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HIGH-POWERED SINGLE-SCREW CARGO LINERS

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Summary

Early in 1945 the Managers of the Blue Funnel Line embarked upon an extensive building programme which included a number of fast single-screw cargo liners of two classes, recognized by the letters "P" and "H". Four of each class entered service, the "P's" serving the Far East and the "H's" Australia, and each is propelled by double reduction geared turbines of comparatively high power on a single screw. This feature is the subject of the present paper, which attempts to amplify information, and record in greater detail than already published, particulars relating to the design, construction, and performance of ships which at the time of their design marked an exceptional advance in single-screw propulsion.

Although the annual returns of post-war shipbuilding feature the continued predominance of single-screw ships, it is significant that there is no indication of any major tendency to extend the upper limit of power normally associated with single-screw propulsion. There does seem a visible trend towards increased power in direct-drive single-screw motor ships which probably indicates general acceptance of the maximum power which the diesel engine has to offer, but for fast cargo-liner tonnage with powers beyond the range of the single diesel engine it still appears customary to favour twin screws. For this class of tonnage, however, it does seem desirable, in view of the advanced stage of development of modern steam plant, to investigate in the design stage the savings inherent in single-screw propulsion. Such savings are reflected in lower initial cost and the cumulative effect of savings in weight, space, lower maintenance charges and the smaller number of personnel, all of which contribute materially towards the earning capacity of the ship, and may be found sufficient to determine the choice of single-screw steam in preference to twin-screw diesel.

The "P" and "H" ships described in the paper are typical examples of single-screw cargo liners propelled by double reduction turbines of high power. In both cases the frequency of service required an average sea speed of not less than 18 knots, and the designers' choice lay between direct-drive diesel engines operating twin screws and a steam turbine installation operating a single screw. The maximum power required was of the order of 15,000 shp, and although at the time when the problem was being explored single-engined ships with powers of this order were unknown it was not considered there was anything experimental in the proposal. On the contrary, it was felt that with modern steam plant reliability could reasonably be guaranteed, reliability being rightly regarded as the major factor, its importance increasing with size and speed, and in the present instance was considered to govern all other factors. The only doubtful matter which remained was the questions of manoeuvrability in ports, harbours, and narrow waters. This could only be assessed upon knowledge already gained with similar size single-screw ships having less power, but it was sufficient to give the reassurance necessary to permit the project to proceed.

The "P" and "H" ships are closely related in design, the main differences applying only to dimensions and distribution of cargo spaces to meet the requirements of their respective services. The propelling machinery is similar in all eight ships, and so also is the general arrangement of accommodation.

Design Features

Dimensions	"P"	"H"
Length overall	515 ft. 6 in.	523 ft.
Length between perpendiculars	478 ft.	485 ft.
Breadth moulded	68 ft.	69 ft.
Depth moulded to upper deck	38 ft. 6 in.	38 ft. 6 in.
Summer load draught ..	30 ft. $7\frac{3}{4}$ in.	30 ft. $11\frac{1}{2}$ in.
Summer load displacement	18,920 tons	19,507 tons
Summer deadweight ..	11,270 tons	11,355 tons
Cargo bale space in tons of 40 cu. ft.	15,450	15,380
Gross tonnage	10,093	10,125
Net tonnage	5,893	5,921

General arrangement of "P" design is shown in Figs. 1 and 2 and "H" design in Figs. 3 and 4. As many of the general arrangement and other features are common to both "P" and "H," it will suffice to describe briefly the essential details of "P" design only.

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General Arrangement Features

No departure is made from the three-island type which is generally characteristic of the company's ships, with six cargo hatches, three forward and three aft of the machinery space. There is an upper and lower cargo 'tween deck of normal height throughout the three forward holds and one 'tween deck of above normal height in the two after holds. No. 3 hold and lower 'tween deck are arranged for the carriage of liquid cargo and No. 4 hold is subdivided into several independent compartments for the carriage of refrigerated cargo at various temperatures. The six cargo hatches are served by a full equipment of derricks and winches, designed and distributed to give maximum efficiency and the maintenance of a high rate of loading and discharge. Altogether there are twenty-six derricks and twenty-four winches, two of the derricks having lifting capacities of 50 tons and 20 tons and the remaining derricks of 5 and 10 ton capacities. All winches are controlled from positions which enable the winchmen to have the loads in view at all stages of lifting and lowering. A satisfactory standard of watertight subdivision is obtained by eight watertight bulkheads, all of which extend to the upper deck.

Accommodation

Crew.—The deck ratings and stewards are European and the engine-room ratings Chinese. The former are berthed in a deck-house on the centrecastle deck amidships and the latter in the poop erection, the company's standard of two ratings per room being observed. Officers in single-berth cabins occupy a deck-house on the airy boat-deck, and may thus enjoy off-duty seclusion from both passengers and other members of the crew. The cabins of both officers and ratings are each completely panelled in selected veneers and liberally furnished and equipped in keeping with the company's high standard of crew accommodation. Commodious recreation rooms are also provided for both officers and ratings.

Passengers.—Following the company's traditional policy, provision for the carriage of passengers is limited to the maximum which the design can comfortably permit without encroaching in any way upon the requirements necessary for the most effective distribution of cargo stowage and its speedy handling. The midship deck-house suffices for the relatively small number of thirty passengers carried, the standard of accommodation comparing not unfavourably with best passenger-liner practice. The cabins, seventeen in number, includes one suite and, situated in a deck-house site, the accommodation is comfortably planned for tropical travel. Public recreation facilities indoor are in keeping with the high standard of accommodation, and ample deck space is available for outside recreation or promenade space.

Form and Model Tests

Model tests were conducted on two models A and B designed for twin-screw propulsion. Model B differed slightly from model A, the modification consisting mainly of a contraction of the breadth from 69 ft. to 68 ft. For these two models, the ship particulars are given in Table I and the resistance results in Table II.

TABLE I

	Model A	Model B
Length b.p. on l.w.l.	480 ft.	480 ft.
Breadth moulded	69 ft.	68 ft.
Draught moulded	29·58 ft.	29·5 ft.
Displacement moulded ..	18,500 tons	18,200 tons
Displacement coefficient ..	0·66	0·66
Prismatic coefficient ..	0·681	0·679
Midship area coefficient ..	0·97	0·973
Longitudinal centre of buoyancy (excluding cruiser stern) aft of amidships ..	7·36 ft.	7·25 ft.

TABLE II

Knots	$\frac{V}{\sqrt{L}}$	Model A		Model B	
		ehp	Ⓒ	ehp	Ⓒ
$14\frac{1}{2}$	0·662	3,305	0·658	3,208	0·648
15	0·685	3,669	0·660	3,575	0·652
$15\frac{1}{2}$	0·707	4,073	0·664	3,980	0·661
16	0·730	4,534	0·672	4,420	0·664
$16\frac{1}{2}$	0·753	5,054	0·683	4,905	0·672
17	0·776	5,609	0·693	5,394	0·676
$17\frac{1}{2}$	0·799	6,188	0·701	5,900	0·678
18	0·822	6,802	0·708	6,527	0·688
$18\frac{1}{2}$	0·845	7,490	0·718	7,340	0·712

The Ⓒ values for model B are less than for model A for all speeds and the results generally are satisfactory, although a smaller Ⓒ value would probably be obtained at the maximum service speed of $18\frac{1}{2}$ knots by fining the forward waterline.

The hull form was re-designed with a single screw stern, model C, and in addition the length of the ship was reduced to 478 ft. b.p. Model C was subsequently re-cut to a modified design, model D, having the body section more U-shape. Both models C and D were tested for resistance, and the resulting Ⓒ values for model D were good at all speeds up to $18\frac{1}{2}$ knots, at which speed the Ⓒ value was $2\frac{1}{2}$ per cent better than for model C. Although improved values are still probable at the higher speeds by further fining of the waterlines forward accompanied by a further movement aft of the longitudinal centre of buoyancy, this step was deemed inadvisable on account of service considerations. Model D was accordingly adopted for the lines of the new ships and for the subsequent screw tests. The body form is given in Fig. 5. For models C and D the ship particulars are given in Table III and the resistance results in Table IV.

TABLE III

	Model C	Model D
Length b.p. on l.w.l. ..	478 ft.	478 ft.
Breadth moulded	68 ft.	68 ft.
Draught moulded	29·5 ft.	29·5 ft.
Displacement moulded ..	18,035 tons	18,025 tons
Displacement coefficient ..	0·658	0·658
Prismatic coefficient ..	0·677	0·677
Midship area coefficient ..	0·972	0·972
Longitudinal centre of buoyancy (excluding cruiser stern) aft of amidships ..	7·2 ft.	7·25 ft.

The resistance experiments were carried out without any turbulence stimulation. Screw propulsion tests were carried out with model D, the general particulars of the screw tested being given in Table V.

In association with the screw propulsion tests, special study was given to the design of sternframe and rudder in view of the importance associated with appendages of the

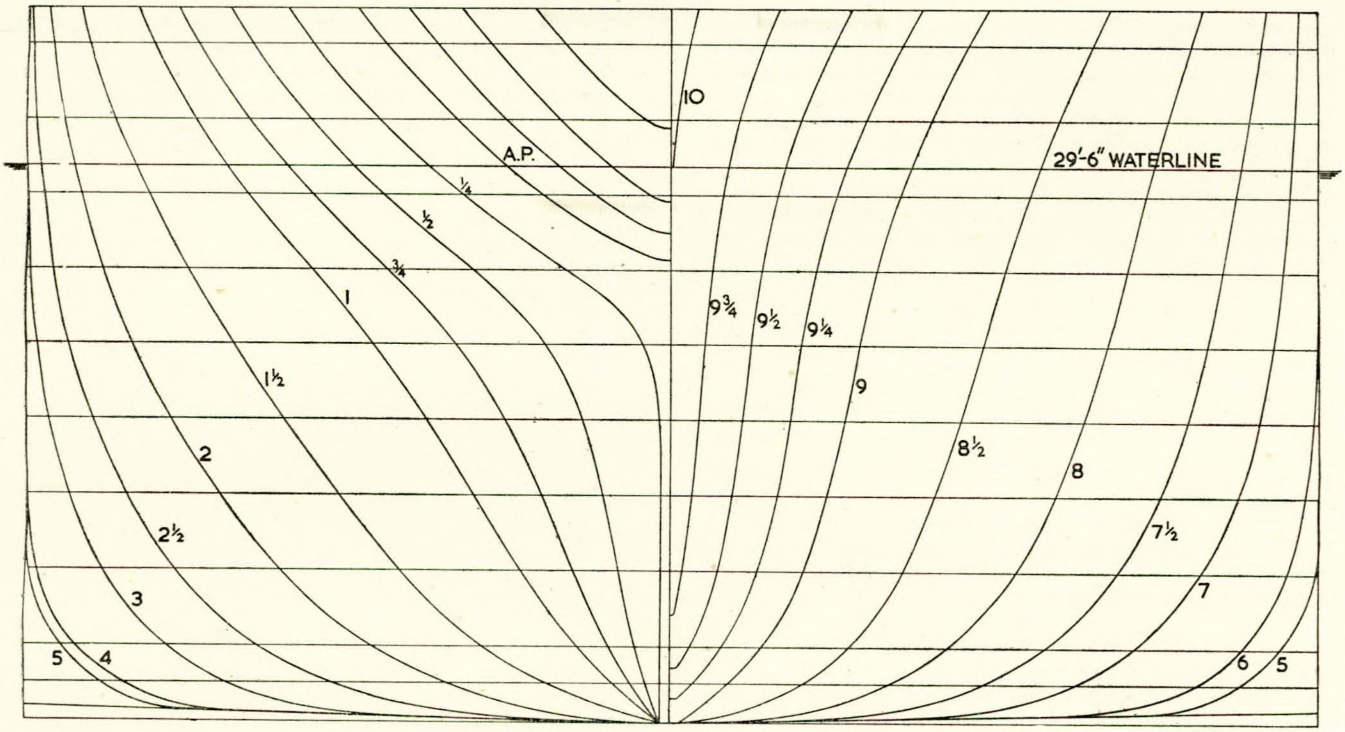

FIG. 5.—"P" CLASS, BODY PLAN.

TABLE IV

Knots	$\frac{V}{\sqrt{L}}$	Model C		Model D	
		ehp	Ⓒ	ehp	Ⓒ
14½	0·663	3,230	0·656	3,135	0·636
15	0·686	3,620	0·661	3,510	0·643
15½	0·709	4,015	0·666	3,900	0·648
16	0·732	4,440	0·670	4,340	0·656
16½	0·755	4,900	0·672	4,795	0·661
17	0·777	5,360	0·674	5,280	0·665
17½	0·800	5,860	0·675	5,800	0·670
18	0·823	6,500	0·689	6,375	0·677
18½	0·846	7,390	0·719	7,180	0·702

TABLE V

Draught moulded	29·5 ft.
Number of blades	4
Diameter	20·5 ft.
Revolutions	106·8
Blade area (ex boss)	189 sq. ft.
QPC	0·689

modern ship and their relation to ship resistance and propulsion. A rudder area of 227 sq. ft. may be considered above the average for the size of ship, but service results have proved excellent and have dismissed any anxiety which may have lingered with regard to manoeuvrability and handling of ship in harbours and narrow waters. Interesting results are given in Table VI of model tests with rudders of varying thicknesses.

These results are significant and indicate more than 3 per cent improvement in the propulsive coefficient for the thin rudder. The improvement is also associated with less wake fraction, less thrust deduction fraction and greater screw efficiency in open water and behind the hull. This may be due to the improved slip conditions under which the screw is working, but the results certainly exhibit a tendency favourable to the fitting of a thin rudder for the specified propulsive conditions. For practical reasons the actual rudder was made about 15 in. thick. The minimum clearance between the trailing edge of the propeller blades and the forward edge of the sternpost is about 9 per cent of the propeller diameter, which is rather more than current prac-

TABLE VI

	Rudder 24 in. thick	Rudder 12 in. thick	Rudder 3 in. thick
Speed in knots	18·57	18·58	18·68
Wake fraction	0·452	0·415	0·393
Thrust deduction fraction	0·227	0·204	0·20
Hull efficiency	1·122	1·127	1·115
Screw efficiency in open water	0·586	0·594	0·596
Screw efficiency behind model	0·613	0·629	0·640
Quasi-propulsive coefficient	0·689	0·710	0·713

TABLE VII

Draught moulded	29·5 ft.
Displacement plated	18,140 tons
Speed	18·5 knots
Effective horse-power	7,180
Quasi-propulsive coefficient	0·710
Shaft horse-power at propeller with 23 per cent allowance	12,440
Revolutions	104

tice, but despite this very slight pitting of the sternpost took place over a small area opposite the propeller blade tips at top and bottom. The pitting is not serious, and after developing in the early voyages is now less active.

