the boil out, acid cleaning and flushing, one had to conclude that that sort of circumstance was fairly common.

Had the ship been operating boilers at the pressure of 2500 lb/in^2, there could be little doubt that turbine performance would have suffered due to blade fouling.

Mr. G. Victory (Member) said that Fig. 1 indicated what they already suspected, that bulk carriers and tankers somewhat in excess of 100 000 dwt were likely to be the main market for the marine turbine of the future. Most of the problems of driving such ships through the water at the relatively high speed dictated by modern economics had been dealt with in some detail. He wished that the author had had time to spare to consider the problem of stopping.

No commercial value appeared to have been placed on the ability of engines to provide astern power at almost 100 per cent of the ahead value, as was available in a Diesel engine. True, full power could not be used effectively in all phases of a "crash stop" manoeuvre, but recent papers on the stopping of large vessels suggested that it was of greater significance over a longer period during the manoeuvre in large ships, than it was in smaller ships, and the safety of the entire ship could depend on its stopping ability.

Mr. Victory thought that stopping ability would have been a good selling point for the c.p. propeller, as its use also reduced the dead-time involved in getting the propeller to turn astern, which was so important in the stopping distance.

In the section on Steam Turbine Design, the author referred to an astern criterion of 75 per cent ahead torque at 65 per cent of the ahead rev/min. Mr. Victory asked what was the source of those figures, as the authorities, who were not vague on this point, usually considered 80 per cent torque at 50 per cent of the ahead rev/min representing 40 per cent of ahead power as an acceptable figure. It would appear that in respect of turbines, this low figure was connected with the desirability of avoiding having to incorporate an astern turbine on the H.P. shaft.

In any case, referring to the one and a half boiler arrangement, even that facility, i.e. of 40 per cent stern power immediately available, appeared to be in some doubt. He assumed that that was similiar to the "get you home" boiler arrangement supplied at the present time on many of the single boiler tankers of 200 000 dwt and upwards. He often wondered if that was a step in the right direction, as it was not feasible to keep the boiler flashed up at all times, and raising steam could well take an hour or more and much could happen in that hour. Under "Boiler Design", however, the author went on to say that the "half-boiler" could be used to supplement the available astern steam, the main boiler being dimensioned solely in respect of the ahead steaming conditions.

Mr. Victory was sure everyone had experienced numerous occasions when the first movement from the bridge had been an emergency full astern, and he wanted to ask what happened to the stopping ability of those large ships, which was already rather suspect, in conditions when the "half boiler" was not available. If it were to be available at all times then surely it would be uneconomic.

He was intrigued at the reference to turbine machinery—steam or gas. He wondered whether that was deliberate, and whether Professor Jung foresaw a day, not too far distant, when the gas turbine would be competitive, at least with the steam turbine, in the power ranges they were considering. Perhaps Professor Jung would refer to that in his remarks.

Mr. R. W. Jakeman, M.Sc. (Associate Member), found the paper extremely interesting, but in some instances most depressing. His depression reached its climax in Fig. 4, where turbine designs throughout the world were subdivided into the United States, Japan, Sweden and others. His remarks were not intended as a criticism of the author, who he thought had presented a fair picture of the present situation, but that was surely a clear confirmation of the fact that Britain, who pioneered the development of the steam turbine, was now becoming a nation of builders under licence. There were, he knew, many notable exceptions to that unhealthy state of affairs, but mercantile marine turbine design was, regrettably, not one of them.

Regarding the various propeller arrangements discussed, he felt that the overlapping propeller system was a doubtful starter. The method of avoiding cavitation damage to the aft propeller shown in Fig. 7 appeared to be rather academic when one estimated typical dimensions for that configuration. For example, considering 25 ft diameter, four blade propellers, rotating at 105 rev/min, on a vessel doing 16 knots, the pitch of the blade tip helices was less than 4 ft, thus at 25 ft propeller centres, that is zero overlap, the blade tips of a correctly positioned aft propeller would have less than 2 ft clearance from the helices of the forward propellor.

At about 18½ ft propeller centres, that is 6½ ft overlap, a correctly positioned aft propeller would be just brushing those helices on both sides. Those figures were based on a very idealized view of the situation, as they did not take into account the non-uniform wake at inlet to the propellers, and as such were undoubtedly optimistic.

In view of that, Mr. Jakeman asked the author what degree of overlap was required in order to gain a reasonable improvement in propulsive efficiency.

The contra-rotating propeller arrangement seemed a much better proposition, on account of its higher potential improvement in propulsive efficiency, and neater design layout. It posed some difficult transmission problems, of which the design of the bearings for the inner propeller shaft was probably the most significant. It was well appreciated that if the inner and outer shaft speeds were identical, a simple journal bearing formed by mating surfaces of each shaft would fail to develop a hydrodynamic oil film.

He believed that proposals had been made to overcome that effect by producing some departure from circularity in one of the bearing surfaces. However, the load line moved relative to both bearing surfaces in that arrangement, and white metal fatigue problems might, therefore, be encountered.

A more satisfactory arrangement was to interpose a free floating ring between the inner and the outer shafts to transfer the load of the inner shaft to the outer shaft. The torque on the inner and the outer surfaces of the ring, due to viscous friction, could be approximately balanced by making the factor length times diameter cubed over clearance the same for each surface.

Very slow rotation of the ring would probably occur in order to balance the actual torques on the inner and outer surfaces exactly.

Alternatively, if the ring was given a slightly non-uniform distribution of mass about its rotational axis, it could be completely prevented from rotating by gravitational force. White metal could be applied to the inner and outer surfaces of the ring, thus virtually eliminating potential fatigue problems. The design, from a hydrodynamic viewpoint, would be virtually identical to a conventional journal bearing.

The contra-rotating gear-box arrangements described by the author raised the question of the optimum drive conditions for contra-rotating propellers. Glover, in a paper given about two years previously, indicated that that optimum was equal to speeds of rotation.

Neither of the arrangements given in the paper necessarily implied that condition; could the author comment on that?

Mr. Jakeman was interested in the comparison of gear-box designs given in Table VIII. That showed a very clear advantage in favour of epicyclic secondaries for the two-cylinder turbine. From the author's comments on that, one gained the impression that it was simply a question of time before his former company would be offering that type of machinery. If that was so, perhaps the author would care to comment a little more on that, and, in particular, if he had any thoughts on the arrangements for inspection and withdrawal of the gear elements from epicyclic secondaries. As many would know, a design of that type was developed by Pametrada under the name Paraplan. Although it now seemed most unlikely that the Paraplan gear-box would ever progress beyond the drawing board, it was at least most gratifying to see the design concept vindicated by the author.

Correspondence

MR. A. F. HODGKIN (Associate Member) wrote that the author referred to remarks he made in September 1963 concerning the improvements then required in all elements of steam propulsion plant and concluded that a competitive potential had been successfully proved. The current position showed a distinct favour for steam propulsion, but how much of this was due to the fact that present power requirements tended not to favour alternative propulsion means? The improvements made by turbine and boiler designers since 1963 were a significant step forward, but did not yet permit a relaxation of effort. A number of recently commissioned vessels had experienced difficulties of one kind or another. Some could be teething troubles associated with new designs, whilst others highlighted the need for improvement in the selection and application of many small items upon which great reliance was placed to preserve the safety of the plant. It was necessary once again to reiterate the need for integrated design where the many individual items comprising a steam plant could be evaluated, chosen and combined to give an overall system acceptable to all parties. A process, seen as a continuation of the co-operation in design, was needed, as suggested by Mr. T. B. Hutchinson.*

It was believed that there was little justification for allowing superheater outlet steam temperatures to greatly exceed 950°F (439°C), if only because there was still insufficient experience with the newer boiler designs. In due course, however, it should be possible to consider 1000°F (538°C) final steam temperature. The limited experience presently available was promising in this respect, and this situation was probably due to the move towards single main boiler propulsion systems. These allowed the main boiler to have more ample proportions than previously, giving the boiler designer the opportunity to use oil firing equipment intended for operation at very low excess air, whilst achieving complete combustion within a furnace which had a very long flame path. Additionally, superheater tubes were much more widely placed, a 4 in gap between tubes in the same row not being uncommon. A boiler having these features was illustrated in Fig. 22, although it was seriously doubted whether "Combustion Engineering" would be pleased to endorse this design, which was, in fact, attributable to Babcock and Wilcox.

* Hutchison, T. B. 1966. "30 000 s.h.p. Unitized Reheat Steam Turbine Design." *Trans. I.Mar.E.*, Vol. 78, p. 109.

Reheat boilers of this same basic "Radiant" type were also available. These were of the single furnace, two-gas passage arrangement, but in contradistinction to that mentioned in the paper (Fig. 23) dampers were not used in high gas temperature zones, so that risk of damper problems were eliminated. Oversurfacing of the boiler was not required and the reheater passage was never completely idle. When steaming astern, all the air registers and the entire furnace was available, and the residual gas flow over the reheater was cooled by superheater surface to a level below the safe working temperature of the reheater tubes. The reheat boiler had similar plan area dimensions as a straight cycle boiler of the same power, but was taller by virtue of the fact that it contained a reheater which of necessity operated at low temperature differences and was, therefore, large in size. It had not been found necessary to invest in additional surface for astern operation, although the efficiency in this mode might be fractionally reduced. It was considered that this boiler, capable of instantly giving maximum astern power, was much preferred to one where the ability to obtain maximum astern thrust was dependent upon the speed at which a second boiler could be coupled in an emergency.

MR. A. F. VEITCH (Member), in a written contribution, commented on the section dealing with propulsive efficiencies and the great improvements which Professor Jung showed to be possible, particularly for large tankers and container ships.

Having been concerned with design of steam turbine propulsion machinery for very many years, he knew only too well how increasingly difficult it was to gain even one per cent when efficiencies of steam turbines and gears were about 85 per cent and 98 per cent respectively. It was not an unusual occurrence to undertake an investigation requiring the aid of specialists in aerodynamics, production methods, stresses involving photo-elastic techniques and various other branches in an endeavour to pick up no more than one per cent overall. This procedure was time consuming and costly with little return. Doubtless, Professor Jung was very familiar with such an exercise, which appeared rather futile when one saw what was possible elsewhere. Perhaps he would comment on this aspect of what effort and urgency he considered desirable to achieve the relatively extraordinary improvements shown in Tables III and V, these were far beyond anything attainable by the engine designer.

Mr. Veitch emphasized the importance of reduced power resulting from improved propulsion efficiency when carrying out investigations. This might seem obvious, nevertheless the engine designer frequently requested for, say, 100 rev/min alternatively 85 rev/min design, power unchanged, hence no allowance was made for any possible cost reduction which the designer might be able to introduce for the lesser powered set of machinery; it was to be hoped that the relative fuel cost estimates were suitably adjusted.

In Table VI it would appear that examples showing a moisture of eight per cent applied to reheat installations and those of 12 per cent were for no reheat, if this assumption was correct, would not the maximum power of one exhaust be relatively greater than shown for reheat compared with non-reheat designs? Had allowance been made for the 15 per cent to 20 per cent reduced steam flow to exhaust resulting from the application of reheat?

Some three years ago, Mr. Veitch was strongly in favour of epicyclic gearing for the final reduction, enthusiasm, however, was not so great elsewhere and it was interesting to note that the ratio of weight and cost of such, as shown in Table VIII, confirmed the claims made for what was known as the "Paraplan" design. He had little doubt that it was only a question of time before epicyclic secondaries became standard practice for higher powered installations. Did Professor Jung agree? Perhaps he would care to make a few relevant comments.

DR. J. F. SHANNON wrote that in Table VIII the author had presented weights and prices of gearing suitable for 30 000 shp at 140 rev/min. For parallel shaft gearing he used through hardened gears at 125 and 95 K factor, and surface hardened epicyclic gearing at 300 K factor. This showed the arrangement with epicyclic gearing to great advantage.

However, the use of through hardened pinions and wheels in parallel shaft gearing had been outdated. Surface hardened shaved or ground pinions with through hardened wheels permitted a higher K factor to be used. It appeared from the author's comments under "Gearing Design" that he was going in this direction.

The author had, however, dismissed surface hardening of both the pinions and wheels on the grounds that the high K factors were limited by tooth bending stress, yet he had accepted epicyclic gearing at a K factor of 300.

It should be made known that the epicyclic gear was more vulnerable to bending stress than the equivalent external gear, because the planet teeth were subjected to reverse bending as against repeated bending on the pinions of the external gear.

Taking the examples in Table VIII, a comparison of a final reduction four-pinion surface hardened wheel with the final reduction four-planet epicyclic gear, assuming for each a gear ratio of 6, a K factor of 300, the same face width to diameter ratio of 1·5 and a bending stress factor for the planet 2/3 that for the external gear, for the reasons mentioned, would give an external main wheel of 84-in diameter and pinions of 14-in diameter with 3 Dp pitch as against an annulus of 78-in diameter and a sun pinion of 15·5-in diameter with 2·5 Dp pitch, for the same reliability against bending stress.

The through hardened wheel diameter at 95 K factor would be 126 in with pinions of 21-in diameter and 5 to 6 Dp pitch.

Clearly it was bending stress that determined the nominal K factor for surface hardened gears and, since external gears were better in this respect than epicyclic gears, surface hardened external gears would not be dismissed on this count.

Probably the greatest full scale testing of both through and surface hardened gears had been carried out by the NAVGRA firms for the M.o.D. (N). From this background it could be said that the greater factors of safety used for surface hardened gears in merchant service gave greater reliability than the present through hardened gears.

Turning again to Table VIII and noting that the diameter of the main wheel for the through hardened gear in the example was 126 in, as against 84 in for the surface hardened wheel and 78 in for the annulus, a considerable reduction in the weight and cost would occur if surface hardened gears were used.

The author had presented the propeller problem admirably and had shown various propeller arrangements for high loads. He had shown an inclination for contra-rotating co-axial propellers, but had pointed out the simpler shafting with other schemes.

The contra-rotating gear shown in Fig. 20, was at first glance the almost perfect gearing arrangement for the purpose, as it made use of the reaction members and could have a high gear ratio. It must be noted, however, that there was a matching problem between the two propellers and the two reaction members which, at best, resulted in reduced efficiency. This could be shown:

Gear: With both reaction members of the epicyclic gear rotating, the velocity equation is:

$$Ns = A/S.\ Na + (A+S)/S.\ Nc.$$

where Ns *and* S = the rotational speed and the diameter of the sun pinion, respectively,

Na and A = the rotational speed and the diameter of the annulus,

Nc and C = the rotational speed and the diameter of the centres of the carrier.

The power ratio is:

$$\frac{\text{carrier or aft propeller shaft power}}{\text{annulus or forward propeller shaft power}} = (A+S)/A.\ Nc/Na.$$

Propeller:

The diameter of the aft propeller must be reduced to come within the tip vortices from the forward propeller. A five per cent reduction was taken by Glover* in his detailed analysis, but it was understood that larger reductions might be necessary to suit seaway conditions. Calculation using data from Glover*, Sinclair/Emerson† and others, gave an estimate of the best matching efficiency of propellers and gear as follows for $Dc/Da = 0{\cdot}95$ and $0{\cdot}90$, and $A/S = 5$.

The power constant for the propellers for the best efficiency with 0·95 diameter ratio at equal speeds is taken from Glover as:

$$\frac{\text{Aft propeller power}}{\text{forward propeller power}} = \frac{DC^5 \cdot No^3}{3620\ 10^5} \Big/ \frac{Da^5 \cdot Na^3}{2920\ 10^5}$$

Results:

$\frac{Dc}{Da}$	$\frac{Na}{Na}$	$\frac{\text{shpc}}{\text{shp}a}$	$\frac{(Bc)^{\frac{1}{2}}}{(Ba)}$		$B^{\frac{1}{2}}$	$\frac{ND}{V}$	Efficiency, per cent	P/D
				forward propeller	4·9	200	63	0·74
0·95	1·38	1·66	1·34	after propeller	6·55	260	55	0·64
0·90	1·59	1·90	1·48	after propeller	7·25	285	52	0·61

Adjustments to the power constants for the propellers would now be required because the aft propeller gained less from the diminished power of the forward propeller. In the

* Glover, E. J. 1966–67. "Contra-rotating Propellers for High Speed Merchant Vessels." *Trans.N.E.C.I.E.S.*, Vol. 83, p. 75.

† Sinclair, L., and Emerson, A. 1968. "The Design and Development of Propellers for High Powered Merchant Vessels." *Trans.I.Mar.E.*, Vol. 80, p. 129.

limit, if equal power constants were used, the efficiencies would be 58 per cent and 55 per cent for 0·95 and 0·90 diameter ratios respectively. These efficiencies would be on the high side.

Thus there was a marked drop in efficiency due to the increased rev/min and smaller diameter of the after propeller. The distribution of power was poor. It must be concluded that this seemingly ideal gear did not allow the maximum efficiency to be obtained with contra-rotating co-axial propellers.

This weakness is inherent also in the scheme shown in Fig. 19, where the aft propeller is driven by the carriers of the first reduction epicyclic gears. This defect can be overcome by driving the aft propeller by the annulii of the two first reduction epicyclic gears.

Contra-rotating gears without epicyclics but with coupled propellers giving equal or any desired speed ratio can give the best efficiency possible with contra-rotating propellers.

MR. C. J. CHARLES wrote that the AP series arrangement of gearing, in which a highly-loaded and complex unit in the form of the primary reduction epicyclic gear, was combined with conventional parallel shaft gearing operating at conservative loadings, had proved to be a sound commercial proposition. It would be interesting to see whether the logical advance, in the form of the final reduction epicyclic, whilst having great technical attraction, would be able to offer the same degree of assurance that the present day shipowner demanded.

Using parallel shaft gearing throughout, with hardened pinions and, in some instances, hardened wheels, it was possible to meet the latest and projected future requirements for tanker and container ship propulsion, without departing from normal standards of accessibility for inspection, whilst at the same time offering reductions on weight and cost comparable with those offered by use of the final reduction epicyclic.

For example Fig. 33 showed a possible machinery arrangement for a twin-screw tanker of say 350 000 dwt using single-cylinder turbines driving through triple-reduction gearboxes. The output power/shaft speed is 22 000 hp and 50 rev/min. The propellers would weigh just over 60 tons each and have a diameter of about 35 ft.

The selection of a very low output speed, with its consequential benefit in terms of propulsive efficiency, had generally been prohibited by a belief that the gearing would be impractical; further, that for a twin-screw ship, propeller diameter would be restricted by the hull form and, in consequence, relatively high revolutions would be inevitable.

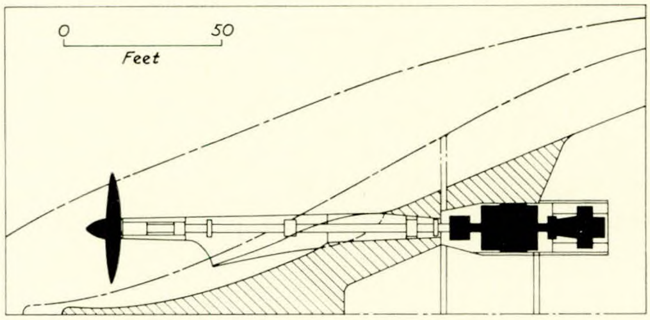

FIG. 33—*Arrangement of single-cylinder turbine machinery in a twin-screw tanker*

Neither of these beliefs was valid when one considered the possibilities that were presented by the configuration of the next generation of tankers, combined with the gearing arrangements now available.

Turning to the other outstanding application for steam turbine machinery, the power requirements for future container vessels would be such as to require division of power between several pinions, perhaps as many as six, meshing with the main wheel. This was not difficult to achieve with either a single or twin turbine input. Fig. 34 showed a gearing arrangement suitable for transmission of 40 000 shp at 140 rev/min for a single-cylinder turbine. The input pinion, which was hardened, meshed with two hardened wheels. These wheels transmitted one quarter of the input power via idler gears to two through hardened wheels. All four primary wheels were connected to hardened secondary pinions meshing with a through hardened main wheel.

The weight of such a gear-box would be approximately 70 tons, which compared very favourably with the weight quoted by Professor Jung for the scheme using a secondary epicyclic gear transmitting 30 000 hp at 140 rev/min (Table VIII). The corresponding gear-box for cross-compound turbines would be slightly larger because of the higher overall ratio to be accommodated, but it would not require the use of hardened primary wheels.

An interesting feature of the new standard range of gear-boxes currently being offered by Mr. Charles' company was the elimination of the conventional horizontal joint through the centreline of the main wheel. The arrangement enabled the adoption of high primary reduction ratios with a consequential reduction in the secondary reduction ratio and hence in the size of the main wheel.

