A MARINE ENGINEERING REVIEW—PAST, PRESENT, AND FUTURE

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Read in London at the Spring Meeting of The Royal Institution of Naval Architects, on March 22, 1960, Viscount Runciman of Doxford, O.B.E., A.F.C., D.C.L. (President), in the Chair.

Summary

The paper begins with a brief review of the state of marine propulsion machinery in 1860, when the I.N.A. was founded, and proceeds to examine the developments which have taken place in the century which has elapsed since then.

The important landmarks in the development of the reciprocating steam engine and steam turbine and cylindrical boiler are outlined, and the introduction and subsequent development of the water-tube boiler. The gradual change from coal-burning to the almost exclusive use oil fuel is referred to. Sections follow on the internal combustion engine, including heavy oil engines, free-piston machinery, and gas turbines. The use of nuclear heat sources to drive steam and gas turbine propulsion units is examined, and this is followed by a reference to the nuclear fusion experiments currently in progress, with their great promise for the future.

The paper ends with a tribute to the Government Departments, Institutions, inventors, and the great body of marine engineers who have all contributed to the progress of marine engineering over the past 100 years.

Introduction

Among the first group of professional papers given before this Institution, in March 1860, was one entitled, "On Various Means and Appliances for Economizing Fuel in Steam Ships," by Robert Murray, Engineer Surveyor to the Board of Trade. This paper began by stating: "The Naval Architect and the Marine Engineer are now so intimately connected in the earnest endeavour of both professions to produce that grand effort of modern science 'a perfect steamship' that I am induced to hope that the subject I have selected for this paper may not be considered out of place though addressed to an Association of Naval Architects." In conclusion the author hoped "that the day is not far distant when the average consumption of marine engines will be reduced to nearly one half of what it is now." His goal, long since passed, was "to induce our marine engines to content themselves with 3 or even $3\frac{1}{2}$ lb. per indicated horsepower per hour."

In the same month one hundred years later the present author has the honour further to develop this theme by first giving in broad outline the changes during one hundred years of marine engineering, summarizing the present state of the art and speculating on future trends.

Machinery Usage in 1860 and at Present

In 1860 when our Institution was founded, of the total of ships under British Registry only about 9 per cent was propelled by steam machinery. Table I shows how rapidly steamships increased until in 1871 they formed nearly one quarter of all ships on the British Register. 1860 was the height of the clipper ship period, the *Cutty Sark* was only built in 1889. Only steam reciprocating engines of various types were in existence then and for nearly forty years to come. The modern position is given in Table II and Fig. 1 showing the main machinery types in relation to the total gross tonnage launched each year. Table II gives the types of machinery in service in 1948 and 1958, from which it will be seen how much the reciprocating steam engine has declined and the diesel and turbine machinery increased in service. The trends can be seen even more clearly in Fig. 1. The heavy-oil engine at 57·2 per cent of all tonnage launched in 1958 is the most important type of propulsion machinery for ships, The corresponding figure for geared turbine machinery is 41·1 per cent. It will be seen that after a long and exciting history the use of the steam reciprocating engine for propulsion is drawing to an end as in 1958 only 1·3 per cent of world tonnage launched was to be fitted with steam reciprocating engines (including sets fitted with exhaust turbines). The corresponding figure for ships launched in Great Britain and Ireland in the same year was 0·29 per cent fitted with steam reciprocating machinery of any type.

Steam Reciprocating Machinery

Historically the first type of marine propulsion machinery to be examined is the reciprocating steam engine. The finest warship in 1860 was H.M.S. *Victoria*, launched the previous year. She was a wooden battleship 260 ft. long, 60 ft. broad, with a displacement of 7,000 tons. Her steam machinery, treated largely as auxiliary to sail power, developed an actual 4,200 indicated horsepower. When the machinery was required the order was "Up funnel and down screw."

A typical merchant steamship constructed in 1860 would be of iron rather than wood and would be more likely to be propelled by a screw than to have paddles. At this date speed increasing gearing between the steam machinery and the screw was becoming obsolete. Any new engines would tend to be direct-acting rather than driving the screw shaft through beams or levers. A very wide arrangement of cylinders was possible, but the first few inverted engines which were to become the standard form for the future were coming into service. A typical steam engine of this date would probably be single expansion rather than compound.

Robert Murray's paper, already referred to, gives particulars of steamships in 1860. There were 50 single expansion steam-engined ships and no compound engines in the list. A similar paper given by him to this Institution in 1865 tabulated particulars of 26 sets of machinery of which 6 showed compound engines and 20 had single expansion types of machinery. The

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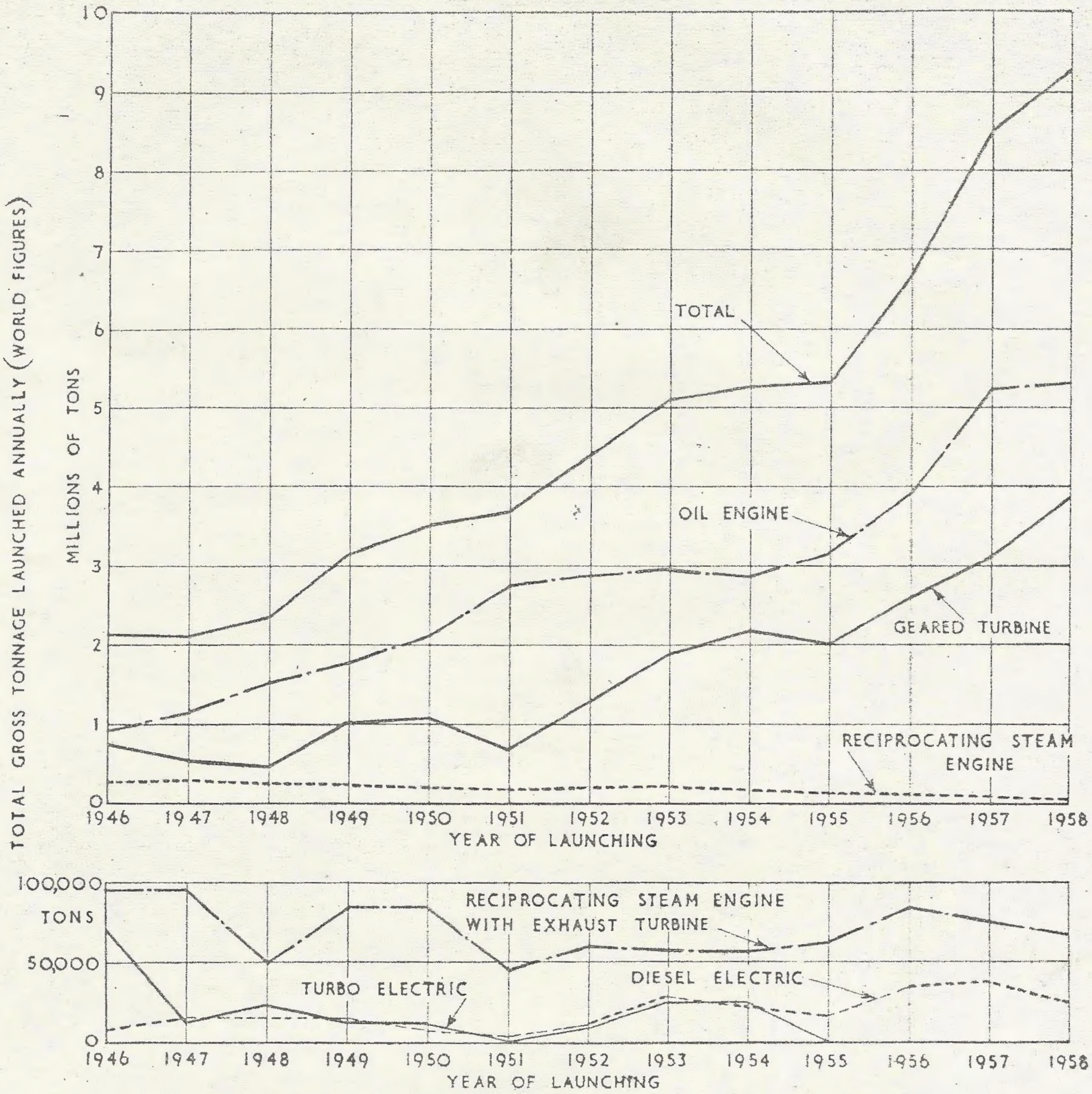


FIG. 1.—TOTAL GROSS TONNAGE OF MERCHANT SHIPS LAUNCHED ANNUALLY

steam pressure in the compound engines was no higher than that used in the single expansion engines which may account for the difficulties experienced at that time in showing a clear gain in all cases for the new compound machinery. In fact a compound steam engine was regarded in the 1860's as being of an experimental nature, but was almost universally adopted in the 1870's.

Superheating was tried experimentally in two naval vessels (*Black Eagle* and *Dee*) in 1856 and was used after 1860 in most designs for naval machinery. For merchant vessels, Robert Murray stated:

> "it may now be considered certain that superheating the steam is desirable for all vessels, which make long voyages, and which expand in the cylinders to any considerable extent" (TRANS. I.N.A., 1860).
>
> "it is found better to limit the heat of the steam at the superheater to 300°, or 320° at the utmost, a higher temperature being detrimental to the engines" (TRANS. I.N.A., 1865).

Surface condensers were patented by Samuel Hall in 1834, but after early trials were abandoned, partly owing to increased cost, partly owing to fouling of condensers and increased corrosion in the boilers (due to absence of scale present when salt water was used). Such condensers were later re-introduced, largely by Humphreys and Spencer. s.s. *Mooltan*, 1859, a P. & O. liner, fitted with a surface condenser gave such satisfactory results that subsequently surface condensers became general.

Fuel consumption was of the order of 3 to 4 lb. of coal per indicated horsepower hour.

In the matter of engine types an Admiralty Committee reported in 1858:

> "of all the variety of engines that have been purchased by Government for our ships of war, the following are so far superior to all others, that no engines of an older make should ever again be put on board:
>
> (1) the single-piston-rod engine, with the connecting rod attached direct to the crankshaft and with a single flat guide,
>
> (2) the engine commonly known as the trunk engine, and patented by Messrs. Penn & Sons,
>
> (3) the double-piston-rod engine."

In 1860, the outstanding merchant ship which had just been completed was the *Great Eastern*. She first put to sea from

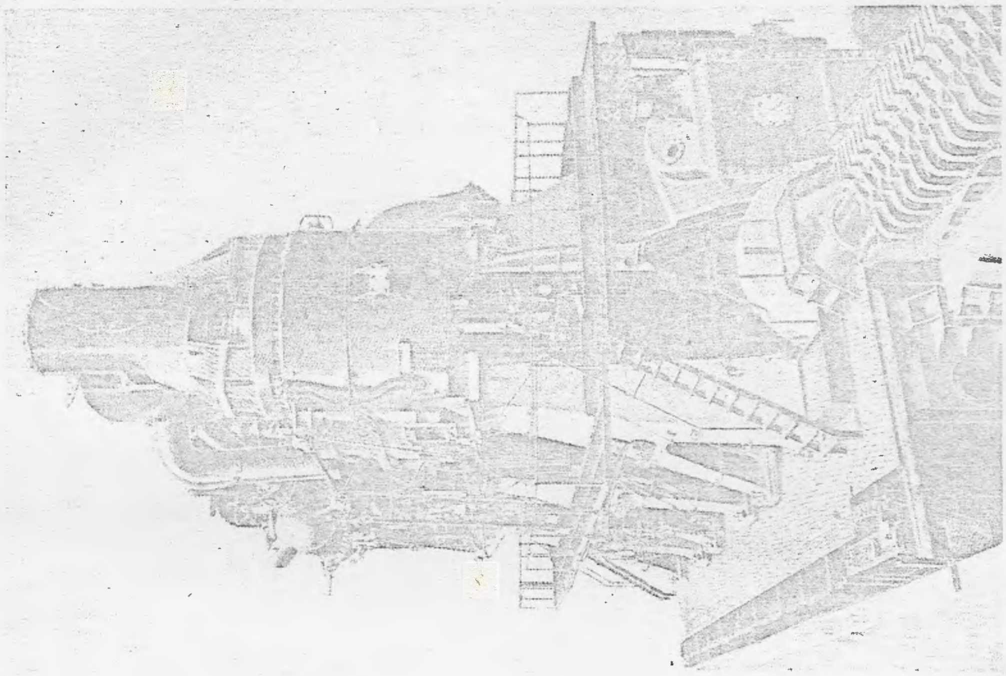


FIG. 3.—FIVE-CYLINDER TRIPLE-EXPANSION ENGINES OF S.S. "CAMPANIA"

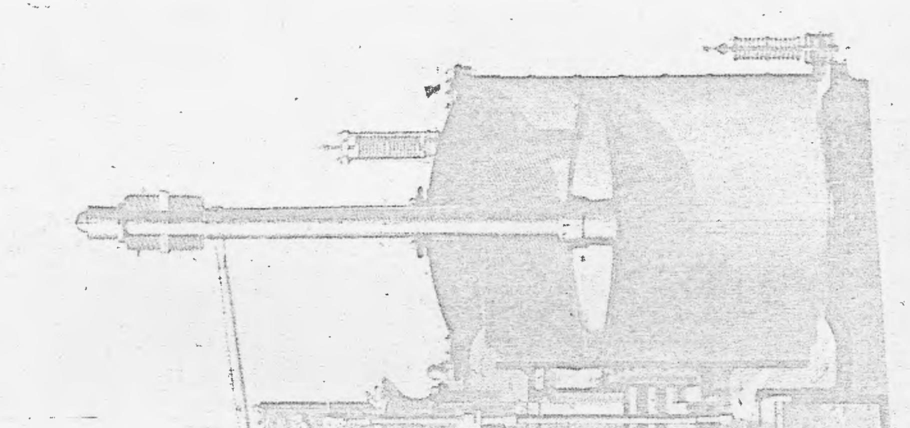


FIG. 2.—SIDE-LEVER PADDLE-WHEEL ENGINES OF S.S. "ARABIA"

Millwall on the Thames on September 9, 1859. Her principal dimensions and machinery details are given in Table III.