The estimate of power for ship based upon the result of the propulsion experiment of model D with improved rudder is given in Table VII.

The propellers actually fitted were designed and manufactured by the Manganese Bronze and Brass Co., Ltd., and J. Stone and Co., Ltd., and have the general particulars given in Table VIII.

TABLE VIII

Number of blades	4
Diameter	20·5 ft.
Blade area (ex boss) (M.B.)	195 sq. ft.
(J.S.)	185 sq. ft.

The propelling machinery provides for 14,000 shp for normal service and an available maximum of 15,000 shp. The service results given later in the paper demonstrate that the required speed for the voyage is well within the compass of the propelling installation.

The Alternative Motor Design

The propelling machinery contemplated consisted of two double-acting Burmeister and Wain engines, two-stroke cycle with airless injection of fuel, driving twin screws, each engine being capable of delivering 6,800 bhp at 116 rpm in service with eight cylinders 550 mm. diameter and 1,200 mm. stroke. The principal hull dimensions of the steam design were also accepted for the motor design and general arrangements of the motor design were adjusted only where necessary to accommodate the twin diesel machinery.

Resistance and Propulsion Tests

Compared with the steam design, the results of the model tests of the motor design are approximately 2 per cent less favourable, which generally accords with expectations.

Cargo Capacity

There is an obvious loss of cargo-carrying space in the motor design due to the larger space required for the propelling machinery and shafting, and this reached the unexpectedly high figure of 800 tons of 40 cu. ft.

Deadweight Capacity

Similarly there is also an appreciable reduction in the deadweight capacity of the motor design, again due to the increased weight of propelling machinery assisted by a slightly heavier hull. The reduction of total deadweight amounts to 650 tons. This loss is also reflected in the reduced weight of cargo which can be carried. Taking a voyage period of ten days for bunkering purposes, the relative figures for the steam and motor designs are given in Table IX.

TABLE IX

Item	Steam design	Motor design
Stores	220	220
Oil fuel	830	570
Fresh water	400	400
Cargo	9,820	9,430
Total	11,270	10,620

Earning capacity being directly related to available cargo-carrying space, the additional increments of cargo deadweight affords the flexibility necessary to make the fullest use of the larger cargo-carrying space of the steamship.

The weight of cargo diminishes as the length of voyage increases, but for all practical purposes the steamship shows a gain up to a voyage length of 10,000 miles or 22½ days steaming at 18½ knots.

Comparative Earnings

A voyage comparison given in Table X assumes a voyage length of 24,000 miles and considers only the two design variables of fuel and tonnage dues. The margin includes all other expenses including depreciation and profit. An index figure of 100 for the steamship is taken as the basis to which all the other figures are directly related.

TABLE X

	Steamship	Motorship
Fuel	13·75	12·00
Tonnage dues	7·53	7·26
Margin	78·72	75·34
	100·00	94·60

Construction Cost

Towards the end of 1945 several shipbuilders were invited to submit tenders for the construction of the two alternative designs, and of the tenders received the most favourable for both the steam design and the motor design revealed that the cost of the latter was higher by about 18 per cent. This, together with the substantial gains in potential earning capacity of the steamship, were major factors in influencing the decision in favour of the steamship and far outweighed any additional cost of fuel consumed by the steamship. This and other economics have since been borne out by voyage results.

Propelling Machinery

For the reasons already given it was decided that the most economic ship would result from the adoption of single-screw turbines of 15,000 maximum horse-power. The machinery installation has been described in the press, and the remarks that follow dwell mainly upon performance and service experiences. The tabulated results of thirty-seven voyages are shown in Table XI.

The feed cycle chosen utilizes bled steam to heat the feed to 240° F.; this relatively low figure was selected in order to keep down to reasonable proportions the size of air heater necessary to obtain 88 per cent boiler efficiency. In actual service, however, the margin of performance specified has enabled us to raise the feed temperature to 280° F. without loss of boiler efficiency and with a consequent gain of about 3 per cent in the overall fuel consumption.

The turbo-generator and main feed pump act as steam consumers for the stand-by or manoeuvring condition, so that an adequate supply of fuel can be fired in the saturated furnace of each boiler under the condition of minimum output. The turbo-generator follows conventional lines with its own condenser; this feature would, however, not be repeated, as it is considerably more efficient to use a back-pressure set and to obtain useful work from heat thrown away in the exhaust steam.

Funnel Design

In common with most shipowners, difficulty with smoke and fumes on deck was experienced with the first ship. A "fancy" design of funnel top shown in Fig. 6A had been fitted and was clearly useless; consequently, later ships were modified to the form shown in Fig. 6B, with the result that under the majority of sea-going conditions this trouble has been eliminated.

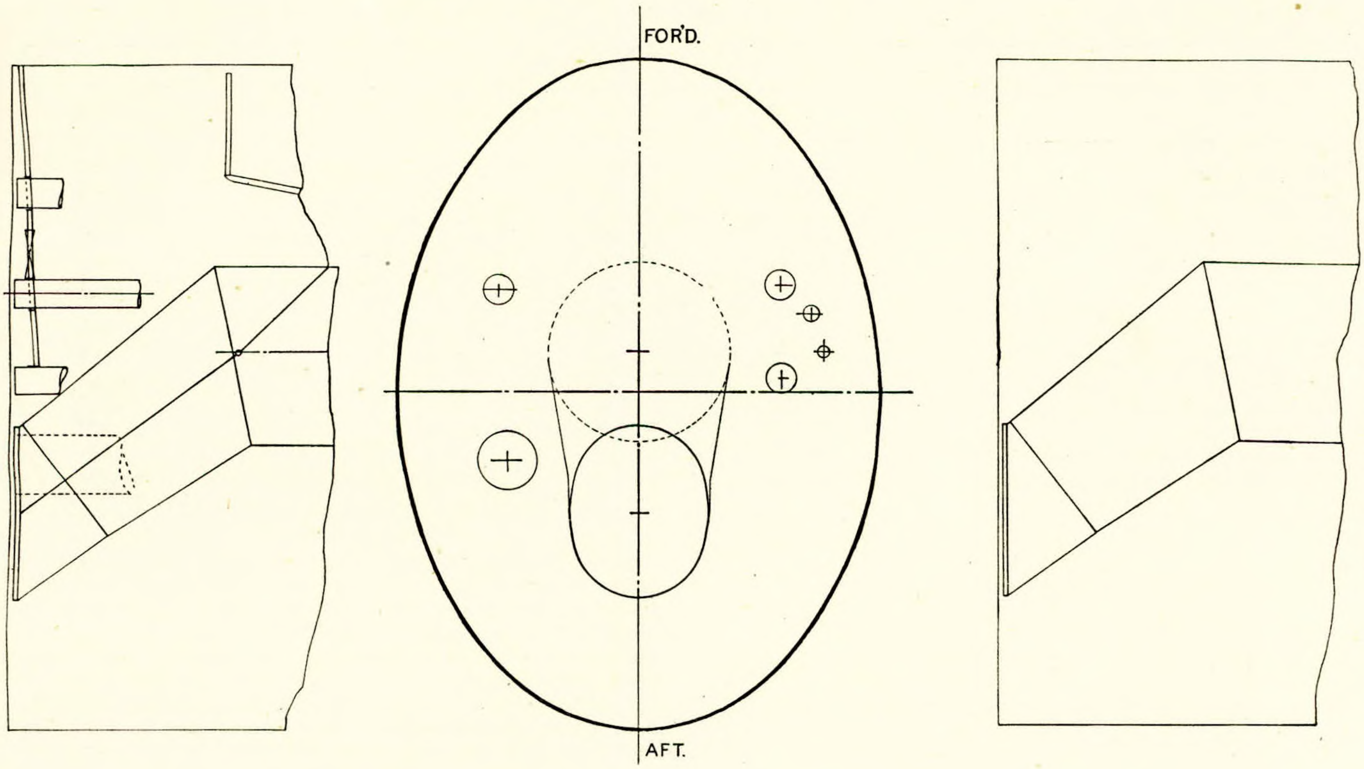

Fig. 6b.—Modified funnel arrangement

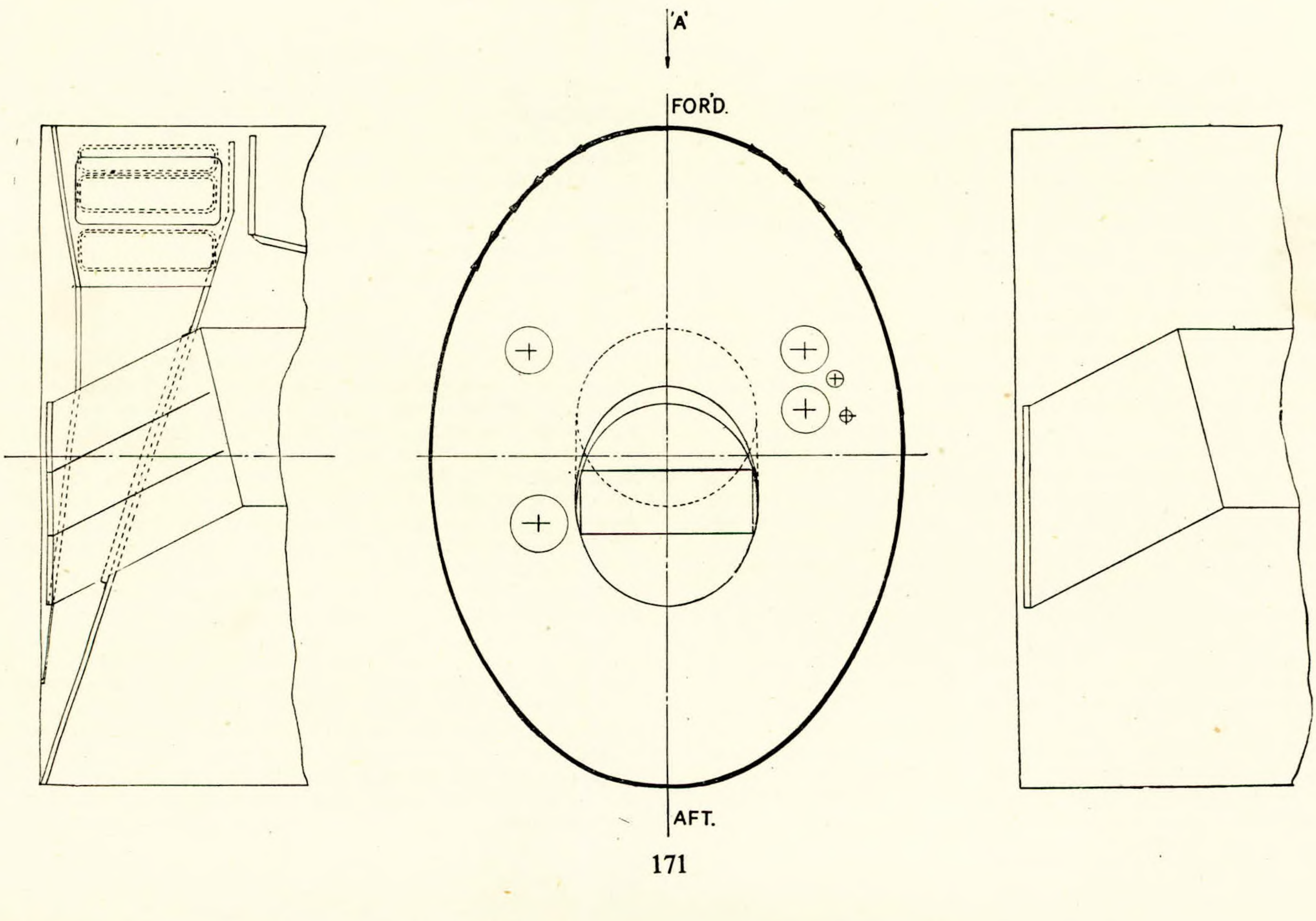

Fig. 6a.—Original funnel arrangement

TABLE XI

Type of burners and number of voyages analysed	Far East			Australia	
	X (2)	Y (10)	A B C (11)	X (6)	A B C (8)
Full speed, miles	21,171	23,094	23,433	24,130	23,595
Total distance, miles	23,186	24,286	25,203	25,496	26,327
Draught, mean	26 ft. 7 in.	26 ft. 8½ in.	26 ft. 9½ in.	27 ft. 3½ in.	28 ft. 5¼ in.
Displacement (mean)	15,947	16,036	16,103	16,697	17,580
Full speed, miles/hour	18·88	18·78	18·771	18·58	18·46
Slip	5·58%	2·78%	5·56%	5·13%	4·54%
rpm	106·19	103·04	105·9	104·34	103·1
shp	13,440	13,334	13,336	13,214	12,925
Fuel, lb./shp/hour	0·576	0·586	0·578	0·584	0·575
Fuel, tons/day (full speed)	83·15	83·456	82·525	82·65	79·64
Admiralty coefficient	317	316·8	317	317	329
Fuel coefficient	51,277	50,415	51,248	50,715	53,421
Ton miles/ton fuel/full speed	86,895	87,011	87,911	90,777	97,869
Ton miles/ton fuel/total distance	86,645	84,814	87,546	88,509	96,213

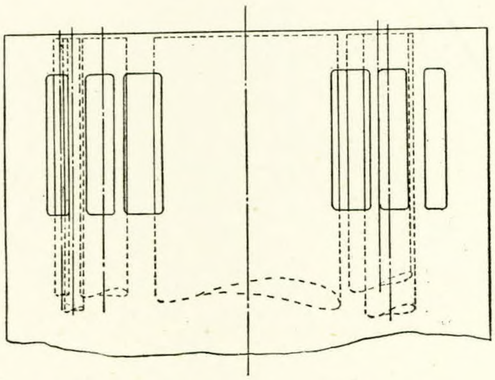

FIG. 6C.—ORIGINAL FUNNEL ARRANGEMENT
View looking in direction of Arrow "A," in FIG. 6A.

Tail-shaft Liners

There have been a number of interesting defects of tail-shaft liners which, although practically confined to these ships, are nevertheless not thought to be in any way the result of the power or service conditions.