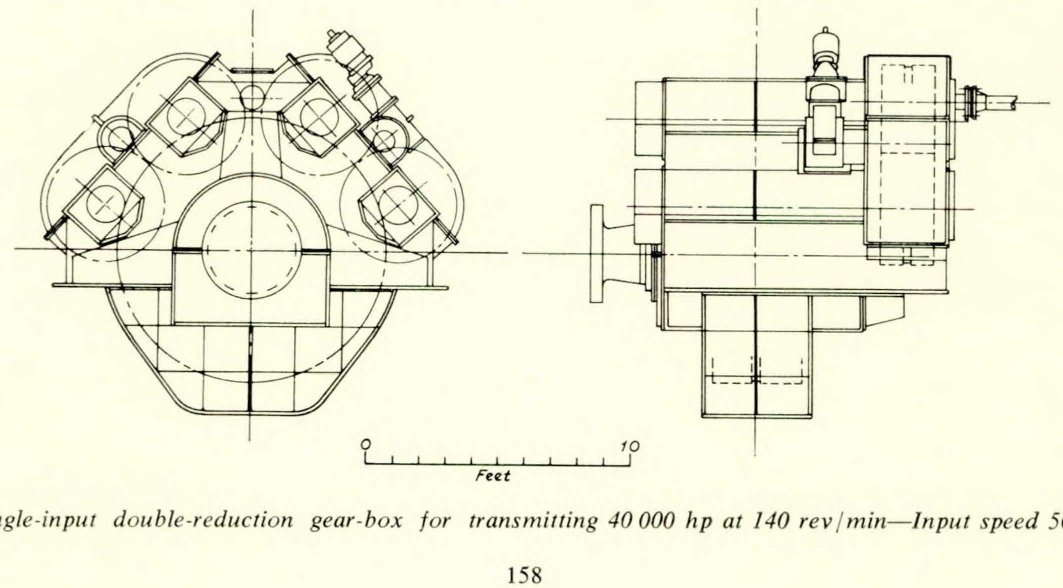

FIG. 34—*Single-input double-reduction gear-box for transmitting 40 000 hp at 140 rev/min—Input speed 5000 rev/min*

Author's Reply

Professor Jung, replying to the discussion, thanked all those who had taken part. To Mr. Falconer he said he understood the situation of the shipowner today—he wanted simple engines and with the present crews, especially on ships which spent a long time at sea, availability was more important than anything else. As fuel price was decreasing and the cost of manhours increasing, high efficiency must be achieved without complications and reduced safety margins.

Mr. Falconer was also right that shipowners and engine builders had to work together to achieve standardized designs of turbines, boilers and auxiliaries. Optimization of the whole system would be necessary in the future. Main engine and auxiliary manufacturers must have service facilities available in all the ports over the world. The shipowner should be able to call on the telephone and get an expert down to deal with any problem. Electronic equipment should not be forgotten. After trials it had happened that it took three months before regulating devices, switch controls, etc. worked. The electronics were marvellous on paper but awful when not working. The author had found that several ships were running without automatic regulating devices because the crew said they could not rely on all those gadgets. So hard work was needed from both the engineers and the shipowners to improve the situation. To try to save fuel and to introduce a lot of complications was asking for trouble.

Table V referred purely to gearing costs and no turbine costs were included.

Replying to Mr. Brown the author said that it was right that higher K factors could be used. Nitrided pinions and higher stresses would come, but he pointed out that he had been called in to see gears which, although they were not highly stressed, still had to be removed. The ship structure was moving, it was not just a rigid machine shop foundation.

One got dirt in oil; one got lots of things in reality. So if one ran with a 500 K factor, the margin was much less than with 300 K. It was expensive to use K factors of 95, but the author said he preferred to sell a somewhat expensive unit and be called old-fashioned, but deliver quality and precision manufacture with some good margins. He thought there was 100 per cent margin in the K factor now used for turbine gears if they were in excellent condition, but this was not always the case. If people saw the ships coming in after several years' service, they would realize why he was sometimes very thankful for the built-in margins in designs. He thought his company could go further in the development of epicyclic gears, especially in the final stage epicyclics.

Mr. Hick's flexible planetary pinion design for epicyclic gearing might be of great interest if it could also be made sufficiently flexible for high torques. For lower torques he understood the design had already been tried. In the low torque field the epicyclic gear had difficulty in competing where cost was the main factor, as in merchant ships.

Higher K factors would also be used at sea in the future for big parallel gears. Nevertheless severe gear failure had happened recently with K factors below 200 and hardened and ground teeth, even with manufacturers of international reputation. Naval duty and merchant marine service were two completely different fields in most respects.

Mr. Young was right in saying that the author was on the conservative side in the comparison of single-cylinder turbines with cross-compound units. 6000 rev/min was not a limiting speed for the gearing or for the turbine, but gave at 10 000 shp, the standard 1080 ft/s tip speed used for the cross-compound design. Epicyclic gearing of the star type had no speed limits. The 140 rev/min propeller speed in Table VIII was chosen as this was a standard container ship speed. With triple reduction on the H.P. side, any practical propeller rev/min could be designed with normal speeds, stresses and dimensions; 50 rev/min was hardly practical in today's ships.

With regard to the gearing for overlapping propellers (Fig. 21), c.p. propellers could be used as well as f.p. propellers.

The torque distribution between the two overlapping propellers would be less unfavourable than with a normal twin-screw arrangement. On cruisers, torque differences up to 30 per cent had been measured at maximum rudder. The gear had to take the loads as any normal gear.

In answer to Mr. Hutchison, the author said he had taken up the optimum speeds of tankers, a very interesting subject. Professor Jung had calculated, many years ago, the optimum speed which gave the lowest transport cost for a 10 000 dwt cargo ship. He was in charge of ship design in 1945 and his company were designing 10 000 ton cargo ships capable of 18–20 knots. The speed giving minimum transport cost was 9 knots. He recently revised his old figures for a 250 000 dwt tanker and came up with the result of 14–15 knots as being the optimum for minimum transport cost. However, the optimum was not very pronounced, allowing variations of ± 2 knots with only small increases in transport cost.

Shipowners had always gained by going to higher speeds than the speed giving minimum transport costs. At times of high freight rates, the speeds had, however, been pressed too far over the minimum transport cost value. The speed giving optimum profit was higher than the minimum transport cost value. However, laws of similarity predicted optimum speeds increasing with size even if the power of the linear dimension was astonishingly low.

Referring to Fig. 29, the author emphasised the simplicity, the good engine efficiency and the low price of the single-screw tanker. With regard to the higher engine cost, the optimum speed of the twin-screw ship was lower than for the single-screw ship and it should therefore require less effective power for minimum transport cost.

The gearing in Fig. 21 had two independent gears as far as erection and lining up was concerned: one for port, including the H.P. gearing, and one for starboard up to the direction changing gear. The long flexible twin shaft between the gear casings had to take care of differences in centres and alignment angles.

As 25 per cent of the total power or 50 per cent of the power of one shaft passed over each of the two H.P. pinion flanks, torque variations of the propellers had to exceed this limit before chattering would occur. If one L.P. turbine should be disconnected, power had to be reduced to about 50 per cent, so reducing tooth load. Seaway and rudder interaction should not be much worse than with twin screw arrangements.

Mr. Hutchison's proposal to give the astern turbine more stages, to get rid of boiler complications caused by the high heat release needed for full astern, was right and sound if the boiler makers could not solve these problems. The author had had, however, excellent experience with astern turbines with smaller separate nozzles and valves for desuperheated steam and full astern power nozzles for superheated steam.

To Mr. Brownlie the author stressed the fact that the comparisons between the single and the compound machine were based on today's somewhat conservative tip speeds of 1080 ft/s. With higher tip speeds, the single-cylinder turbine would come out better, giving cost advantages at somewhat over the 20 000 shp limit given by the author. However, with the designs available and even with final epicyclic gears, little gain had been found by the author's company. From the security standpoint, single-cylinder turbines were mainly of interest for twin-screw ships of not too great powers, but here one would meet fairly heavy competition from the geared medium speed Diesel.

The author congratulated Mr. Jones on having been able to get orders for several million horsepower of first reduction epicyclics and to the excellent record in performance and in quality. Mr. Jones's company was one of the exceptions in British industry; they had delivered promptly according to the dates promised.

The author and Mr. Jones had both thought that after the enormous Suez boom, resulting in orders of two million horsepower, the expansion would slow down, but it was still going on. New container ships were planned with powers up to 2 × 60 000 shp per ship. There would be lots of epicyclic gearing developments for their two companies to do in the coming years, especially with regard to final gears.

Professor Jung was completely in agreement with Mr. Preston that before taking all extra costs and complications with twin shafts, the single conventional screw should be optimized. The author thought that even ships up to 400 000 dwt could with advantage have single screws, as Tokyo Tankers' new ship indicated. Powers up to 50 000 shp with fairly low rev/min were conventional solutions but still of interest.

The author was not in favour of using extreme admission steam data but thought that with experience, stationary trends could be carefully followed, with due regard to economic restraints. For greater powers, reheat would pay in future plants. The silica problem with high steam pressures was definitely not yet solved in practice, as Mr. Preston had pointed out, and gave the reason for being careful with too rapid an increase of steam pressure.

Mr. Victory had made the important point that tankers had to be stopped in a reasonable time and distance. It had been found in the early Stal-Laval designs that more astern power was available than could be utilized by the propeller: they had reduced the astern power more and more towards the internationally accepted 80 per cent/50 per cent value. As the astern torque at constant maximum steam flow was linear with an approximate factor:

$$(35 \times n_{\text{max-astern}}\ \text{rev/min})$$

the torque would be 75 per cent at 65 per cent rev/min and 81 per cent at 50 per cent rev/min and 105 per cent at 0 per cent rev/min. The 75 per cent/65 per cent specification was more reasonable as a characteristic value. Usually the propeller was spinning at low ahead speeds so it was a very complicated problem to optimize the astern manoeuvre.

He proposed a connexion to the half boiler for increase of the astern steam flow because in the first Esso ships with de Laval turbines the temperature was 950°F (512°C) and at that time they were afraid to admit the superheated astern steam directly into the cold astern turbine. The astern turbine was provided with two valves, one for saturated steam direct from the desuperheater and one for superheated steam. This arrangement with two different manoeuvring valves had worked without any difficulty for many years. The author meant that the half boiler should always be on steam with heat from the main boiler. It should always have pressure and be ready to go on line. With automatic devices that was not too complicated.

However, there was still the great problem of stopping supertankers. If one compared with a car which had a weight of one ton, the power available for manoeuvring was one-tenth of one horsepower. If one tried to stop a car without brakes with such a small motor, one would understand the problem. Supertankers must be limited in speed in all crowded surroundings as speed changes were so slow. The c.p. propeller was one solution. The stopping distance was reduced to about 60 per cent with a c.p. propeller; 20 000 horsepower c.p. propellers were in service.

On the matter of gas turbines Professor Jung said that the gas turbine had a long way to go before it got into the tanker business. For the transportation of gas, the situation was favourable. For some container ships the gas turbine could also be competitive. The Americans had some experience with *Callaghan* and although it was not a very economic solution it was doing very well. If one went in for high powers for transports where speed paid, gas turbines would definitely be competitive.

Mr. Jakeman had discussed the cavitation problem with overlapping propellers. It was not possible to place the blades of the two propellers so that the eddies would not impinge upon the aft propeller's blades at some power. The tip cavitation eddy path from a blade through the water changed with the speed. It was however possible to get the propeller tolerably free from eddy at full power.

Overlaps at 0·1 to 0·32 D were used with the Munk-Prohaska model tests. The best combination was outward turning screws with 0·23 D overlap.

The working group in Sweden had made half-scale tests with contra-rotating bearings without any sleeves; grooves had been made in the outer bearing only to generate a rotating oil film. Tests up to 300 lb/in^2 showed normal behaviour of the bearing. Professor Jung thought that the floating sleeves were an unnecessary complication.

Professor Jung thought final stage epicyclics would come soon. He believed that Mr. Jones might know more about the inspection problem. Inspection should be possible without taking the whole planet gear out. It should be possible to inspect it visually and with special equipment. Mr. Jones was asked to give his opinion on the matter.

Mr. Jones said, at this point, that his company would not offer anything to any ship builder until they had demonstrated very practically exactly how one would examine it and deal with it *in situ*.

With regard to torque distribution, the arrangement for contra-rotating propellers in Fig. 19 could be designed for any torque value that the propeller designers asked for by using different second reduction ratios. The opinion among propeller designers seemed to be that under all circumstances the aft propeller should not have more torque than the fore propeller to achieve the optimum efficiency.

The author had found in principle the Paraplan layout to be ideal. However, big extrapolations were not liked by shipowners who had to take the risks for ships costing £5–10 million.

Dr. Shannon's contribution with regard to contra-rotating gearing shown in Fig. 20 was of great interest. The author fully agreed with the conclusion that an arrangement with a torque distribution of 45–50 per cent on the aft propeller and optimized propeller speed should give a better performance. The arrangement in Fig. 19 as well as the parallel gearing could be designed for any torque and speed distribution. Dr. Shannon's and his company's contribution to the surface hardened parallel gearing were very well known and would no doubt also be introduced into the merchant marine. However, with the torques and powers asked for, too rapid increases of dimensions with today's manufacturing methods for hard gearing should be avoided.

Mr. Veitch had rightly pointed out that the exhausts for the reheat units (with 8 per cent moisture content, Table VI) could be used at higher powers than non-reheat exhausts because of lower steam flow from the boiler and the greater amount of bled steam. The figures for the uprated 3000 rev/min marine reheat blade gave a power increase of 50 per cent with about 30 per cent larger exhaust area.

As already mentioned, the higher the torque, the more competitive was the epicyclic design. It would only be a question of time and experience before the final stage epicyclic went to sea. The boiler problems taken up by Mr. Hodgkin were of great importance. The author very much regretted the mistake made when Fig. 22 was taken from the journal *Combustion Engineering* and then incorrectly attributed to an American competitor of Mr. Hodgkin's company. The author would very much like to see a development towards a simple reliable reheat boiler design with low costs as proposed. As powers would soon exceed 50 000 shp on many ships, it was probable that the next design generation of marine steam plants would have reheat.

Mr. Charles was as much in favour of the hardened and ground pinions and bull wheels as Dr. Shannon. Looking at the dimensions given for shaft horsepower: rev/min values of 440, it was obvious that great weight savings could be achieved in comparison with the conservative AP design. The costs for the different designs would, however, show much smaller differences, if any.

In the final epicyclics, high K factors and nitrided teeth could be used in the smaller sun wheels and planetary wheels, and normal conservative K factors for the internal through-hardened annular rings, which should give favourable cost figures.

The gear design described (Fig. 34) without a horizontal centre line joint, was of great interest. The contributions from the British gear producers showed fresh approaches and absolute freedom from the conservatism which had often been a marked characteristic of older designs.

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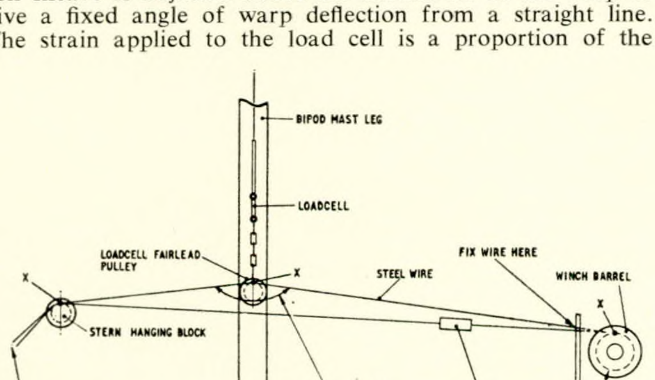
* Patent Specification.

Warp Load Meters

The warp load meter was first used aboard a British stern trawler, *Junella*, in late 1963. This was fitted by the Industrial Development Unit of the White Fish Authority and its purpose was to study the performance requirements of a trawl winch by means of measurements of warp tension.

The use of such a device to monitor the warp tension to give warning of fasteners continuously and to provide additional information about the warp strain in various situations of trawling, was thought to be of commercial benefit; subsequent trials proved this to be the case.

The warp load measuring system consists of two load cells, one suspended each side of the mizzen mast with a sheave attached, through which the warp passes from the winch drum to the towing sheave. The height of the load cell sheave is adjusted above the normal lie of the warp to give a fixed angle of warp deflection from a straight line. The strain applied to the load cell is a proportion of the tension in the warp, and changes in tension produce changes in the electrical resistance strain gauge in the load cell. The resistance in the load cell is continuously monitored in the wheel house, the result being displayed on a meter which is calibrated directly in tons of tension in the warp.



Hanging block with load cell and run of warp on stern trawler

When stern trawlers were first discussed, one of the many advantages put forward for them as compared with side trawlers was improved warp life. On a stern trawler, each warp has to pass through only one running sheave, and that at an obtuse angle, before entering the water, compared with the several sheaves at acute angles and the towing block aft through which the side trawler's warps pass. The warp wear brought about by hauling and shooting was therefore much reduced on the stern trawler.

Once the trawl is shot and the gear is settled, both warp tension meters will read four to five tons in fine fishing weather. A control is adjusted to read about two tons above the usual towing pull. Should the trawl snag, and the tension in each warp reach about seven tons, a light will flash indicating an overload in one or both warps. A further optional refinement is the fitting of a siren which operates when excessive tension is measured, immediately drawing one's attention to the meters.—*Drever, C. and Ellis, G., World Fishing, November 1968, Vol. 17, pp. 34–35.*

Stern Freezer Trawler with Unmanned Engine Room

Built by Ferguson Brothers (Port Glasgow) Ltd., a member of the Scott-Lithgow Group, *St. Jasper* is under construction to the order of Thomas Hamling and Co. Ltd., Hull.

Principal particulars are:

Length, o.a.	231 ft 0 in
Breadth, moulded	39 ft 0 in
Depth, moulded	24 ft 6 in
Deadweight	500 tons
Trials speed	14 knots

Propulsive power for *St. Jasper* is provided by a British Polar six-cylinder two-stroke M66T-type engine capable of developing 2400 bhp at 250 rev/min maximum continuous output—2000 bhp at 225 rev/min normal rating. It is a uni-directional trunk-piston engine directly coupled to a stainless steel c.p. propeller of Liaaen manufacture.

Since the British deep water fishing fleet has been suffering from a continuing decline in recruitment of both fishermen and engineers, the White Fish Authority carried out a design study for a manpower-saving automation system in conjunction with the English Electric Co.'s power and marine division. The study was undertaken by English Electric on a specification supplied by the Industrial Development Unit of the W.F.A., on the 1000 g.t. stern freezer trawler *Criscilla*. From this study a design for automated engine room machinery in a similar sized vessel was produced and this is being incorporated in *St. Jasper*.

Electrical power in this new vessel is provided by a five-cylinder, 625-bhp British Polar SF 15RS engine driving a Laurence Scott and Electromotors alternator at 750 rev/min which gives an output of 468 kW at 440 V, 50 cyc. A smaller set comprises a three-cylinder, 415-bhp British Polar SF 13RS engine driving a 750 rev/min alternator capable of producing 280 kW. For harbour duties a six-cylinder Gardner LX-type engine developing 80 bhp drives a 55 kVA alternator at 1000 rev/min.

All the main engine and auxiliary pumps are electrically driven except for the main hydraulic pump which is driven through a clutch from the free end of the main engine. Power from this pump is taken to the trawl winch and windlass, both of which are manufactured by A/S Hydraulik Brattvaag. An emergency electrically driven hydraulic pump of 80 hp is also provided.

The main engine is designed to burn residual fuel of up to 400 sec Redwood No. 1.

The machinery supervisory system proposed will include extensive alarm scanning capable of supplying data on what has happened, pinpointing the cause of the fault and indicating the course of action to be taken. Operation of all engine room machinery will also be centralized in the control room as mentioned earlier.

At the centre of the information system will be a small computer designed to receive all data from the plant which will then be sorted out and the relevant action taken to inform the staff by alarms and recording of any fault as it occurs. For control purposes, separate electronic circuits are to be fitted, each covering a particular automatic operation or sequence control.

The main information devices will comprise a mimic display panel of the main systems of the plant fitted with a warning light for each monitored point. Print-out of all recordings will also be provided for with audible annunciation in suitable areas of the ship.—*Motor Ship, December 1968, Vol. 49, pp. 451–452.*

Experimental Evaluation of Heat Transfer Using Thermal Oil as Heat Transfer Medium

Little is known about the heat losses from double-bottom tanks. To obtain the necessary information, a test was carried out on such a tank of an 8000-dwt freighter. The tanks of this ship were heated by means of thermal oil; the tank was loaded with low viscosity fuel oil. Instrumentation was fitted to measure the heat flow through the tank areas, mean oil temperature, temperature of heating coil and temperatures in adjacent spaces. Furthermore the heat supplied by means of the thermal oil was determined.

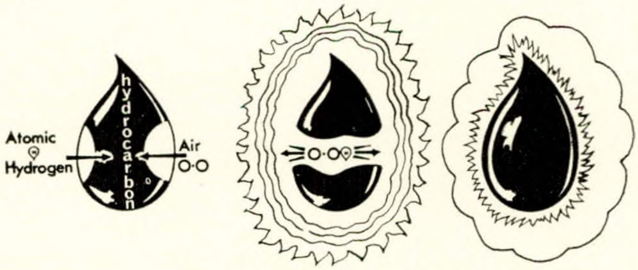
The test yielded much valuable information, including heat-transfer coefficients for all tank areas and for the heating coil. The heat loss from the tank bottom proved to be very high. A simplified method for calculating the heat losses from the frames made it possible to obtain a tallying heat balance. Based on the information obtained, the heat loss for heavy fuel oil was calculated.