It may be remarked that the *Great Eastern* remained the largest ship in the world until 1899 when Messrs. Harland & Wolff built the *Oceanic*. Figures for this ship are compared with those for the *Great Eastern* in Table III, and show clearly the vast improvements in steam machinery effected in forty years.

It must be remembered, however, that the *Great Eastern* was a very exceptional ship and that in following out the changes in the reciprocating steam engine it is necessary to select those which represent the most typical as well as the largest of those in service at the appropriate date. On this basis the steam engines for the paddle steamer *Scotia* should be mentioned. The ship, built in 1861, was the last paddle-driven steamer built for the Cunard Line and was also the last built for ocean-going. She held the Blue Riband of the Atlantic from 1862–67. Her machinery was built by Robert Napier & Sons at Glasgow, and similar machinery built by him is shown in Fig. 2 (further particulars are given in Table IV). In Fig. 2 the steam is shown exhausting from the bottom cylinder to the jet condenser, while the top cylinder takes steam. This side-lever engine had two cylinders 100 in. diameter × 144 in. stroke and developed 4,632 ihp, and if not the largest is among the largest simple expansion steam reciprocating engines for marine propulsion. In most instances in Table IV the largest engine of a type has been chosen as far as possible to illustrate the current position, as new types of engine are usually built first in small sizes and may take some time to become established.

Among the largest sets of compound engines which I have been able to find were those fitted in the *City of Rome*, 1881, which had three pairs of tandem cylinders, H.P. 43 in. diameter, and L.P. 86 in. diameter by 72 in. stroke driving three cranks, developing 11,890 ihp with 90 lb./in.^{2} pressure in the boilers. This represents the largest compound engine of this type. Before that, however, there had been many compound engines fitted in very important ships. For example, the first ship built for the Orient Line by John Elder & Co., the *Orient* (1879), had a three-crank compound engine

$$\frac{\text{H.P. cylinder 60 in., 2 L.P. cylinders 85 in.}}{\text{60 in. stroke}}$$

developing 5,400 ihp at 75 lb./in.^{2} pressure in the boilers.

Some of the largest cylinders ever built were fitted in compound engines at about this time. For example, in 1873 the Mail Steamer *City of Chester* was fitted with compound machinery having an H.P. cylinder of 72 in. diameter and an L.P. cylinder of 120 in. diameter, the stroke being 66 in. With steam at 80 lb./in.^{2}, 4,600 ihp was developed at about 47 *rpm*. Sister ships *City of Richmond* (1873), Todd & McGregor, Glasgow, and *City of Berlin* (1874), Caird & Co., Greenock, also had L.P. cylinders of 120 in. diameter associated with H.P. cylinders of 76 in. and 72 in. respectively. Even larger was the compound machinery of the Imperial Russian Frigate *General Admiral*, which had an H.P. cylinder of 92 in. and an L.P. cylinder of 130 in., with a stroke of 48 in. The engine was designed to develop 6,300 ihp at 65 *rpm* with a working pressure of 60 lb./in.^{2}.

This L.P. cylinder is the largest diameter of steam cylinder known to the writer, although larger cylinders were used in the hot air engines of the *Ericsson*. These engines had four cylinders of 14 ft. (168 in.) diameter × 72 in. stroke, but developed only 300 ihp at 9 *rpm* with a mean effective pressure of 2 lb./in.^{2}. Although the coal consumption in terms of ihp was good for the time (1·87 lb./ihp/hr.), the engine was a commercial failure.

The *Britannic* and *Germanic* (1874), built by Harland & Wolff, obtained their engines from Maudsley, Son & Field of London. The machinery was of the double compound direct-acting type having 2 H.P. cylinders 48 in. driving in tandem with the 2 L.P. cylinders of 83 in., the stroke being 60 in.

The *Umbria* and *Etruria*, built by Messrs. John Elder & Co. in 1884 for the Cunard Company, were the last large single-screw steamers built for the Atlantic run, and had the highest powered compound engines of all. H.P. cylinder 71 in. diameter with 2 L.P. cylinders 105 in. diameter by 72 in. stroke developing 14,321 ihp at 69·9 *rpm*.

Triple expansion engines were introduced earlier by Dr. Kirk when employed by John Elder & Co. in the ship *Propontis*, 1874. This ship developed trouble in a water-tube boiler which was well before its time, and it was not until 1881 that the next triple expansion engine of his design was fitted in the *Aberdeen*. The machinery was built by the firm R. Napier & Sons in which Dr. Kirk was then a partner. The engines had cylinder sizes

$$\frac{\text{H.P. 30 in., I.P. 45 in., L.P. 70 in. dia.}}{\text{54 in. stroke}}$$

and worked at a pressure of 125 lb./in.^{2}.

Particulars of the twin-screw triple expansion machinery fitted in the *City of Paris* and *City of New York* built by J. & G. Thomson, Clydebank, is shown in Table IV. The cylinder sizes were:—

$$\frac{\text{H.P. 45 in. dia., I.P. 71 in. dia., L.P. 113 in. dia.}}{\text{60 in. stroke}}$$

developing on twin screws 20,117 ihp at 90·8 *rpm*. This appears to be the largest L.P. cylinder diameter used at sea in any triple-expansion machinery. Incidentally, these ships introduced the fashion of twin-screw machinery on the Atlantic run.

In 1892 the Elder Yard (The Fairfield Shipbuilding & Engineering Co.) produced most important machinery for the Cunard liners *Campania* and *Lucania*. The machinery consisted of twin-screw 5-cylinder 3-crank triple expansion engines comprising 2 H.P. cylinders per engine 37 in. diameter, 1 I.P. cylinder 79 in. diameter, and 2 L.P. cylinders 98 in. diameter × 69 in. stroke developing a total of 31,050 I.H.P. at 84 *rpm*. Fig. 3 taken from *Engineering* gives a good impression of this machinery.

After these ships, large British passenger ships to be fitted with reciprocating engines were the *Oceanic*, 1899, and *Caronia*, 1904. The *Ivernia*, 1899, built by C. S. Swan & Hunter and engined by The Wallsend Slipway & Engineering Co. Ltd., was the first ship to use quadruple expansion machinery on the Atlantic run. These ships, were, however, overshadowed by large German liners all built by the Vulcan Company at Stettin. For convenience the main particulars of the machinery of these ships are tabulated in Table V. These German ships held the Blue Riband of the Atlantic for nearly ten years until knocked out by the turbine-engined ships *Mauretania* and *Lusitania* in 1907. The *Mauretania* then held the record for 22 years.

Another large single-screw triple expansion engine was that fitted in the P. & O liner *Caledonia* (1894). She had a three-crank triple engine consisting of 2 H.P. cylinders, 33 in. diameter, in tandem with 2 L.P. cylinders 84 in. each driving separate cranks. The I.P. cylinder, 69 in. diameter, drove the centre crank. The stroke was 72 in. and the machinery developed 11,000 ihp with a boiler pressure of 170 lb./in.^{2}. The eccentric sheaves were about 4 ft. 6 in. diameter being fitted on the couplings connecting her three single throw crankshafts which made up the engine.

Triple expansion engines were introduced into the Navy in 1885, usually in the form of a 4-crank engine and were used up to the general introduction of steam turbines. Quadruple expansion engines were not used.

Fig. 4 shows graphically the difference between naval and merchant reciprocating machinery in 1900. The naval machinery has higher pressure (300 lb./in.^{2}), higher *rpm* (120) and light construction. In contrast the *Deutschland* machinery, W.P.

220 lb./in.^{2}, of greater power (36,940 ihp), but lower *rpm* (77·4) is much larger and heavier, but more economical, the figures for ihp produced per ton of engine being 12 and 6·86 respectively for the two engines.

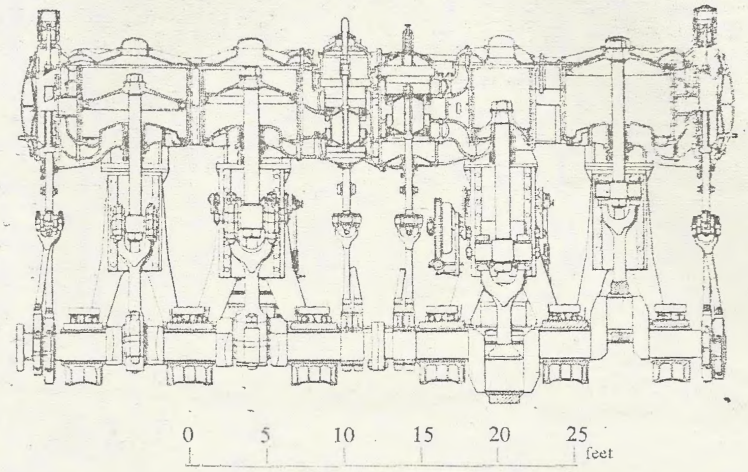
The last large naval reciprocating engines fitted in a British warship were built by Scotts' Shipbuilding & Engineering Co. in 1909 for installation in H.M.S. *Defence*. The machinery developed 27,000 ihp at 125 *rpm* at a piston speed of 1,000 ft./min. Destroyers with reciprocating engines worked up to 400 *rpm* with a piston speed of 1,200 ft./min.

Later reciprocating machinery after the general introduction of steam-turbine machinery was only fitted in smaller British warships, e.g. the *Flower* class sloops in the 1914–18 war, and the corvettes and frigates in the 1939–45 war.

After the giants built at the turn of the century the steam reciprocating engine declined in importance except in the case of combination machinery having reciprocating machinery driving the wing shafts exhausting into an L.P. turbine driving a centre screw.