On the dock trial of one ship of the class, the liner was found stopped whilst the shaft was still rotating. This had occurred during a change of direction of the shaft. The shaft was changed, but there was no indication of a possible cause on either liner or lignum vitae bearing. The trials of the ship were then carried out satisfactorily and the ship was loaded at Glasgow and Birkenhead. A dock trial before finally sailing for the Far East was satisfactory, but when the ship increased speed in the river the liner suddenly began to slow down and stopped. At the same time it moved forward due to the pressure of the rubber ring on the end of the liner. The ship was dry-docked with full cargo and the spare shaft borrowed from another ship. Once again there was no indication of the reason for this defect arising. In neither case was the liner apparently heated unduly until *after* slipping had commenced. The liner was tested on its return to the makers and was found to be quite tight on the shaft when cold, withstanding a dead load of 120 tons. On warming up, however, it was found that about 25° F. difference in temperature between the steel shaft and gun-metal liner was sufficient to remove the interference fit that had been allowed. As a direct result of this experience, we are now requiring the use of an interference fit of 0·5 to 0·75 thousandths of an inch per inch diameter for this service.

Lately another unusual defect has come to light. Owing to the continued failure of stern gland packings in one ship, the shaft was drawn for inspection and was found to be considerably roughened in all places where the weight of the shaft did not actually bear on the lignum vitae. There were apparently two layers of scale deposits, the outer one hard and the inner softer and reddish in colour; the liner had in fact "destannified." No reason has so far been found for this, although other shaft liners in this and other classes of ships have been found to be affected similarly. Measurements of the potential difference between the shaft and ship show that the shaft is positive to the ship with a voltage varying from 0·2 volts at full speed to zero when the shaft is stopped.

Propellers

At the time of their design the propellers for these ships represented a step into the unknown. It is particularly satisfactory therefore to record that the designs produced by the Manganese Bronze and Brass Co., Ltd., and J. Stone and Co., Ltd., have been free from serious troubles. There has been slight roughening of the backs of the blades of both designs. After developing rapidly for two or three voyages it now appears to be static, and it is to be hoped that no further trouble will ensue.

Gear Wear

After the trials of the first ship it was apparent that all was not well with the gearing, even though the full designed load had not been used. The secondary pinions and the main wheels which had the 7/10 in. deep tooth form were marked to a significant degree; opinions differed as to the designation of the failure. Most experts called it "scuffing," but others, including ourselves, believe it was "wear" due to general penetration of the film rather than local failures. At each subsequent examination of this and succeeding ships it was apparent that the hand dressing done was improving the load-carrying capacity of the gear. About a year after the first ship went into service it was discovered that there was insufficient tip relief on the teeth. Then followed an interesting series of tests on the gearing of these ships. At this stage four ships were in service; all showed wear to a marked extent. The next two ships to go into service had increased tip relief, provided in one case by change in design of the hobs and in the other by a post hobbing process developed for this purpose by Vickers-Armstrongs, Ltd., of Barrow. The former was and still is entirely free from this type of wear, whilst the latter has only developed

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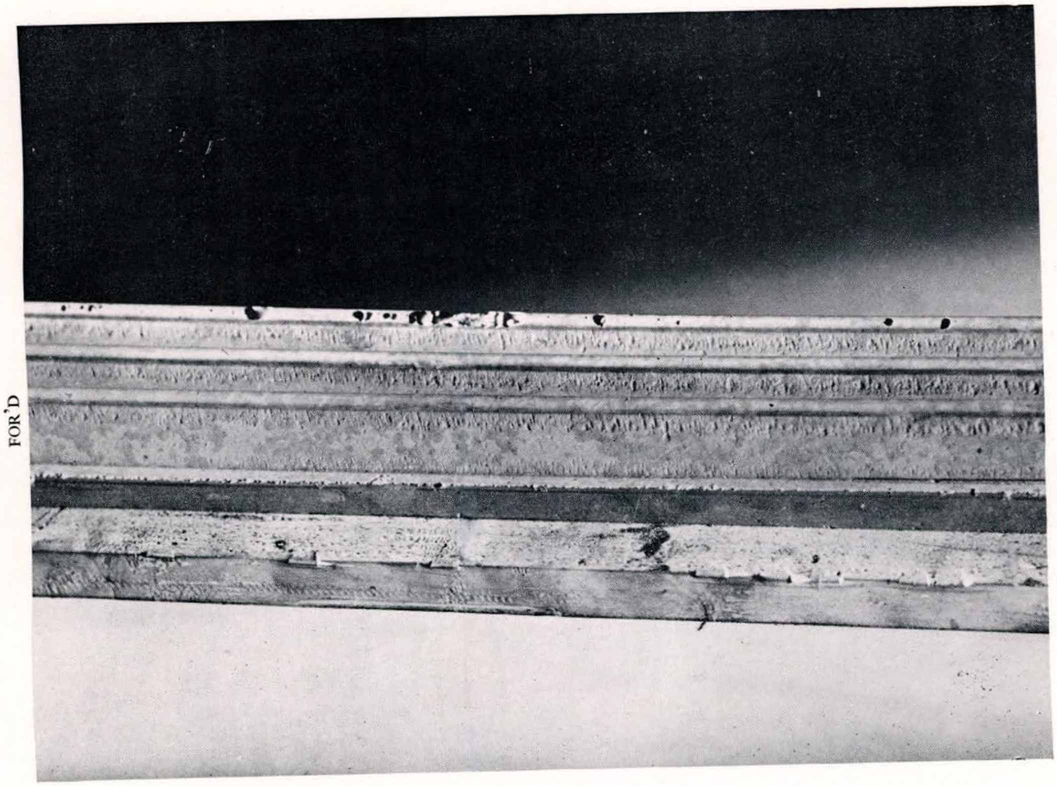

FIG. 7.—SULPHUR CAST OF AHEAD FACE OF MAIN WHEEL—FORWARD HELIX. MAY 1950

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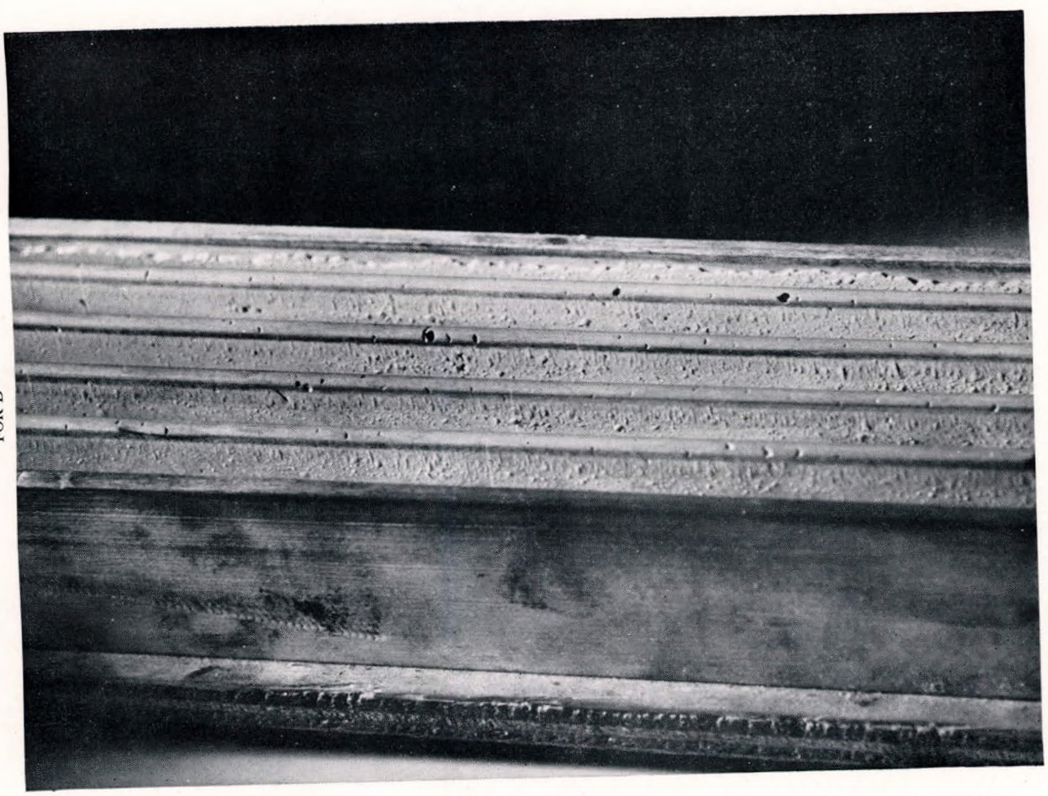

FIG. 8.—SULPHUR CAST OF AHEAD FACE OF MAIN WHEEL—FORWARD HELIX. OCTOBER 10, 1951

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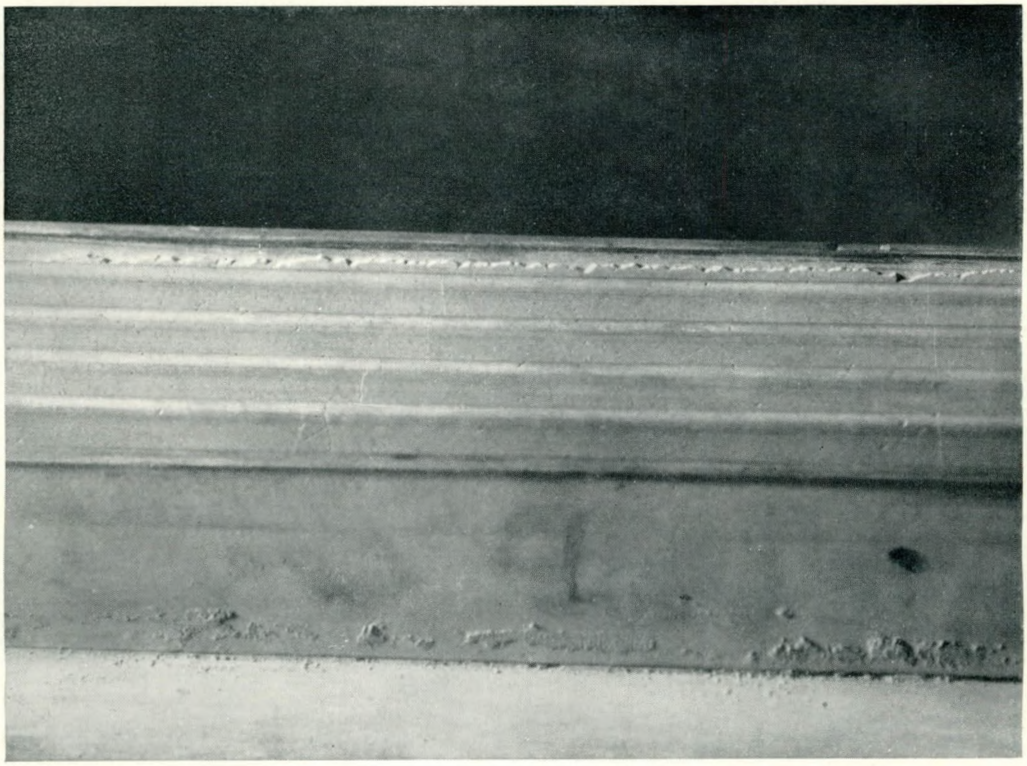