For the whole of the measuring period (48 h), total heat losses from the tank surface were computed. As far as surfaces to which no heat-flow meters had been fitted, the heat flow was obtained from readings of a meter fixed to a comparable surface and the measured temperatures on both sides of the wall obtained. Actually, these surfaces are of minor importance. Heat flow through the part of the bilge that had an inclination less than 45° to the vertical, was taken as the average of the values for side shell and bottom.—*van der Heeden, D. J., Shipping World and Shipbuilder, October 1968, Vol. 161, pp. 1635–1637; November 1968, pp. 1779–1785.*

Diesel Engine Combustion Stabilizer

During the past few years a considerable amount of research has been carried out in developing the Houseman-Vapo combustion principle to overcome the problems associated with the use of heavy fuels in Diesel engines.

It is known that certain iron compounds added to the fuel improve combustion and modify the melting temperatures of ash residuals of combustion, but in the past these iron compounds were extremely expensive and furthermore were toxic and therefore unsuitable and uneconomical for general use. A new non-toxic low cost liquid which contains an oil-soluble organic iron compound has been developed, the significance of which can be illustrated quite simply by an explanation of auto-ignition and combustion in a Diesel engine.



Combustion and its control

If compression ignition is to be adopted, as distinct from spark ignition, petroleum gases alone cannot be used as a fuel; small injections of liquid fuel (gas oil) are needed for the ignition and combustion of petroleum gas in the combustion chamber. Petroleum gases have only six carbon atoms or fewer in their molecules and will not crack under Diesel engine conditions, but the small amount of gas oil with larger carbon chains will crack momentarily and take up molecular oxygen in the air charge to form hydrocarbon peroxides which, being unstable, explode violently when heated and compressed, so setting off the ignition necessary for the combustion stroke. In other words, it is the hydrocarbon peroxides which initiate and govern the nature and velocity of the combustion stroke in any Diesel engine.

Unfortunately, residual heavy fuel oil is a mixture of liquid and solid hydrocarbons (asphaltenes) with varying chain lengths, and to ensure complete combustion or oxidation throughout, any pre-ignition of the light shorter chain hydrocarbons should be prevented before the heavy long chain hydrocarbons have had time to spread out and fill the combustion chamber, and to reach their proper ignition temperature. Partial combustion will inevitably mean overheating in local zones, for instance piston crowns, and the presence of unburnt carbon particles which tend to deposit in the exhaust system or find their way out to the atmosphere as smoke. Pre-ignition is also considered contributory, if not the major cause of piston ring failure due to metal fatigue.

Intensive research has shown that modification of the hydrocarbon peroxide formation in combustion can be achieved by applying the Haber-Weiss mechanism*. This suggests that by introducing an organic iron compound with the fuel injections, the hydrocarbon peroxides are converted to hydrocarbon alcohols. By this means the peroxide explosive tendencies are delayed until controlled ignition takes place due to the glowing atomic iron vapour distributed throughout the combustion chamber. At the compression temperature the iron vapour will glow spontaneously, irrespective of the air present. This reaction has a further important effect in taking up the atomic oxygen and by this means the formation of sulphur trioxide is reduced, as the sulphur dioxide compounds of combustion require atomic oxygen to form sulphur trioxide which, in the lower temperatures of the exhaust system, condenses to form corrosive sulphuric acid.

The diagram illustrates combustion and its control. On the left, a petroleum molecule is visualized as an oil drop; when the atmospheric oxygen, O-O, and atomic hydrogen, H, reach each other and combine, an explosion results. This starts a chain reaction of cracking, with organic peroxide formations and subsequent explosion in all directions. In the centre is shown what happens in uncontrolled combustion; atmospheric oxygen is pushed away from the petroleum particle by explosion pressure waves at 700 m/sec. Carbon and hydrogen in excess form unwanted endothermic reactions and atomic oxygen, O. On the right, the Houseman-Vapo process has been adopted and combustion is controlled by an iron reaction. No explosion occurs as the O-O-H organic peroxide is decomposed spontaneously to water and molecular oxygen by one electron transfer, Fe′ to Fe. No atomic oxygen is present in the flame zone but atmospheric oxygen is present in excess, close to the petroleum particle. Waves emanating from combustion are soft and adiabetic and travel at 20 to 30 m/sec.—*Marine Engineer and Naval Architect, December 1968, Vol. 91, pp. 517.*

Large Swedish-built Oil/Ore Carrier

An ore/oil carrier of 106 600 dwt has been built by Götaverken for the Turnbull Scott Shipping Co. Ltd., London, and given the name *Flowergate*.

A sister ship is also under construction at the Arendal yard for the Grängesberg Co., Stockholm.

Principal particulars are:

Length, o.a.	830 ft $0\frac{5}{8}$ in
Length, b.p.	807 ft $1\frac{1}{16}$ in
Breadth, moulded	131 ft $2\frac{13}{16}$ in
Depth, moulded	67 ft 9 in
Draught, summer	49 ft 9 in
Total cargo capacity, oil ...	4 605 000 ft^3
ore ...	1 956 000 ft^3
Total clean water ballast capacity	585 000 ft^3
Deadweight	106 600 tons

Oil cargo is handled by three vertical turbine-driven pumps of centrifugal type, each rated at 2500 ton/h (water). These pumps are associated with a common evacuator installation and individual gas separators and a condenser for exhaust steam. All this equipment is in the pump room, which is amidships. A cargo control room in which the remote control panels for pumps and valves are installed is located above the pump room. Tank cleaning is carried out by means of Gun-Clean apparatus, fixed permanently in the case of wing tanks and mounted in port as required for the centreline compartments.

The propelling machinery in *Flowergate* comprises an eight-cylinder Diesel engine of Götaverken large-bore design developing 17 600 bhp at 115 rev/min, corresponding to a loaded speed of 15·1 knots. The engine, which has a bore of 850 mm and a stroke of 1700 mm, is arranged to run on fuel having a viscosity of 3500 sec Redwood No. 1 at 100°F.

The machinery is highly automated, including bridge control of engine and KaMeWa propeller and is intended for operation with a partly unmanned engine room. The extensive instrumentation and monitoring arrangements for the main engine and auxiliaries, the main engine control console and the main switchboard are incorporated in a soundproofed control room at upper platform level. The equipment includes an order logger.—*Shipping World and Shipbuilder, February 1969, Vol. 162, pp. 309–310.*

Bladeless Propeller

The bladeless propeller described is an interesting device in which the energy interchange takes place between a primary and secondary fluid. In this case the primary fluid is pumped through a central boss which is caused to rotate by the emission of this fluid through specially orientated slots. The swirling downstream motion of the primary then creates the necessary pressure forces on the surrounding secondary fluid to cause the two fluids to flow through the interior of the surrounding duct. This propeller can be a single-phase device, i.e. water as primary and secondary fluids, or as a two-phase device with a gaseous primary and water secondary fluid. A number of experiments have been performed on this device and the author concludes that its use in marine application is very promising.—*Fao, J. V., Seventh Symposium on Naval Hydrodynamics, 25th–30th August 1968; Shipping World and Shipbuilder, December 1968, Vol. 161, p. 1915.*

Increasing the Application of Ship Research

During the last few years substantial changes have taken place in the organization of work being undertaken by the British Ship Research Association. These changes are all intended to increase the application of results in the shipbuilding industry.

Historically, these changes began with the merger in 1962 between the old British Shipbuilding Research Association and the old PAMETRADA which placed the considerable facilities of the Wallsend research station at the disposal of the new B.S.R.A.

During this period the decision was taken to cease all direct research and development on main propulsion engines for ships, whether turbine or Diesel, leaving such work to be done by the manufacturers concerned. This decision, and the bringing to completion of a large number of small research projects, has made it possible to concentrate effort on fewer items of research and to inter-relate these in a helpful way which is designed to ensure that the shorter-term problems of the industry receive priority and that results are made available quickly to the industry.

Changes have also been made to the organizational structure of the association and research is now prosecuted by four main divisions having the titles of: naval architecture, production, marine systems and computer.

To improve the quality of the results needed in the assessment of ship performance, new types of recording instrumentation are being developed and installed on board ship. EMDAS, or Electronic Marine Data Acquisition System, which is a system recently developed to record data in digital form from transducers, is to be installed in three ships. The first will be a large tanker, *British Argosy*, while the second will be a fast cargo liner.

Propeller-induced vibration can give rise to considerable trouble in the operation of ships of very full hull form and an intensive effort has been made to overcome the problem on a project basis with co-operation from N.P.L. and the University of Newcastle upon Tyne.

A B.S.R.A. working group set up to survey methods of ship structural design by computer has recommended the full three-dimensional analysis of the structure of a large tanker using finite element computer methods.—*Hurst, R., Shipping World and Shipbuilder, February 1969, Vol. 162, pp. 123–124.*

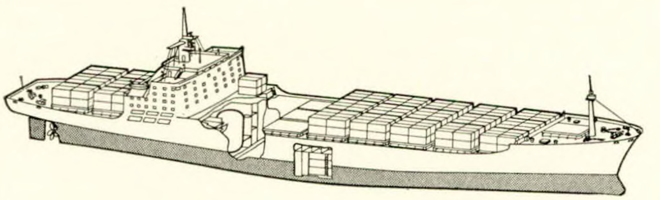
* Haber, F. and Weiss, P., Proc. Roy. Soc., 1939, Vol. 147, p. 233.

Design Study for Nuclear-powered Container Ship

Vickers Ltd., in conjunction with the United Kingdom Atomic Energy Authority has completed design studies on the application of nuclear power to all these types of vessel, but, although such studies provide valuable data, it was not, until recently, possible to show that nuclear propulsion was an economic concept.

The study summarized in this account is based on a refrigerated container vessel for the United Kingdom–New Zealand trade route via the Panama Canal.

The conventional vessel is an oil-fired steam turbine refrigerated container ship of dimensions similar to those building today. The machinery and accommodation is aft, 1100 containers can be carried, 880 of these are refrigerated and the remaining 220 are ordinary containers carried on deck.



Proposed nuclear container ship

Matched against this conventional vessel is a nuclear-powered container ship. The following parameters are constant for both ships:

Ship speed	24 knots
Shp	40 000
Displacement	40 000 tons

Principal particulars are:

	Conventional	Nuclear
Length, o.a.	755 ft 0 in	790 ft 0 in
Length, b.p.	710 ft 0 in	740 ft 0 in
Beam, moulded ...	105 ft 0 in	105 ft 0 in
Depth, moulded ...	60 ft 0 in	60 ft 0 in
Load draught B.O.K.	32 ft 0 in	31 ft 3 in
Service speed	24	24
Round trips/year ...	6·35	6·59
Refrigerated containers at 14·3 tons gross	880	900
Ordinary containers on deck at 18·25 tons gross	220	380
Total containers ...	1100	1280
Crew numbers ...	40	44
Oil fuel	6700 tons	100 tons
Total cargo dwt ...	16 600 tons	19 805 tons
Load displacement ...	40 000 tons	40 000 tons

For use on merchant ships the pressurized water type reactor recommends itself as it is extremely stable.

The design of pressurized water reactor that has now evolved is known as the BPWR (burnable poison water reactor). It is of the integral type and has been chosen because of its simplicity and low capital cost.

When the main propulsion machinery is developing 40 000 shp the present reactor core, which weighs about seven tons, is required to produce 113 MW of heat (385 m Btu/h), i.e., an overall thermal efficiency of 28 per cent for the cycle which has been adopted. The core has a life of over 1200 days at full power, which is sufficient to meet the most demanding service such as will be required by high-speed container ships.

The weight of the nuclear steam generating unit complete with fuel for 1200 full power days of operation is about 1000 tons including shielding.

The nuclear steam-generating unit, i.e., the integral reactor and boiler, is situated forward of the machinery spaces in its own compartment which extends from the double bottom through the main deck to the deckhouse above, in the top of which is an access hatch for refuelling. A cofferdam isolates the reactor compartment from the forward cargo spaces. Collision barriers are situated between the reactor compartment and the ship's side.

Turbine stop valve steam conditions are 585 lb/in^2 at 570°F. The machinery consists of a two-cylinder cross-compound steam turbine driving a single, fixed pitch propeller via double helical, single tandem, double-reduction, articulated gearing. The main turbines have a maximum service shp of 40 000. The distribution of power in the turbine is approximately 50 per cent in each of the H.P. and L.P. sections. In the L.P. cylinder, due to the low steam conditions, there is an integral centrifugal-type water separator which maintains the steam wetness at a tolerable level in the later stages of the L.P. turbine. At the forward end of the L.P. turbine there is the astern turbine which develops 80 per cent of the maximum ahead torque at 50 per cent of the maximum ahead rev/min.—*Gaunt, I. A. B. and Wilkinson, G. R., Manchester Association of Engineers Paper; Shipbuilding and Shipping Record, 29th November 1968, Vol 112, pp. 707–709.*

Low-volatile Refined Oil Carriers

Built at the Kure yard of I.H.I., *Esso Bangkok*—more recently joined in service by her sister ship *Esso Bombay*—is claimed to be the first bulk carrier ever designed with 27 cargo oil tanks separated into four groups which allow the ship to carry simultaneously a number of low-volatile cargoes such as benzine, xylene, toluene, etc., in addition to normal refined products.

With an engine room planned and equipped for periodically unmanned operation, I.H.I.-Sulzer-engined *Esso Bangkok* and her sister ships are of the well deck design with bridge, all accommodation and machinery aft. This class has the Esso bulbous bow, a cruiser stern and has the entire external surfaces of the hull, all cargo tanks, decks and the interior and exterior of all pipes and valves coated with Humble Rust-Ban.

Principal particulars of *Esso Bangkok* are:

Classification ABS +A1 (E) "Oil Carrier" and +AMS

Gross register	12 994·19 tons
Deadweight capacity	21 056 tons
Length, o.a.	170·80 m
Length, b.p.	161·00 m
Breadth, moulded	23·47 m
Depth, moulded	12·12 m
Draught	9·10 m
Cargo oil capacity	28 701 m^3
Fuel tank capacity	1336 m^3
Fresh water capacity	167 m^3
Ballast water capacity	1571 m^3
Speed, max.	15·3 knots
service	14·9 knots
Main cargo pumps	$4 \times 682\ m^3/h$
Stripping pump	$1 \times 91\ m^3/h$
Complement	31 men

The nine main cargo tanks are subdivided to form 27 tanks, which are arranged in four groups each served by its own cargo pump, but so arranged to allow total flexibility of pump selection. Each group is fitted with a drop line for rapid loading. Each main cargo pump is fitted with a "Vac-strip" system, which automatically strips the cargo oil by the vacuum pump and main oil pump. As a result, discharge time is said to be reduced. A small standby conventional stripping pump is also fitted.

The I.H.I.-Sulzer six-cylinder RD68 engine has a normal continuous rating of 6480 bhp at 130 rev/min and a maximum continuous output of 7200 bhp at 135 rev/min, at which the m.i.p. is 9·8 kg/cm^2.

Control of the engine can be from the bridge by a single lever for direction and speed from the side of the engine or from the acoustically insulated, air-conditioned control room within the engine room at camshaft level. In this compartment, the main console is arranged in 18 sections, covering manoeuvring, monitoring, Diesel generators, cooling water circuits, lubricating oil circuits, etc., each equipped with the necessary gauges, meters and visual and audible alarms. Such alarms are provided for 33 temperature points, 26 liquid levels, 39 pressure points, 39 points for stopping auxiliary machinery and seven miscellaneous items, which makes a total of 154 points.

In addition, there is a K-logger to record 40 readings. A "first-alarm" system is included in the control console to indicate which item failed in the line first, should that cause a pyramid effect of machinery stoppage.

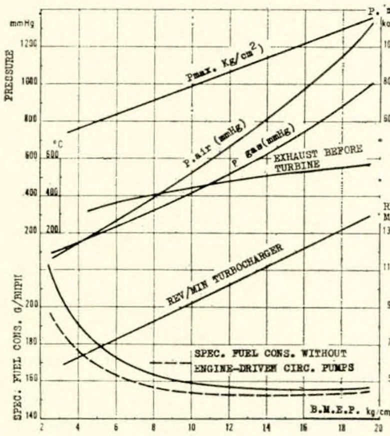
Since *Esso Bangkok* is designed to carry a number of refined products the cargo lines are provided with double valves where required. In addition, all sluice valves and angle valves are fitted with Teflon seats or phenolic foam resin disc inserts to prevent oil leakage.

A low-pressure hydraulic system using Rockwell pumps and motors is provided for driving the deck machinery.—*Motor Ship, December 1968, Vol. 49, pp. 439–443.*

S.E.M.T.-Pielstick Engines

The first production engine of the S.E.M.T.-Pielstick PC3 design has been on test at the St. Nazaire works of Chantiers de l'Atlantique, developing 900 bhp/cyl at 450 rev/min. By any standard this vee-type four-stroke, turbocharged unit is a tremendous achievement, one which inevitably raises the question of how it will affect other types and forms of prime mover. It is essentially a new unit in that it has larger cylinder dimensions and is capable of attaining greater outputs than most medium-speed Diesel engines yet in production.

This is an engine with a cylinder bore of 18·9 in, a piston stroke of 20·45 in and a swept volume of 94·1 litres. Although in the early stages the PC3 will be offered with a continuous service rating of 765 bhp, it will in due course be 855 bhp at 460 rev/min, with a mean indicated pressure of 11·0 kg/cm².

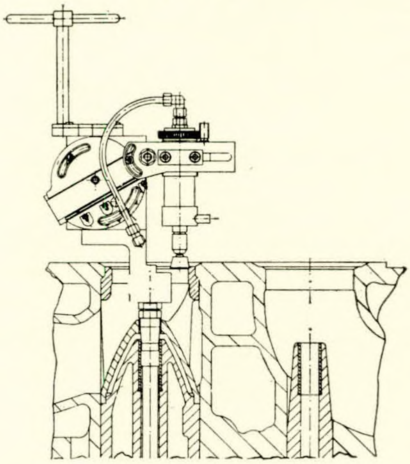


Testbed results of the 12PCV3-480 engine with two engine-driven circulating pumps—operating characteristics at 440 rev/min

Assuming the maximum continuous rating as 850 bhp/cyl, a 12-cylinder PC3 engine will give 10 200 metric bhp with a weight of only 110 tons, while an 18-cylinder unit of this type would deliver 15 300 bhp. Bearing these figures in mind, it will be seen that medium-speed geared-Diesel propulsion has entered a new phase and that the PC3 engine emerges as the most serious challenger to the slow-speed Diesel engine, now capable of producing from a single 12-cylinder unit outputs of 45 000 bhp and more. PC2 engines have been specified for such large ships as the 130 000 dwt tanker of 25 100 bhp ordered by A/S Nynas Petroleum of Stockholm (part of the Johnson Group). This ship has been specified for propulsion by three Swedish-built 18-cylinder vee-form PC2 engines running at 485 rev/min and driving a KaMeWa c.p. propeller through a reduction gear-box with a take-off drive for a 960 kW 1800 rev/min alternator and two cargo oil pumps, each of 3500 hp at 1100 rev/min.—*Motor Ship, February 1969, Vol. 49, pp. 40–42; 45–46.*

Valve Seat Grinder

The grinding-in by hand of cylinder head valves on auxiliary and medium-speed engines can often be a time-consuming maintenance job, particularly where the valve seats cannot be removed from the cylinder head. To save time and labour Chris Martin AB, has added to its range of large valve seat grinding machines a new model for use with valves of 70 to 240 mm diameter.



Valve seat grinder

It is a portable self-contained unit arranged with an adjustable locating spindle which is fitted into the valve guide and held in place by tapered sleeves. The grinding head is arranged with an off-centre stone and diametrically opposite this is the adjusting and control head which also has the operating handle fitted on it. When in operation the whole head assembly is swung round the locating spindle by hand. A typical cut would be about 0·1 mm per operating cycle.

Almost any seat angle can be accommodated and an interesting feature of the unit is that it is designed to produce a minute convexity on the seat to give an initial tangent contact within the face width. The finish produced is said to be such that further work with grinding paste is unnecessary. Furthermore, should it be necessary to adjust the width of the valve seat, this can be accomplished by skimming the top plane of the seat with the machine in place.

The unit operates from a compressed air supply at a working pressure of 100 lb/in² which gives the grinding stone a speed of 16 000 rev/min. It can be used on cast iron, Stellited faces or any hard metal alloy such as nickel-chrome-steel.—*Motor Ship, January 1969, Vol. 49, p. 528.*

Russian Ice-breaking Cargo Vessels

The construction of ice-breaking merchant vessels was started in the U.S.S.R. because of the need to provide adequate transport facilities for the country's rapidly developing northern areas. Practice had shown that ordinary ships, even those strengthened for navigation in ice, cannot tackle this task, their speed behind ice-breakers being too low. In this connexion when drawing up plans for ice-breaking transport vessels, their lines, strength of the ice belting and the power of propulsion machinery were reviewed.

An adequate ballast system was necessary to provide proper immersion and to protect the propeller in the light condition. The retention of good running qualities in clear water, and acceptable specific cargo capacity, were also a requisite.

On this basis, ratios were established between dimensions, hull form and strength, and also the required propulsion power to the vessel's increased ice-negotiating ability.

Limiting loads for the forward, 'midship and aft parts of the ship's framing were accepted at 300, 150 and 225 ton/m^2 (with no account taken of the contribution to strength of stringers) and 375, 200 and 300 ton/m^2 for the shell plating.

A comparison of power plants showed that the Diesel-electric d.c. installation was the only suitable one for navigation in ice and plans were drawn up for such a plant with two separate circuits. Constant power, in spite of changes in propeller torque, could be ensured by electro-automation as could reliable operation with frequent start-ups and reverses without considerable regeneration of power, and also a reduction in the propeller electric motor speed with a sharp increase in torque on the screw without overloading the Diesel prime movers.