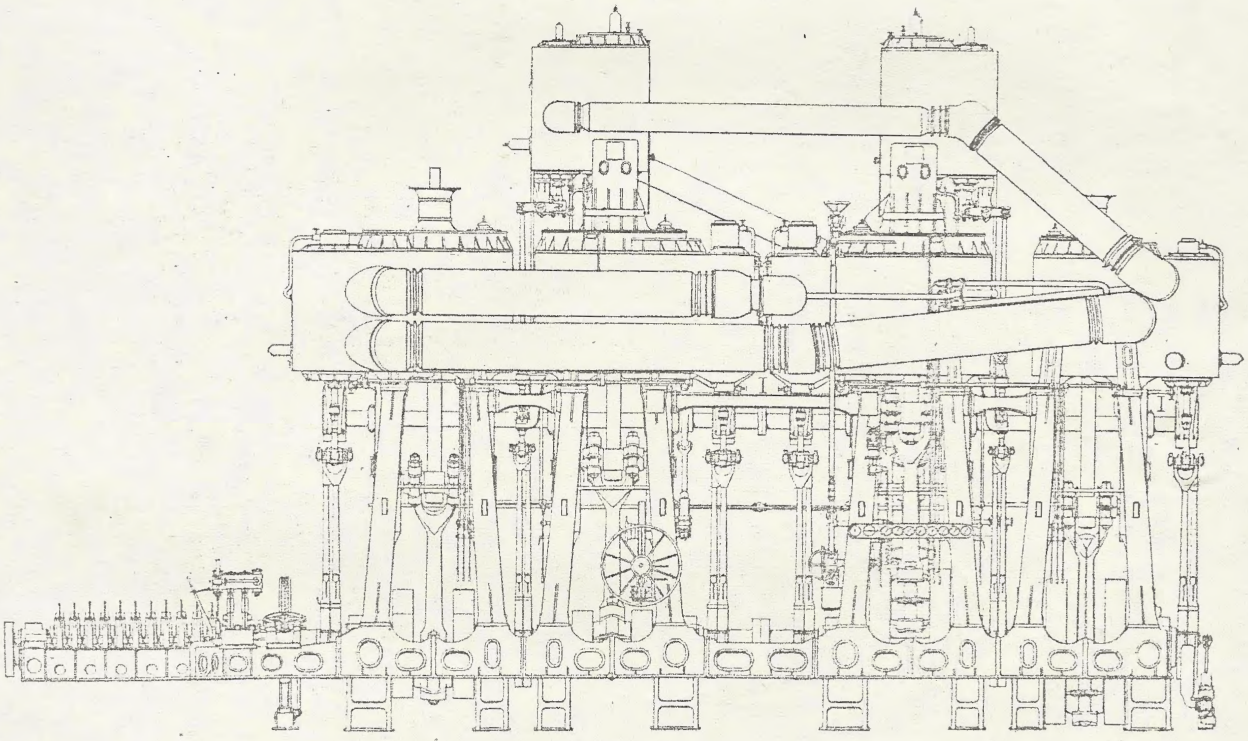
Messrs. Harland & Wolff were the great exponents of this drive, building the *Laurentic*, 1909, and *Olympic* and *Titanic*, 1910. The designed power of ships was 2 × 17,000 ihp from the wing reciprocating engines and 17,000 shp at 165 *rpm* from the turbine driving the centre screw. The reciprocating engines were 4 crank triples, H.P. 54 in., I.P. 84 in. and 2 L.P.'s, 97 in. by 75 in. stroke, and ran at 77 *rpm*. A sister ship, *Britannic* (1914), had the same power. These L.P. turbines are again referred to in the section of the paper dealing with steam-turbine machinery.

In summary, Table IV shows that although the boiler pressure



H.M.S. *King Alfred*
4-crank triple-expansion engine
30,000 ihp at 120 *rpm*

Scale for both diagrams



T.S.S. *Deutschland*
6-cylinder 4-crank quadruple expansion engine
36,940 ihp at 77·4 *rpm*

FIG. 4.—COMPARISON OF HIGH-POWERED NAVAL AND MERCHANT SHIP RECIPROCATING ENGINES OF 1900

rose from 24 psig. to 225 psig., the equivalent effective pressure referred to the L.P. cylinder(s) only rose from 20 to 43 lb./in.^{2}, which meant an earlier cut-off and hence more "expansions" in the cylinders to produce economy. The corresponding piston speed rose from about 300 to 1,000 ft./min. in merchant ships in the period. Even more important, however, is the change in fuel consumption. At the beginning of the period the average figure was about 4 lb. of coal per ihp per hour, and at the end before the final extinction of the steam reciprocating machinery as the usual type, it achieved a coal equivalent consumption of 0·875 lb./ihp per hour in engines of comparatively low ihp. To achieve the latter figure, however, an exhaust turbine has to be fitted as well as the use of superheat and reheat.

Indicator cards are shown in Fig. 5 for two engines about the beginning of the period under consideration:

(*a*) from a single expansion double-acting beam engine similar to that shown in Fig. 2, and

(*b*) from an inverted or steam hammer engine in the ship *Thunder* (1859). The much better expansion line with superheated steam is apparent.

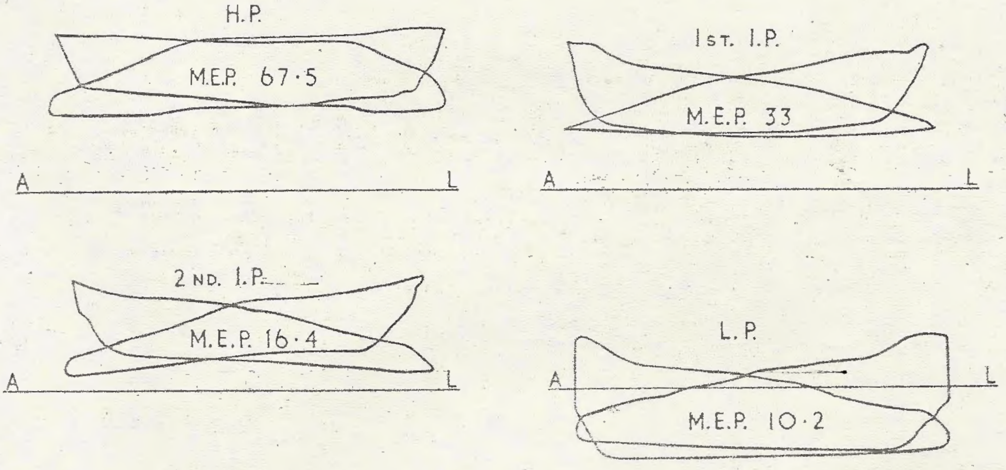
The same figure shows the top and bottom cards from a quadruple expansion engine (1920).

The subsequent Fig. 6 shows the combined diagram for the quadruple expansion engine contrasted with the previous cards and a temperature entropy diagram for these examples, which clearly shows the change in heat drop handled by the machinery from 1860 to its decline.

Before being reduced to near extinction, great efforts were made to improve the economy of the reciprocating steam engine by reheating, and use of an exhaust turbine in many forms. The machinery for *Baron Ardrossan*, the last item in Table IV, shows how much was achieved in economy. It was, however, the end of the development as pressures could not go much higher (225 lb./in.^{2}) as cylinder lubrication involved using boilers with plenty of capacity, hence cylindrical boilers. With piston rings working on cast-iron liners in the cylinders, temperatures were limited (750° F. max.) and the exhaust turbine utilized economically all the energy in the exhaust steam from the reciprocating engine.

(A) TYPICAL SIMPLE EXPANSION ENGINE CIRCA 1860
2 CYLINDERS 71" DIA. X 36" STROKE
2001 I.H.P. AT 63½ R.P.M.
STEAM CONDITIONS 20 P.S.I.G. SATURATED

(B) S.S. THUNDER 1859 (SIMPLE EXPANSION ENGINE)
2 CYLINDERS 55" DIA. X 36" STROKE
696 I.H.P. AT 50 R.P.M.
STEAM CONDITIONS 13 P.S.I.G. & 350° F.

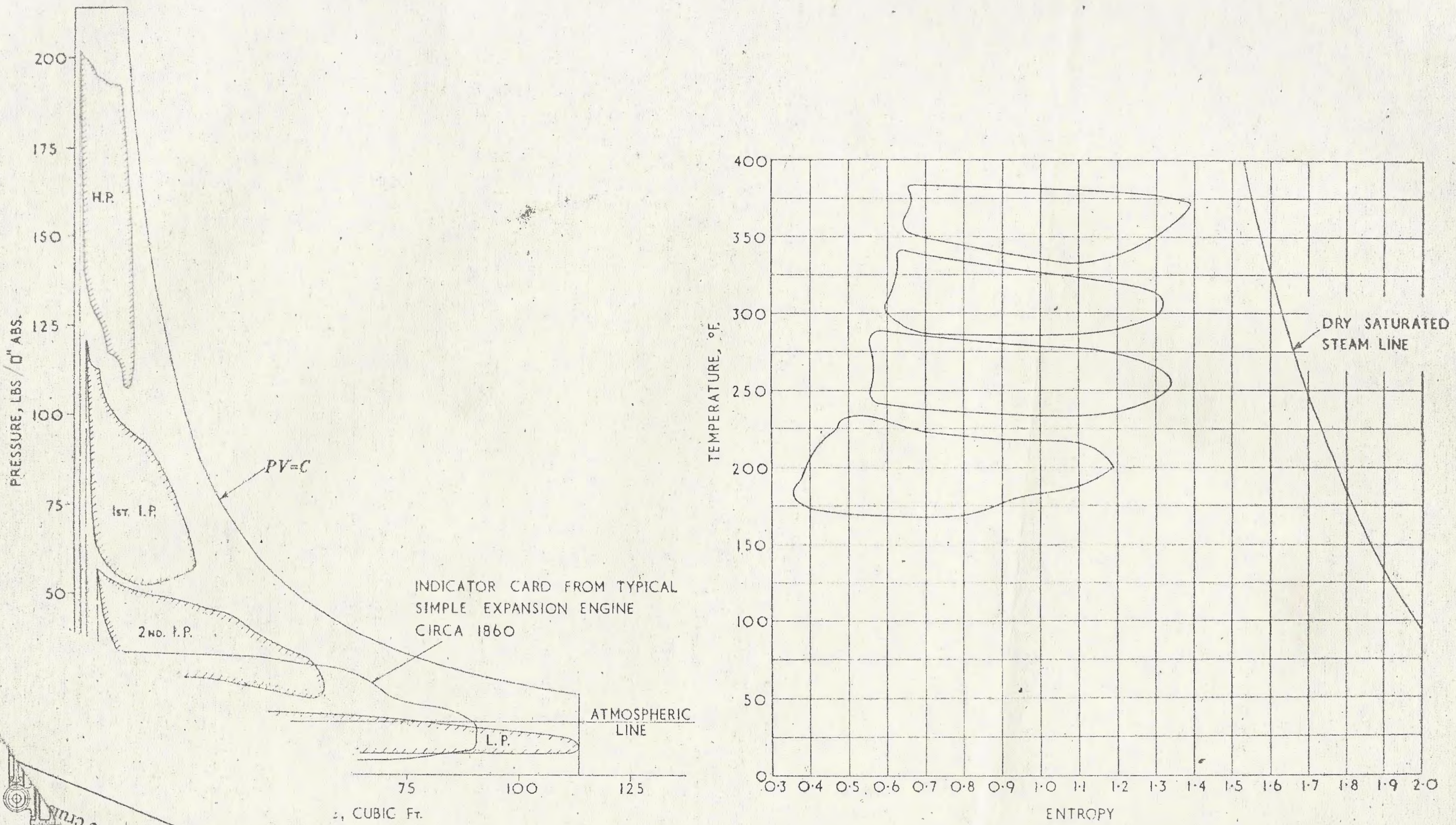


(C) STARBOARD ENGINE OF T.S.S. NARIVA 1920 (QUADRUPLE EXPANSION)
CYLINDER DIMENSIONS H.P. 23" 1st I.P. 33" 2nd I.P. 47" L.P. 67" X 51" STROKE
TOTAL I.H.P. ON TWO SCREWS 5952 AT 95 R.P.M.
STEAM CONDITIONS 200 P.S.I.G. SATURATED

FIG. 5.—RECIPROCATING STEAM ENGINE INDICATOR DIAGRAMS

Marine Steam Turbines

The pioneer worker in the development of marine steam turbines was Sir Charles Parsons. Many others worked in the same period, particularly Dr. De Laval in Sweden, Zoelly in Switzerland, Rateau in France, and Curtis in the United States



BINED INDICATOR DIAGRAM AND CORRESPONDING TEMPERATURE-ENTROPY DIAGRAM FOR QUADRUPLE-EXPANSION STEAM ENGINE
(Starboard engine of T.S.S. *Nariva*, 1920)

of America, but it was some time before marine turbines of any size incorporating their ideas were produced.

The first turbine-engined ship was the famous *Turbinia*, built in 1894 by Sir Charles Parsons, which, to begin with, gave a disappointing performance. She was re-engined in 1897 and then developed a speed of 34·5 knots with turbines of the axial flow type driving three shafts. At this speed, the combined horsepower was about 2,000 shp. *Turbinia* took part unofficially in the Diamond Jubilee Naval Review at Spithead in 1897, and directed attention to the suitability of turbines for the propulsion of high-speed warships.

The destroyers *Cobra* and *Viper* were built 1898–1900 and were fitted with steam-turbine machinery developing 12,300 shp. The fastest run of H.M.S. *Viper* (1900) was 37·11 knots. These ships were lost in tragic circumstances unconnected with the installation of turbine machinery. At this dark period (1901) the Clyde passenger steamer *King Edward* was built (250 ft. long, 650 tons displacement), showing great courage not only in the Parsons Marine Steam Turbine Company and Wm. Denny, the shipbuilders, but even more in the shipowner, Captain John Williamson. She was followed by a sister vessel, *Queen Alexandra*, in 1902. They were followed by the *Queen* in 1902, for the cross-Channel service between Dover and Calais, and many other cross-Channel vessels driven by steam turbines followed.