FIG. 9.—SULPHUR CAST OF THE ASTERN FACE OF MAIN WHEEL. OCTOBER 1951

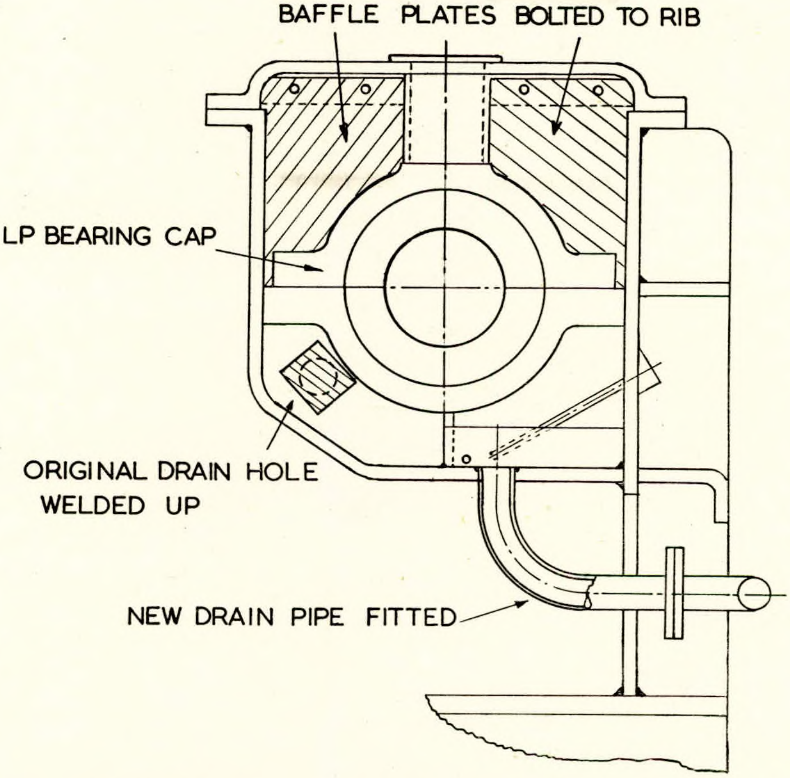

FIG. 10.—MODIFICATION TO FORWARD LOW PRESSURE PINION BEARING

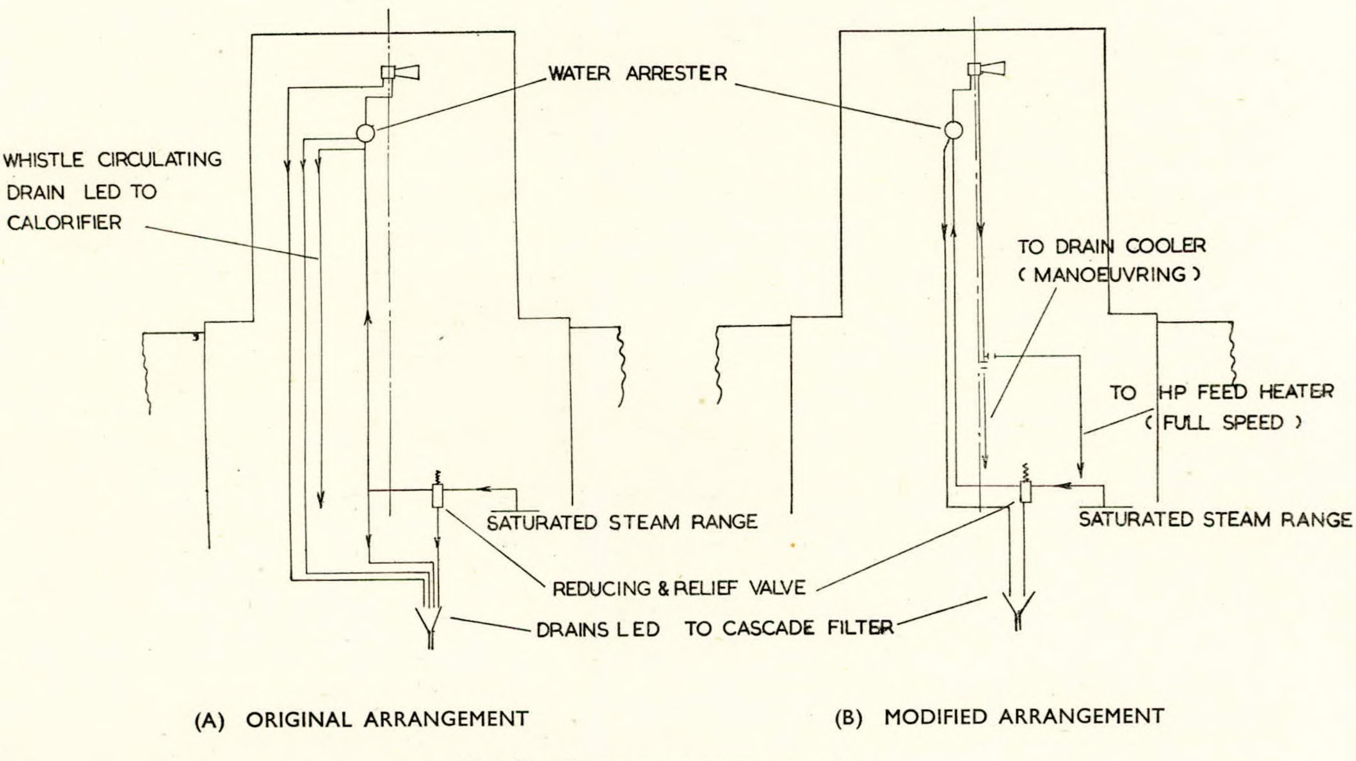

FIG. 11.—CONNECTIONS TO STEAM WHISTLE

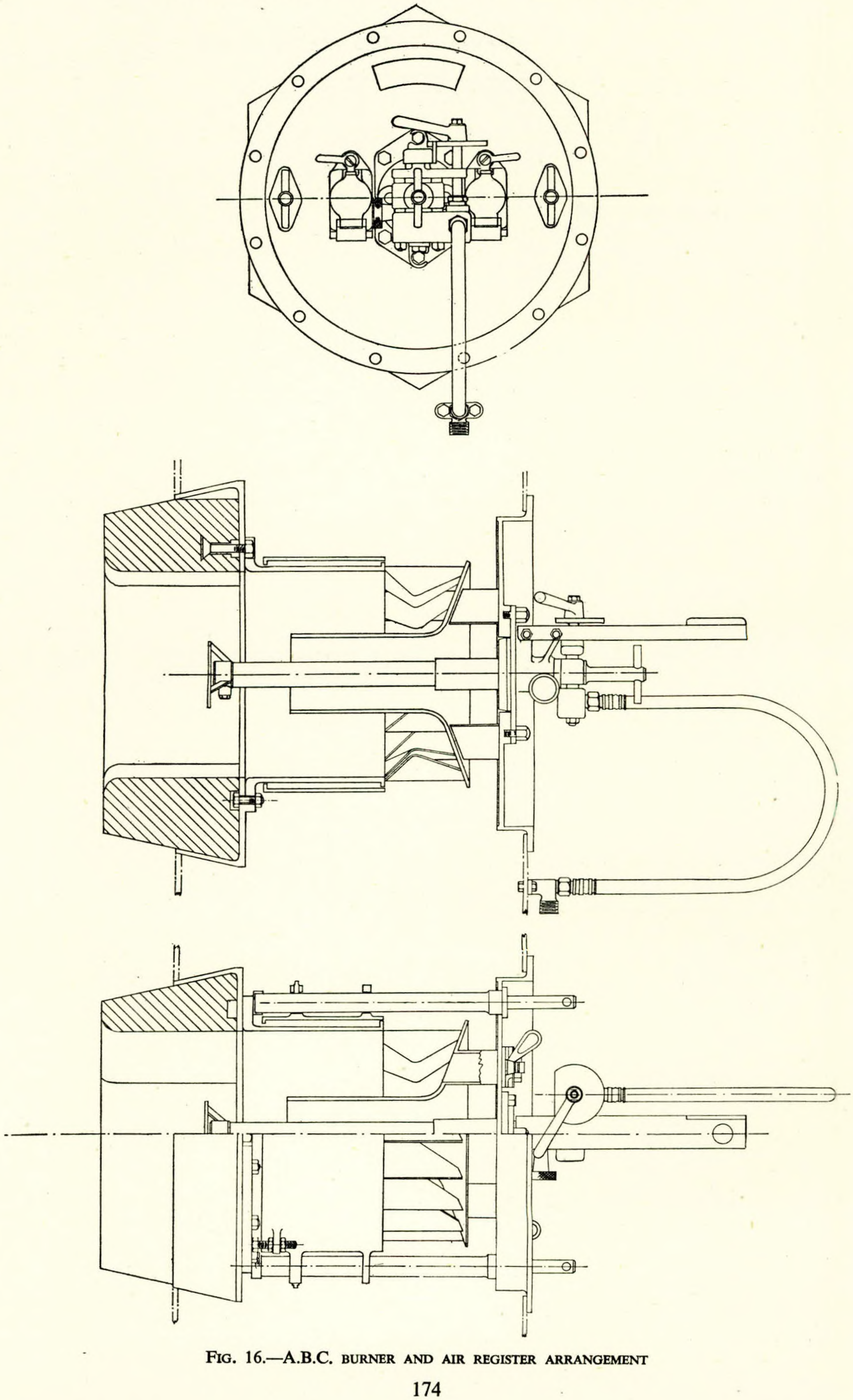

FIG. 16.—A.B.C. BURNER AND AIR REGISTER ARRANGEMENT

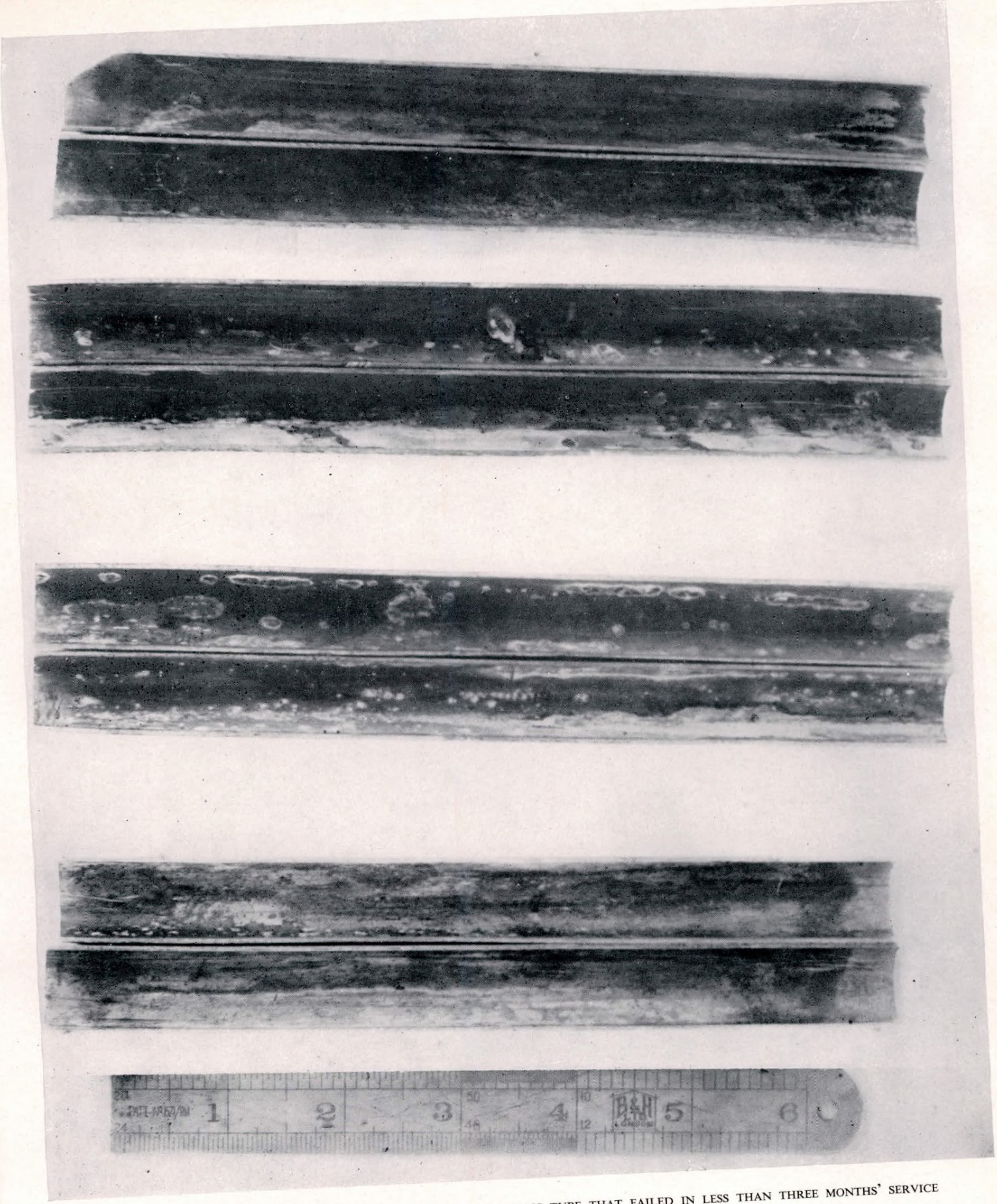

FIG. 12.—TWO-FOOT LENGTH OF PERFORATED ALUMINIUM-BRASS TUBE THAT FAILED IN LESS THAN THREE MONTHS' SERVICE

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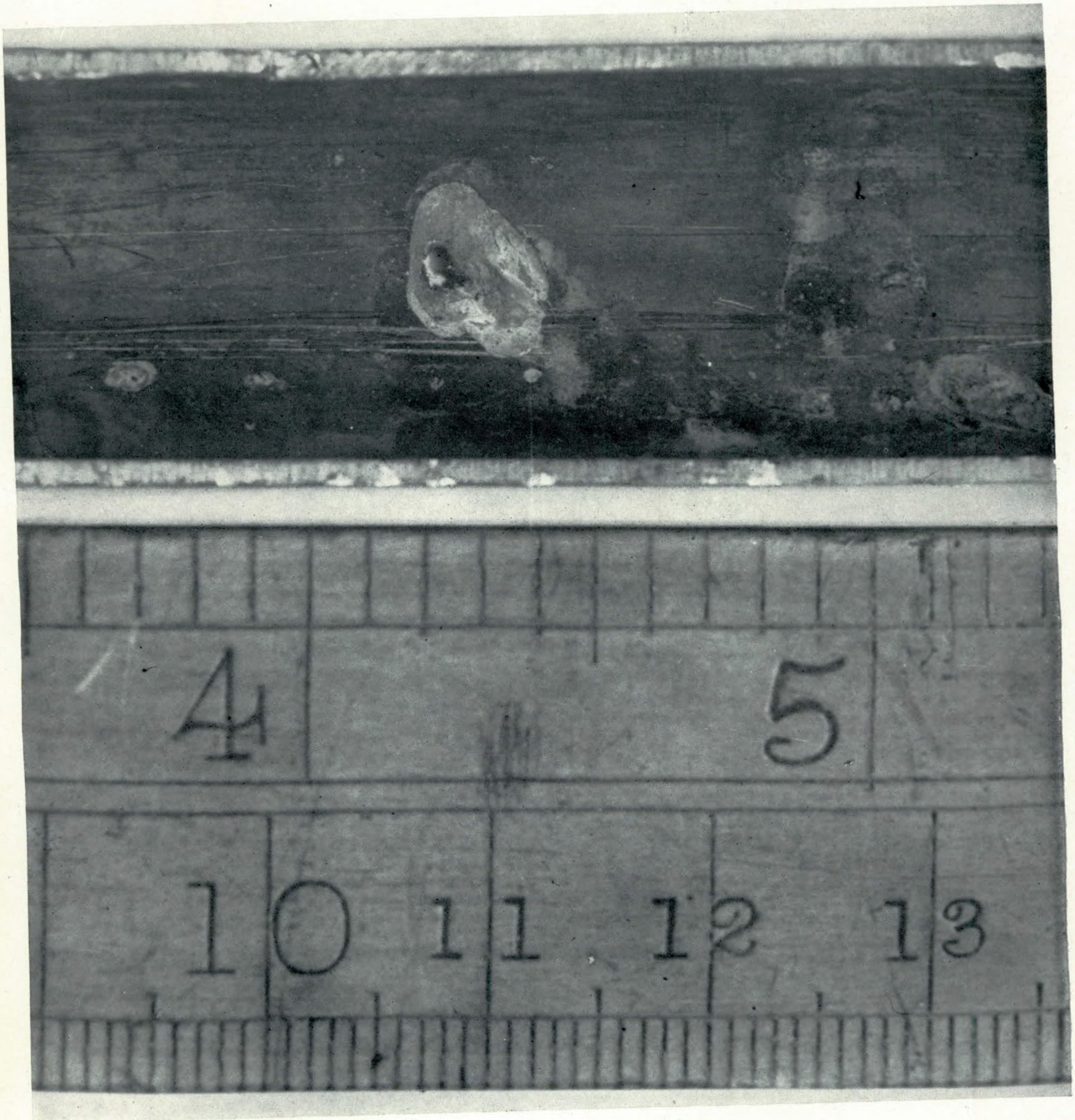