All this was accepted when orders were placed for ice-breaking transport ships of *Lena* series. The operation of them has confirmed that they can sail along the Northern Sea Route at higher speeds than conventional vessels on the run, making considerable use of the power of their propulsion units on regular trips, and conducting other ships without recourse to permanent ice-breaker support.

Good manoeuvring qualities at low speeds enabled *Lena* series ships to sail clear of ice obstacles, which ensured an increase of 1·5 to 2 knots in the ship's average speed. When conducting ordinary transport vessels, with varying conditions of operation of the propulsion machinery and the reverses taking much of running time, the high manoeuvrability of the propulsion installation and the rational choice of power were confirmed. At the same time it was established that to pass through heavy ice, it is necessary to generate far more kWh/mile of electricity than when sailing in moderate ice conditions.

Practical experience revealed, however, that ships of that first series had insufficiently strong framing, which limited their speed in ice fields. There was no side protection in way of the engine room.

These drawbacks have in the main been eliminated in the series of ice-breaking transport ships of *Amguema* type now built in the U.S.S.R.

Diesel-electric powered *Amguema* is a single-screw, two-deck, full scantling ship to ULRu/1 class of the U.S.S.R. Shipping Register with four holds, forecastle, bridge superstructure, a cruiser stern and ice-breaker type stem.—*Shipbuilding and Shipping Record, 6th December 1968, Vol. 112, pp. 745–746.*

Increasing the Output of Trailing Dredgers when Working in Compacted Fine Sand

In recent years, a comparatively high proportion of the operations for which trailing-suction hopper-dredgers have been used have involved the shifting of fine sand. This is perhaps because of the ease—and thus regularity—with which fine sand is deposited in coastal areas. But there is also the phenomenon that sand grains tend to become finer in areas where trailing dredgers operate regularly and the reason for this is that the material settles less easily in the hopper than coarse sand, so that more of it is carried off via the overflow and thus remains in the area. Much of this fine sand is compacted, and thus not only is there the problem of getting it into the hopper, but also the problem of diminishing intake via the drag head.

The Mineral Technological Institute (M.T.I.), Delft, has for a number of years carried out investigations and tests aimed at increasing the efficiency of trailing-suction dredgers. Many experiments have been conducted with a view to determining the optimal hopper design and filling system for operations in fine sand, i.e., a configuration affording the least possible overflow losses. Related experiments have been carried out in an effort to develop a drag head giving a satisfactory level of production in highly compacted fine sand. The latter project forms the subject of this article.

The primary objective of the research programme was to increase production without incurring any significant rise in the resistance offered by the drag head. Initially, heads fitted with a variety of vibrating knives and rotating disc-type cutters were employed, however this line of development was abandoned when it became clear that the problems of wear which might be expected to arise with such devices outweighed the increases in production.

In the meantime, scientists at the Versuchsanstalt für Wasserbau und Schiffbau in Berlin achieved considerable success with vertical water jets mounted in front of the suction inlet as a means of decompacting sand. Similar experiments had earlier been carried out in the United States and elsewhere, but these had been unsuccessful owing to incorrect positioning of the jet nozzles. Taking the results of earlier jet tests as a starting point, the M.T.I. embarked on further experiments. These were based on the following theoretical considerations:

1) The introduction of water into the spaces between the compacted sand grains. This has the effect of enlarging the volume of the mass, causing the distance between the individual grains to increase. The mass is thus decompacted and can be sucked up without difficulty.
2) The degree of penetration of the water and the position of the nozzles together determine the depth to which the mass can be decompacted. Higher pressure (and thus greater jet velocity), and a higher flowrate (and thus a greater volume of water) each serve to increase penetration. However, greater penetration can also be achieved by the use of shielded jets, the nozzles of which are below the surface of the sand when in the operating position. The depth of the sand thus loosened, in turn, forms an indication of the production attainable with a drag head drawn across the bed at a given speed.
3) The direction of the jets of water and the dimensions of the jet guards together determine the force required to draw a drag head across the bed at a given speed.
4) The ultimate objective being an optimal configuration, the tests must serve to determine the combination of variable factors which affords the most favourable result.

Experiments were carried out in the tank using the following drag heads:

a) Dutch head as normally employed;
b) Californian head as normally employed;
c) Dutch head fitted with seven vertical jets;
d) Dutch head fitted with six small shielded jets;
e) Dutch head fitted with nine small shielded jets;
f) Dutch head fitted with six large shielded jets;
g) Dutch head fitted with nine large shielded jets;
(and while this article was being written)
h) Californian head fitted with seven vertical jets.

The best results, in terms of production, trailing power requirements and jet force, were obtained with the Dutch head fitted with six small shielded jets. In terms of production alone, the Dutch head with six large shielded jets produced slightly better figures, but only at the expense of substantially greater trailing power. In view of this, we shall confine ourselves here to reference to the results obtained with the Californian head (normal type, b) and the Dutch head in normal form (a), with vertical jets (c) and with six shielded jets (d).—*Hadjidakis, A., Port and Dredging, 1968, No. 60, pp. 4–8.*

Hapag Lloyd Container Ship

The first of a series of four 20-knot container ships recently entered the North Atlantic container trade of the Hapag-Lloyd Container Line, a group formed by two long-established shipowners operating fast liner services from North European ports to New York and other east coast ports of the U.S.A. The two companies concerned are Hamburg America Linie (Hapag) and Norddeutscher Lloyd.

The service has been inaugurated with *Weser Express* and *Elbe Express* built by Bremer Vulkan and Blohm and Voss respectively.

Principle particulars of *Elbe Express* are:

Length, o.a.	561 ft $0\frac{1}{4}$ in
Length, b.p.	508 ft $6\frac{1}{4}$ in
Depth, moulded	47 ft $10\frac{1}{2}$ in
Breadth, moulded	80 ft $4\frac{3}{4}$ in
Freeboard draught	25 ft $10\frac{3}{4}$ in
Corresponding deadweight ...	11 350 tons
Displacement	18 485 tons
Service speed	20·0 knots

Elbe Express is a single-deck ship with a two-deck forecastle and all accommodation and machinery arranged aft of the four container holds. The hull design also incorporates a bulbous fore-foot and a bowed-type of transom stern; stabilizers of the B. and V.-Siemens Electrofin type are also fitted. The ship is built to the Germanischer Lloyd ✠ 100 A4(E) + MC(16/24) classification and it is interesting to note here that the MC(16/24) notation indicates that the engine room may be operated in the unmanned state for 16 hours per day.

To obtain the maximum stowage for containers underdeck, and bearing in mind the weather conditions frequently met in the North Atlantic, *Elbe Express* has been constructed with the widest possible holds and correspondingly large hatch covers. Indeed, the largest holds, Nos 3 and 4, are 20·24 m (66 ft 5 in) wide in a total ship's breadth of 24·5 m (80 ft 4 in). Similarly, these hold lengths have been built to the maximum dimensions to accommodate two rows of 40-ft containers in each hold.

During the early design stage of *Elbe Express* it was decided that the minimum economical number of 20 ft × 8 ft × 8 ft containers to be carried by the ship in the North European/U.S.A. trade was not to be less than 730. Thus the capacity of the ship was designed around this figure and in fact the equivalent of 732 20-ft containers can be carried. However, since containers of the 40-ft size are becoming increasingly popular in the U.S.A. and will fit in well with the German road and rail transport organizations, the ship is arranged to carry a mixture of both 40-ft and 20-ft containers.

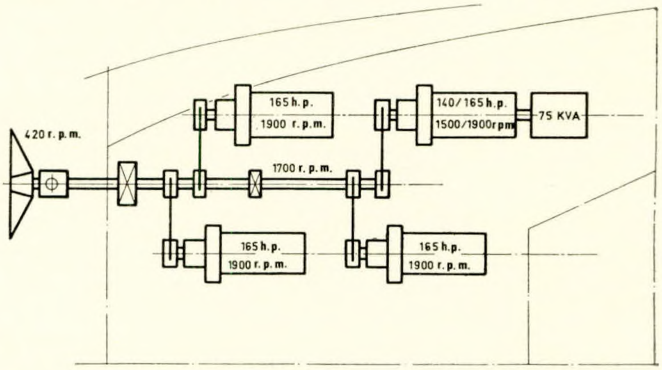
Elbe Express is powered by a nine-cylinder B. and V.-M.A.N. engine of the K9Z78/155E type having a stroke of 1550 mm and bore of 780 mm and a maximum continuous rating of 15 750 bhp at 122 rev/min. However, in a fully loaded condition in calm weather it is anticipated that the output will be 13 640 bhp to give a service speed of 20 knots. It is arranged to burn heavy fuel of up to 3500 sec Redwood No. 1 and is directly coupled to a fixed-pitch propeller manufactured from Alcunic alloy.

Since the vessel's classification carries the MC(16/24) notation, a high degree of automation and remote control of the machinery is fitted. Thus, the main engine is arranged for remote control, by an A.E.G. electro-pneumatic system, from either the bridge or the machinery control room: mechanical control from the latter is also fitted. The remote-control system is actuated by telegraph transmitters both in the wheelhouse and control room with the mode control being selected from either control positions: emergency override of this is provided.—*Motor Ship, December 1968, Vol. 49, pp. 408–412.*

Heavy and Bulk Cargo Carrier to Operate out of Confined Ports

Lady Jane, built by T. van Duijvendijk's Scheepswerf for B. v.d. Laan Scheepvaart- en Handelmaatschappij, is an interesting vessel both in purpose requirement and design conception.

The owners' requirements were for a vessel capable of carrying both heavy and voluminous cargoes between loading and discharge points having a limited depth of water. Stability would require to be such that the vessel should be able to carry about 1000 tons of cargo in the shelter deck which should be open from forward to aft and have a clear



Lady Jane

height of at least 3 m. A multiple machinery installation was required for assured continuity of operation and a high degree of manoeuvrability was an essential. A minimum dwt of 1300 was specified yet, for economy of operation, the gross tonnage had to be kept below 500.

Principal particulars are:

Length, o.a.	249 ft 3 in
Length, b.p.	229 ft 7 in
Breadth, moulded	42 ft 8 in
Depth to m.d.	9 ft 6 in
Depth to s.d.	20 ft 4 in
Draught	9 ft 4 in

The vessel has been arranged with separate part and starboard superstructure with clear drive-through deck space between. The wheelhouse is carried on the starboard superstructure with an auxiliary wheelhouse, providing all-round visibility for close manoeuvring, atop the main command space.

The propulsion installation is an unusual one comprising two Schottel SRP300 directable propeller units each having a maximum drive torque of 275 kgm and being powered, by way of Turner Poly-vee belt drive transmission, by three DAF six-cylinder type DP-680M Diesel engines each delivering 165 bhp at 1900 rev/min—a total shp of 450 at 1660 rev/min, which is only 70 per cent of the SRP300's permissible load. A further two similar prime movers drive Heemaf 75 kVA three-phase alternators through Wülfel-Elco flexible couplings; the generators can, however, be driven from the Schottel shaft in the event of their driving engine failing.

The prime movers, with integral electrically operated friction couplings, are resiliently mounted on a common bedplate and drive a Cardan shaft which carries the Poly-vee sheave which provides a 1·1 : 1 reduction in rotational speed; a further 4 : 1 reduction is embodied in the Schottel gearing so that the Van Voorden 1500 mm diameter four-bladed propellers turn at 415 rev/min.

All engines are fitted with Woodward governors to ensure synchronization. Automatic protection is provided against high cooling water and exhaust gas temperatures and low l.o. pressures. Control and monitoring of the machinery is from a control room in the engine room with duplication of control stations on the bridge and top deck.

The Schottel propulsion-steering unit project through the bottom of the vessel aft, and a skeg and fin has been added to eliminate any possibility of fouling the propellers with ropes or wires. Each unit is provided with its own steering system, a hydraulic unit operating at up to 150 kg/cm^2 and providing a maximum output of 8 hp—capable of rotating the propulsion unit through 180°, from ahead to astern thrust, in nine seconds.—*Shipbuilding and Shipping Record, 8th November 1968, Vol. 112, pp. 602–603.*

Atlantique-built Tanker 'Magdala'

The hull of *Magdala* is divided into the following spaces—fore peak, bunkers, sea water deep tank, cargo tanks, pump room, machinery space and after peak.

Principal particulars are:

Length, o.a.	324·6 m
Length, b.p.	310·6 m
Breadth, moulded	47·2 m
Draught	19·0 m
Gross register	105 296 tons
Net register	76 175 tons
Deadweight	212 000 tons

Cargo spaces comprise 15 compartments formed by two longitudinal bulkheads and four watertight athwartship bulkheads. The 248 428 m^3 of cargo spaces are distributed into centre tanks Nos 1, 2, 3, 4 and 5 and lateral port and starboard tanks Nos 1, 2, 4 and 5. No. 5 centre tank also incorporates a watertight compartment for use as a slop tank during tank cleaning.

Clean ballast, to a total of 39 250 tons, is carried in No. 3 side tanks, fore peak, forward deep tank No. 3 side tanks and two tanks each side of the pump room.

On departure after discharging and during the tank cleaning period, tanks Nos 1 and 4 are ballasted and the washing water discharged into the slop tank, the latter incorporating an adequate heating system to accelerate settling.

Magdala's pumping equipment comprises four KSB 3500 m^3/h horizontal centrifugal pumps driven by a Stal-Laval Atlantique geared turbine; two duplex vertical steam reciprocating stripping pumps of 350 m^3/h capacity; and two 400 m^3/h ejectors. Each pump draws from the tanks through a 700 mm main pipeline.

Two different products can be carried simultaneously, one in the side tanks and the other in the centre tanks, discharge lines from pumps 1 and 4 and those from pumps 2 and 3 being connected in parallel in the pump room, discharging to the loading manifolds through two 800 mm main lines located inside the tanks. Cargo is taken through the manifolds to two 800 mm direct down pipes and, from there, through 700 mm main tank lines. Connexions in No. 3 centre tank allow immediate loading of the wing tanks through the lines running to tanks Nos 1 and 4, while discharging overside the ballast from tanks Nos 2 and 5. The stripping system incorporates a 300 mm circuit.

Propulsion is by a type AP 36/27 Stal-Laval Atlantique geared turbine developing 28 000 shp at 85 rev/min. Steam at 63 kg/cm^2 510°C is supplied by a Foster Wheeler-Atlantique ESD 111 boiler producing 100 tons/h and a 30 tons/h stand-by unit.

Steam production is normally ensured by the main boiler (pressure 63 kg/cm^2—superheat 515°C). The main steam supplies directly the propulsion turbines, the turbo-generator and a feeding reducing-valve of the H.P. bleeding header. The pressure in the H.P. bleeding header ranges from 27 to 30 kg/cm^2 when the propelling power increases from 20 000 to 28 000. It serves as required, the turbo-feed pumps, the turbine-driven forced draught fans and the air ejectors.

Then comes the M.P. bleeding header which supplies the exhaust header at 2·5 kg/cm^2. The latter receives the exhaust steam from the turbine driven auxiliaries and supplies the de-aerator, the gland seals and the L.P. bleeding header. The L.P. bleeding header at 0·7 kg/cm^2 receives the exhaust steam from the turbo-generator and supplies the L.P. heater and the evaporators. In the event of steam excess, this header can discharge either to the main condenser or to the auxiliary condenser.—*Shipbuilding and Shipping Record, 15th November 1968, Vol. 112, pp. 632–635.*

Cruise Liner for Norwegian Owners

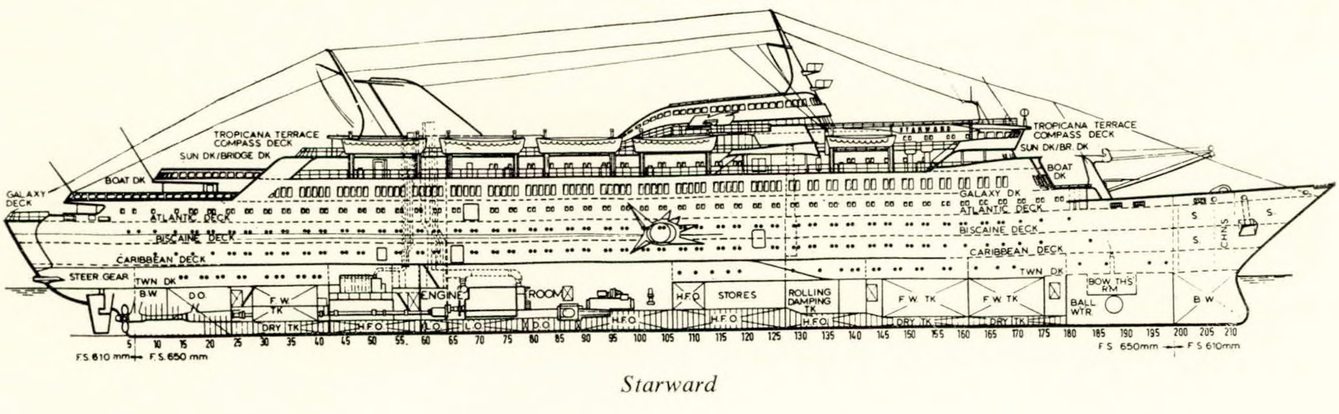
The German shipbuilding firm of A.-G. Weser, Bremerhaven, has delivered passenger and car ferry *Starward* to Kloster's Rederi A/S of Oslo. This 13 100-g.r.t. vessel is a larger version of *Sunward*, 8667 g.r.t., which was delivered to the same owners in 1966.

Principal particulars are:

Length, o.a.	524 ft $9\frac{5}{8}$ in
Length, b.p.	452 ft $3\frac{7}{8}$ in
Breadth, moulded	73 ft $3\frac{7}{8}$ in
Depth to freeboard deck ...	28 ft $2\frac{1}{2}$ in
Depth to A-deck	45 ft $7\frac{1}{8}$ in
Draught, maximum	20 ft 4 in
Deadweight, approximately ...	2100 tons
Gross tonnage	13 100
Machinery output	17 380 hp
Speed, maximum	21·5 knots
Passengers	750

Starward will cruise in the Caribbean from Miami via Nassau in the Bahamas to Kingston, Jamaica, on seven-day trips. She is, however, suitable for longer cruises. In addition to having stabilizers of the Denny-Brown AEG fin type this new ship also has passive roll-damping tanks.

Cargo can be loaded either by means of the stern door



Starward

and ramp, or through a hatch measuring 7 m × 7 m on the forecastle deck: there are two Atlas mast cranes of 10 tons capacity each, which is sufficient to enable 20 ft containers to be brought on board. The trailer deck can accommodate about thirty 40-ft trailers or about 200 American-type motor cars in the lower deck and suspension deck.

The hydraulically-operated car deck fitted below A-deck consists of two ramps, 11 sections and two pontoon covers, one of which is fitted with a turntable. The stern ramp, with a loading capacity of 40 tons, is hydraulically-operated and secured. There are also seven side doors with hydraulic locking arrangements.

The propelling machinery in *Starward* consists of twin M.A.N. 16-cylinder four-stroke Diesel engines, type V8V 40/54, fitted with turbo-charger and intercooling. Each engine has a continuous output of 8690 hp at 400 rev/min and is designed to run on heavy fuel having a maximum viscosity of 1500 sec Redwood No. 1 at 100°F. Each engine drives a KaMeWa variable-pitch propeller at a speed of 225 rev/min through A.-G. Weser reduction gearing. In order to improve manoeuvrability a bow-thrust unit running at 870 rev/min and powered by a 1000-hp electric motor has been installed. —*Shipping World and Shipbuilder, February 1969, Vol. 162, pp. 175; 177; 181; 183.*

Hitachi-built Container Ship

The first container ship to be built by Hitachi Zosen has been delivered from the Innoshima yard. She is 25¾-knot *Kashu Maru*, to be powered by a 12-cylinder, 840-mm-bore Hitachi-B. and W. engine of 27 600 bhp. Of 14 800 dwt she will operate in the Yamashita-Shinnihon Steamship Co.'s service between Japan and Los Angeles/San Francisco and is designed to carry 504 20-ft containers in the holds with, during the summer months, an additional 224 containers stowed on deck, making a total of 728 containers.

To make available the maximum capacity for containers, the machinery space is arranged "semi-aft"; i.e., with four container holds forward of the engine room and one aft. The ship is also designed to carry a total of 88 refrigerated containers.

The hull design is, in effect, a double-hull arrangement. To augment the strength of the hull with its wide-hatch arrangement, the underdeck sides of the 'midship part of the hull are constructed with longitudinal box girders. Utilizing this structure, the ship is designed with a double hull, which provides ample space for fuel oil and water ballast.

A bulbous bow has been incorporated following extensive studies of effects in the hull design and, in order to improve stability in heavy seas, the double hull and double bottom are used as anti-rolling tanks to offer protection to the containers and to assist in maintaining schedule.

Principal particulars are:

Length, o.a. ... (approx.)	188·00 m
Length, b.p.	175·00 m
Breadth, moulded	25·70 m
Depth, moulded	15·30 m
Designed full-loaded draught ...	9·10 m
Gross register ... (approx.)	16 500 tons
Deadweight capacity (approx.)	14 800 tons
Capacity of container holds	728 (8 ft × 8 ft × 20 ft containers)
Trials speed (max.)	25·75 knots

—*Motor Ship, December 1968, Vol. 49, p. 416.*

Ship Halves Joined Afloat in the U.S.S.R.