The next destroyer was the *Velox* with reciprocating cruising engines as well as turbine machinery for full power, followed by the *Eden* with cruising as well as main turbines.

In 1902 the Admiralty decided to adopt Parsons' turbines in H.M.S. *Amethyst*, one of four third-class cruisers. She was very successful in comparison with her sisters *Topaz*, *Sapphire*, and *Diamond* fitted with reciprocating machinery, when tried in 1904.

The first turbine-engined liners were the *Virginian* and *Victorian* in 1904. These ships were driven by three 3-bladed propellers direct coupled to one high-pressure turbine on the centre shaft (60 tons) and two L.P. turbines on the wing shafts (100 tons each). The H.P. turbine had a mean diameter of 68·75 in. at inlet and 74 in. at exhaust. The corresponding figures for the L.P. rotors were 95·75 in. to 107 in. At full speed of 19 knots the *rpm* of the turbines was 250 and the total shp developed was 12,700.

In 1904 the Cunard Company decided to adopt turbine drive for their new ships, *Mauretania* and *Lusitania*. In this same year the Cunard Company ordered the steam reciprocating-engined *Caronia* and decided that her sister ship, *Carmania* (21,000 shp), should be driven by turbine machinery. With the same boilers *Carmania* proved to be half a knot faster than her sister.

In 1905 an Admiralty commission recommended the fitting of steam-turbine machinery to the *Dreadnought*. The ship was built at speed and commissioned in 1906. Her turbines developed 23,000 shp at 320 *rpm* (displacement 17,900 tons) and drove the ship at a speed of 21·25 knots. The next British naval ships, *Indomitable* and *Invincible* (1908) had machinery developing 42,000 shp.

And so the power of destroyers, cruisers, battleships had all risen above anything which could be developed in such ships by reciprocating engines. As a matter of interest, some other warship installations of high power with direct-turbine drive are given below:

Ship	Year of design	Total shp	Number of shafts
Lion (battle cruiser) ..	1909	70,000	4
Malaya (battleship) ..	1912	75,000	4
Repulse (battle cruiser)..	1914	112,000	4

Repulse had the largest direct-drive turbine machinery fitted in naval ships. She had four shafts, the wing shafts having the I.P. ahead turbines and H.P. astern turbines in a common casing. The inner shafts were driven by the H.P. ahead turbines driving in tandem with the L.P. ahead and astern turbines, again in common casings. The first row wheel in the H.P. turbine had a Curtis stage with three moving blade rows 126 in. P.C.D. Some further Curtis wheels were followed by reaction blading right up to the L.P. exhaust which had "wing" blading 21 in. long at 151 in. P.C.D. and the *rpm* 275. On the inner shafts the overall length from the forward end of the H.P. turbine to the shaft coupling at the aft end of the L.P. turbine was 42 ft. 6 in.

We have to return to the *Mauretania* and *Lusitania*. They were commissioned in 1907 and the turbines developed 70,000 shp on 4 shafts at 180 *rpm*. There are 6 turbines in all, 2 high-pressure turbines placed in the wings and 2 low-pressure turbines driving the inner shafts for going ahead, and two high-pressure units for going astern. These latter turbines are coupled to the inner shafts forward of the L.P. turbines. These ships represented the highest powered merchant ships propelled by direct-coupled steam turbines until the *Majestic*, formerly *Bismark* of 1914.

By the kindness of the Cunard Company, Figs. 7, 8, and 9 show the direct-coupled turbines fitted in the *Aquitania* in 1913. They developed 60,000 shp total on 4 shafts. They are chosen to illustrate the highest state of development of the direct-coupled turbines before they were displaced by the advent of gearing in high-powered ships. In this ship the port-wing shaft is driven by the H.P. ahead and H.P. astern turbines which are arranged in separate casings. The starboard wing shaft is driven by I.P. ahead and H.P. astern turbines, the inner shafts being driven by identical L.P. ahead and astern turbines, mounted in a single casing. This was the normal arrangement of running and provided the greatest expansion of steam passing in series through the H.P., I.P., and L.P. turbines before passing to the condenser. Fitted in this ship, however, was a special valve arrangement to enable boiler steam to go direct to the I.P. ahead turbine as well as the H.P. ahead turbine which then exhausted direct to the associated L.P. turbines on the inner shafts. The H.P. drum diameter was 9 ft. 2 in. and the L.P. last row blade height 20·75 in. at a P.C.D. of 177·75 in. The overall length of the L.P. ahead and astern casing was 53 ft. 6 in.

In this same period the largest L.P. turbines operating on the centre shaft in conjunction with reciprocating engines in the wings were those fitted to the *Olympic* (1910) and her sisters. The turbine developed a maximum of 19,270 shp and 17,000 shp in service at 165 *rpm*. The turbine complete weighed 639 tons, the rotor alone weighing 155 tons. The overall length of the turbine was 49 ft. 8 in. with a bladed width of 14 ft. 11½ in. There were 6 expansions taking steam at 10 lb./in.^{2} absolute, and expanding to 28½ in. vacuum with a 30 in. barometer. The initial blade height was 16 in., last blade row (wing-blade settings) 26½ in. with a mean diameter at exhaust of 176·5 in.

In the period from 1901 to 1914 blade speeds did not greatly change, rising from 70 to 80 ft./sec. for H.P. turbines, and 110 to 135 ft./sec. for L.P. turbines. Inlet pressures only rose from 150 lb./in.^{2} to 200 lb./in.^{2} saturated stream, but consumption improved from 1·7 lb. of coal/shp hour to 1·12 all purposes in large installations as a result of a number of detailed improvements in the parts.

In all these direct-drive turbines, the turbine blade speeds were much too low for high efficiency, and the large turbines would have suffered severely from distortion had superheated steam been used. They were difficult enough to heat up. The propeller speeds were also much too high for maximum efficiency and the designs formed a rather unsatisfactory compromise.

In 1892 Dr. De Laval in Sweden had developed very elegant

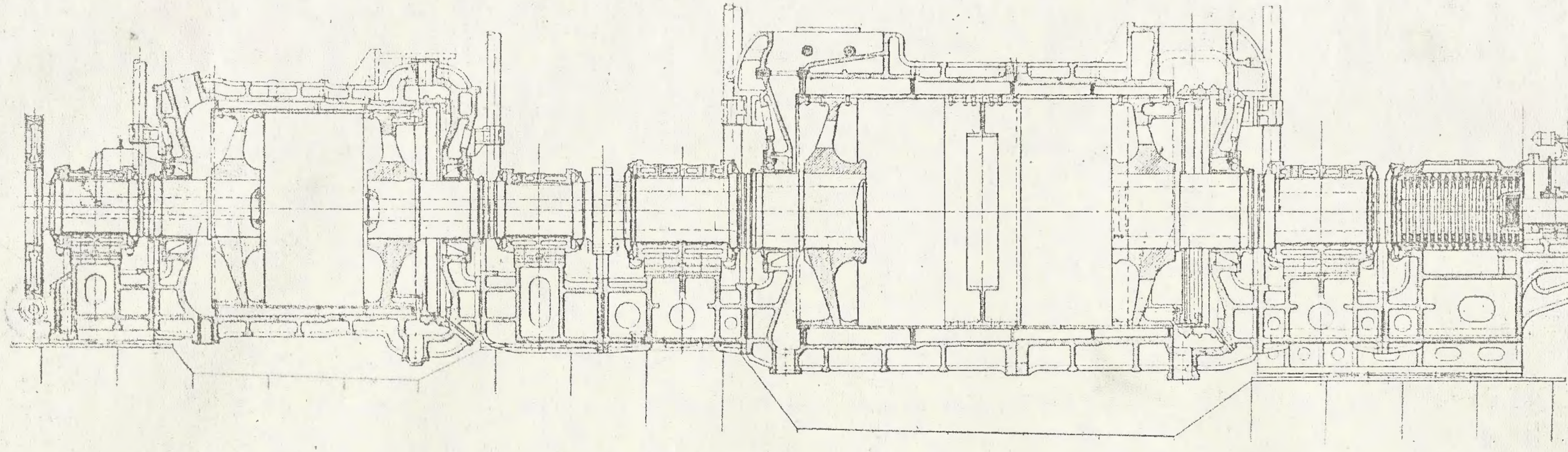


FIG. 7.—H.P. AHEAD AND H.P. ASTERN TURBINES FOR S.S. "AQUITANIA"

0 5 10 15 20 ft.

Scale for both diagrams

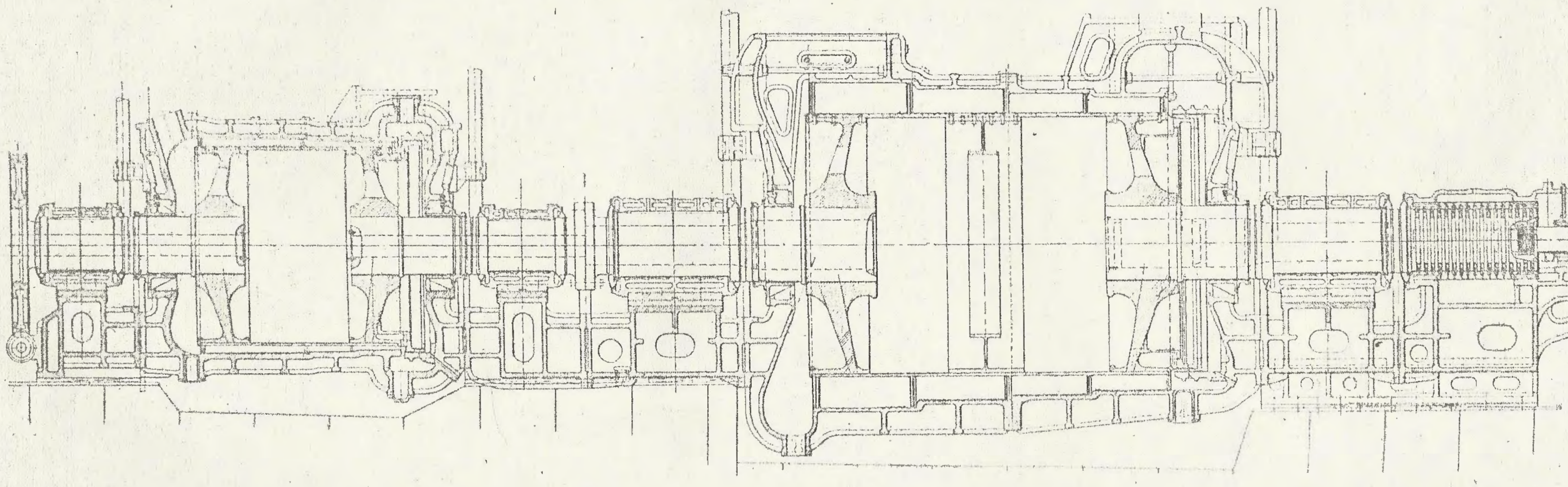


FIG. 8.—I.P. AHEAD AND H.P. ASTERN TURBINES FOR S.S. "AQUITANIA"