FIG. 13.—ENLARGED VIEW OF PERFORATION IN ALUMINIUM-BRASS TUBE

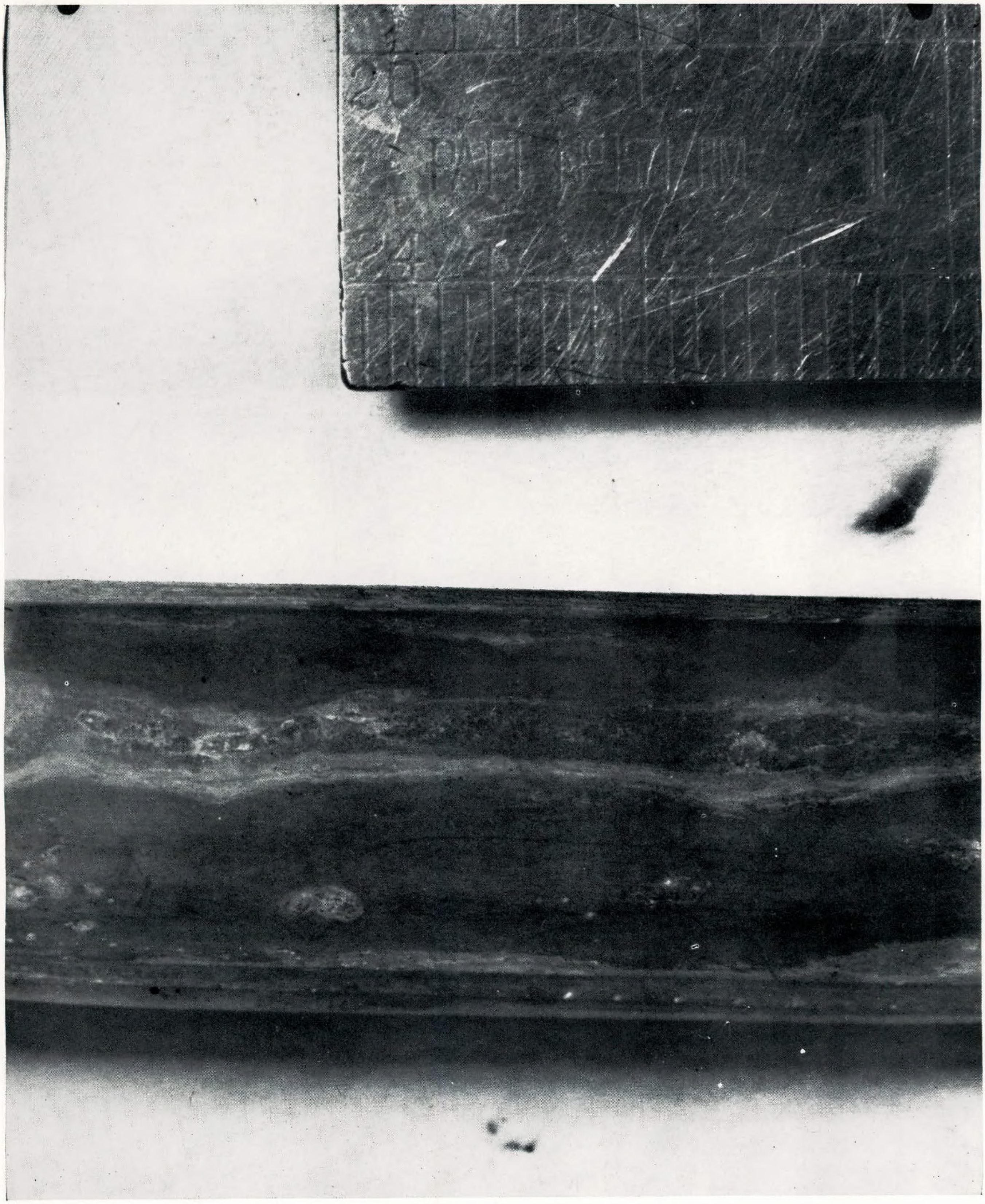
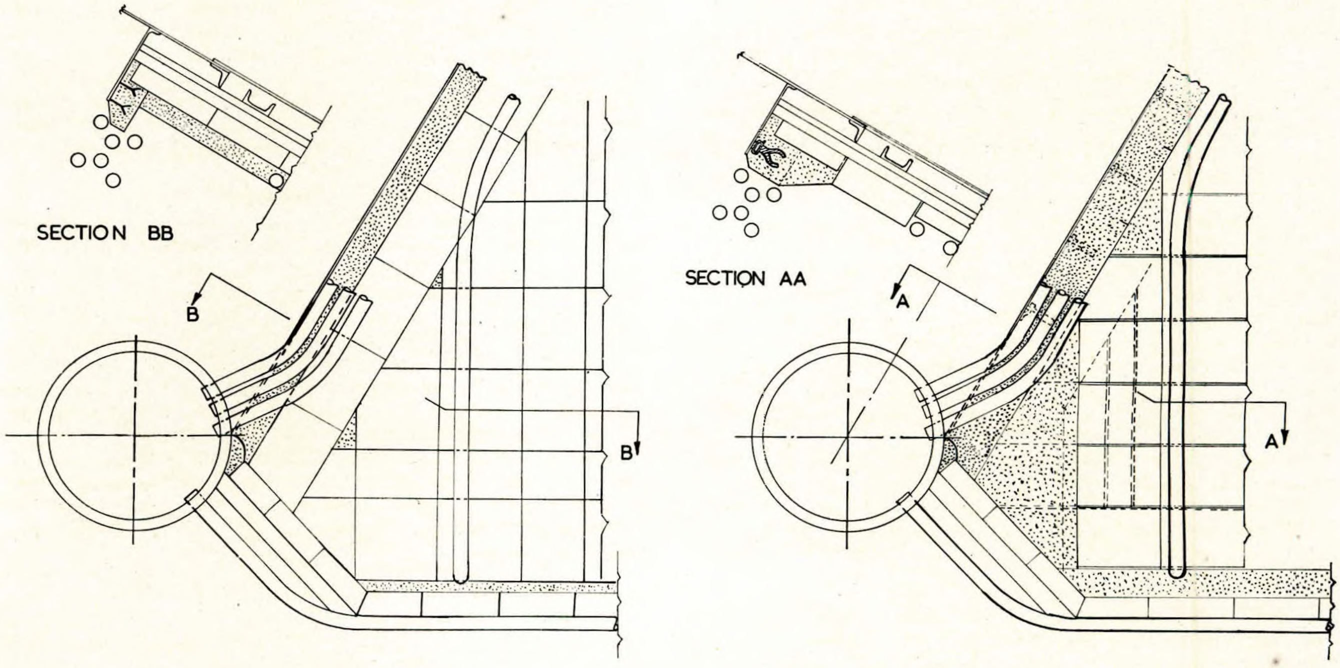


Fig. 14.—Enlarged view of general deterioration of aluminium-brass tube



Original | Modified

FIG. 15.—MODIFICATIONS TO CORBEL WALL

a patch of insignificant dimensions which has not increased after the first voyage.

Similarly no wear has occurred on one set of post hob treated gears which had been cut on the same machines as two worn sets of gears that had not been given the extra tip relief. Figs. 7, 8, and 9 show the type of wear as recorded by sulphur casts from the actual wheel.

Windage of Main Gear Wheel

From the first ship onwards the turbine lubricating oil consumption was excessive. Despite attention to numerous leaks, it remained high, and was eventually traced to oil creeping along the l.p. turbine shaft and being thrown out as a fine mist. Nothing unusual appeared to be involved in the design of the bearing and it was difficult to find the reason for this occurrence.

It was then found that there was a considerable build-up of pressure on the l.p. side of the gear case, presumably due to windage of the teeth of the main gear wheel. Attempts to relieve this pressure by venting proved vain and resulted in oil mist and vapour spreading more generally over the engine room. Attempts to prevent the pressure rise by eliminating the ingress of air failed because no indication of the points of entry of air could be found.

The trouble was finally cured by fitting a baffle over the bearing cap and by modifying the drainage arrangements from the pedestal (Fig. 10). The shorter and straighter pipe for oil returns has eliminated the trouble completely.

Steam Whistles

Following on traditional practice, it was decided to fit a steam whistle in these ships with an air whistle as stand-by. The Tyfon was chosen in view of our satisfactory experience with their air whistles over a period of years. This was installed as shown in Fig. 11A. No serious difficulty was experienced with the whistle *per se*, but it took nearly two years to solve the problem of the water that from time to time prevented the whistle from blowing reliably. There appeared to be no logical reason for the difficulties experienced and it was only the chance passage of two superintendent engineers to Singapore that gave them the time and facilities to solve the mystery. Once the drain system was modified to that shown in Fig. 11B no further trouble was experienced and the steam whistles are in every way as reliable as the air whistles of motorships.

Main Condensers in the "Helenus" Class

The condensers of the "Peleus" class have been entirely free from defect, but those of identical design in the four Australian ships of the "Helenus" class have all had tube failures of the types shown in Figs. 12, 13, and 14. The tubes that have failed have been randomly distributed over the tube nest and failures have not been confined to the first part of the tube from the inlet end.

One tube proved to be of Admiralty brass instead of aluminium brass. These comments do not apply, however, to the bulk of the failures, some eighteen in all.

No adequate explanation of these failures has been forthcoming, though it may be associated with water contamination in Australian harbours; reference to trouble in power stations from this source has been found in the *Transactions* of the Institute of Metals.

Condenser Circulating Water Inlet Pipes

In common with other shipowners, some difficulty has been experienced with the erosion of the sea-water pipes to the main circulating pumps. We have not had to renew any pipes, as when this defect was first found we embarked on a programme of coating these pipes with sprayed rubber, and perforation did not therefore occur. This has been done by P. B. Cow and Co., Ltd. (Peradin division) and has proved entirely successful in eliminating this trouble. All future steam ships will have their pipes so coated before they enter service.

In this connection it may be of interest to mention a similar trouble in the motor ships, where severe erosion of the sanitary pump casing was found to be due to the specification of an unduly high head. The pump was required to discharge 15 tons/hour at 100 ft. head; in service the head was limited to between 50 and 60 ft. with the result that the quantity pumped was about 45 tons/hour. The high resultant velocity in the suction pipe caused air impingement attack on the casing. The cure in this case was a shorter and more direct lead of the suction pipe and an increase in diameter.

Oil Fuel Combustion

With the early ships difficulty was experienced in getting better than 10·5 per cent CO_2 with consequent loss in efficiency. A change was made to the oil fuel registers and burners made by Associated British Combustion, Ltd., with an immediate gain in economy of about 2 per cent. In addition, brick-work defects (except in way of the corbel) and soot deposits in the superheaters, etc., have been so far reduced as to be no longer significant. Indeed, it is now common to run from Liverpool to Singapore and beyond without needing to change a burner for cleaning.

The success of the change in oil-burning technique has in fact shown up a defect in the boiler design, as it is not possible to maintain a burner alight in the saturated furnace and at the same time achieve the designed 850° F. final steam temperatures.

	Number of burners per boiler		°F pressure	Air pressure	Steam temperature
	Saturated	Superheat			
Before modification ..	2	4	190	5·5	850° F.
After modification ..	1	4	260	8·0	820° F.

Control of Main Engines

Probably the biggest problem that has not yet been solved is the control of the main engines under the "Stop" condition. As is well known, it is essential to move turbines nominally stopped, in order to avoid distortion and thereby cause damage to the blading. Our practice is to keep the turbine turning continuously at the lowest possible revolutions until it is essential to stop the propeller for the safety of the ship or "finished with main engine" has been rung. The lowest steady speed is 6 rpm, corresponding roughly with 1 knot; if, however, the shaft stops, the initial kick necessary to start the engine again is not less than 15 rpm or 3 knots, which puts a great strain on ropes and personnel controlling movements into docks, etc.

In new construction a design of turning gear has been produced that enables the engine safely to be "barred" by turning gear even though steam is on the engines, but the adaption of this to existing ships has taken a long time, and with current manufacturing delays it is problematical when the first unit can be fitted to one of the existing fleet.

DISCUSSION

Commander R. B. Cooper: I am a combustion engineer, and therefore will not presume to occupy time in discussing aspects of the paper other than combustion. It does seem to me that we have here an example of what can be achieved by attention to fuel economy. Most of us to-day are concerned with fuel economy, but I believe we tend to think mainly in terms of solid fuel because it comes so near to our everyday lives. To-day, of course, oil fuel is the life-blood of shipping.

We know that the costs of all fuels have gone up, and we are well aware of how the qualities of solid fuels have deteriorated; but I am not quite certain whether we are fully aware of how oil fuel has deteriorated or might deteriorate in the future, and the author's remarks are very interesting concerning the improvement obtained by changing the design of the oil fuel burners. I believe I am right in saying that the previous design of burners was one which had been in use for some time before the war. All oil fuel burner manufacturers have, since the war, realized that they must pay close attention to design principles, and have produced new types of burners to deal with modern conditions.

We see here economic ships with efficient boilers with economizers and air heaters, and yet a saving of very nearly 2 per cent of fuel was made.

In planning a conversion careful consideration has to be given to all features of design if the desired result is to be achieved.

The size and shape of furnace and the characteristics of the fans are most important items. Although the fuel pressure from the pumping unit is often considered to be a limiting factor as to whether conversions are worth while, it should be borne in mind that a modern and good design of atomizer will give better combustion even with a low oil fuel pressure than from the older designs.

Another point is not to go for absolutely theoretical maximum efficiency, but bear in mind that quite a number of boiler-room personnel to-day are untrained and require a system which is simple to operate and maintain. This applies particularly to periods of lighting up or manoeuvring. It is during these occasions that most of the damage to boilers takes place.

I think the owners are well placed in that in my opinion their engineers are extremely good and their boiler-room crews follow out instructions received.

It is interesting to see how putting up the draught loss and thereby increasing the air velocity for mixing the air with the oil achieved better combustion, and that by making full use of the fans available a total overall increase in efficiency was obtained from these ships.

It will be seen from the table of superheat temperatures that the superheat was down after conversion. This is obviously due to the fact that combustion was being completed in the furnaces with less excess air.

Burning efficiency and superheat temperatures are very much interlinked, and I have found that as a rule excess air will influence the temperature profoundly up to half power, whereas above that power it is usually late combustion which gives rise to unduly high superheat temperatures.

The point made by the author with regard to running from Liverpool to Singapore without changing an atomizer for cleaning is interesting, as this is undoubtedly allied to the remarks that maintenance and soot deposits in superheaters have been so far reduced as to be no longer significant. Running continually with a clean undamaged atomizer is very important in order to achieve optimum combustion, as it is the small percentage of large drops which comes from a dirty or a damaged atomizer which produce the ill effects.

A well-designed atomizer, however, must be precision made to very accurate limits with a knife-edged orifice, and is of course very easily damaged on cleaning. It is therefore important to introduce into the oil fuel register a device whereby the atomizer rarely becomes dirty, and this is no doubt being achieved in the burner described, but we all know that with the best of intentions mistakes can occur, and it is thought that even greater efficiency could be obtained even with the burner in question by renewing atomizers more frequently.

It should be the aim of oil fuel burner manufacturers to mass-produce if possible a standard type of atomizer, precision made, at such a cheap price that owners would never bother to clean them. Even with the most careful cleaning, atomizers can become damaged in such a way that poor combustion is achieved thereafter.

The oil fuel register should be designed such that the atomizer will steam for three weeks to a month without changing or cleaning and it is then renewed.