The first welding afloat of a vessel in the U.S.S.R. recently took place at the small Volodarsky shipyard at Rybinsk on the Upper Volga, the vessel being 394 ft dry cargo carrier *50 Years of Komsomol.*

Construction of a ship of this size at the Volodarsky yard necessitated a great deal of preparatory work, not only at the shipyard itself but by designers of the Shipbuilding Technology Research Institute.

While the slipway capacity at the building yard could only cope with a vessel of this size in two sections, the experiment was considered to be of considerable value in increasing the potential of shipyards without greatly adding to existing facilities.

The preliminary work involved was very considerable with the thickness and form of the weld being of primary interest. A wide variety of samples were tested to destruction in the laboratory to provide an eventual solution which was satisfactory in every way.

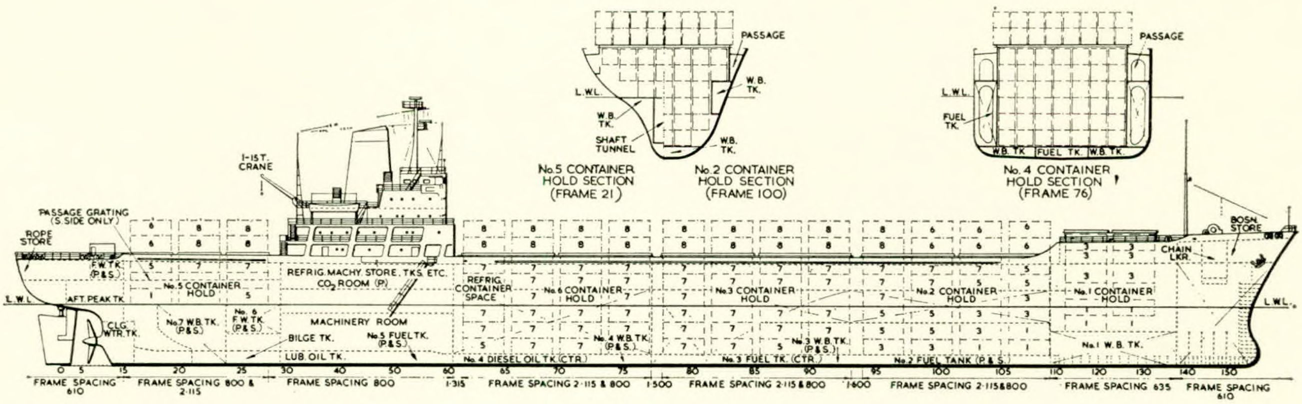
The majority of examples of welding afloat so far carried out have made use of caissons which, in effect, have provided a working tunnel round the hull.

This particular method could not be employed in this instance and, instead, a simle but effective method of hermetically sealing the joint area for welding was evolved.

Before this stage, however, the problem of the alignment of the two half-ships had to be solved particularly as, although precise arrangements for lining up by means of guide strips and locking pins were made before launching, the weather conditions of the area together with the large sail area of the sections were not normally favourable for such an operation.

Despite the complexity of winches, pulleys and wire ropes employed to effect a union between the two halves, the linkup was successfully completed within three hours.

Finally, welding was begun. One-sided welding from the inside was employed, with a back formation of the weld joint provided by a copper shoe, having an oval recess to



Hitachi-built Kashu Maru

impart the required shape to the fused metal, moving outside the hull in concert with the welding head inside. Only the succession of welding was changed. Usually, welding proceeds from bottom upwards, the weld being moved to either side. In this case it was just the opposite. The topsides were welded first and the bottom of the ship followed. This precedence was dictated by the necessity to provide the vessel with the earliest stiffness to oppose stresses on the ship imposed by sharp fluctuations of water level. As soon as welding was completed, the seam was tested by X-ray techniques.—*Shipbuilding and Shipping Record, 10th January 1969, Vol. 113, p. 52.*

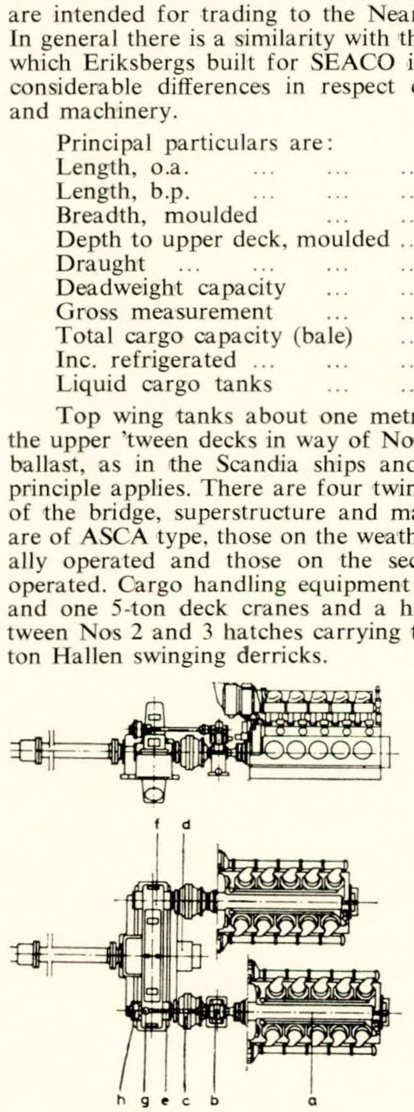
New Swedish Vessels

Eriksbergs Mek Verkstads have launched the 12 425 dwt cargo motorship *Isfahan*, first of a series of four 12 425 dwt vessels for the Swedish Orient Line and the Swedish East Asiatic Co. both members of the Brostrom group. These are intended for trading to the Near, Middle and Far East. In general there is a similarity with the class of Scandia ships, which Eriksbergs built for SEACO in 1966/67 but there are considerable differences in respect of cargo space, hatches and machinery.

Principal particulars are:

Length, o.a.	458 ft 0 in
Length, b.p.	425 ft 0 in
Breadth, moulded	70 ft 6 in
Depth to upper deck, moulded ...	42 ft 1 in
Draught	31 ft 0 in
Deadweight capacity	12 425 tons
Gross measurement	9600 tons
Total cargo capacity (bale) ...	565 000 ft³
Inc. refrigerated	35 000 ft³
Liquid cargo tanks	17 000 ft³

Top wing tanks about one metre wide are arranged in the upper 'tween decks in way of Nos 2 to 4 holds, for water ballast, as in the Scandia ships and the same "open ship" principle applies. There are four twinned hatches all forward of the bridge, superstructure and machinery. All the covers are of ASCA type, those on the weather deck being hydraulically operated and those on the second deck mechanically operated. Cargo handling equipment consists of three 12-ton and one 5-ton deck cranes and a heavy mast arranged between Nos 2 and 3 hatches carrying two 22-ton and one 100-ton Hallen swinging derricks.



a) 10PCV2 engines
b) step-up gear for generators
c) main Pneumaflex couplings
d) KaMeWa control gear
e) Cardan shaft
f) twin-input
g) footstep bearing
h) Spiroflex coupling on 1800 rev/min generator shaft

Main machinery arrangement for new Swedish vessels

The propelling machinery comprises two 10-cylinder Eriksbergs-Pielstick engines of 4000 bhp at 428 rev/min arranged at 3·75 m centres and driving a 125 rev/min stainless steel KaMeWa propeller through Lohmann and Stolterfoht Pneumaflex clutch-couplngs and twin-input combining gears. Interposd between the starboard engines and its clutch-coupling is a step-up gear providing a power take-off for an 1800 rev/min 750 kW alternator. This arrangement which also permits a main engine to drive the alternator in harbour without making the rest of the transmission "alive" will be seen from the sketch. Total output is 6900 shp giving a service speed of 16·7 knots on 24 ft draught.—*Marine Engineer and Naval Architect, December 1968, Vol. 91, p. 529.*

Tanker Repairs Effected without Gas-freeing

With the successful completion of repairs at Falmouth to BP Tanker Company's 67 000 dwt *British Commerce* which had previously carried a cargo of crude oil, a new trend in tanker repairs has been set in the United Kingdom.

After discharging her cargo of crude oil at Wilhelmshaven, the vessel proceeded to Falmouth and on passage completed inerting all cargo tanks by means of her own inert flue gas system, which maintains a safe condition within the cargo tanks. On arrival at Falmouth the vessel was accepted within the Yard of Silley Cox and Co. Ltd. without cleaning and gas-freeing and the repairs, including extensive engine refit and underwater cleaning of the hull, were completed within seven days.

The underwater cleaning was very successful, the clear water at Falmouth Docks being a distinct advantage in ensuring the complete removal of marine growth. The exercise was undertaken by Silley Cox and Co. Ltd in close co-operation with BP Tanker Co. Ltd, the local trade unions, the factory inspectorate and industrial petroleum chemist.—*Shipbuilding International, December 1968, Vol. 11, p. 38.*

Numerically Controlled Propeller Milling Machine

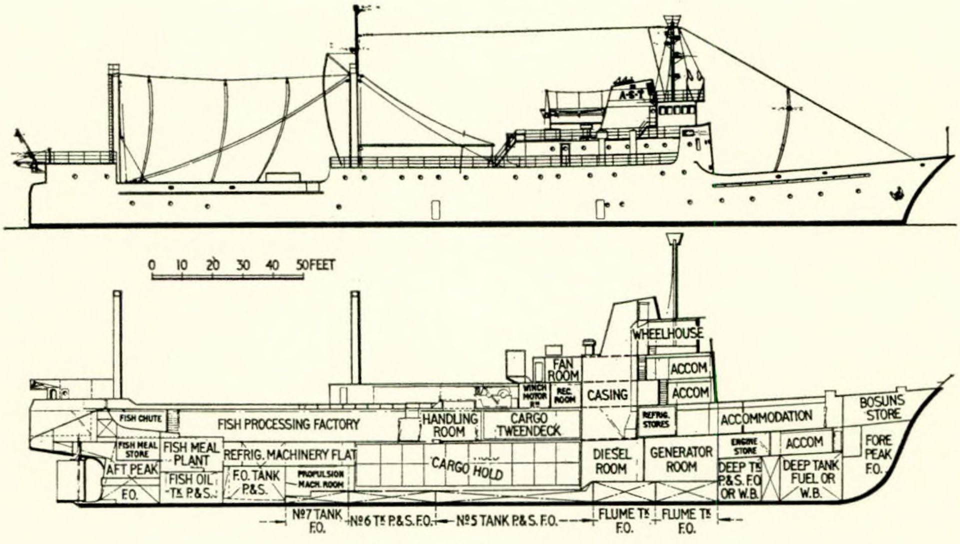
One of the most advanced numerically-controlled machine tools yet built, capable of the simultaneous milling of two propeller blades with provision for continuous path numerical control in nine axes, has recently been delivered by Maschinensabrik Froriep to Karlstad Mekaniska Werkstad for manufacture of KaMeWa c.p. propeller blades.

Known as the "Spheromill P", the machine holds the workpieces on a rotating clamping fixture, similar to the headstock and faceplate with a centre lathe. Hydraulic rams on the faceplate support the workpieces from behind to balance cutting forces. Two identical milling heads are fitted, one for each blade. They move horizontally and axially on the carriages and can also be swivelled both in the vertical and horizontal axis, so that the milling cutter is correctly aligned with the profile of the workpiece. The nine axes under numerical control are faceplate rotation and, independently for each milling head, horizontal and axial transverse of the carriage and vertical and horizontal swivelling of the milling heads. The control used is a GE 105C, a five-axis control whose functions are shared between the two milling heads and the faceplate. The machine eliminates the need for a detailed drafting and for making templates, thus saving many man-hours and reducing lead time. It is capable of working on stainless steel blades.—*Shipbuilding and Shipping Record, 15th November 1968, Vol. 112, p. 644.*

First American Stern Factory Trawlers

The advanced stern factory trawlers *Seafreeze Atlantic* and *Seafreeze Pacific*, built by Maryland Shipbuilding and Drydock Co. for American Stern Trawlers Inc., a subsidiary of American Export Industries, were delivered on schedule in November/December, 1968.

Seafreeze Atlantic was designed to operate as a completely independent trawler, catching and processing the complete catch into fish fillets in wholesale packs, bagged fish meal and fish oil. The net was to be handled over the stern and brought inboard via a stern ramp. The proposed maxi-



Lead ship Seafreeze Atlantic

mum voyage was 90 days. It was envisaged that the catch from other trawlers would be taken aboard, if available.

As the general arrangement shows the normal arrangement for Diesel-electric stern trawlers, with the Diesel generator room forward and the propulsion motor room aft of the cargo hold, has been followed. The refrigeration machinery is arranged on a flat in the propulsion motor room. A complete factory space containing all the machines to head, gut, fillet, skin, freeze and pack the fish is located on the second deck aft with a fish meal plant below the second deck, aft of the propulsion motor room. Fresh water is carried in the tanks outboard of the stern ramp between the second and upper decks. Forward of the factory space is an insulated and refrigerated 'tween deck cargo compartment.

The trawlers are built to ABS requirements for Class A1(E) Fishing Service, Ice Strengthening C; A.M.S. for a scantling draft of 19 ft.

Electrical power for main propulsion and all ship's service requirements is provided by three Diesel engine driven generators located in the forward machinery spaces. Two AEG 1000 kW constant current d.c. generators are driven, through torsionally flexible couplings, by General Motors 12-cyl 12-645-E2, Roots blower scavenged Diesel engines, rated continuously at 1500 bhp at 900 rev/min. One AEG 670 kW constant current d.c. generator is driven, through a torsionally flexible coupling, by a similar GM eight-cyl. 8-645-E2 engine, rated continuously at 975 bhp, at 900 rev/min.

Two constant current d.c. motor-driven 500 kVA ship's service alternators furnish all ship's service power requirements.

Control of the main propulsion system and ship's service system is grouped in the generator room at the propulsion control switchboard and the ships' service a.c. generator control switchboard. Local control of the speed and sense of rotation of the main propulsion motors is provided at the propulsion control switchboard. The engineer's signal and alarm panel, the Diesel engine alarm panels, the Diesel engine tachometers and governor speed control motor switches are located at the ship's service a.c. generator control switchboard, serving to bring all vital propulsion and ship's service control and montitoring to a single control centre.

Two AEG 1500 hp (metric) constant current d.c. propulsion motors drive the single reduction gear pinions through rubber bushing type flexible couplings.

The single reduction gear is of the double helical type with pinions arranged in a horizontal plane through the low speed shaft centreline. The reduction gear reduces the motor speed from 1000 rev/min to 185 rev/min.

A four-bladed solid manganese nickel aluminium bronze propeller, of 11·75 ft diameter and 9·17 ft pitch has been designed to provide a good compromise between the free running and trawling conditions of the ship.—*Shipbuilding and Shipping Record, 10th January 1969, Vol 113, pp. 43–48; 52.*

Polish General Cargo Ship

Last year witnessed the introduction of several new types of general cargo ship from Polish shipyards, among which the Gdansk Shipyard delivered the 12 500 dwt B40 type to Russian owners and the 6000 dwt type B448 to Norwegian owners.

In the latter part of the year the Szczecin Shipyard completed m.v. *Zakopane*, the first of the B446 series of general cargo ships. The vessel is designed to carry refrigerated cargo, edible oils and 16 containers, each of 20 ft × 8 ft × 8 ft stowed on the upper deck and is also suitable for the carriage of grain in bulk.

Zakopane has been designed at the Szczecin Branch office of the Ship Design and Research Centre and built according to the rules and under Lloyd's Register of Shipping supervision for the class 100 A1 ✠LMC ✠RMC.

The hull is strengthened for navigation in ice to class 1. Of all-welded construction, the vessel has two continuous decks with five cargo holds, one of which is refrigerated, with the machinery and accommodation located between Holds Nos 4 and 5. Edible oil tanks which have no internal stiffening, are located in the centre of Holds Nos 1 and 5. Hatches Nos 1, 2, 3 and 4 on the upper deck and No. 5 on the superstructure deck are all equipped with steel hatch covers of Macgregor single-pull type operated by electric winches. Steel flush hatch covers of the hydraulically operated multi-folding type are fitted to Nos 2 and 3 hatches on the tonnage deck which is suitable for operation of fork lift trucks.

Cargo handling equipment comprises four goalpost masts, six pairs of mechanical derricks of 2½ tons lifting capacity when working in union, two pairs of mechanized derricks of 5/2½ tons lifting capacity each, two 15-ton and one 40-ton heavy lift derricks. The derrick reach beyond the ship's side is 19 ft 8 in. All cargo winches are a.c. machines and operate on a Ward-Leonard system.

A temperature of −4°F (−20°C) is maintained in the refrigerated hold by means of a Freon 12 refrigerating plant using cold air circulation of up to 100 air changes per hour. The non-refrigerated holds are force-ventilated at 12 air changes per hour and distant reading and recording arrangements are provided for cargo hold air temperature and humidity.

Water ballast amounting to some 900 tons is carried in side deep tanks in Holds Nos 1 and 5 and in double bottom tanks whilst a further 820 tons of water ballast may be carried in the cargo oil deep tanks. The bilge and ballast systems and the fuel oil transfer systems are remote-controlled from a panel located near the main engine manoeuvring stand.

Propulsion machinery comprises a Cegielski-Sulzer type 6RD68 Diesel engine of 7200 bhp at 135 rev/min and suitable for operation on heavy oil with a viscosity of 1500 sec Redwood No. 1 at 100°F.

Principal particulars are:

Length, o.a.	442 ft 11 in
Length, b.p.	403 ft 6 in
Breadth, moulded	58 ft 1 in
Depth to upper deck	34 ft 1 in
Depth to tonnage deck	23 ft 7 in
Draught to freeboard mark ...	25 ft 1 in
Deadweight at this draught ...	7350 tons
Trial speed	17·2 knots

—*Shipbuilding International, February 1969, Vol. 11, pp. 16–17.*

Shipping Company's Experience with S.E.M.T. Pielstick Engines

The Johnson Line fleet consists of 41 ships of which 11 are powered by S.E.M.T.-Pielstick engines. All eight of the newbuildings now on order will be Pielstick-engined.

For the marine engineer, one of the most rewarding aspects of multi-engine drive is the flexibility afforded in the search for new solutions to auxiliary power problems. The 130 000-ton tanker at present under construction for the Johnson Line demonstrates original thinking in her machinery arrangements. The three main engines (Eriksberg-built S.E.M.T.-Pielstick machines each developing 8370 bhp at 485 rev/min) will drive a single KaMeWa propeller through common gearing, one of the engines being aft and two forward of the reduction gearing. The main generator is driven from the common gearing, the small auxiliary generator being required only when the ship is neither at sea nor pumping cargo. The cargo oil pumps are also powered by the main engines via the reduction gearing and long shafts running beneath the forward engine pair and through the forward engine room bulkhead.

The ships of *Rio de Janeiro* class, the first Johnson Line vessels to be fitted with Pielstick engines, were run initially on marine Diesel fuel. In 1960, however, the first trials were carried out with fuel having a viscosity of 200 sec Redwood No. 1 and about a year later 600 sec Redwood No. 1 was introduced. The reason for choosing this grade was its general availability at the ports used by the ships, added to which it could be handled by the existing fuel heating arrangements in the ships.

Apart from minor running-in troubles, which have been cured, the engines have proved to run very well on this fuel and at the moment fuel of 1000 sec Redwood No. 1 is specified for the newest ships. But, in the case of the new tanker, and the newbuildings, provision is made for the use of all grades up to 3500 sec Redwood No. 1 and there seem to be no objections to this.

The company's records of engine performance for the Pielstick-powered ships show that a fuel consumption of about 155 gal/bhp-h is a realistic figure, while lubricating oil consumption is around 0·8 gal/bhp-h. The Johnson Line is satisfied that these are good figures.—*Peterson, P. E., Motor Ship, Special Survey, February 1969, Vol. 49, pp. 51–54.*

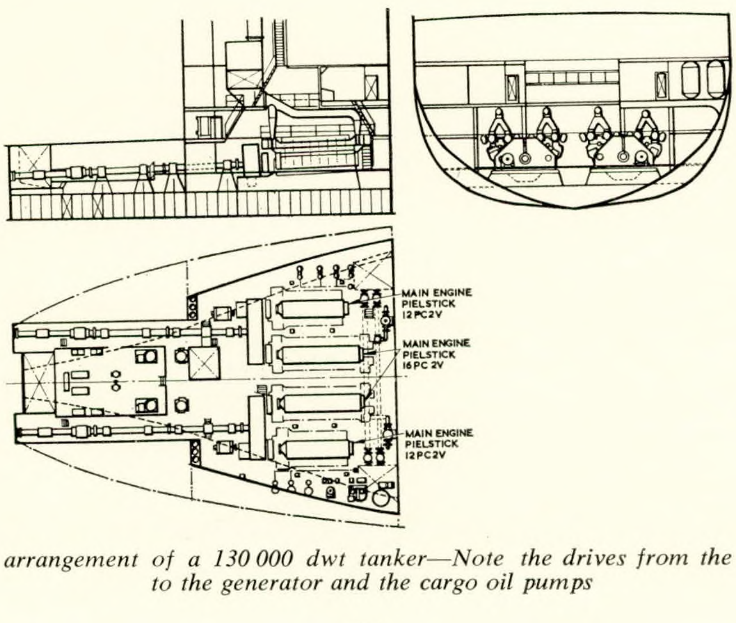
Digital Computer Applications in Marine Sciences

Digital computers are becoming more and more prevalent in the broad category of marine sciences. Their applied use and the technology of digital logic is revolutionizing the methods for observation, collection and evaluation of data obtained from the marine environment.