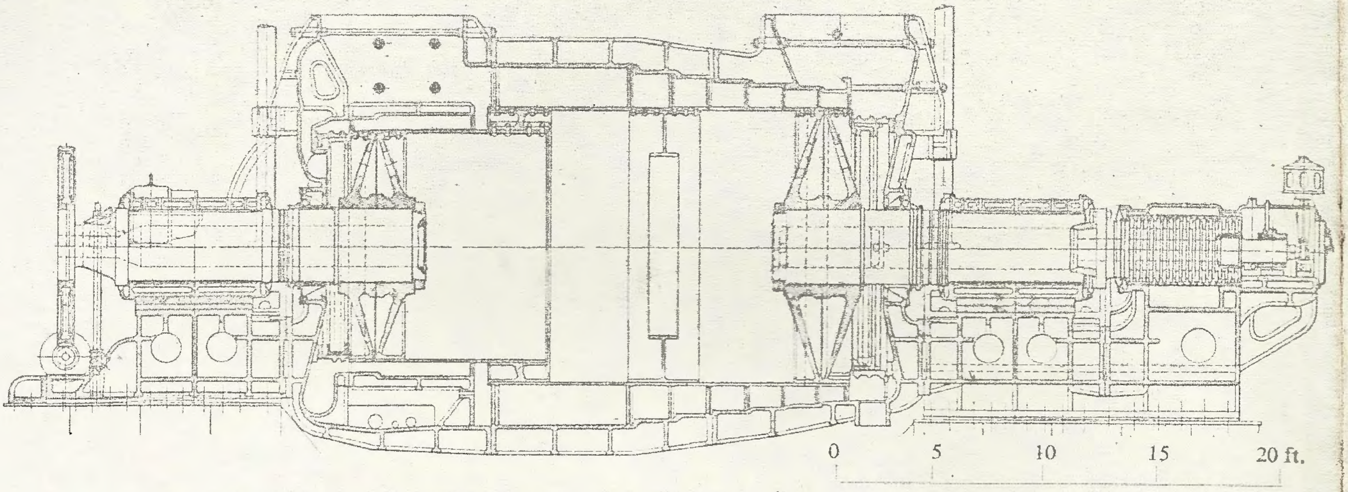


FIG. 9.—L.P. AHEAD AND L.P. ASTERN TURBINES FOR S.S. "AQUITANIA"

double helical gearing to reduce the revolutions per minute of his single-stage turbine wheels driving separators. He built a reversing turbine in the years 1892–3 to drive a launch in Lake Mälaren. The turbine developed 15 hp at 16,000 *rpm* and this speed was reduced by double helical-double reduction gearing to 330 *rpm* at the propeller shaft. Sir Charles Parsons constructed a similar turbine with reduction gearing in 1897 to drive a launch. The power developed was 10 at 19,600 *rpm*, the propeller shaft speed being 1,400 *rpm*. The complete turbine driving a pinion meshing with 2 wheels, each driving a separate propeller shaft, is still at the Science Museum, South Kensington. The idea of gearing in association with steam turbines lay dormant until the problem of applying turbine machinery to low-speed merchant ships had become important. In 1909 the Parsons Marine Steam Turbine Company bought the *Vespasian*, a 275 ft. long merchant ship with reciprocating machinery. The machinery was put into good working order and measurements of consumption made. Single reduction geared turbines were then (1910) fitted in place of the reciprocating engines, the boilers, propeller shafting, and thrust block remaining as before. Fig. 10 compares the *Vespasian* turbines of 1,100 shp (1,675 *rpm*) delivering the power at 74 main shaft *rpm* with a modern double reduction set of turbine machinery developing 5,500 shp per shaft at 120 *rpm*. In the case of the turbine machinery fitted in *Vespasian*, the consumption was 13·5 per cent less than that of the piston engines.

Only 10 years later, about 18,000,000 shp was transmitted through gearing in warships and merchant ships. Luckily the Michell thrust block was available as no longer could the steam and propeller thrusts be nearly balanced with an adjusting block to take the difference. Gearing required that the propeller thrust be taken by an adequate block able to support the whole of the propeller thrust.

The first geared turbines in the British Navy were fitted in the twin screw destroyer *Badger*, built in 1911. The high pressure and cruising turbines were both geared to the L.P. turbines which were direct-coupled to the propellers. The reduction ratios were 5·05 and 3·36 to 1 between the cruising turbine and the H.P. turbine respectively and the L.P. direct-coupled turbines.

Gearing allowed the turbine revolutions to rise to those required by efficiency considerations and the turbine could be divided into stages of expansion according to the heat drop to be handled determined by conditions at inlet and exhaust. The way was therefore clear for increases in pressure and temperature at turbine inlet.

In 1926 the Parsons Marine Steam Turbine Company built the first high-pressure high-temperature machinery installed in the Clyde passenger steamer *King George V*. The turbines were very successful, but her running was marred by burst tubes in her boilers (two types were fitted). The boiler pressure was 550 lb./in.^{2} at 750° F. total temperature.

Many installations followed, among them being the machinery for the *Empress of Britain* (1931), built by Messrs. John Brown Company. The total power was 60,000 shp with a steam pressure of 425 lb./in.^{2}, at an initial steam temperature of 725° F. This gave her a fuel consumption as low as 0·57 lb. of oil/shp hour for all purposes.

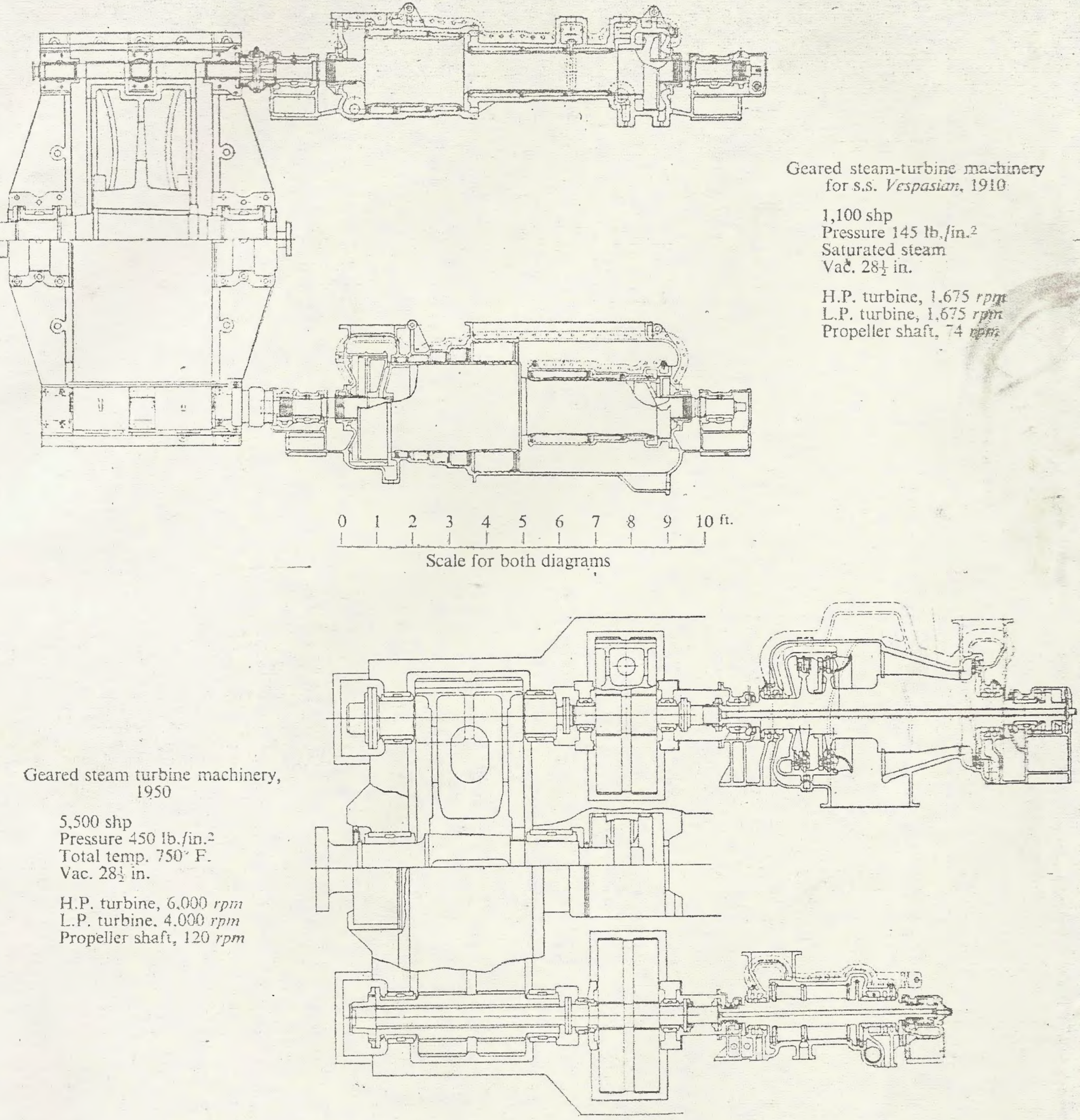
Important ships like the *Queen Mary* (1936) and *Queen Elizabeth* (1939) of 158,000 shp followed, and the story is brought up to date in Table VI which shows some figures from the new reheat liner *Empress of Britain*, built by The Fairfield Shipbuilding & Engineering Company, and the *Pendennis Castle* built by Messrs. Harland & Wolff. It will be seen that fuel consumptions can now be below 0·500 lb. of oil per shp hour all purposes, in tankers and similar ships providing the "hotel" load or load additional to all propulsion purposes, including lighting and steering, is kept to a reasonable figure.

Fig. 11 shows an H.P. ahead and astern turbine designed to operate at 850 psig and 1,050° F. at superheater outlet. The ahead turbine has nozzle control at 75 per cent power, 87·5 per cent, and 100 per cent power. An overhung H.P. astern wheel is fitted. The full power *rpm* are 5,557.

Fig. 12 shows the corresponding L.P. ahead and astern turbines in a common casing. The turbines are of impulse construction, but naturally have considerable reaction towards the L.P. end of the ahead turbine. Bleeding for feed heating takes place between stages 2 and 3 and stages 4 and 5. Two water shedding stages are fitted before the exhaust row. Full power *rpm* are 3,658. The turbine is designed for a vacuum of $28\frac{1}{2}$ in. The main shaft *rpm* are 108 and full power is developed at 22,000 shp.

Fig. 13 which shows an $H - \phi$ diagram for turbines summarizes turbine heat drops and efficiencies from the *Turbinia* through the *Lusitania* to the *Empress of Britain* 1957 with reheat, and the prototype turbine with simple expansion from a datum of 815 psig and 1,035° F. at turbine inlet. The heat drops vary from 183 B.Th.U./lb. of steam flow in *Turbinia* to 515, and efficiencies from 56 to 85 per cent.

Table VI thus summarizes the main particulars for outstanding steam-turbine installations used in merchant ships. Some installations of smaller power are included as they utilize high inlet conditions in relation to the date of construction. The difficulty has not been what to include, but the necessity to be strongly selective or the table would have been of inordinate length.



Geared steam-turbine machinery for s.s. *Vespasian*, 1910

1,100 shp
Pressure 145 lb./in.^{2}
Saturated steam
Vac. 28½ in.

H.P. turbine, 1,675 *rpm*
L.P. turbine, 1,675 *rpm*
Propeller shaft, 74 *rpm*

Geared steam turbine machinery, 1950

5,500 shp
Pressure 450 lb./in.^{2}
Total temp. 750° F.
Vac. 28½ in.

H.P. turbine, 6,000 *rpm*
L.P. turbine, 4,000 *rpm*
Propeller shaft, 120 *rpm*

FIG. 10.—GEARED STEAM TURBINE MACHINERY FOR S.S. "VESPASIAN" COMPARED WITH TYPICAL GEARED STEAM-TURBINE MACHINERY OF 1950

Naval Turbine Machinery

The next Table, No. VII, showing naval machinery is shorter because so much less is published about it. The consumption is seen to improve from 1·74 lb. of coal/shp per hour in 1904 to 0·608 lb. of oil/shp hour in 1931. Taking the specific consumption as proportional to water rate the YEAD. 1 machinery would have a corresponding fuel rate of 0·59. This machinery has also a particularly good fuel rate at cruising power which did not apply in pre-war steam-turbine machinery for naval use. Drawings would be required to show the simultaneous improvements in weight and space.

Just as naval reciprocating engines developed into high speed, short stroke, highly-rated engines, so too turbine machinery, particularly when geared, became highly specialized to meet naval requirements. Such requirements include economy at low powers, coupled with manœuvrability to high powers and back again with great frequency when required. Light weight consistent with very high battle reliability—this involves the need to have inbuilt ability to sustain shock loads and battle damage. Accessibility on occasion has to give way to the overriding requirement of minimum space.

The later Table, No. VIII, shows some high-powered geared turbine machinery in British warships with two examples from the present-day building of the United States Navy, as they represent the highest powered propulsion machinery per shaft ever fitted at sea.