On the average vessel, one burner will consume about 8 tons of fuel per day which costs nearly £50. If by renewing atomizers a saving of only 0·5 per cent were made, then the atomizer would pay for itself in a week, without taking into account the lowering of maintenance bills. It would be interesting to see the amount of savings which could be effected by renewing atomizers at shorter intervals than at present.

It has been mentioned that the soot deposits being reduced is due to the atomizers remaining clean for a long period. If, as a result of a dirty sprayer, there is only 1 per cent of large drops coming from the atomizer and that these large drops are only half burnt when they reach the cold surfaces of the tubes, it will be found that something like 5 cwt. of soot and sticky oil deposits would form in the boilers in the course of a week with the present-day grade of fuel oil and with the average size of boiler. Thus it will be seen how essential it is to keep the condition of atomizers good at all times, as a very small nick in the orifice or a tiny piece of dirt will easily produce 1 per cent of large drops in the oil spray.

In conclusion, as it is very possible that however much the oil companies strive to preserve quality, the grade of oil fuel may become worse in the future, it is believed that the savings which have been made by the conversion in these ships can be achieved in most other ships, particularly in any new construction, if the boiler and burner designers would work in the closest of co-operation instead of separately as is usually the case to-day.

Mr. H. N. Pemberton (*A.M.I.N.A., M.I.Mar.E.*): The paper is of considerable interest to-day, when large power single-screw tankers are projected both in Europe and the U.S.A. These ships, up to 40,000 tons dw., transmitting 20,000–22,000 shp through a single screw, present certain operational problems of which so far we have little experience; for example, large diameter propellers of low rpm are necessary for good propulsive efficiency at high powers, and this brings in train difficulties peculiar to large propellers associated with stern frame design in ships having propelling machinery aft. In these cases adequate propeller aperture clearance is essential in order to avoid excessive propeller excitation, torsional, lateral and axial, of the dynamic system comprising turbines, gearing, shafting and propeller.

The experience of our research department at Lloyd's has shown that, where aperture clearance is less than about 6 per cent of the propeller diameter, there is danger of serious torque and thrust variations of such proportions as might cause trouble with shafting, bearings and gearing. In addition, substantially large transverse forces become available for causing vibration of the ship's hull.

Apart from the question of manoeuvrability (and in this connection the fitting of special power-assisted rudders might be considered), it would seem that the propeller, stern frame, and rudder comprise the governing factors in the maximum power which can conveniently be used in single-screw ships, particularly with engines aft. No other serious engineering factor seems to arise.

I would like to say a word on the tailshaft liner problem mentioned by the author. The tailshaft in question is of 23¼ in. diameter, and it may be that the shrinking of liners on shafts of such a size gives rise to manufacturing difficulties. However, similar trouble, namely, the slackening of the liner, has occasionally occurred with new ships having shafts of normal size, and it is generally attributed to insufficient clearance in the stern tube bush, or to the swelling of lignum vitae when the fitting-out period has been unduly prolonged.

It may be that with very large shafts the bush clearances and the size of water grooves will not be adequate if based on established practice for normal diameter shafts. This trouble has also been associated with the use of laminated plastic bearings, and it is generally understood that a very ample supply of forced water lubrication is essential with these synthetic bearings. I would like to know whether the author has experienced similar trouble with bearings of this type?

With regard to the shrinking of liners on tailshafts, the author states that as the result of this experience his company now requires an interference fit of 0·5 to 0·7 thousandths of an inch per inch diameter. This lines up with representative American and British practice, and I would like to know what shrinkage allowance was made in the case of the liner mentioned in the paper which became slack.

With reference to the tailshaft liner which the author states had been "destannified"—which I presume means that the tin had been taken out of the material—I think the explanation for this is that an electro-chemical process had been set up which caused selective corrosion of the tin in the alloy and also the solution and redeposition of copper. It is a similar process to dezincification, which attacks copper alloys containing silicon, aluminium, tin, arsenic, and manganese; and in fact, since tin is a noble metal as compared with zinc, I wonder whether the case mentioned in the paper was really one of dezincification. Perhaps the author could supplement his remarks by giving us the complete composition of the material, both before and after the trouble had occurred.

It is unusual for this type of corrosion to occur with the zinc and tin contents normally used for tailshaft liners, and so far as the electro-chemical interaction between the alloying elements of the liner is concerned I understand that tin has an inhibiting effect and does not therefore promote or encourage cathodic reaction. I am not an authority on the subject and am therefore open to correction. This, however, I do know, that tin is cathodic to iron and steel, and therefore the potential difference between the shaft and the ship's structure mentioned by the author would, I think, lead to corrosion of the steel and not to the destannification of the brass. I would like to have further details of this trouble, for the information would be valuable. We at Lloyd's Register have a very active engineering research department, which has a wide experience in investigating troubles in the machinery of ships, and even though the author confesses to being mystified by this example of destannification (and he has certainly mystified me), there is no reason why the matter should be allowed to remain a mystery.

Professor L. C. Burrill, M.Sc., Ph.D. (*Member of Council, I.N.A., M.I.Mar.E.*): I should like to thank Mr. Dickie and his company for having so kindly agreed to the publication of all the technical details of these important ships, thus laying themselves open to a general discussion of decisions which they made some five or six years ago.

In his opening remarks Mr. Dickie said that this was a courageous development, and the success of their pioneering venture is proved by the fact that their lead has been followed by others; so that such high-powered single-screw vessels are now relatively common practice.

I am interested mainly in the propeller side, although I realize that the propellers for these vessels represented only one small problem among the many which confronted the superintendent engineer and others concerned when they undertook this difficult task. At the time when the design was first considered the normal power of 14,000 bhp on a single screw was a relatively new venture and represented a definite step forward, although the propeller makers had previously had some experience with powers not greatly inferior on several ships for both British and foreign owners.

There had been some built in Sweden for Norwegian owners, notably the *Talabot* and *Tamerlaine* for Wilhelmsen-wilk, and there was also the m/v *Olna* and her sister vessel, built by Swan, Hunter and Wigham Richardson and Co., Ltd., for the Anglo-Saxon Petroleum Co., Ltd.

Consequently they were able to advise the owners with some confidence and some knowledge of previous installations.

In fact, it is of interest to note that in this instance the owners, builders, engineers, and propeller manufacturers co-operated for a period of two years before the final decision was taken. Twin-screw and single-screw alternatives were worked out in some detail, and the question of optimum revolutions for both cases was looked into; even the question of built versus solid propellers was examined in detail.

It is also very interesting to note that these preliminary investigations, which were made on the basis of systematic propeller data, were confirmed not only by the comparisons later made in the tank, as between the three-bladed and four-bladed designs, but also by the performance figures deduced from the voyage records of the actual ships.

The design of a single-screw propeller of this size and power was, however, not solely a matter of academic considerations; it involved a great deal of common sense, experience and compromise, in addition to the purely hydrodynamic problems which could be approached by calculation. For example, the question of physical size and weight introduced new problems of handling, slewing and fitting the propeller, and the question of the intensity of loading on the tailshaft liner was also of extreme importance.

For this reason, the final propeller was made smaller than the optimum diameter indicated by the normal approach to the propeller problem. The actual diameter is 20·5 ft., whereas it should have been about 22 ft. Furthermore, less blade surface was adopted than would be indicated by the usual calculations; the surface indicated was more than 200 sq. ft. and the actual surface is 195 sq. ft. Steps were also taken to reduce weight by carefully examining the strength of the blades at each radius. It is of interest to mention this, because obviously the same conditions will obtain in the future, when larger powers are required.

These decisions were taken as the result of close consultation and despite the design limitations I have mentioned, the performance is extremely good, and the propellers of these vessels have stood up remarkably well to the high loading they have been carrying in service.

Again, it is of interest to mention that the blade sections are of aerofoil type throughout the length of the blades, and despite the heavy loading it has not been found necessary to turn to a less efficient type of section than usually employed.

The problem of rudder design for single-screw vessels is not a new one, but here we have some information which is worthy of careful consideration. There are some who prefer large and thick rudders behind single screws, which propeller designers do not favour, owing to the interaction between rudder and screw, and in this case the thinner rudder, which interferes less with the propeller, is shown by the author's tests to give higher propulsive efficiency.

Mr. F. McAlister (*M.I.N.A., M.I.Mar.E.*): The author's company have long been known for their enterprise, and it is pleasing to see Mr. Dickie carrying on the traditions of the Blue Funnel Line in such a worthy manner.

My particular interest in this paper is the choice in the mode of propulsion, and I would like to remark on the way the problems of history keep repeating themselves. In the early days of marine engineering slow-running paddle-wheel engines were geared up to the first screw propeller vessels. Machinery development proceeded quickly until in 1879 the largest vessel was the *Arizona*, 450 ft. long, single screw of 6,350 ihp; in 1881 came the *City of Rome*, the Inman liner, 560 ft. long, single screw of 11,900 ihp, followed in 1885 by the *Etruria* and *Umbria*, Cunard liners 501 ft. long, single screw of 14,500 ihp and 19 knots. These ships had large compound engines with the l.p. cylinders 105 in. in diameter. Further advances in size and power of ships inevitably produced twin-screw propulsion with consequent disadvantages in propulsive efficiency and manoeuvring from rest. These disadvantages were offset by the more obvious advantage of additional security by multiplication of power units and, no doubt, under the then metallurgical and mechanical limitations imposed upon engine-builders it is probable that accidents occurred from time to time which amply confirmed the decision of the prudent shipowner in this respect.

But all this was over half a century ago. In the middle of the 1930's a great deal of attention was being paid to the forward development of high-powered single-screw vessels, as the manifold advantages of this mode of propulsion were being increasingly appreciated.

In Japan, 1932 brought the *Fujisan Maru* of 7,200 bhp, 1934 the *Komaki Maru* of 7,600 bhp. In 1935 appeared the *Toa Maru* of 8,000 bhp, in 1938 came the *Kinka Maru* of 9,200 bhp, and later in 1938 the *Itukusima Maru* of 10,000 bhp, all single-screw motor-driven vessels. Contemporary development in U.S.A. in this period produced the *Hawaiian Planter* class of 8,500 shp, the *Albany* of 9,000 shp, and in 1941 the *Corsicana* of 12,000 shp, all single-screw turbine-driven vessels.

In this country moderate development proceeded up to the 6,800 shp Ministry vessels during the war and the four Canadian Pacific Beavers of 9,000 shp in 1944.

At the end of the war the author's company made their decisions embodying single-screw turbine-driven vessels of 15,000 shp.

Now my own company have designed and manufactured propellers up to 50,000 shp with pleasing success, but there is a difference between multiple screws and single screws in that the wake behind a single screw is much more varied than behind the multiple screw shaft bracket. It is this large variation in wake which brings propellers of about 15,000 hp designed with reasonable area, thickness and other normal features up to the borderline of trouble. I am therefore pleased to see amongst Mr. Dickie's comments that the propellers are reasonably satisfactory for their duty.

It is probably true to say that many a fine ship is out of date on the day of her launch, the implication being that shipbuilding and engineering progress rapidly and improved designs are already in preparation. The point of my comment is that we are talking to-day in 1952 of Mr. Dickie's and his company's views in 1945 on the step forward to 15,000 hp single-screw vessels, whereas to-day designing offices and shipyards are concerned with 20,000 hp single-screw vessels and to-morrow possibly higher powers.

Propellers for such vessels can easily reach 45 tons in weight, requiring a total melt of up to 70 tons of bronze, and such propellers are bound to cause more than usual thought in the consideration of design, material and manufacture. If in the years to come the answers to the present movement in single-screw vessels are as satisfactory as the author's admirable summation of his step into the dark in 1945 then we shall all have cause for congratulation.

Mr. S. Archer, M.Sc. (*M.I.N.A., M.I.Mar.E.*): The paper offers material for discussion on a great many provocative subjects, but for brevity I shall confine my remarks to the question of gear wear (Figs. 7, 8, and 9).

One of the chief difficulties with gearing defects such as pitting, wear, and scuffing is to know with certainty whether the trouble is progressive or static, and in most cases reliance is placed upon "eye memory," sketches, and/or evidence from the amount of "pick-up" in the magnetic filters. It is, therefore, interesting to note the author's use of sulphur casts. This material, however, suffers from a number of drawbacks. It is messy and requires just the right amount of heat in melting. Furthermore, it is brittle when set and usually needs some mechanical stiffening such as a core wire. Fortunately, a much more convenient material is now available which is applied cold and takes an extremely faithful impression. It is a synthetic resin of the semi-thermosetting type known as "Marco" and sets in less than half an hour with an initial expansion of under 1 per cent and subsequent contraction of 6 per cent. [Here the speaker showed some sample casts from a main reduction gear which had been heavily overloaded in service (about 25 per cent overload on power).]

The author believes that the defects shown in Figs. 7, 8, and 9 should be classed as "wear" due to general penetration of the oil film rather than local failures. Fig. 7, however, appears to show typical heavy scuffing near the roots and lighter scuffing near the tips, and to that extent, relative to the involute profile, the defects may legitimately be regarded as local failures, undoubtedly due to insufficient tip relief on pinion and wheel teeth, as borne out by subsequent successful experience.

[At this point the speaker showed some slides of pinion teeth illustrating typical scuffing of the local failure type and indicating that where scuffing does occur it most frequently leads to accelerated wear.]

Most reputable companies to-day fit torsionmeters as standard equipment to their turbine-driven ships and thereby safeguard the gearing and shafting from serious overloading, at least as regards the mean torque loading. There are, however, a considerable number of owners and, in particular, those operating American war-built tonnage who do not fit such an appliance, and, in consequence, cases of serious gear tooth wear, pitting and scuffing caused by over-driving are by no means uncommon. The typical results of such treatment can be seen from the "Marco" casts exhibited.

Of the various torsionmeters in use to-day probably the best known is of the null-reading differential transformer type in which a needle has to be brought to zero by the rotation of a graduated scale drum carried on a screw, the amount of which rotation is proportional to the shaft twist and hence the mean torque. This instrument has many advantages but suffers from the limitation that it does not indicate torque variation directly.

There is good reason to believe that many of our gearing troubles are initiated when operating conditions are such as to impose heavy fluctuations of loading on the gears, as under torsional vibration or when pitching heavily in a seaway. It will be appreciated that in the latter case, owing to the high inertia of the turbine rotors and gears, practically the whole of the propeller torque variation will be transmitted to the gears with negligible change of speed. Thus, under severe conditions with the propeller periodically losing and regaining immersion, the load on the gears may well vary from almost zero to full-load torque, perhaps several times a minute for long periods.