Perhaps the most fundamental problem at sea is that of position fixing. Locating the ship is of prime importance for the most obvious of reasons and, for this, the problem of position-fixing embraces all of the disciplines of marine science.

The computer fits into the complexity of today's various electronic navigation methods as a collator and mathematical problem solver. For instance, the U.S. Navy's Transit satellite navigation system provides high precision, high accuracy, all-weather, worldwide position-fixing capability. However, with the limited number of satellites that are now in orbit, a position of this calibre can be obtained approximately every 70 minutes of time.

Loran and Decca coverage are quite extensive in the well-travelled shipping routes of the world but leave much to be desired in the remote positions of the earth and away from major seaports. Auxiliary short range electronic systems



Engine room arrangement of a 130 000 dwt tanker—Note the drives from the main gearing to the generator and the cargo oil pumps of high precision, such as Raydist, Decca, Hi-Fix and Shoran allow the hydrographer to survey the coastal areas with reasonable accuracy but they do not have worldwide coverage available on a continuous basis.

In addition, numerous charts, tables and interpolation schemes must be carried out for each type of electronic aid-to-navigation used.

The small, compact computer of today is helping to solve the position determination by a system such as shown in the block diagram.

Essentially, the above system solves the equations necessary for the computation of latitude and longitude. It then plots the track of the ship on the X-Y recorder in a compatible chart projection system. Geographic location determines which electronic navigation system is used.

The system must use a mass storage device with a relatively fast transfer rate. It has the primary function of a programme library that includes the programme routines necessary for each type of electronic navigation used. It also contains plotting, charting and a host of housekeeping programmes.

As a natural outgrowth of the above system, a great deal of attention is now being given to automating coastal and deep sea charting. With the inclusion of an interface to record horizontal sextant angles and one to record sonar pulses, the ship can automatically chart the ocean as it steams along a predetermined track.

It can use the customary method of sextant angles for close shore surveying or use the available electronic systems for either close shore or off-shore soundings.

It is possible to input the predetermined track into the computer and have the ship's helmsman steer a computed course that contains corrections for vessel drift, due to wind and current.—*Under Sea Technology, October 1968, Vol. 8, pp. 35–37.*

Japanese Salvage Tug

Salvage tug *Koyo Maru*, built at the Shimonoseki Shipyard and Engine Works of Mitsubishi Heavy Industries for Japan Ocean-Going Tugboat Co. Ltd has been designed and equipped for towing disabled mammoth tankers which serve on the Japan–Persian Gulf route where the majority of Japanese tankers operate.

Special design consideration was given to the vessel's towing capability which can develop a maximum bollard pull of 80 tons with two main engines producing a total output of 9000 bhp; sufficient to tow a 150 000-ton vessel at seven knots or a 300 000-ton vessel at five knots in adverse weather conditions.

After a careful analysis of resistance, propulsion and wave tests carried out at the Nagasaki Ship Experimental Tank, the final hull lines were drawn to give a fine block coefficient enabling a free running sped of 17 knots to be maintained.

To obtain sufficient stability under various weather conditions, likely to be encountered on towing voyages from the Persian Gulf to Japan, and particularly when loaded with heavy salvaging equipment on the upper decks, the hull breadth was made as wide as possible. In addition, the oil and water tanks are divided into small sections to reduce the free-surface effect. Under full load conditions a metacentric height of about 120 cm has been achieved.

To assist manoeuvrability freely and quickly under arduous conditions, a bow thrust unit is installed capable of turning the vessel on the spot in four minutes. When proceeding at five knots, the time required for turning can be reduced by about half through the combined use of the bow-thrust unit and by applying the correct rudder angle.

The main propulsion machinery consists of two M.A.N. R9V 40/54 turbocharged Diesel engines, each developing a maximum continuous output of 4500 bhp at 400 rev/min. Both engines drive a single shaft via a Vulcan fluid coupling and reduction gear which rotates the propeller shaft at 155 rev/min. The Mitsubishi-KaMeWa four-bladed c.p. propeller is made of SCS2 stainless steel giving a substantial weight saving and a smaller diameter shaft than a conventional bronze casting.

Towing equipment consists of a winch having a capacity of 35 tons and a hawser holding power of 150 tons, and a 35-ton rotating hook. The winch can accommodate 600 m of 65 mm diameter wire.—*Shipbuilding and Shipping Record, 3rd January 1969, Vol. 113, pp. 15–16.*

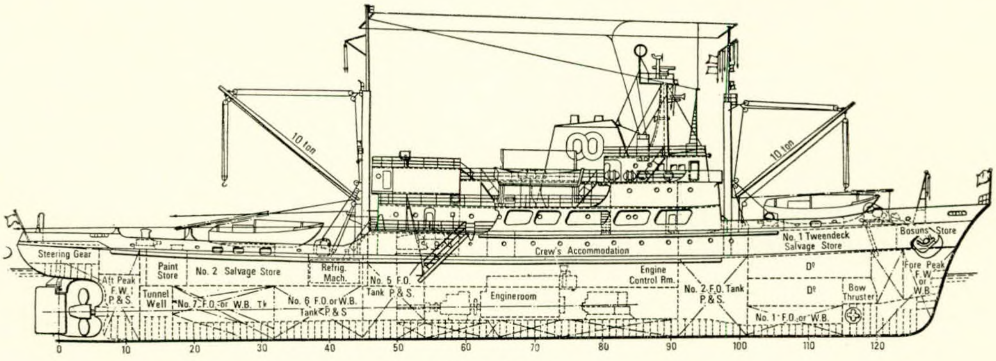
Solidification of Fusion Welds in Face-centred Cubic Metals

This investigation on face-centred cubic metals was initiated as a continuation of the studies of the solidification mechanics of fusion welds. The primary objective was to compare the solidification behaviour of fusion welds made in face-centred cubic metal with that previously observed for body-centred cubic materials.

It was shown, by X-ray crystallographic studies and metallographic examination of gas tungsten-arc welds made in two high purity nickel heats and a copper single crystal, that solidification proceeds by epitaxial nucleation and growth from the partially melted heat-affected zone grains.

This investigation revealed that the competitive growth process which occurs during the solidification of ingots and castings also occurs during solidification in fusion welds in face-centred cubic metals. It was also shown that the competitive growth process is not limited to adjacent grains, but can also involve twinned structure competing with the parent or adjacent grains during growth.

This investigation also revealed that the relationship between the characteristic easy growth direction and the maximum temperature gradient (which is essentially normal to the instantaneous solid-liquid interface) determines the degree of preferred orientation in the weld fusion zone in face-centred cubic metals. This observation is of considerable



Koyo Maru

practical significance, since preferred orientation of the grains in the fusion zone arising from the competitive growth process should cause the weld to exhibit anisotropic properties. Thus, the transverse and longitudinal properties of welds produced with a tear drop shaped puddle should differ significantly due to the predominant transverse growth direction associated with this puddle geometry. On the other hand, in the case of welds made with an elliptical puddle the grains tend to be hook-shaped and to curve into parallelism with the welding direction as they approach the weld centreline. —*Savage, W. F., Lundin, C. D. and Chase, T. F., Welding Jnl, November 1968, Vol. 47, pp. 522s–526s.*

Minimizing Container Damage

The design and manufacture of containers is a fundamental factor to their life and performance and there are an increasing number of types now being produced. Although currently all cargo containers are being built to I.S.O. requirements, there is a wide variation in the design and manufacture of even box types with end doors only and in regard to containers having side loading facilities, the variations are even more numerous. Most of these incorporate moving or removable parts which can be damaged. Tilts and other types incorporating nylon or canvas covers are, by their very nature, vulnerable to damage.

The non-standardization of lifting fitments and particularly the absence of fork-lift pockets for empty lifting on many containers presents serious obstacles to correct handling. In many instances where it is usual to deal with the movement of containers by the use of fork-lift trucks, equipment is used for the handling of containers unsuited to such lifting with the result that their floors are seriously damaged. This type of damage is very costly to repair since the underfloor members are usually severely distorted.

It is known that quite frequently "Fork Pocket Holes" are punched into the sides of containers by fork-lift operators and, in these instances, the consequential damage is even more alarming. Invariably floor damage arising from the misuse of fork trucks cannot be repaired on site and the additional expense of transporting the damaged container to and from a convenient repair shop is then incurred.

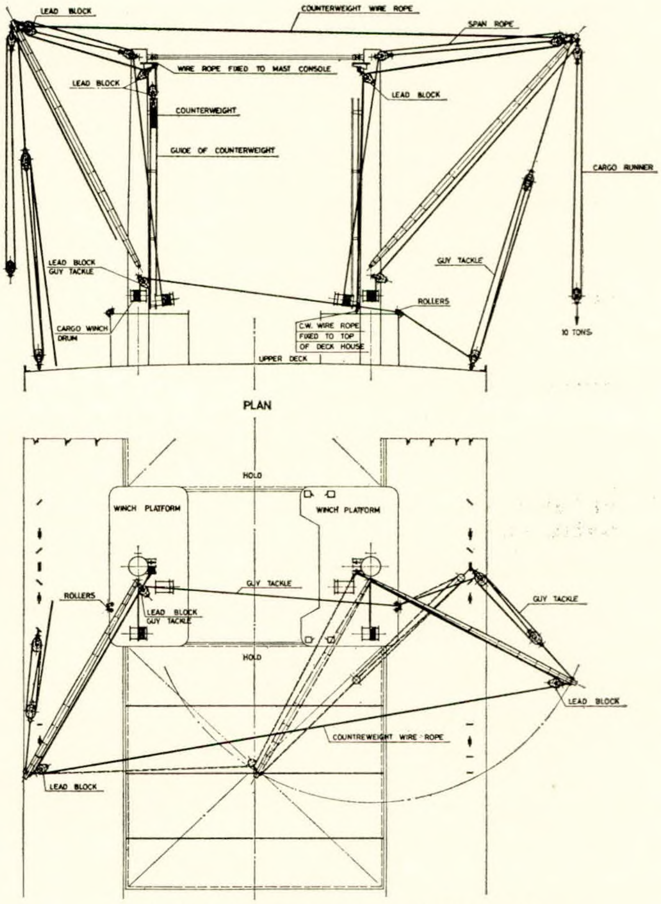
Although standards have been laid down internationally, the fitting of I.S.O. corner castings has, on certain containers, presented serious handling problems in terminals. Instances on record show that corner castings have been set out of alignment so that they could not be handled by the cranes. On some occasions corner castings are reported to have broken away from the containers with quite drastic and costly results.

The materials from which containers are made will naturally affect their life and also the damage rate. At the moment, compensating factors at work seem to equate the value of each cladding material, if due account is taken of the life expectancy. On sea routes, corrosion could prove to be an important factor against the merits of certain types of container, depending upon the degree of anti-corrosive treatment which has been given.—*Hobbs, H. C., Shipping World and Shipbuilder, February 1969, Vol. 162, pp. 164–165.*

Bulk Carrier from Yugoslav Shipyard

The ship described in this article is a bulk-carrier of 30 110 dwt built by "3 Maj", Brodogradiliste, Rijeka, Yugoslavia. The owners are Welsh Carriers Ltd. of Newport, Mon. This new ship *Welsh Minstrel*, is identical in construction to the bulk carrier *Alida Gorthon*, which was built at the "3 Maj" shipyard in 1963 for the Swedish shipowners Rederi A/B Gylfe, Helsingborg.

Welsh Minstrel has Diesel propulsion and a speed of 15·5 knots.



Rigging arrangement of winches and derricks in Welsh Minstrel

Principal particulars are:

Length, o.a.	645 ft $0\frac{1}{4}$ in
Length, b.p.	600 ft $0\frac{1}{4}$ in
Breadth, moulded	75 ft $1\frac{1}{2}$ in
Draught, summer (Load Line 1966)	35 ft $7\frac{5}{8}$ in
Depth to upper deck	47 ft $5\frac{7}{8}$ in
Deadweight	30 110 tons
Gross tonnage	18 775·99
Lightweight	
Machinery output	10 500 bhp at 119 rev/min
Cargo capacity	
grain	1 357 655 ft^3
dry cargo	22 885 ft^3

This vessel was designed primarily for the transport of such bulk cargoes as grain, coal and ore, as well as for the carriage of general cargo. In considering the strength of the ship, some 14 possible loading or ballast conditions were examined with a cargo stowage factor ranging from 12 ft^3/ton to 47·9 ft^3/ton. Furthermore, a detailed study was made into the question of grain loading. Departure and arrival conditions were examined for stowage factors ranging from 45 to 65 ft^3/ton. Each of the ten-ton derricks has an outboard reach of 4·5 m at an angle of 45°. Each pair of derricks is equipped with union purchase rig for lifting loads of up to four tons.

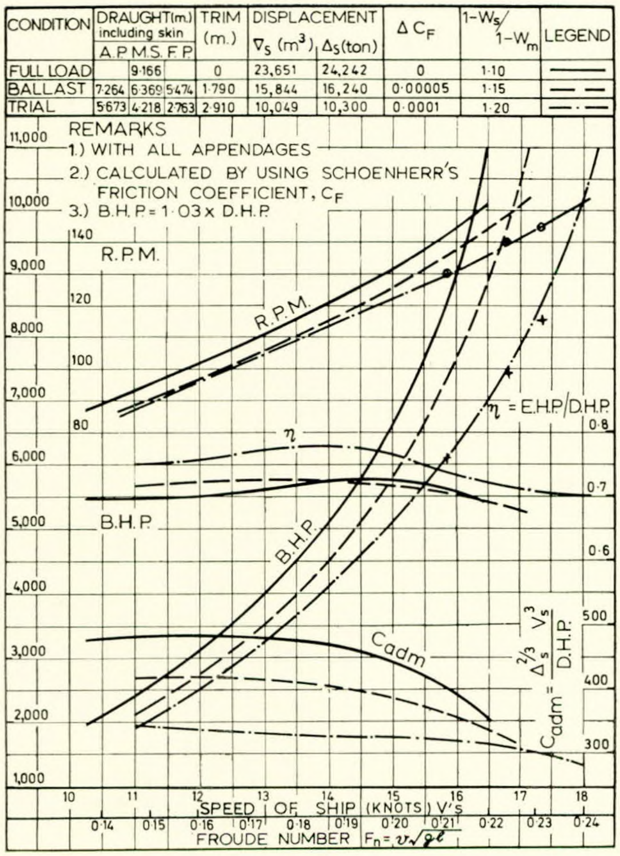
As there are no slewing winches acting on the derrick guys, movement of a single derrick is made by means of one cargo winch and a counter weight (see figure). The counterweight system makes it possible to load and unload cargoes of up to ten tons in weight by means of one derrick and two cargo winches.

On the warping head of No. 1 cargo winch there is a runner for hoisting the cargo. On the warping head of No. 2 winch there is the outboard slewing guy of the same derrick, by means of which the derrick can be moved outboard. In this way the counterweight on the opposite derrick is lifted.

The propelling machinery in *Welsh Minstrel* consists of a "3 Maj" Sulzer type 7 RD 76 two-stroke crosshead Diesel engine fitted with two Brown Boveri turbochargers. This engine, which was built at the shipyard, is rated at 10 500 bhp at 119 rev/min. It has been designed for operation on heavy fuel of a viscosity not higher than 3500 sec Redwood No. 1 at 100°F (38°C).—*Shipping World and Shipbuilder, February 1969, Vol. 162, pp. 295–300.*

Japanese-built Bulk Carrier

Launched at the Kasado dockyard, Japan, *Pacific Defender* is an 18 600 dwt bulk carrier built for Pan American Bulk Carriers Inc., Liberia. The vessel is managed by the Island Navigation Corp. Principally intended for the carriage of packed timber or grain, this single-screw ship can also carry coal and scrap steel cargoes. Its main machinery



Trial results for Japanese-built bulk carrier under three loading conditions

comprises a Uraga-Sulzer 7 RD 68 supercharged cross-head type Diesel engine with a maximum output of 8400 bhp at 142 rev/min. This drives a four-bladed manganese bronze propelled through shafting supported in four bearings. The propeller has a diameter of 16 ft 9 in and aerofoil section blades. Two turbochargers of the IHI BBC-VTR 500 type and an auxiliary boiler of the oil-fired vertical Cochran type are fitted.

Principal particulars are:

Length, o.a.	520 ft 0 in
Length, b.p.	487 ft 6 in
Breadth, moulded	74 ft 2½ in
Depth, moulded	43 ft 3½ in
Design draught, moulded summer	30 ft 0 in
Block coefficient	0·767
Gross tonnage (U.S. Rules) ...	11 000
Deadweight	18 600 tons
Cargo capacity	
grain	943 000 ft³
bale	920 000 ft³
packed timber, under deck ...	8 000 000 bf
packed timber, on deck ...	2 250 000 bf
Timber height, on deck	
average	15 ft 9 in
maximum at after end ...	16 ft 4¾ in
Machinery output, maximum ...	8400 bhp at 142 rev/min
Normal service speed	about 15 knots

The trial results, uncorrected for wind and tide effects, for runs at 75, 90 and 100 per cent of engine output have been plotted on the curves shown in the accompanying figure. These are the results of tank tests with the model of the ship when run at three loading conditions, and give details of Admiralty coefficient and efficiency, as well as bhp and rev/min, against speed and Froude number. When considering the position of the sea trial results compared with the tank prediction it should be remembered that the trial speeds given are uncorrected for wind and tide; it should also be noted that the model test trial condition does not correspond exactly with the ship trial condition. The owners state that the final trial speed curves correspond exactly with the model test prediction in the trial condition. It is claimed that the guarantee speed was exceeded by more than a knot during the sea trials.—*Shipping World and Shipbuilder, July 1968, Vol. 161, pp. 1086–1089.*

Service Results Obtained with the PC-type Engine

Results have been obtained in service on some hundred S.E.M.T.-Pielstick PC-engined ships belonging to about 20 owners. Figures given are mean values, between which individual results lie with the usual dispersion.

The information given also takes into account experience acquired by a set of engines which to date have totalled eight million hours service, including two and a half million hours on heavy fuel.

The specific fuel consumption of the PC engines as a function of the power per cylinder and of the revolutions is given in Fig. 1. These results apply to an engine with an oil pump driven by the engine and to a fuel having a calorific value of 10 100 kcal/kg.

The majority of ships operate traditionally at continuous ratings of from 85 to 90 per cent of maximum power. Under these conditions, the specific fuel consumption is around 150 gal/bhp-h. Where the water pumps (sea water or raw water pump, fresh water pump) are also driven by the engine, the consumption is increased by about four gal/bhp-h.

Fig. 1 shows that the operating zone, which can be covered by maintaining full throttle, extends from full speed down to about 60 per cent of full speed. For a power plant with two engines on each shaft, it is, therefore, possible by operating with a single engine, to maintain a specific consumption of the same order as that for full speed down to below half the ship's speed.

A comparison of the costs of the various fuels which can be used in Diesel engines reveals such an advantage in the use of the heavy fuels that S.E.M.T. has been working since 1953 to solve the problems raised by their use in the PC-type engines.

Heavy fuel means those fuels which ships can take on in

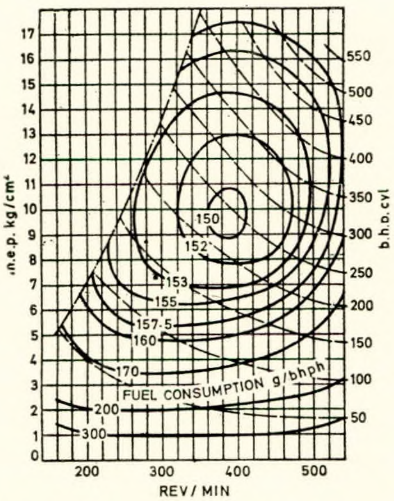


FIG. 1—*Specific fuel consumption of the PC-type engine as a function of output/cylinder and rev/min*

all ports in the world. However, S.E.M.T. recommends fuels with characteristics not exceeding:

Viscosity ...	3500 sec Redwood No. 1 at 100°F
Sulphur content	3·5 per cent
Conradson index	12 per cent

As a result of systematic trials, S.E.M.T. has solved the problems presented by the use of these fuels.

Experience with over 150 PC-type engines operating on heavy fuel, some of which have exceeded 35 000 hours service, has shown that fears of rapid contamination of the lubricating oil of trunk-piston engines by the acid combustion products were not justified. This is proved by the total absence of corrosion on all the engines in service, and is due to the value of excess combustion air, to the quality of combustion in PC engines, to the good sealing of the rings and to the progress made by lubricating oil manufacturers, whose products, which include additives giving the oil a high base number, ensure neutralization of the acids which may be formed at the level of the rings. These oils also contain detergent dopes which disperse the deposits which may form during combustion, and thus influence the wear of the liners.

Long-term maintenance of the lubricating oil quality enables the same charge of oil to be retained indefinitely: only topping-up is required to maintain the required quantity. Obviously, oil changes necessitated by an accident, or renewals which may be decided on at the time of important work dictated by the engine maintenance schedules, are not taken into account.