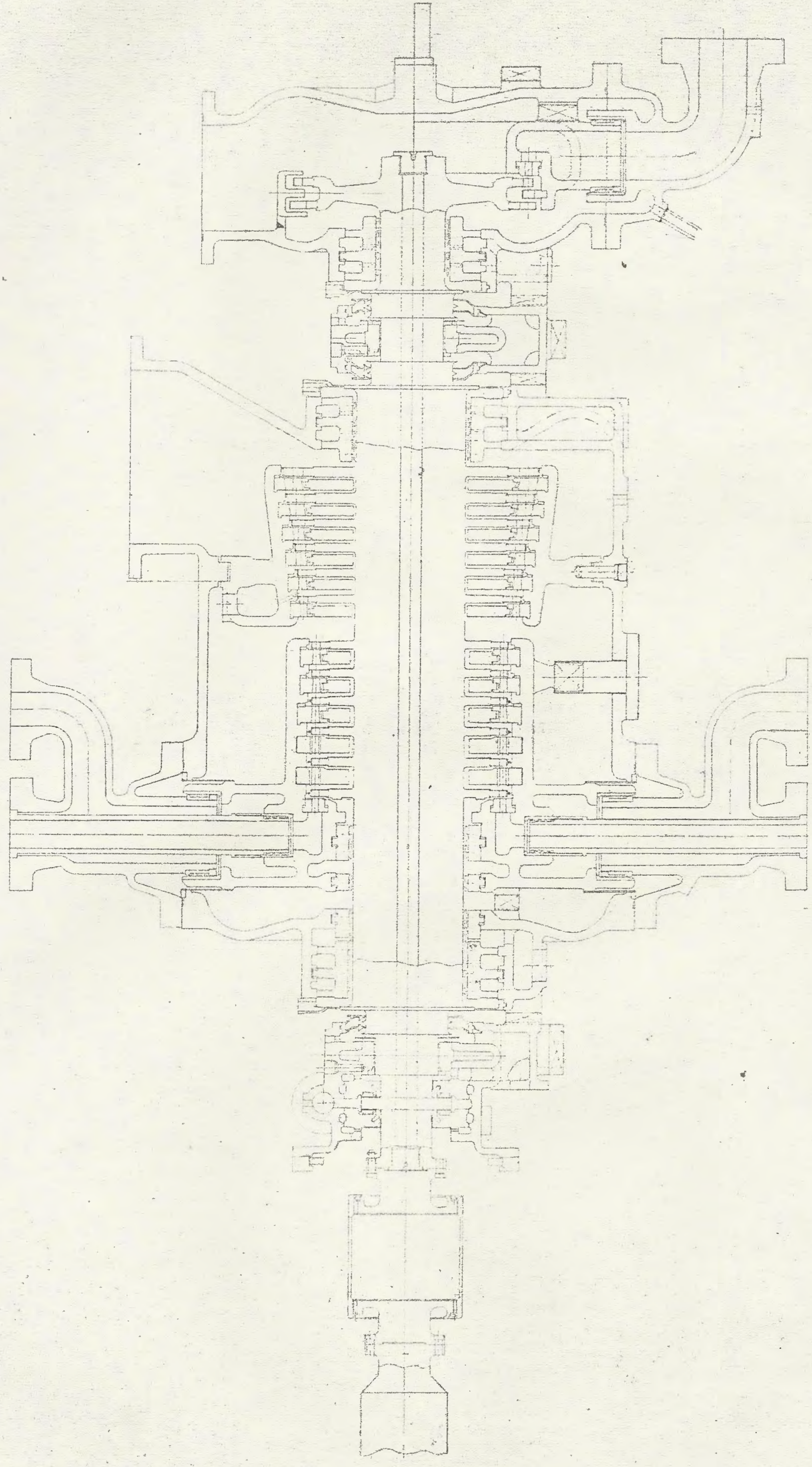


FIG. 11.—H.P. TURBINE FOR PAMETRADA PROTOTYPE MACHINERY FOR ADVANCED STEAM CONDITIONS

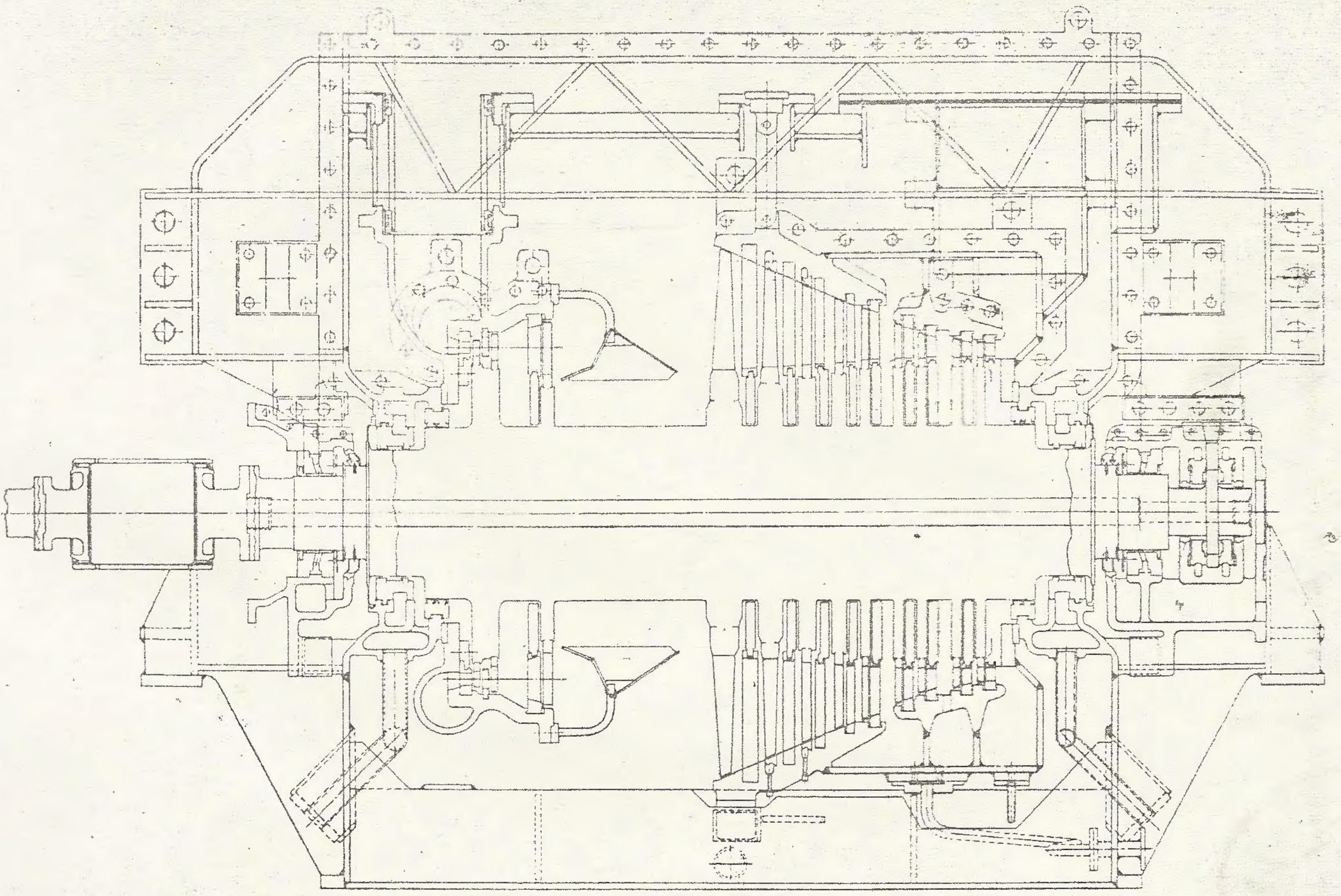


FIG. 12.—L.P. TURBINE FOR PAMETRADA PROTOTYPE MACHINERY FOR ADVANCED STEAM CONDITIONS

Steam Turbo-Electric Machinery

Another well-tried method of speed reduction between steam turbines and screw shafts is by means of electric propulsion. For high-powered sets a.c. current, 3-phase, alone is used. If, for example, the turbine drives a two-pole alternator and supplies current to a 36-pole motor, when running at synchronous speed, this is equivalent to an 18 to 1 reduction gear. It utilizes the excitation current like a clutch working across the air gap between rotor and stator in the propelling motors. This reduction ratio should be compared with figures of nearly 60 to 1, used in high-speed geared turbine machinery in association with low main shaft *rpm* at the present time. In reversing, two phases are changed over and the motors driven as squirrel cage motors until synchronous conditions are established. The efficiency of the alternator is about the same as double reduction gearing (97·5 per cent), but the overall conversion, including the provision of cooling fans for alternator and motor and excitation for the motor, produces a figure of about 92–94 per cent.

Table IX gives some notable installations. The aircraft carriers *Lexington* and *Saratoga* built in 1927 represent the last high powered installations built for the United States Navy.

When employed at Messrs. Alexander Stephen & Son, the author was employed on the construction of the machinery of the *Chitral* (1925), one of a class of ships ordered by the Peninsular and Oriental Line, consisting of twin-screw quadruple expansion engines and cylindrical boilers. In 1927 this progressive shipping company changed over from such orthodox and even a little old-fashioned machinery built to a very high specification by ordering turbo-electric machinery in association with water-tube boilers of working pressure 375 lb./in.^{2} and 700° F. total temperature for their new ship *Viceroy of India*. This ship was very successful and was the forerunner of her "white sisters," the *Strathnaver* and *Strathaird*.

The largest merchant ship building at the present time is the *Canberra* for the same owners, which is being fitted with turbo-electric propelling machinery developing 85,000 shp.

Turbine Reduction Gearing

Gearing which is in general use in turbine propelled ships has the same look from that used in Dr. De Laval's launch engine through the *Vespasian*, the first full-scale propulsion set to be used at sea, to the latest form of double-reduction gearing produced today. The accuracy and consequent noise reduction has, however, increased greatly over the years. Noise figures taken 12 in. from the gear-boxes can be as low as 90 db. Lloyd's "K" value in Table X gives a measure of the surface loading which, coupled with the tooth deflection due to the combined effects of torque, bending, and shear, gives the stress picture after the tooth form and pitch has been chosen. The "K" value has also risen with the passage of the years.

Fig. 14 represents machinery for H.M.S. *Grafton*, built by the Parsons Marine Turbine Company to the order of Messrs. J. S. White & Co. The design of the turbine was made by the English Electric Company and the gear-cases were designed by David Brown & Company. This example is chosen as it illustrates a locked-train double-reduction gear driven by the main turbine with a third reduction gear from the cruising turbine coupled to the main gearing by way of a clutch. The main thrust block integral with the gear-case can be seen aft of the main wheel after bearing.

The majority of gears are of the parallel shaft type with a few epicyclic gear-boxes transmitting up to 8,000 shp per box.