Another suspected source of trouble, particularly scuffing, has been the sudden or heavy-handed operation of the main steam controls during manoeuvring causing excessive tooth loads at relatively low speeds when oil films have not built up sufficiently, i.e. under conditions approaching boundary lubrication. This is particularly liable to cause damage during the early life of the gears before they are properly run in.

Most existing torsionmeters unfortunately give the engineer on watch no indication of the magnitude of such torque variations, and for some time now it has been felt that there is a real need for a direct-indicating torsionmeter which could be used to supplement the normal meter and give to the engineer at the control station a visual warning, for example by dial and pointer, of the instantaneous values of the torque fluctuations. It may be of interest therefore to state that a well-known British torsionmeter manufacturer has at the present time a prototype instrument of this nature on experimental service, and the indications are distinctly hopeful that it will before long be possible to go into commercial production with this instrument. It is expected that accuracies of the order of $\pm 2\frac{1}{2}$ per cent should be obtainable, although when used as a warning device in conjunction with the conventional null-reading instrument such high accuracy would not, of course, be essential.

The author's views on this point would be much appreciated.

Mr. H. F. Sherborne, M.C., M.A. (*Assoc.I.N.A., A.I.Mar.E.*): There are two sections of the paper to which I would like to refer. The first is under the heading "Condenser Circulating Water Inlet Pipes." This is a subject on which a lot has been said and written, particularly since the war. I have no criticism to make, but I do ask a very genuine question with reference to the coating of the circulating water inlet pipes with sprayed rubber. I have seen this done in certain parts of the circulation systems in some of H.M. ships, notably the corrugated part of the condenser circulating system trunking, and I understand the results have been quite satisfactory. I should, however, like to know whether these pipes to which the author refers are fabricated from sheets or are solid drawn? In either event, is the rubber applied before or after fabrication and bending?

It is becoming almost universal practice to-day for owners to specify one or another of the corrosion-resisting alloys, instead of copper, for pipes carrying salt water on board ships. This seems to be a highly desirable and prudent precaution. On the other hand, if rubber coating is going to make use of the special alloys unnecessary, we want to get to know about this clearly as soon as possible. The problem is really one of equipment, because whereas there is plenty of capacity in this country for making all the copper tubes that are likely to be required, providing we can get the metal from which to make them, the manufacture of these special alloys is a very different proposition, and if they are going to be further and more widely specified, non-ferrous metal manufacturers will have to make considerable additions to their plant in the way of extrusion presses and melting furnaces.

The other item to which I want to refer is the most interesting account of the comparative behaviour of the condenser tubes in the "Helenus" and "Peleus" class ships respectively. In propriety I must declare an absence of interest in the matter. I am not responsible for the manufacture of the tubes in either class of ship. In these circumstances, and presuming there is no fault to be found with the composition or structure of the alloy, I think I can properly adopt these tubes. As one who has been very actively concerned with the service of condenser tubes for the last quarter of a century I think it is my duty to say a word or two if only in the interests of a sense of proportion. I say nothing of the one tube which proved to be of Admiralty brass instead of aluminium brass—the less said about that unhappy error the better. I do not know how many aluminium brass tubes are in service—one firm alone has sent out over twenty million since first making them—so it is an indirect testimony to their great worth when seventeen aluminium brass tubes fail in four ships and there are three pages of pictures of them in the TRANSACTIONS.

I do hope there is not going to be another epoch of trouble in the condenser tube world. I am just old enough to have been in the thick of the last one—we got over that with the new alloys, and shipowners and marine engineers were able to go ahead with their plans and designing on the basis that they would not have to worry about a tight condenser, at any rate so far as the metal of the tubes was concerned. This was so as far as ships were concerned—the position was not so clear with power stations using estuarine water. Unfortunately it is generally true to say that estuaries and harbours generally are tending to get fouler and fouler whilst ships remain in such places longer. I referred to this yesterday on commenting on Mr. Basil Sanderson's paper and it is not controverted.

The tubes illustrated in Mr. Dickie's paper from the "Helenus" class ships look to me to be suffering from corrosion of the oxide pitting type due to pollution.

The up-to-date position may well be illustrated by a few brief quotations from a paper read before the Institute of Metals by three very great authorities on the subject—"Corrosion and Related Problems in Sea-Water Cooling and Pipe Systems in H.M. Ships," by Slater, Kenworthy and May, *Journal of the Institute of Metals*, Vol. 77, Part 4, 1950.

Thus, at page 310:

"Later work led to the development of more resistant alloys, and these have proved so successful for condenser tubes that unless and until changes in design bring about a marked increase in the severity of the operating conditions the condenser tube problem may be considered to be virtually solved."

And page 317–18:

"Occasional failures in aluminium brass are usually due to deposit attack, assisted by decomposing organic matter and by excessive local turbulence where a foreign body has lodged."

This reference to excessive local turbulence is not relevant to the matter at present under consideration. In any event intense impingement attack due to excessive local turbulence only penetrates the tube when the partial obstruction lodges in the very first days, even hours, of the tube's life.

Thus, at page 318:

"Moreover, in some of the pipes the conditions may be almost stagnant for long periods, during which fouling organisms may develop to an astonishing degree and by their eventual decomposition liberate various harmful sulphur compounds into the water. During stagnant periods such compounds readily encourage pitting or larger areas of corrosion."

This reference is under the sub-heading of "Auxiliary Piping" but is not irrelevant to the subject.

Finally, at page 329 is the following:

"Stagnant periods when marine muds activated by decomposing organic matter settle on the metal surfaces and destroy the natural protective films, are yet another of the major problems."

By the same post as my print of this paper came also the 32nd Annual Report of the British Non-Ferrous Metals Research Association, dated April 1952. The following paragraph occurs under the sub-heading "Corrosion in Condenser Systems":

"The aluminium brasses, which have deservedly gained an excellent reputation for resistance to impingement attack, occasionally pit in waters heavily contaminated with organic matter, and this subject is among the items of further work now in hand."

I thought the condenser tube problem was finished with so far as ships were concerned (not power stations), and that it had become a very interesting subject for students to show what we had to contend with in the past. The position, however, appears to be that corrosion of condenser tubes has raised its ugly head again, and the question is, how widespread is it? Before the war I can recall only one case of this pollution corrosion, and that was in a tug, *Socony* 12, belonging, as its name indicates, to the Standard Oil Company of New York. That tug was working continually in dirty water and her aluminium brass tubes suffered from this type of trouble—corrosion due to pollution. That was the only known instance. During the last eighteen months there have come to light to my knowledge not more than half a dozen cases, of which the one described by the author is the latest.

Neither aluminium brass nor cupro-nickel is impervious to this type of corrosion. It only seems to occur in new ships or after a re-tubing—that is to say, when the pollution conditions are present before the tubes have had sufficient work in salt water to give them a thorough protective film.

We in the non-ferrous world are extremely grateful to Mr. Dickie for letting us all know about this incident. No one denies that pollution is increasing and that ships stay longer in polluted waters.

We await developments with some little anxiety as there is not yet any metallurgical answer to this type of trouble.

Professor E. V. Telfer, D.Sc., Ph.D. (*Member of Council, I.N.A., M.I.Mar.E.*): The first question I would raise concerns Mr. Dickie's remarks on the area of the rudder. Why should he be worried about the rudder area being too *big*? I cannot see why this should be troublesome, unless he is thinking that it will take too long to put the rudder over, as compared with a rudder of less area. I have devoted considerable attention to rudder design, and in a paper on seakindliness (N.E.C. Inst., 1938) I gave a formula facilitating the determination of rudder area; the figure obtained from my formula is exactly the value Mr. Dickie has used, and I am pleased that he has found the rudder to be highly satisfactory. I do know of cases where excess areas have given trouble, and probably the author has had similar experience.

My next question concerns Table XI, and I want to ask about the second column, Y (10), which gives the lowest apparent slip of 2·78 per cent and the lowest rpm. That means automatically that, if the propellers are the same, the power should have been 8 per cent less than in all the other cases, whereas in fact it is not. The power is roughly about the same as in the other cases. Why is this?

The final column in that table, A B C (8), gives the next lowest slip, 4·54 per cent. There you have the highest Admiralty coefficient; and this simply arises from the fact that the ship has the greatest mean displacement.

My final point arises somewhat out of the remarks of previous speakers. I feel that the experiments carried out by Mr. Dickie to decide whether the single-screw ship suffers any penalty as compared with the more normal twin-screw ship are not nearly sufficient to decide the issue. I have felt for a long time that the normal model comparison need not give the true ship comparison between the single-screw and the twin-screw types, because the beneficial effects of wake are much more evident on the single-screw model than on the twin-screw model, and the model benefit of the single screw may disappear or be reversed on the full-size ship.

It is still more essential, in view of the unusually high powers we are now proposing to put through single screws, to develop the efficiency of the twin-screw design. Shortly before the war I had to consider a programme of twin-screw ships, and I advised my clients that by a fundamental change in bossing design a definite propulsive economy could be obtained. I discussed the matter with Dr. Baker, who naturally felt that it was quite impossible to improve upon the normal Teddington design with the bossings lined off to avoid all hull cross-flow. However, our proposals were tested, and it was soon evident that the standard design could be improved. We eventually were able to improve the propulsive coefficient by amounts varying from 6 to 8 per cent. My clients were naturally very pleased. Their managing director came down to the Tank and, expressing his pleasure to Dr. Baker, suggested that he and his co-directors before adopting the improved design would naturally like to have from the Tank some guarantee that these results will be obtained in the ship. Dr. Baker almost exploded and said: "We never guarantee anything."

Thus the possibility of progress which could have affected hundreds of twin-screw ships subsequently built stopped at that moment. I do hope, however, that the author will be encouraged to return to the improvement of twin-screw design, because I feel that the single-screw is being forced to unnecessary high limits and for an efficiency gain which may be more apparent than real.

The Chairman (Viscount Runciman) (*President I.N.A.*): It gives me particular pleasure personally to propose a very hearty vote of thanks to Mr. Dickie for his paper, because I started my shipowning career in the office of Alfred Holt and Company. I there learned first that they are probably the best shipowners of the world, and second that they know it! Now, to some extent, we know it too, and we must be very grateful to Mr. Dickie for his paper and for the discussion it has provoked; he has made a contribution of lasting value to our TRANSACTIONS.

Written Contributions to the Discussion

Mr. George Wood (*M.I.Mar.E.*): The author has generously given a fund of most useful information, that built upon recent sea-going experience.

Referring to the funnel design, it is interesting to note that the author, while claiming a solution to the smoke nuisance with the modified funnel, only claims freedom from the trouble under the majority of sea-going conditions. The designer is to-day, however, often called upon to produce a funnel shape to give freedom under all sea-going conditions, as the nuisance for an odd short period can quickly soil the decks, etc.

Particulars are not given of the smoke exit speed, which, according to present-day theory, is the important factor, and can only be sufficiently great to give a desirable s/v ratio at the expense of increased I.D. and/or F.D. fan power. It would be interesting to learn if the greatest possible smoke exit velocity was obtained by using the available margin of fan power.

The experience of the tailshaft liner becoming loose may be taken to indicate, as the author suggests, that it is desirable on such large shafts to use greater interference fits between shaft and liner than have been adopted in the past on smaller size shafts. One is, therefore, convinced of the author's wisdom in suggesting the need for the interference fit to be 0·5 to 0·75 thousandths of an inch per inch diameter of shaft.

The author has referred to the high power transmitted to the single screw, and this, together with the comparatively low revolutions resulted in a large shaft diameter. Maybe the experience should be taken as a pointer also that the design of larger diameter stern tube bearings may require, among other special attention, more effective cooling.

The experience quoted of gear wear may be considered to show how the advantage of excellent and accurate gear hobbing can be lost if sufficient tip relief is not provided. There now appears to be some evidence that extra tip relief is desirable on gear teeth of the deep tooth standard. The fact that some gears with deep teeth have been running for many years without trouble, although admittedly cut less accurately, may possibly be due to the fact that more hand work used to be done on teeth than is done to-day, i.e. hand work tends in general to increase the amount of tip relief.

The reference to the trouble originally experienced with the sluggish response in operating the steam whistle is very opportune because this experience is, according to reports, all too common. While the solution was found by altering the drain system, one is still aware of the constant water loss which could be reduced if the size of steam supply pipes were smaller. Whether these steam supply pipes are unduly large or the actual steam consumption does give a maximum allowable steam speed, one is led to observe that the proportions of energy given by the steam to the energy of sound produced is remarkably great, and it may be that, as in the case of producing sound in musical wind instruments, a smaller operating pressure would be equally effective. The experience quoted does bring into relief the as yet unsolved problem of obtaining a steam whistle giving constant ready response in operation combined with a low steam consumption.

The trouble experienced with condenser tubes, apart from the inadvertent fitting of Admiralty brass tubes, appears to be more puzzling in the absence of such troubles with tubes of the same material fitted in older ships and subject to the same sea-water conditions. One is naturally led to wonder whether, the material being the same, process in manufacture could be a factor in the problem.

Mr. W. Sampson (*Vice-President, I.Mar.E.*): The author's most valuable paper might be summed up as describing a very well matched installation. Obviously, from the propeller to the boiler, the steamships described by the author have their machinery units thoroughly matched and operating at equally peak efficiencies.

Referring to the boilers under the heading "Oil Fuel Combustion," the author shows how significant is the perfection or otherwise of the burning of the fuel, and he states that the success of the change in oil-burning technique has in fact showed up a defect in the boiler design. For the purpose of record it might be stated that the boilers were designed correctly on the known performance of the burners originally chosen for these ships, and the fact that a redistribution of the fuel burning between the two furnaces was found necessary when the very noticeable improvement following the fitting of the new burners was made did mean that very little fuel needed to be burnt in the outer furnace of the boilers when meeting the required superheat conditions, but what the author terms a defect in the boiler design might have been corrected by the fitting of a smaller output burner in the saturated furnace, thus enabling it to be kept alight.

The author deserves to be thanked for bringing to light how serious a matter the combustion of fuel really is. Shipowners cannot expect anything but deterioration in the quality of the fuel to be supplied in the future. Increased impurities can be expected, and we are still a long way off from perfecting oil burners which can burn satisfactorily the very many grades of fuel oil that will be supplied on normal trading routes, and boiler design will improve only as oil fuel burning technique improves.

Mr. S. A. Smith, M.Sc. (*M.I.N.A., Member of Council, I.Mar.E.*): This paper has provided most useful data on this subject, and it may be of interest to note that the P. & O. Company, when considering their post-war building programme, were faced with somewhat similar problems, and their inquiries and investigations took place about eighteen months previous to similar investigations by the Blue Funnel Line.