Trials carried out with PC 1 engines showed that residual fuels led to much greater piston groove wear than when operating with Diesel oil. A study of the condition of the grooves showed that there was an erosion caused by particles of ash imprisoned between the ring and the groove, this action no doubt being complemented by a chemical attack. The corresponding faces of the rings, although they remained bright, also showed considerable wear.

Following the studies made, new arrangements enabled a highly satisfactory result to be obtained. The wear of the grooves was reduced in a proportion of approximately 10:1 and is thus much less than was previously found on pistons without ring carriers used for operation with Diesel oil.

Operating with residual fuel quickly showed burning of the valve seats after running times varying from 500 to 1000 h, whereas when operating on Diesel oil, the valves were removed only after 5000 or even 10 000 h.

To improve the life of exhaust valves, S.E.M.T. has adopted the following measures:

1) rotation of exhaust valves during operation;
2) use of face hardenings with better resistance to corrosion by vanadium and sodium salts;
3) reducing the intensity of encrustation with ash.

These measures are perfectly satisfactory for those fuels which do not have high vanadium and sodium contents.

For fuels which do have high vanadium and sodium contents, it was necessary to try to lower the temperature in the region of contact between seat and valve so as to prevent the fusion of the vanadium and sodium salts:

a) by reducing the temperature of the valve seats;
b) by cooling the valves themselves.

The first of these aims was achieved by increasing the excess of scavenging air and increasing the opening speed of the exhaust valve, which brings about a reduction in heating of the exhaust valves at the time of opening.

The second aim (use of cooled valves) necessitated extensive technological studies which resulted in the development of a valve, a drawing of which is shown in Fig. 2. It has been proved that cooled valves can be maintained in position for 6000 h.—*Motor Ship, Special Survey, February 1969, Vol. 49, pp. 50–52.*

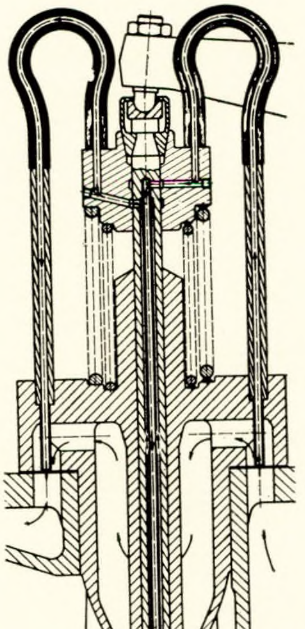


FIG. 2—*Sketch showing the method adopted for cooling the exhaust valves*

Ship Model Correlation Procedure

The present work was carried out in order to investigate the reasons for, and as far as possible to remove, the uncertainty in the correlation of resistance and propulsion test results to the full-sized ship. The measurement results on which this work is based were obtained through the evaluation of extensive data.

The obvious first step was to compare the results produced by the most usual methods of calculating the correlation of model tests to a full-sized ship with the results of actual measurements made on board. The result of this investigation showed that the methods of correlation used at present are unsatisfactory. This gave the impetus for the conception and working out of new and better methods for:

a) the correlation function of the viscous resistance;
b) the correlation function of the wake;
c) the basis for the correlation of the thrust deduction fraction.

To discover correlation functions between model and ship test results, it was necessary to bring the results used on to some basis which would enable meaningful comparisons to be made. For this purpose corrections are required to the test results of both model and full-sized ships. These are treated separately in the original work in clearly demarcated sections. Thus, the resistance corrections for the model test results and the corrections required in consequence of a slight change in propeller pitch are given. These corrections, applied separately, influence the initial data of the model tests, such as efficiency factors, rates of revolution, thrust deduction fractions etc. The change in efficiency, the change of rate of revolution and the change in thrust deduction fraction as a function of propeller load are also treated in the unabridged work. A summarized description is also given of the use of all corrections mentioned up to now in the most usual methods of correlating model test results to the full-sized ships (Froude, Schoenherr, I.T.T.C. and Schulz-Grunow). The slight changes in the full-sized ship results required by the additional resistance due to wind and shallow water, if these are to be taken into account, are also considered in the original work.

Following from this preliminary work, a new function for the correlation of the specific viscous resistance co-efficient of the model to the ship has been developed and is presented in the form of correlation lines. This correlation function is dependent on the ship's form. Furthermore, the wake correlation function between model and ship has been developed. The correctness of the wake correlation function is tested by evaluating the thrust measurements. The development of this correlation function became possible only after the correlation lines for the specific viscous resistance coefficient had been found.

In addition, the thrust measurements made it possible to work out the basis for the correlation of the thrust deduction fraction. The correlation from model to ship of the thrust/torque relationship proved to be independent of scale. The effect of scale on the relative rotative efficiency has also been referred to briefly.—*Chirila, J., Shipping World and Shipbuilder, February 1969, Vol. 162, pp. 323–328.*

Marine Gas Turbine

The gas turbine benefits from two inherent advantages:

1) the ability to produce very great power for low weight;
2) a low requirement for supervision and on-board maintenance.

In addition, gas turbine development is financed by the aircraft industry, and marine versions are redesigns of aero-engine forms.

Major disadvantages are a high fuel consumption and the inability to use residual fuels due to blade corrosion. In the short term, these problems are considered likely to inhibit the general employment of the gas turbine for marine propulsion, but three factors will contribute to its success in the more distant future:

1) the cost of residual fuel. At present, marine operators are only paying a price representing recovery costs, as most deep-sea vessels bunker near to oil sources. As recovery costs increase, the extra cost of distillate fuels will become less marked. Currently, fuel for a gas turbine costs 50 per cent more than that for a heavy fuel engine. This differential may well drop to about 20 per cent within 10 years.
2) the decreasing importance of fuel costs as a fraction of total operating costs. This will only apply to certain classes of vessel, but container ships would benefit by the higher speeds offered. Engine replacement could be achieved in the course of a normal discharge loading period.
3) gas turbine development. The specific problems are protection against sea salt in both fuel and air, fuel cleaning, higher tolerance of blading materials to extreme temperatures and increased thermal efficiency. Work is progressing on all these problems and improvements can be expected in the fairly near future.

Recent experiments on protoype aero-engines have shown that an increase in cycle temperature from 1600 to 1850°F can result in a 32 per cent increase in specific power and a 15 per cent improvement in fuel consumption. Maximum gas temperatures of 2250°F have also been demonstrated, so that great improvements in marine gas turbines may be anticipated.—*Motor Ship, January 1969, Vol. 49, p. 459.*

Giant Oil Tanker

An oil tanker of 312 000 dwt has been built at the Nagasaki Shipyard and Engine Works of Mitsubishi Heavy Industries Ltd, for the Bantry Transportation Co. of Liberia, an affiliate of National Bulk Carriers Inc., U.S.A.

A unique feature of the steering control system is the provision for the twin rudders to be angled outboard from 0–8° by the officer on watch, which increases the vessel's steering ability at slow speeds. The Bech reverse spiral test was used to establish whether the ship was stable or unstable in her steering characteristics.

Principal particulars are:

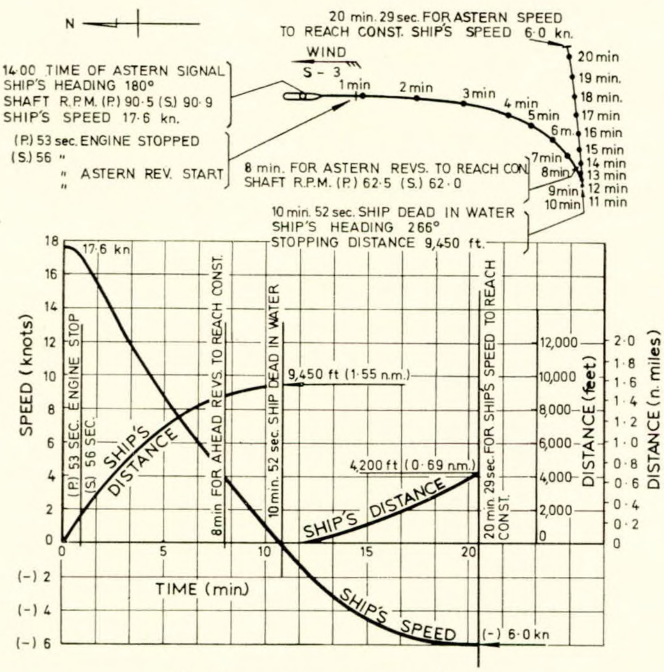
Length, o.a.	1135 ft 2 in
Length, b.p.	1082 ft 8 in
Breadth, moulded	174 ft 10 in
Depth, moulded	105 ft 0 in
Draught, extreme	79 ft $2\frac{1}{8}$ in
Freeboard	25 ft 11 in
Deadweight	312 000 tons
Lightweight	48 963 tons
Gross tonnage	149 608·58
Cargo capacity:	
oil	399 630 m^3
water ballast	46 828 m^3
fuel oil	14 479 m^3
Machinery output:	
m.c.r.	37 400 hp at 93 rev/min
n.c.r.	34 000 hp at 90 rev/min
Service speed, average	16·3 knots

The propelling machinery in *Universe Kuwait* consists of two sets of I.H.I.-General Electric type, cross-compound, steam turbines with reduction gearing, each having a normal output of 17 000 hp at 90 rev/min, and a maximum continuous rating of 18 700 hp at 93 rev/min. The rotating parts of these turbines were made by General Electric of U.S.A., and the casings, pipework and final assembly work were carried out by I.H.I.

Steam is generated in two I.H.I.-Foster Wheeler ESD type two-drum watertube boilers, each having a normal evaporation of 125 660 lb/h and a maximum of 165 350 lb/h. The steam conditions at the superheater outlet at normal evaporation are 780 lb/in^2 and 959°F (515°C).

This ship exhibited very good, stable steering characteristics; undoubtedly due, in large measure, to the twin-screw twin-rudder arrangement adopted in the design. The vessel also has excellent stopping characteristics. In a crash stop astern test, the ship was stopped in the water in just under

11 minutes from an initial speed of 17·6 knots. Even for the ballast condition this is a remarkable performance for a ship of this size. As can be seen from the plot of the vessel's position for successive minutes after the astern order was



Crash stop astern test of giant oil tanker in ballast

given, the ship tended to veer to starboard. This veering, in an arbitrary manner, is not uncommon, and once the propeller is going astern, the ship's course cannot be influenced by rudder angle changes. This is not so critical in a twin-screw where it may be assumed that a certain amount of manoeuvring can be accomplished using the propellers. This would probably be achieved at the expense of overall stopping distance.—*Shipping World and Shipbuilder, November 1968, Vol. 161, pp. 1746–1751.*

Two-part Shipbuilding Method

A new method for building very large ships has been developed by the Nederlandsche Dok en Scheepbouw Maatschappij (NDSM), Amsterdam. The main feature of the method is that the ship is built in two parts, which are launched separately, then joined while afloat by welding from inside a caisson.

To obtain a good fit when joining the two parts, requirements which must be met are:

1) all butts of the division must be situated in a flat plane, which is perpendicular to the longitudinal axis of the ship;
2) cross sections of both the fore and after parts of the ship must be identically equal; this means that corresponding linear dimensions must be equal and that corresponding angles must be equal.

A pentagonal prism was the most important instrument used with the measuring technique chosen. It has the property to reflect light at a 90° angle with very great accuracy. The light beam which entered the prism was aligned parallel to the longitudinal axis of the ship. By rotating the prism around an axis, which corresponds with the centreline of the entering light beam, the reflected light beam describes a plane which is perpendicular to the ship's axis. This reference plane was situated about 1½ ft away from the hull in order to clear all constructional parts and staging. Marking off was done by measuring back from the reference plane at a large number of points.

As a source of light, a simple laser was used, because of two important advantages it has to offer:

1) a very narrow beam of great intensity and definition which is easily visible at night;
2) monochromatic light by which it is possible to obtain by placing a finely slotted plate in the beam, an interference pattern consisting of concentric circles: thus it is possible to locate the centreline of the beam very accurately.

The laser which was used had an output of only 0·3 mW. It was placed on deck some 300 ft from the joint and above eye level to eliminate any danger of eye damage, which is always a possibility when a laser is used and when one can look directly at the source of light. To check the alignment of laser and prism, sights which also showed deflexions due to temperature differences were placed between them. The sights were aligned at night, because temperature differences are smallest then, and for the same reason marking off was also done at night. In the daytime deflexions of several millimetres were found.

The measuring technique described above, allowed marking off to be carried out with an accuracy of ±1 mm. Cutting to size was done with the utmost care to maintain the required accuracy. To obtain the best possible results, all important linear dimensions in the divisional plane were measured twice, using steel tapes. Both the shipyard's personnel and the consultants took measurements. As each group used different reference points, the results were totally independent. For all measurements the same steel tapes were used so as to eliminate as far as possible inaccuracies due to differences in temperature and humidity. This is rather important in view of the time lag of two to three months between measuring the fore and the after parts of the ship.—*Shipping World and Shipbuilder, February 1969, Vol. 162, pp. 333–339.*

Computers in Shipbuilding

With the development of computers, new fields are being exploited for optimum planning and labour saving. Nippon Kokan is presently applying a series of system programmes upon inquiries from customers, as a function of sales activities, in the following manner:

1) In order that an early offer can be made to inquiries, data of the ship's type and dimensions are first applied to prepared standard ship designs to minimize input data (Nippon Kokan has developed various economical types ranging from 20 000 to 350 000 dwt).
2) Under a preliminary engineering programme, calculations are made for preliminary lines, hydrostatic data, capacity, trim, stability and longitudinal strength, and results analysed.
3) Concurrently, midships scantling and weight estimation are computed.
4) The approximate cost estimation and speed calculation are carried out in parallel.
5) Operational economy of the ship is also computed and the calculations of the above system programme are iterated for several cases. Comparative analysis of the results is then made.
6) The optimum type and dimensions, specifications (including data of the machinery department), arrangement and hull form are decided.
7) If necessary, the computer programme of the preliminary engineering calculation is repeated and with minor adjustments, a final decision is made on the optimum type of ship. The offer is then made on the basis of this decision.

8) When the order becomes definite, fairing is partially made to the preliminary lines and detailed drawings are then developed.

In this manner the optimum type of ship satisfying the requirements of the customer is scientifically determined in a short time, allowing the deployment of more manpower capacity to other work such as research and development including extending the scope of standardized components, etc.—*Toyama, K., Shipbuilding International, December 1968, Vol. 11, pp. 40–44.*

"Oiling" the Waters

The addition of trace amounts of some long-chain polymers to water has been shown to reduce, dramatically, hydrodynamic skin friction under conditions of turbulent flow. This phenomenon is potentially of great interest to the Navy where the short-term bursts of extra speed which could be obtained by its use from existing propulsion machinery represent an attractive tactical advantage.

The mechanism whereby this reduction in skin friction occurs is not fully understood, but laboratory tests have shown that the polymer must be soluble in water. The structure should be linear, with a minimum of side branching and the molecular weight should be greater than 50 000 and preferably in excess of one million. Polyethylene oxide and polyacrylamide can have these properties and are both very effective drag-reducing agents. The investigation of the basic physical and chemical properties of the polymer solution to the boundary layer of a ship or weapon is under active consideration at the Admiralty Materials Laboratory, Holton Heath, Poole, Dorset.

Several factors besides the chemical structure are important when selecting the polymer; these include particle size, packing fraction, rate of solution, viscosity and stability of solution. Equipment has been developed to measure these parameters as well as the drag reduction obtained from polymer solutions.

Towing tank tests with models have demonstrated that the skin friction resistance of a hull can be reduced by 30 per cent when a dilute solution of polyethylene oxide (MW 4×10^6) is ejected from the model. This result has been sufficiently encouraging to justify a full-scale ship trial and AML is collaborating with Admiralty Experiment Works, Haslar, on this project. H.M.S. *Highburton*, a coastal minesweeper, has been chosen for these trials. Because it is impossible to store on board a sufficient volume of dilute solution for an effective trial, a device has been developed to prepare solutions continuously from stored polymer powder and sea water and to eject this at controlled rates and concentrations from slats sited at different positions around the hull. By this means it is hoped to determine the optimum conditions in which these polymers may be used to improve a ship's performance.—*Marine Engineer and Naval Architect, January 1969, Vol. 92, p. 11.*

Ship Surgery at Hong Kong

A major example of ship surgery was recently completed by the Taikoo Dockyard and Engineering Co. of Hong Kong Ltd., for the Union Steamship Co. of New Zealand Ltd.

The Henry Robb-built general cargo vessel *Poolta* was the subject of the surgery which has resulted in the ship being increased in deadweight by 1000 tons by the addition of a 60-ft long 'midship section. At the same time, *Poolta* has been extensively re-equipped to suit her for a new rôle as a subsidiary container vessel to work with the two new roll-on/roll-off vessels now building at Caledon, Dundee, in the Sydney–Hobart trade.

The ship was slipped at the Taikoo repair yard and the fore end was chocked-up with heavy timbers, transferring the weight of the forward section to the surrounding hard. The ship was then cut just forward of the 'midships superstructure and the aft section, still resting on the slipway cradle, was pulled away from the fore section a distance of 60 ft. A new section of the same length was then erected in prefabricated units and welded in place.

Flume stabilizer tanks were incorporated in the new section and MacGregor hatch covers were fabricated and fitted to the new 56 ft $\times$ 30 ft 6 in hatch.

When completed in 1959, the deadweight of *Poolta* was 2200 tons but the new section has enabled the deadweight to be increased to approximately 3183 tons.—*Motor Ship, January 1969, Vol. 50, p. 521.*

Shaft Alignment System for Two-part Ship Construction

In conventional ship construction procedures, the propeller shafting is aligned before installation of the main engine, after completion of the aftermost section of the hull structure. This method is still widely employed. However, at the Chiba shipyard of the Mitsui Shipbuilding and Engineering Co. Ltd., the fore and aft sections of a ship are separately built and completed in dry dock and later joined to make the finished ship. Because the ship is virtually completed in dry dock there is no need for the vessel to undergo the usual time-consuming and expensive bottom-cleaning procedure.

To retain this advantage of a two-part hull construction system, it is necessary for the shafting and main engine to be installed and aligned during construction of the aftermost section.

The first step is to mark out the position of the main engine holding-down bolts; the correct distance is taken from installation drawings and the positions are marked after setting up the ship's bottom. At the same time the horizontal distortion of the main engine foundation is measured with a water level gauge.

The main engine foundation chock liners are then machined with a portable miller; the bed plate, columns and cylinder blocks are then set up in their required positions. It is interesting to note that this procedure is undertaken at an early stage after keel laying, usually about one month later.

The intermediate shafting, propeller shaft, plummer blocks and thrust bearing are brought on board and ranged alongside the shaft line for convenience of sighting before the bedplate is installed.

A base point is set up at the forward end of the engine crankshaft in line with the bored-out main journals of the main engine. (It should here be mentioned that the engines referred to in this article are generally of the Mitsui-built B. and W. VT2BF design which has suitable main pin borings: the new B. and W. KEF and KFF machines do not normally have hollow pins but Mitsui proposes to bore a 75 mm-diameter hole in the pins of its versions of this engine type so as to follow the yard's established alignment procedure.)