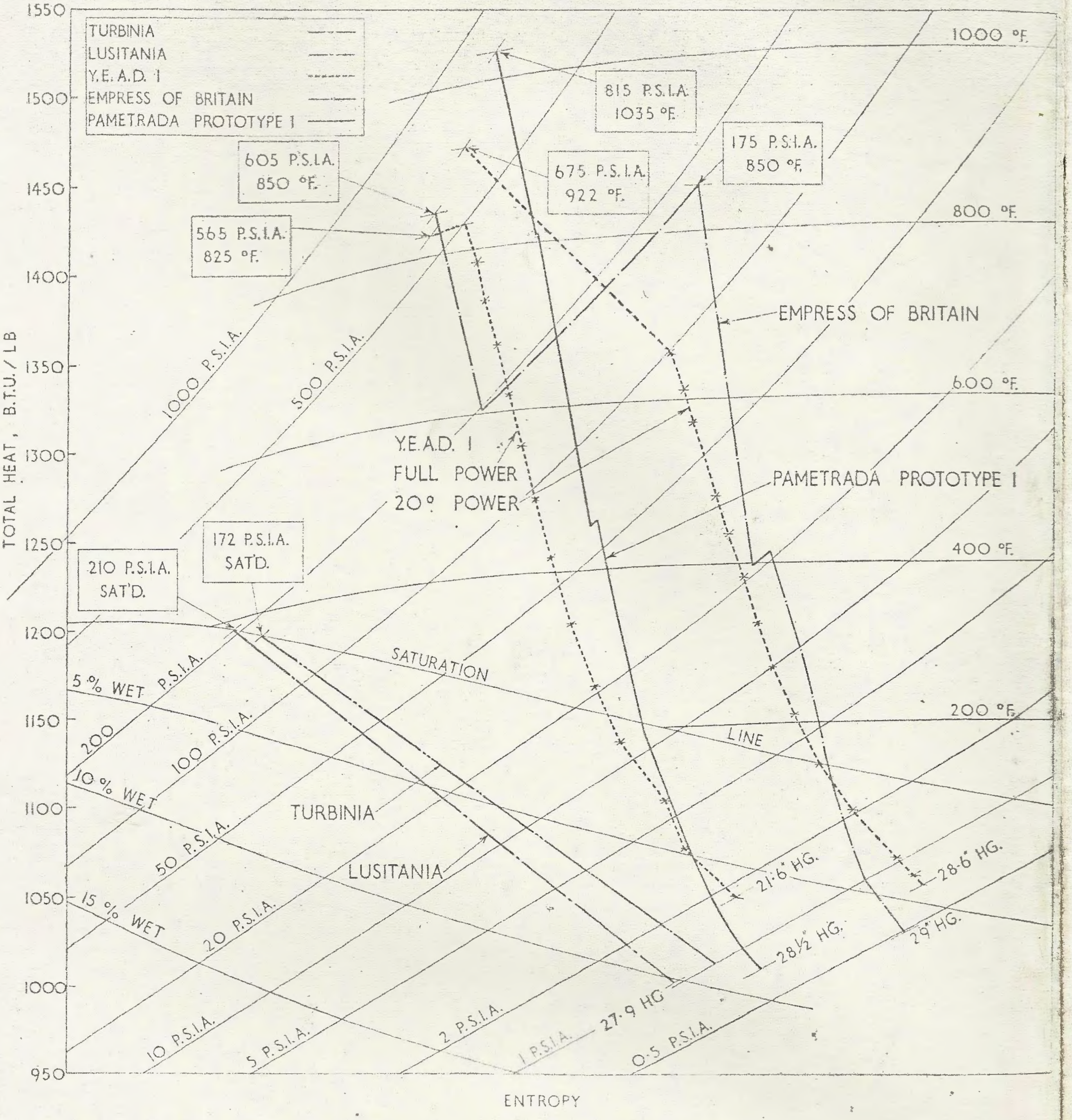


FIG. 13.—HEAT-ENTROPY DIAGRAMS FOR TURBINE MACHINERY

Reversing and Manœuvring Mechanisms

Epicyclic gears can be designed with suitable brakes fitted to allow the propeller to be driven ahead and astern with unidirectional power input. The principal problem is the design of a braking system to absorb the kinetic energy of shafting and propeller and, above all, that of the ship.

Epicyclic gearing is light and involves relatively cheap production equipment, even taking into account the use of hardened and ground gears, as the diameters are small. It is, however, difficult to dismantle, and many parts cannot be monitored in operation.

Variable pitch propellers have been fitted to a few ships and results are favourable. Cost is, however, very high, and it would appear preferable to have controlling gear accessibly mounted within the hull.

One form of hydraulic transmission which has been in service at sea in the *Auris* is shown in Fig. 15. The unidirectional turbine drive is coupled to the driving member of the ahead hydraulic coupling by a quill shaft and flexible couplings. The efficiency of the hydraulic coupling is about 96–98 per cent.

A torque reversing coupling is fitted at the end nearest to the turbine so that the quill shaft is available for all the ahead running of the transmission system.

The efficiency of the torque convertor is now about 68 per cent.

FIG. 14.—MODERN NAVAL STEAM-TURBINE MACHINERY SHOWING LOCKED-TRAIN DOUBLE-REDUCTION GEARING WITH TRIPLE-REDUCTION CRUISING TURBINE TRAIN

Manœuvring is carried out by filling either coupling as required with interlocks to prevent overspeeding. Heat generated in the couplings is dissipated in the oil cooler.

When "Full Away" is given, oil at a pressure of about 10 lb./in.² is admitted to the centre of the shaft and enters the small volume behind a clutch plate where the spinning of the clutch by the fluid flywheel generates very high pressures (about 4 to 5,000 lb./in.²). The clutch moves along splines without torque until it engages with the clutch plate. The slip to be taken up on engagement is only that due to the fluid flywheel, say 4 per cent, and when driving the only loss in the transmission is the normal one due to the gearing and shafting (2–2½ per cent of the power transmitted).

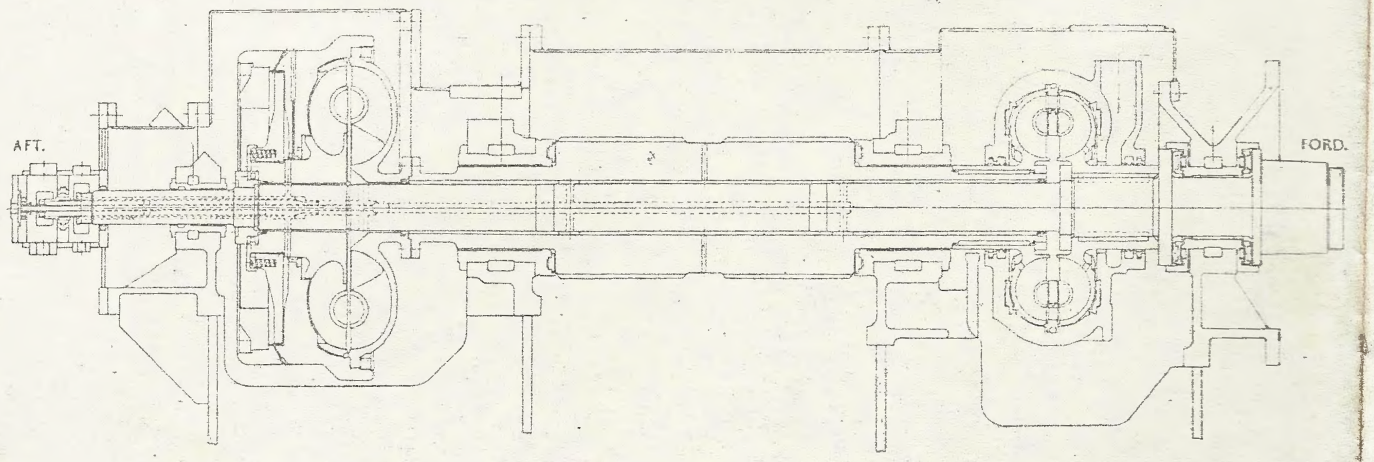


FIG. 15.—PAMETRADA HYDRAULIC REVERSING TRANSMISSION

Future Developments of Steam-Turbine Machinery

The question of manœuvring without the use of astern turbines will lead to great improvements in the steam turbine itself, as no cool astern turbines will be required to take the thermal shock of hot steam. At 950° F. considerable experience at sea has shown that no attemperation is required, but in the search for further economies, temperatures will go still farther. The Pametrada prototype set now building takes steam at 1,050° F. from the superheater outlet, and it would appear that metals will be available in the near future to allow steam of about 1,500° F. to be used. Such temperatures would be used in a high-speed top turbine which would handle the hot steam in a turbine of small physical dimensions which would then exhaust at present-day inlet temperatures to a single casing turbine without astern elements. The turbines would be unidirectional and manœuvring would be carried out by one of the means already mentioned, the hydraulic system being preferred.

That 1,500° F. is not an entirely fanciful temperature to aim for is proved by papers such as that by M. A. Levinstein, American Society of Metals, 1959, on the development of oxidation resistant coatings for molybdenum turbine blades. One such blade with four coatings:

(*a*) a chromium electro deposit to provide a diffusion type bond,

(*b*) a nickel electro deposit to improve ductility,

(*c*) a fused nickel-silicon-boron coating as a bonding layer for the subsequent nichrome cladding, and

(*d*) a 0·006 in. brazed layer of "nichrome" hard surfaced with a chromium tungsten cobalt boron alloy,

has already been produced.

Blades were tested in a turbine at 1,800° F. steady state and under cyclic conditions at 1,900° F. and 2,000° F. Of 96 blades tested for fifty hours under steady-state conditions, two failed, due to damage to the coating. In the cyclic tests the cause of failure was thermal fatigue.

In considering machinery for ships in relation to conditions used in power stations on land it must be remembered that the size is very different. The largest power per shaft in any ship in the world, including the latest nuclear aircraft carrier, *Enterprise*, is 75,000 shp. Land sets are now up to 550 MW in size in dual tandem sets, roughly equivalent to 700,000 shp. Even small land turbine-driven sets are much larger than the average used in marine propulsion units (about 10,000 shp). Supercritical pressures cannot therefore be employed for marine boilers and turbines, as the parasitic losses in the small volume flows due to high pressure and small output would swamp any possible gain from the extremely high pressure.

With such considerations in mind, Table XIV, originally produced in 1953 (26th Thomas Lowe Gray Lecture before the Institution of Mechanical Engineers), still applies, and several fuel points in the table have since been checked in service. Compare, for example, the specific consumption figure for the *Empress of Britain* (1956) in Table VI with that in the second line of the table.

The steam turbine has therefore still a considerable development to take place in the future as a marine propulsion unit, and it will remain the main propulsion unit for any machinery to develop high powers above, say, 30,000 shp per shaft. Reheating of the steam will be seen to be an important factor in improvement in the future. If two stage reheating should be arranged for, this could be carried out in heat exchangers close to the turbine using a special closed circuit of fluid which is heated in a section of the boiler, but is isolated from the steam circuit so that pressures and temperatures could be determined only from reheating considerations without complicating the main steam cycle conditions.

Development of the Fire-Tube Boiler or Cylindrical Boiler

The earliest marine boilers were large box-like structures with internal rectangular furnaces and long, narrow rectangular-section flues. The material was either copper or wrought iron, copper being more expensive, but having a longer life.

Changes from this design were necessitated by the steady increase in steam pressures, typical figures being:

	lb./in.²
1830	5
1840	10
1850	20
1860	25

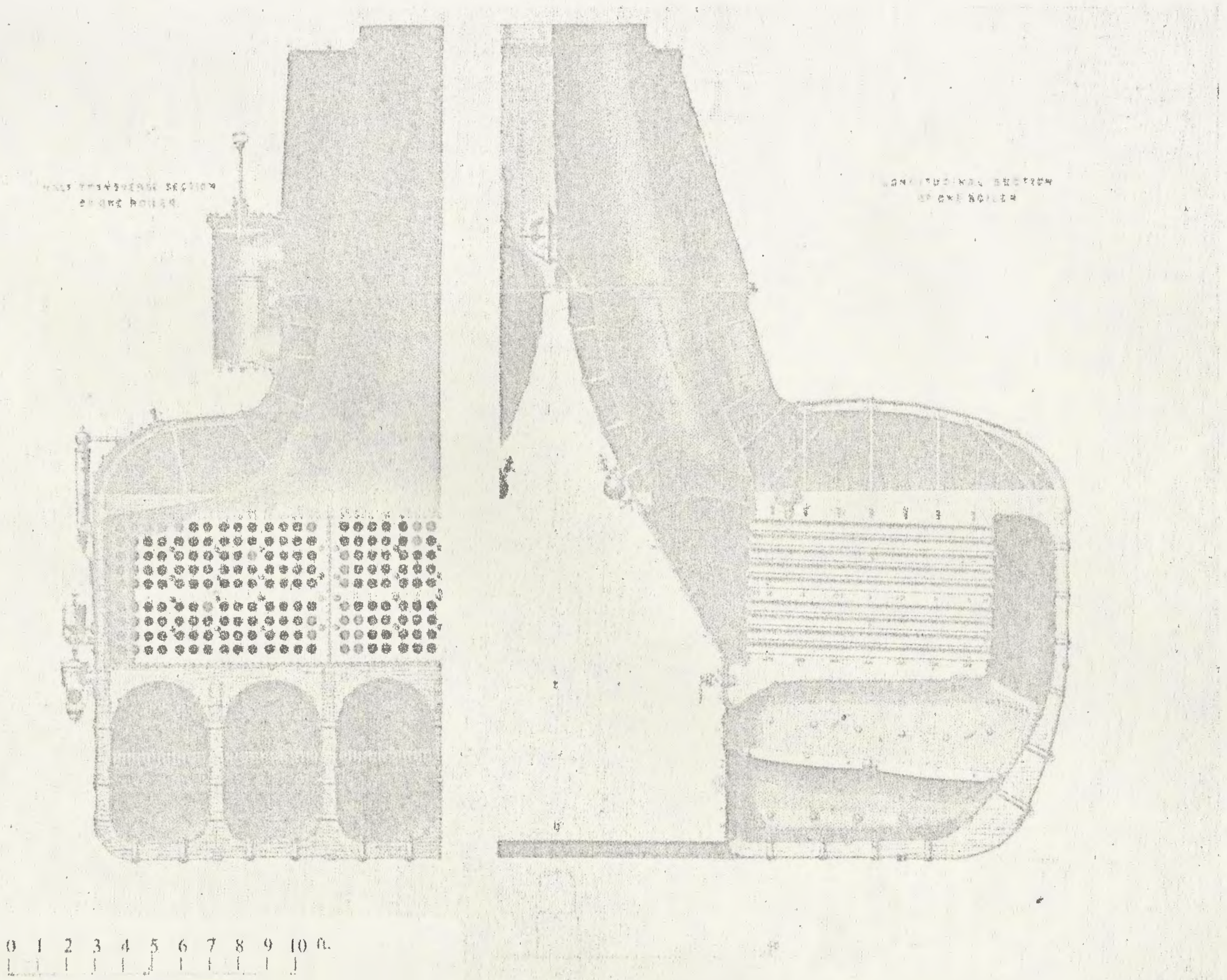


Fig. 16.—Early marine return-tube flue type boiler (s.s. "Arabia")

To cope with the higher pressure, the rectangular flues were first replaced by tubes, giving the multitubular box boiler. The tubes were either on the same level as the furnaces ("through-tube boiler") or above them ("return-tube boiler"). By 1860 this change was already complete in naval vessels and was well under way in merchant ships. Fig. 16 shows the return-tube box boiler of 1862, working pressure 25 lb./in.^{2}, of the type fitted to *Scotia*. The inclusion of part of the flue as heating surface inside the boiler to dry the steam produced is a noteworthy feature.