Early in 1944 investigations were made into the relative running costs in similar ships propelled by single-screw turbines and twin-screw diesels, and in December of that year machinery specifications were forwarded to builders and the order for two turbine ships was awarded to Vickers-Armstrongs, Ltd., in March 1945.

Four ships were actually required, and in order to obtain equitable deliveries and to suit existing shipyard conditions it was necessary to accept two ships with twin-screw diesel propulsion, and Barclay Curle & Co., Ltd., were awarded the contract.

Four ships were, therefore, ordered, two with single-screw turbines, *Surat* and *Shillong*, and two with twin-screw diesels, *Soudan* and *Somali*. It is a well-known fact that single-screw propulsion is more efficient than twin screw as there is no bossing to push through the water, and this adds an additional power in the region of 12 per cent for similar forms of ships. As the designed speed and power of these ships was such that single-screw diesel machinary was out of the question the relative costs, etc., of single-screw turbines and twin-screw diesels had to be discussed, and factors leading to the preference of steam propulsion were based on the following calculations.

These ships were designed for the Far Eastern run, and calculations were made to cover a 22,000 miles voyage, a China coast voyage and a United Kingdom coastal voyage at an average round voyage speed of 18 knots at 15,750 tons displacement.

(*a*) *Machinery and Oil Fuel*

	Twin-screw diesel	Single-screw turbine
Weight of installation ..	1,743 tons	1,220 tons
Oil fuel for 6,395 miles ..	1,290 tons	1,955 tons
Total	3,033 tons	3,175 tons

(*b*) *Oil Fuel and Lubricating Oil Consumption*

	Twin-screw diesel	Single-screw turbine
Total fuel consumption ..	2,960 tons	4,020 (blr.) 278 diesel
Lubricating oil including diesel generators ..	40 galls./day	13 galls./day

At the price of oil at the time in question these costs worked out at £8,220 for diesel and £9,023 for steam, giving an advantage of £803 per voyage in favour of diesels.

Repair and survey costs were then discussed and it was estimated that these gave an advantage to steam by £460. The net saving in favour of diesel machinery was, therefore, £343 per voyage or £1,029 per year.

To offset this cost there was a saving in favour of steam plant in the initial cost, and also a reduction in personnel, namely two engineer officers. It was later found that due to the type of machinery eventually installed—especially in regard to the evaporating plant—that a saving in space equivalent to 450 tons of cargo was made in the steam ships above that which was originally estimated.

The principal dimensions of these ships and those later laid down for the Blue Funnel Line are given here, and it will be seen that these are almost identical, each company, however, keeping to its own recognized practice regarding crew and passenger accommodation.

	Steam "S" single screw	Diesel "S" twin screw	Blue Funnel "P" single screw
Length bp	490 ft.	490 ft.	478 ft.
Breadth moulded ..	67 ft.	67 ft.	68 ft.
Load draught ..	29 ft. 6 in.	29 ft. 5 in.	30 ft. $7\frac{3}{4}$ in.
Draught—upper deck	—	34 ft. 6 in.	38 ft. 6 in.
Block coefficient ..	0·65	0·653	—
Load displacement..	18,000	18,270	18,920
Gross D/W ..	11,590	11,100	10,000
Net D/W	9,345	9,271	—
Cargo bale	588,000	589,000	618,000
Maximum sea speed	18 knots	18 knots	18 knots
Normal shp ..	11,000	—	14,000
Maximum shp ..	13,000	13,600	15,000

A comparison of the main propulsion machinery of the three classes of ships can now be set out and is as follows:—

Steam "S"

Normal propelling power 11,000 shp with a maximum of 13,000 at 125 rpm. The h.p. turbine driving through double reduction and the i.p. and l.p. through single reduction gearing.

The steam supply from two W.T. boilers of Foster Wheeler design working at 525 lb./sq. in. and 850° F. superheat. The normal boiler output to be 58,500 lb./hr. with a maximum of 84,000 lb.

Generators: Three 6-cyl. 325 kW at 325 rpm and 220 volts.

Propellers: 18 ft. 6 in. diameter, 16 ft. 3 in. pitch, 160 sq. ft. surface, four-bladed solid.

Diesel "S"

Twin-screw reversible opposed piston two-cycle engines to maintain 13,600 bhp (87 per cent mechanical efficiency) at 116 rpm. Each engine of six cylinders 670 mm. diameter and 2,320 mm. combined stroke.

Generators: Three 6-cycle 325 kW at 325 rpm and 220 volts.

Propellers: 17 ft. 6 in. diameter, 17 ft. 6 in. pitch, 115 sq. ft. surface.

Blue Funnel "P"

Normal service power of 14,000 shp with a maximum of 15,000, the drive being through double reduction gearing.

Generators: Turbo (no rating given).

Propellers: 20 ft. diameter, no pitch given, surface approximately 190 sq. ft.

Model tests were carried out and the following results obtained for an equivalent of 18,000 tons displacement at 17 and 19 knots.

	Steam "S" (P. & O.)	Diesel (P. & O)	Blue Funnel "P"
Speed	17	19	18·5
ehp	4,941	7,730	7,180
Wake, per cent	30·5	28·4	41·5
Hull efficiency	1·117	1·106	1·127
Relative rotative efficiency	0·998	0·991	—
Qualified hull efficiency	1·114	1·1097	—
Propeller efficiency in open	0·632	0·624	0·594
Quasi-propulsive efficiency	0·705	0·685	0·710
Speed length rates	0·768	0·858	0·845

The figures on the right are those obtained from models of the Blue Funnel "P" class ships given in the paper.

From the principal dimensions, model results, etc., a graph of the speed against ehp was drawn up for both steam and diesel ships, and by comparing the following figures it can be seen that for equal speeds the ehp for steam propulsion is lower than that for diesel propulsion.

Steam		Diesel	
Speed	ehp	Speed	ehp
13	2,384	13·2	2,385
14	3,014	14·3	3,100
15	3,755	15·4	4,050
16	4,571	16·5	5,510
17	5,459	17·6	7,410
18	6,555	—	—

The first ship to be delivered was the twin-screw diesel vessel *Soudan*, and trials took place over the Skelmorlie measured mile. When the single-screw *Surat* was ready for trials she sailed over the same course under similar weather conditions, the wind being force 3 on the beam for the *Soudan* and force 4 in the case of the *Surat*. The trial results are set out below:—

	Surat	*Soudan*
Mean draught	16 ft. 7 in.	16 ft. 10½ in.
Displacement	9,350	9,565
Revolutions	127	118
shp	12,860	13,490 ihp = 11,710 shp
Speed	20·12	18·97
Speed length ratio	0·908	0·857
Slip, per cent	1·22	6·5
A.C. on shp	281·1	262·7
A.C. on shp at 18·97 knots	281·1	262·7

These results again bear out the contention that for equal speeds the hp for single-screw steam propulsion is lower than for twin-screw diesels. This is apparent, for in the former the additional resistance of the bossing is eliminated.

A fuel consumption trial followed the run over the measured mile, the *Soudan* for 12 hours' duration against 9 hours for the *Surat*, which figures compare favourably.

	Surat (9 hours)	*Soudan* (12 hours)
Revolutions	122·13	114
shp	11,449	11,937 ihp (9,675 shp at 81 per cent)
M.E. fuel	16 tons	18·5
Gen. fuel	16·62	19·45
Lb./shp/hr.	0·5814	0·305 (ihp) 0·377 shp

These ships have now been on continuous service to the Far East for several years, and a comparison has been made and average figures for the two ships covering a period of two years (six voyages) is quoted.

	Surat	*Soudan*
Total distance at sea	25,923	25,334
Average draught	24 ft. 5 in.	24 ft. 2 in.
Average displacement	14,455	14,505
Average speed	16·7	16·07
Average slip	3·5	9·5
Average revolutions	108·2	102·8
Average shp	8,450	8,950
Fuel coefficient	51,449	71,104
Fuel/day	56·3	34·84
Admiralty coefficient	327	275
Fuel, lb./shp/hr.	0·62	0·363

It will be seen that the *Surat's* average speed is almost 1 knot more than that of the *Soudan*. This is accounted for by the fact that when running into heavy weather diesel machinery is governed by its m.i.p., and cannot be driven, whereas turbine machinery is not affected to the same extent.

Repair and survey costs were also compared over the same period (six voyages) and these bear out the original conception that the running costs of a turbine are less than those of diesel by £300–£400 per voyage.

	Surat	*Soudan*
	£	£
Main engine repairs	647	3,254
Auxiliary repairs	1,731	1,024
Boiler repairs	5,059	87
Dry dock—		
Propeller, etc., repairs	708	1,061
Tail shaft repairs		
Gen. repairs	1,529	1,674
Main engine survey	13	3,582
Auxiliary engine survey	—	740
Boiler survey	1,638	—
Dry dock—		
Propellers, etc., survey	179	184
Tail shaft survey		
Gen. survey	1,127	900
Number days	845	788
Total costs	19,207	20,089
Rate/day	22·7	25·5
Stores rate/day	6·5	7·6
Stores and repairs	29·2	33·1

The total cost includes the main machinery repairs and survey costs as set out above, and also such items as electric wiring, ventilation, deck, machinery, pipes and connections, etc. Lubricating oil is included in the stores.

Author's Reply

Commander Cooper and Mr. Sampson have both pointed out the importance of the combustion equipment in the economic operation of a steam ship. The author entirely agrees with their views and would only add that it is essential for the oil-burning equipment to be simple to operate, and for the design of the boiler to be closely related to the requirements of the oil fuel registers when burning the most difficult fuel in the range from 50 to 6,500 seconds Redwood I. Only in this way can foolproof combustion equipment be provided.

The author generally agrees with Mr. Pemberton's comments regarding the probable limiting factors in single-screw propulsion, but would also like to draw attention to the points made by Mr. McAlister; indeed, it seems quite probable that the sheer physical problem of making and handling the propeller will set a limit to the wide application of very high powers on a single screw.

The tailshaft liners that have slackened have all been forced on with an interference fit of between 4 and 7 thousandths of an inch by a press without heat. No trouble has been found with liners shrunk on by heating as these have a greater interference fit, nor has any undue wear of the lignum vitae been found to suggest that the cause could have been lack of cooling or "lubrication."

The experience with plastic materials has been confined to those cases where no additional water supply was provided. Wear has appeared to take place in fits and starts, as no wear occurred for perhaps nine months and then in three months $\frac{1}{8}$ in. took place. All these bearings have now been replaced by lignum vitae. The heat generation during this rapid wear has not caused a liner to move on the shaft. The only other case of the movement of a liner in the Blue Funnel fleet was a Sam ship when a rubber stern tube lining was used without additional water cooling. Wear down was rapid and, when after about 8,000 miles sudden rapid heating took place, the liner stopped and moved up the shaft under the influence of the rubber ring.

With regard to the destannification of liners, the phenomenon has now been found in varying degrees on every liner that has been examined. All these liners are standard gunmetal of the following composition range:—

Zinc	1·77–2·00
Tin	9·40–10·7
Lead	0·06–0·4
Nickel	0·0–0·5
Iron	0·06
Phosphorus	0·016–0·03
Copper balance	

Some have been sand cast and some centrifugally cast; within the limit of our observations it would be true to say that the effect is somewhat more pronounced on centrifugally cast liners. The author welcomes Mr. Pemberton's offer of the assistance of the Research Department in solving the mystery.

Mr. Archer's comments on the gearing are of interest and the recommendation for Marco is particularly valuable. He differs from the author somewhat in nomenclature of gearing wear, but this was not unexpected. His comments on torsionmeters are of value, but the author feels that what is required in this field is a torsionmeter that is robust enough to stand up to service conditions and to maintain its accuracy within $\pm 2\frac{1}{2}$ per cent. This is far from being achieved yet—as indeed it is far from being achieved with more essential equipment such as pressure gauges and thermometers.

The main injection pipes to which Mr. Sherborne refers were fabricated from copper sheet and had been in service up to about two years. The rubber coat was applied to the finished pipes and was carried round to form the jointing material on the flange faces. The isolated failures of the aluminium brass condenser tubes in the ships voyaging to Australia continues and the Far Eastern ships continue to be free from these defects. Whilst the author agrees that one swallow does not make a summer, a series of isolated condenser tube failures can cause considerable cost and delay in steam ships.

In reply to Professor Telfer, in Table XI, columns 1 and 3, X (2), and ABC (11), the propellers are supplied by the Manganese Bronze and Brass Co., Ltd., while the second column Y (10) are all of propellers made by J. Stone and Co., Ltd.

From complete voyage results we find that Stone's propellers under service conditions give 103/104 rpm and Manganese Bronze 106 rpm.

Columns 4 and 5, X (6), and ABC (8), are results of voyages with propellers of both firms.

In reply to Mr. Wood, the funnel gas exit velocity was raised to about 120 ft./sec. by the modifications; with the additional air pressure required for the oil fuel registers this was the practical limit to the fan capacity. With regard to Mr. Wood's comments on the condenser tubes, the author emphasizes that it is only the ships voyaging to Australia that experience the trouble. Similar ships building at the same time but running between the United Kingdom and Far East do not experience the defects.

Mr. S. A. Smith's comments in the comparison with the P. & O. Company's *Surat* class are interesting, but he has misinterpreted the data regarding the Blue Funnel ships. The maximum sea speed of the "P" and "H" class ships is of course considerably in excess of 18 knots.

The author is surprised to see the high revolutions of the P. & O. single-screw steam ships and presumes that this speed was chosen in order to retain single reduction gearing for the i.p. and l.p. turbines.

The rating of the turbo-generator fitted in the "P" and "H" class ships is 550 kW, but the peak sea load seldom exceeds 400 kW. There is thus adequate margin of power for all seagoing purposes.

The following voyage figures may be of interest in comparing the results of the motor ship *Soudan* with two Blue Funnel motor ships.

	Glengarry (twin screw)	*Bellerophon* (single screw)	
Total distance at sea	24,599	26,103	21,378
Average draught	26 ft. $1\frac{1}{2}$ in.	25 ft. 1 in.	25 ft. 9 in.
Average displacement	14,943	13,098	13,514
Average speed	17·42	16·07	16·29
Average slip	5·8	− 1·9	− 3·2
Average rpm	105·55	106·5	106·7
Average bhp	11,440	6,698	6,688
Fuel coefficient	66,180	80,660	87,935
Fuel/day	48·46	27·9	27·89
Admiralty coefficient	280	336	367
Fuel, lb./bhp/hr. (all purposes)	0·396	0·389	0·389