After setting up, the forward base point is adjusted for height relative to the keel and the distance from the ship's sides, in the usual manner. The aftermost base point is also set up by conventional means using an electric light sighting source. Two intermediate slits are arranged at the stern frame boss forward surface and at the after end of the engine crankshaft. Following the setting up of the base points, piano wire of 0·6 mm-diameter is stretched tightly between both slits and the height and centre of the thrust bearing, the line shaft bearings and the main engine foundations are checked and confirmed. The aftermost sighting slit is located at the outboard end of the stern frame and a 500 W light source is provided for optical sighting; this enables the stern boss fore and aft centres to be determined and for the machining marks to be scribed on the boss surfaces.—*Motor Ship, December 1968, Japanese Shipbuilding Supplement, p. 63.*

Wave Patterns on the Surface of Hydrodynamic Cavities

In experiments on cavities behind various axisymmetric headforms, a pattern of waves or ripples with crests parallel to the separation line was observed on the cavity surface just downstream of separation. A theoretical analysis suggests that this pattern results from amplified instabilities in the separated laminar boundary layer on the cavity surface.—*Brennen, C., N.Ph.L. Ship Division, August 1968, Rep. No. 121.*

Mechanical Retrieval of Waste Oils and Solids from Water

This paper discusses the application of rotating cylinders for the recovery of oil spills from inland waterways and protected harbours, and for the retrieval of waste oils from waste water. It also discusses the general design details of permanent, fire-retardant, oil retention floating booms for marine and industrial applications.—*Clyne, R. W., Lubrication Engineering, November 1968, Vol. 24, pp. 514–520.*

Video Tape Recording of Ultrasonic Test Information

A video tape recorder has been converted into a wide band instrumentation recorder. The "A" scan from the ultrasonic tester is directly recorded together with the operator's voice giving the location, transducer position and interpretation of test data. An oscilloscope is used for the playback. The circuitry necessary to couple the output of the ultrasonic tester to the tape recorder is described.—*October 1968, Ship Structure Committee Rep. SSC-189.*

SG Iron Casting Examined from the Foundry and Material Aspects

The production of a simple machine component is used to illustrate the advantages yielded by close collaboration between designer, stress analysis and fatigue expert, metallurgist and foundryman. In addition the path is mapped out from a given design to the finished casting. The knowledge gained on deformation capacity and structure shows that a slightly undercoolable iron has inferior deformability to a stable iron.—*Michels, G. and Modl, E. K., Sulzer Technical Review, 1968, No. 3, pp. 133–140.*

Thermal Ratchet Mechanism

Thermal ratcheting is the cyclic growth of a structure that is subjected to successive cycles of heating and cooling. The mechanism of thermal ratcheting in a simple two-element assembly is investigated. The conditions required for non-ratcheting captive thermal cycling and for ratcheting cyclic growth are defined and a method is developed for computing the magnitude of the plastic strains and the cyclic growth.—*Burgreen, D., Trans. A.S.M.E. Jnl Basic Engineering, September 1968, Vol. 90, pp. 319–324.*

Evaporation of Water Droplets in Superheated Steam

There are many circumstances in steam power engineering in which water drops exist temporarily within a superheated steam atmosphere. It is sometimes of importance to know the probable lifetime of a drop, especially if its continued existence presents a threat, as for example, to rotary machinery in its future path. The experimental procedure consisted essentially of suspending the test drop in a fine horizontal glass fibre within a steam flow of controlled velocity, pressure and superheat and observing the rate of evaporation.—*Lee, K. and Ryley, D. J., Trans. A.S.M.E. Jnl Heat Transfer, November 1968, Vol. 90, pp. 445–451.*

Theoretical Investigation of Heat Pipes Operating at Low Vapour Pressures

A one-dimensional analysis of a compressible vapour flowing within the evaporator section of a heat pipe is presented. Comparisons between the theoretical results and existing heat pipe data show that the presence of gas-dynamic choking can limit the heat transfer capacity of a heat pipe operating at sufficiently low vapour pressures.—*Levy, E. K., Trans. A.S.M.E. Jnl Engineering for Industry, November 1968, Vol. 90, pp. 547–552.*

Ocean Engineering

This lecture is in two parts. The first points out the need for teaching and research in ocean engineering and describes briefly a scheme that has been set up in an attempt to meet a part of the country's needs. The second examines some of the wider problems of British industrial involvement in marine technology. The obstacles to progress are substantial and some suggestions are offered on overcoming them.—*Bishop, R. E. D., Forty-first Thomas Lowe Gray Lecture, 1969, I.Mech.E. Paper No. L4/69.*

Evaluation of Heat Transfer in Dry Cargo Ship's Tank

Little is known about the heat losses from double bottom tanks. A test was carried out on such a tank of an 8000 dwt freighter. The tanks, loaded with low-viscosity fuel oil, were heated by means of thermal oil. The test yielded much valuable information, including heat transfer coefficients for all tank areas and for the heating coil.—*Van Der Heeden, D. J., International Shipbuilding Progress, January 1969, Vol. 16, pp. 27–37.*

Model Tests on Contra-rotating Propellers

This paper presents the results of open water tests with a systematic series of contra-rotating propellers, consisting of a four-bladed forward screw and a five-bladed aft screw. Based on the open water test results, contra-rotating propeller systems were designed for a tanker and a cargo liner. Comparative tests have been carried out with the tanker and the cargo liner both equipped with contra-rotating propellers and with conventional screws.—*Van Manen, J. D. and Oosterveld, M. W. C., International Shipbuilding Progress, December 1968, Vol. 15, pp. 401–417.*

Testing of Weldmetal from the Aspects of Brittle Fracture Initiation

A method of testing weldmetal for its sensitivity to brittle crack initiation is described. The method is based upon considerations derived from experience to date and feasibility by industrial laboratories.—*Van den Blink, W. P. and Nibbering, J. J. W., International Shipbuilding Progress, January 1969, Vol. 16, pp. 3–13.*

Turbine Flow Problems of Binary Cycles Employing High Density Fluids

High density working fluids such as the fluoro-carbons can be used with advantage at the low temperature end of power generation cycles. Two applications are examined:

1) a gas turbine binary cycle, which offers a compact plant of good efficiency for moderate load factors only;
2) a steam turbine binary cycle.

—*Horn, G., Norris, T. D. and Whybrow, J. F. T., 1969, I.Mech.E. Paper No. P8/69.*

Application of Pulse Converters to Four-stroke Diesel Engines with Exhaust-gas Turbocharging

A simple form of pulse converter has become widespread in recent years, chiefly for eight and 16-cylinder engines. The development of the pulse converter in its present form is briefly described and examples are illustrated. The principle of operation is explained with the aid of curves of pressure in the cylinders and exhaust ducting of various engines and under different operating conditions.—*Meier, E., Brown Boveri Review, 1968, Vol. 55, No. 8, pp. 420–428.*

Computer Programme for the Gas Exchange Process in Turbocharged Diesel Engines

Numerous problems concerning the turbocharging of Diesel engines can be solved with step-by-step calculations of the gas exchange, viewed as a quasi steady-state process. With the aid of electronic computers such calculations can be made quickly and reliably. The theoretical and experimental principles behind a straightforward computer programme are summarized and the important question of evaluation is discussed.—*Ryti, M., Brown Boveri Review, 1968, Vol. 55, No. 8, pp. 429–439.*

Effect of Reynolds Number and Clearance in the Centrifugal Compressor of a Turbocharger

Measurements on turbochargers and experiments using special-purpose test rigs are constantly yielding extensive information on turbocharger compressors as built by the authors' firm. This brief article deals with a test rig suitable for fundamental, detailed investigations. A closed flow circuit makes it possible to alter the Reynolds number independently of the Mach number.—*Schmidt-Theuner, P. and Mattern, J., Brown Boveri Review, 1968, Vol. 55, No. 8, pp. 453–460.*

Practical Approaches to Ship Vibration

This paper considers the problems of ship vibration from the angle of a practising engineer and covers topics on hull vibration, stern slamming, torsional vibration, vibrations of equipment on board and instrumentation as well as vibration standards. Typical examples are also given to illustrate the problems involved. It is suggested that further investigations should be carried out on the slamming of the stern of trawlers.—*Ojak, W., European Shipbuilding, 1968, Vol. 17, No. 5, pp. 68–83.*

Research on the Hydraulic Coupling

A short account of the origin of the hydraulic coupling is followed by a description of a production model and its method of operation. Tests have been carried out both on a transparent plastic model and on the full-scale production coupling. From the author's experience of the design and application of hydraulic couplings, a mathematical analysis which uses conventional design formulae and general rotodynamic theory is made. Dimensionless parameters and empirical laws are established for the prediction of design performance.—*Rolfe, G. H., 1968, I.Mech.E., Paper No. P12/69.*

Experimental and Analytical Investigation of the Gas Exchange Process in a Multi-cylinder Pressure-charged Two-stroke Engine

The results are given of a comprehensive investigation of the pressure and temperature diagrams in a multi-cylinder two-stroke engine during the gas exchange process. A six-cylinder turbocharged two-stroke Diesel engine was fully instrumented to record transient pressures in three cylinders, in the scavenge belt and in a number of positions in a specially designed exhaust pipe. Transient temperature records were taken in the exhaust pipe.—*Benson, R. S. and Galloway, K., 12th February 1969, I.Mech.E., Paper No. P14/69.*

Current Instrumentation Development and its Effects on Engineering Practice

This paper describes the development of indicating and controlling instruments from manually operated systems. The use of feedback in instruments with its attendant improvement in frequency response and reduction in load imposed upon the system being measured is illustrated with examples of mechanical and pneumatic measurement systems. The advantages gained by the employment of electronic amplifiers are outlined.—*Brown, J. and Reid, D., 10th December 1968, I.E.S. Paper.*

Analysis of Slamming Phenomena on a Model of a Cargo Ship in Irregular Waves

Model tests in irregular head waves were analysed with special emphasis on the slamming phenomena endured by a cargo ship in the ballast condition. The conditions leading to slamming were evaluated. By means of theoretical computations of pitching and heaving motions and their phase angles in regular waves, relative motions and relative velocities could be evaluated at any station aft of the forward perpendicular. —*Ferdinande, V., International Shipbuilding Progress, November 1968, Vol. 15, pp. 373–387.*

Experimental Study of Constant Pressure Turbocharging

In this paper the authors describe several scavenging air control systems intended to improve performance at low load in the case of a constant pressure system. Information is also given concerning an experiment and its analysis concerned with the use of a constant pressure system in parallel with an injector system as applied recently to Kawasaki-M.A.N. KZ engines.—*Hashimoto, H. and Kobayashi, H., Jnl Mar. Engineering Soc. Japan, 1968, Vol. 3, No. 6, pp. 391–398.*

Control Systems for Magnetic Bearings

Magnetic bearings with control circuits are very attractive because of their frictionless characteristics, but analytical research carried out in this field has so far been unsatisfactory. The authors carried out a theoretical and experimental study of the problem. First a mathematical model was constructed by means of linearizing procedures, next, the root-loci were calculated on a high speed computer and the results were then compared with experimental data obtained on an actual model.—*Shimizu, H. and Taniguchi, O., Bulletin Japan Soc.Mech.E., 1968, Vol. 11, No. 46, pp. 699–705.*

Technological Progress in Water Pollution Abatement

As a result of recent intensified local and national emphasis on environmental pollution abatement, mounting attention is being directed toward controlling ship and water craft wastes. Legislative trend at every level of government clearly points towards eventual adoption of strict standards for all classes of ship and water craft. The Secretary of the U.S. Navy has required that pollution be controlled by the operation of naval ships. A comprehensive programme to control the wastes discharged from ships has been initiated.—*Singerman, H. and Kinney, E. T., Naval Engineers Jnl, October 1968, Vol. 80, pp. 795–801.*

Design and Fabrication of Deep-diving Submersible Pressure Hulls

This paper describes design requirements and fabrication procedures used in pressure-hull construction of the Deep Quest (DQ) and the Deep Submergence Rescue Vehicle (DSRV). The bisphere pressure hull for DQ utilized 18 per cent nickel maraging steel at a yield strength level of 175 kg/in^2, while the trisphere pressure hull required by DSRV was fabricated from HY-140 at 140 kg/in^2 yield strength.—*Garland, C., 13th–16th November 1968, S.N.A.M.E. Paper No. 5.*

Qualification of Gas Turbine Engines for U.S. Navy Shipboard Use

The author emphasizes the need for thorough proof testing of gas turbine engines prior to shipboard use. He describes some of the gas turbine testing programmes in which the U.S. Navy has been involved and the type of testing that will be required in the future. The need for proof testing of some kind has always been recognized. In many instances, lack of adequate funds, timing and other exigencies have forced the U.S. Navy to use engines without an adequate testing programme.—*Carleton, R. C., Trans. A.S.M.E. Jnl Engineering for Power, October 1968, Vol. 90, pp. 361–368.*

Thermal Cycles and Straining Effects in a Multi-pass Butt Weld

This report gives a description of the welding sequence, welding conditions and temperature distributions involved in the fabrication of a standard B.W.R.A. wide plate test specimen. In addition, the results of microhardness measurements and COD tests on small scale specimens extracted from the welded joint at notches are compared with those for notched plain plate specimens giving hot straining treatments to simulate the conditions existing at the notch tip in a wide plate test.—*Dawes, M. G., British Welding Jnl, November 1968, Vol. 15, pp. 563–570.*

Progress in the Development of Gas-dynamic Pressure-wave Machines for Pressure-charging Diesel Engines

Brown Boveri are developing a device for pressure-charging Diesel engines which operates on the principle of a gas-dynamic pressure-wave machine. Measurements with a pressure-wave machine and a mechanically driven charger on a Diesel engine have shown that with the pressure-wave machine, just as with the mechanically driven charger, increasing engine torque can be achieved over a wide range as the engine speed falls. Specific fuel consumption with the pressure-wave machine is better than with the mechanically driven charger.—*Wunsche, A., Brown Boveri Review, 1968, Vol. 55, No. 8, pp. 440–447.*

Pulse Converters on Two-stroke Diesel Engines

When two-stroke engines are pressure charged using the pulse system, higher mean pressures require methods of charging which permit continuous admission to the exhaust gas turbine. Possible ways of creating optimum charging conditions by means of suitable exhaust ducting are discussed. The chief factors to be considered are discussed and the practical benefits are illustrated by results obtained with an operational example.—*Zehnder, G., Brown Boveri Review, Vol. 55, 1968, No. 8, pp. 414–419.*

Medium-speed Diesel Engine Noise

This paper reviews past work on Diesel engine noise with particular reference to medium speed engines. A survey carried out on a large number of British and European turbo-charged and water cooled Diesels having bores in the range 2¾ in to 33 in clearly indicated at least two main sources. Qualitative analysis suggested these to be due to combustion and piston transverse motion in the liner clearance space.—*Bertodo, R. and Worsfold, J. H., 4th December 1968, I.Mech.E. Paper No. P6/69.*

Patent Specifications

Deck Structure of Ships with Very Wide Hatches

This invention relates to the deck structure of ships with very wide hatches. Fig. 1 shows a ship's hull structure (11) having a very large hatch opening (12) and narrow side decks (13). These side decks are of cellular box construction, each having a lower plated deck (14), an upper plated deck (15) and intervening longitudinal girders (16).

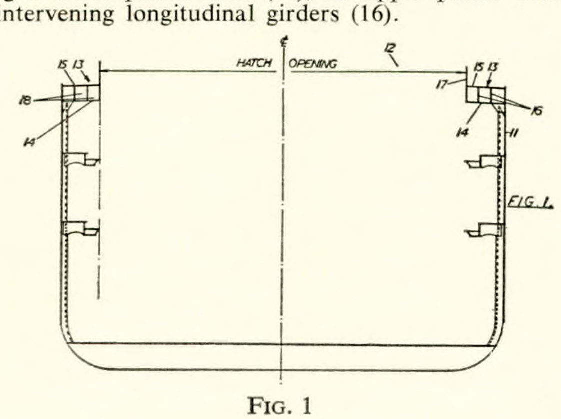


FIG. 1

The distance of the lower one (14) of the double decks (14), from (15) above is equal to the normal working clearance

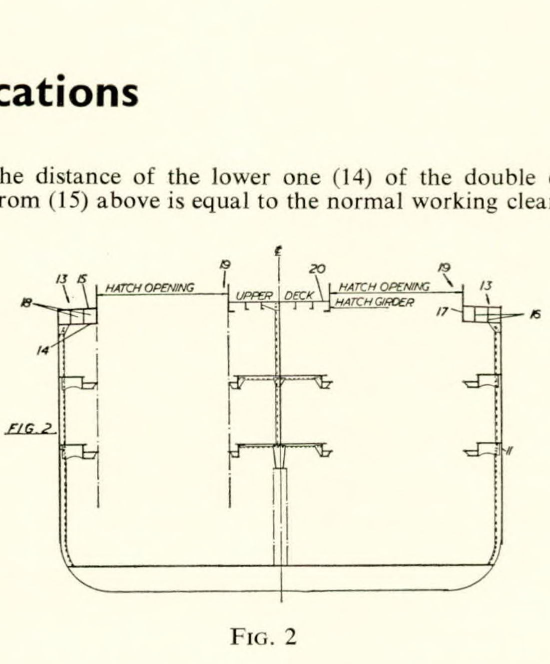


FIG. 2

height that would be present if a conventional single deck were fitted instead of the double deck. The hatch coaming (17) extends down to the bottom of the lower deck (14) and forms an inboard member of the deck box structure. The spaces (18) within the box deck structure may be arranged to carry water ballast.

In Fig. 2, the structure is similar except that there are two hatch openings (19) on either side of the centreline and a central upper deck section (20) of conventional construction between them.—*British Patent No. 1 138 691 issued to Burness, Corlett and Partners Ltd. Complete specification published 1st January 1969.*

Control System for Twin-engine Marine Power Unit with Single-screw Shafting

The power unit in Fig. 1 comprises two engines (A) and (B) connected by couplings (E_A) and (E_B) to a reduction gear (Re) and rotating a single shafting (L) to which a screw is connected.

Fig.2

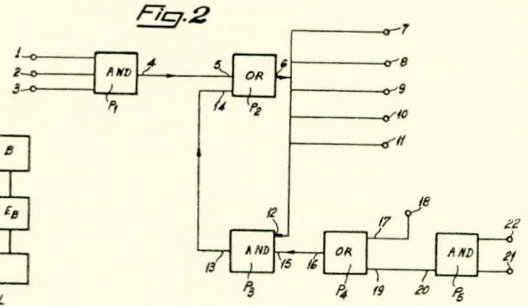
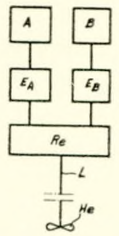
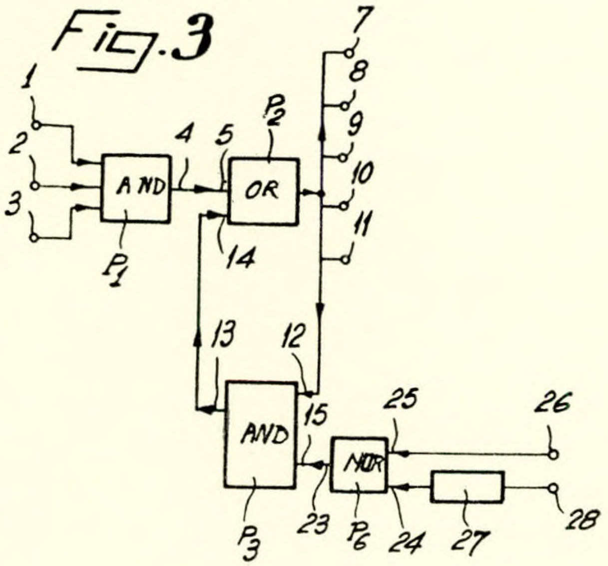


Fig.1



When the vessel is cruising normally, the two engines rotate in the same direction and drive the screw (He). When the ship is manoeuvring, the two engines (A) and (B) rotate in opposite directions, so that engagement or disengagement of the couplings (E_A) and (E_B) enables the screw to be rotated in either direction.

Fig.3



If the switch is in the manoeuvring direction and one of the engines breaks down, it is dangerous to remain in that position, since the screw can then be driven in only one direction. The system according to the invention as shown in Figs 2 and 3 obviates these drawbacks. They show the system in the form of "AND", "OR", "NOR" logic circuit diagrams or gates. An "AND" gate gives a signal at its output only if all its inputs are fed and such a gate corresponding to a group of relays of which all the operating contacts are in series. Similarly, an "OR" gate gives a signal at its output if any one of its inputs is fed and corresponds to a group of relays whose operating contacts are all in parallel. A "NOR" gate gives an output signal if none of its inputs is fed and corresponds to a group of relays whose non-operating contacts are all in series.— *British Patent No. 1 139 025 issued to Société Financière et Industrielle des Ateliers et Chantiers de Bretagne. Complete specification published 8th January 1969.*

Bows of Ships

This invention relates to a bow for ships having a forwardly projecting bulb or bulge. According to Fig. 1, the underwater stem portion (16) has underneath the most forward point (20) a straight or substantially straight portion (23), forming an angle of 20° to 40° with the vertical.

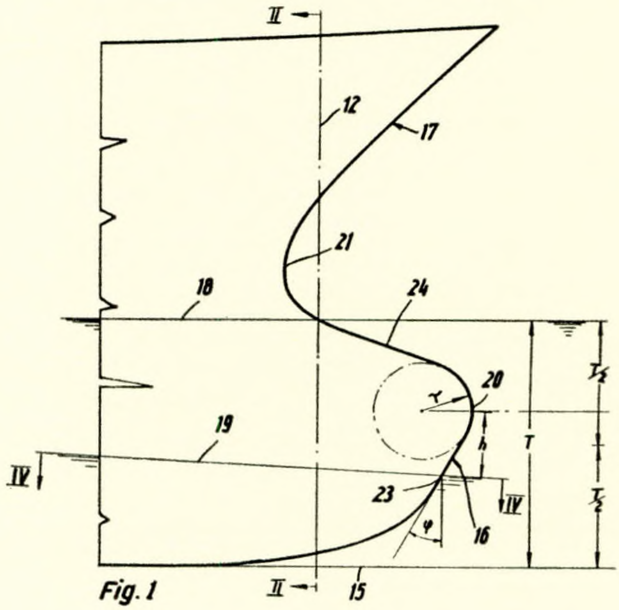


Fig. 1

The part (24) of the stem extending on the upper side of the bulbous bow has also a straight or substantially straight portion forming an angle of 15° to 30° with the horizontal. This angle should never exceed 35°. Since the stem always has a reversing point, on the upper side (24) of the bow bulb, the inclination of the tangent of reversal may be regarded as a measure of the inclination of the stem on the upper side (24). The bow according to the invention is also particularly suitable for ships travelling in frozen seas. In this case, the water line inlets are formed sharp in order to cut the ice reliability. This sharpened portion is required only between the ballast waterline and the load waterline, where the stem, having the profile according to the invention, is constructed after the manner of a coventional icebreaking stern.

The improvements made according to the invention, namely the lower power input required with constant speed or the higher speed with the same power input, are far above 10 per cent with deep loading. If a ship's model equipped with such a bow is given a large positive trim, a further surprising saving in power is achieved, resulting in an improvement of up to almost 30 per cent.—*British Patent No. 1 141 954 issued to Maierform Trust Reg. Complete specification published 5th February 1969.*