The box boiler was not suitable for pressures higher than 30–35 lb./in.^{2}, and the adoption of higher pressures forced the change to a cylindrical shell (although some boilers were for a time built with oval shells). The return-tube boiler with a cylindrical shell built in 1862 constituted the first Scotch boiler. The first boilers of this type 10 ft. diameter × 14 ft. 6 in. long, working pressure 50 lb./in.^{2}, were fitted by Randolph Elder & Co. (today The Fairfield Shipbuilding & Engineering Co. Ltd.) in the *McGregor Laird*.

Scotch boilers became widely used in the 1870's with the general adoption of higher steam pressures in association with compounding. The change from iron to steel took place in the late 1870's, and the corrugated furnace was patented by Samson Fox in 1877.

Forced draught was first used by Hawthorn for large ships in 1881, and was introduced for large naval ships in 1882. The Howden system was evolved about 1880 and applied in the *New York City*, 1884.

Later changes involved size and details of construction to allow sizes such as the following double-ended Scotch boilers to be built:

	Diameter		Length		Pressure	Date
	ft.	in.	ft.	in.	lb./in.^{2}	
Deutschland ..	16	7	20	4	213	1900
Kaiser Wilhelm II	19	5	20	10	225	1903
Aquitania ..	17	8	22		195	1912

For naval vessels the Scotch boiler was superseded by the water-tube boiler between 1890 and early 1900, primarily owing to the poor flexibility of the Scotch boiler, but also to reduce weight and space.

For large merchant ships the change-over began in the 1920's (e.g. the *Franconia*, 20,000 tons, completed in 1923, had Scotch boilers, while the *Duchess of Bedford*, completed 1928, had water-tube boilers).

For smaller merchant ships the Scotch boiler continued to be used in association with steam reciprocating machinery up to the present time, although the number of such installations has decreased steadily and is now very small.

Prudhon Capus and Howden Johnson Boilers tried to combine the advantages of the water-tube boiler and the cylindrical boiler, but had only a short vogue about 1925 to 1935.

It is quite an interesting fact that the large Scotch boilers built for pressures as high as 225 lb./in.^{2} will float if openings for mountings are appropriately blanked off. The fact that these large and heavy parts can be floated is used even at the present day. For example, the heat exchangers for Bradwell, 20 ft. diameter × 100 ft. long, weighing 200 tons each, went by sea from Thornaby-on-Tees to Bradwell-by-Sea, the drums merely being put into the water and towed down.

Some Data on the Development of the Water-Tube Boiler

Early Water-Tube Boilers

Successful water-tube boilers were not developed until the end of the nineteenth century. Prior to this, water-tube boilers of various designs were fitted in ships from time to time, but all gave trouble in service and were later replaced. The usual difficulties were internal corrosion, poor circulation, and poor accessibility for cleaning and repair. Presumably the use of impure and oxygenated feed-water was one of the more important causes of these early failures.

Some early installations were:

Thetis	1857–8	Rowan boilers; working pressure 120 lb./in.^{2}.
Murillo	1860	Williamson boilers; working pressure 90 lb./in.^{2}.
La Biche	1856	Early Belleville boilers with vertical coils.
Argus	1861	
St. Barbe	1861	
H.M.S. *Chanticleer*	1865–70	Cochrane boilers.
H.M.S. *Oberon*		
H.M.S. *Audacious*		
H.M.S. *Penelope*		
Wanderer	1878–80	Perkins boilers with high working pressures (e.g. *Anthracite* 350 lb./in.^{2}).
Irishman		
Anthracite		

The later design of Belleville boiler having horizontal tubes was fitted in the *Hirondelle* in 1872, and subsequent to 1880 was adopted for ships by Messageries Maritime and later in French warships.

Adoption for Naval Vessels

The naval application began with torpedo craft (torpedo boats and the later developments, torpedo gunboats and torpedo-boat destroyers), where the need was to obtain the maximum power from compact and relatively lightweight machinery. Previously locomotive-type boilers had been used, but these had not been satisfactory.

For larger naval vessels they gave promise of improvement over cylindrical boilers, which were unsatisfactory owing to the very limited extent to which they could be forced or could accept rapid changes of power without giving trouble.

For a number of years it was the practice to fit new designs of water-tube boilers experimentally in smaller vessels before adopting them for larger ships. Some dates are:

1885 First Thornycroft boiler fitted in Torpedo Boat No. 100.
1891 *Speedy* (torpedo gunboat) fitted with Thornycroft boilers.
1892 *Sharpshooter* (torpedo gunboat) fitted with Belleville boilers (Trials 1894).
1894 *Powerful* and *Terrible* (cruisers) fitted with Belleville boilers (Trials 1896).
1896 Belleville boilers adopted for new construction, cruisers and battleships.
1896 *Sheldrake* (torpedo gunboat) fitted with Babcock & Wilcox boilers.
Seagull (torpedo gunboat) fitted with Niclausse boilers.
1902 Admiralty Boiler Committee recommended the Babcock & Wilcox, Niclausse, Dürr and Yarrow large-tube boilers as suitable for use in large naval vessels. After further service experience the first and last of these were found to be the most suitable.
1906 Following similar service trials the three-drum small-tube Yarrow and White-Forster boilers were found to be most suitable for light cruisers and destroyers and were adopted for new construction.

Boiler pressures became stable at about 250 lb./in.^{2} towards the end of this period and did not alter appreciably for the next 25 years.

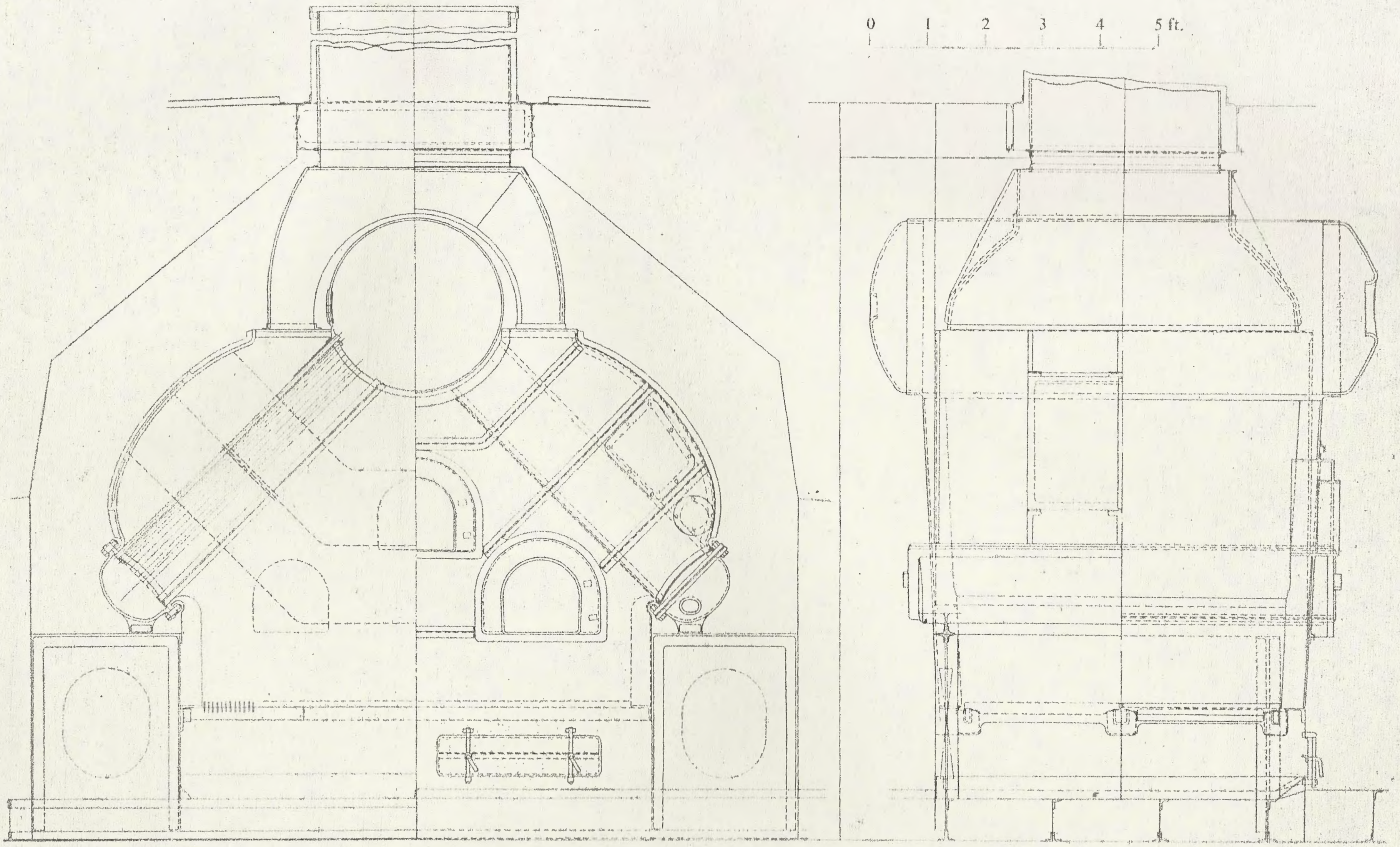


FIG. 17.—EARLY YARROW WATER-TUBE BOILER

From about 1918 the three-drum small-tube boiler became standard for all naval vessels, and became known as the *Admiralty Boiler.* Fig. 17 shows an early Yarrow boiler with D-shaped water pockets and cylindrical steam drums, working pressure 180 lb./in.^{2}. For nearly 40 years this formed the prototype for all naval boilers in important ships.

Adoption for Merchant Vessels

The change from cylindrical boilers to water-tube boilers in merchant ships took place first in passenger vessels during the mid-1920's, as is illustrated by the following figures for passenger liners given by Denholm Christie (*N.E.C. Inst.* of *E.* & *S.*, 1934–5):

Year of completion	Number of liners listed with steam machinery	Number with cylindrical boilers	per cent
1885–1920 . .	32	30	94
1921–5 . .	15	14	91
1926–30 . .	10	1	10
1931–5 . .	15	1	7

The change to water-tube boilers was usually accompanied by an increase in boiler pressure from around 200 lb./in.^{2} to around 400 lb./in.^{2}.

The types of water-tube boiler used in this country were mainly the Babcock & Wilcox, Yarrow, and Johnson boilers. The Wagner boiler was widely used in German vessels.

New Types of Water-Tube Boiler Developed Between the Wars

A better understanding of the principles of heat transmission in boilers, coupled with the desire to use higher steam pressures for reasons of economy, led to the invention of a number of new types of water-tube boiler during this period. They were (and are) used to some extent in land power stations, and a few were installed in ships, but none have been at all widely used at sea. The chief types are as follows:—

(*a*) *Benson Boiler.*—This was the first "once through" boiler. There is no steam drum. It was at first thought that this system could only be used if the pressure exceeded the critical pressure, but this was later realized to be unnecessary, and later boilers operated at lower pressures. Marine installations were:

Uckermark	1931	250 atmos./460° C. (one boiler only fitted in place of a cylindrical boiler).
Potsdam	1934	90 atmos./470° C.

(*b*) *Loeffler Boiler.*—This boiler is essentially a superheater, comprising radiant and convective sections. Part of the superheated steam passes to the engine, and the remainder to a drum where it evaporates the pre-heated feed-water. A steam-circulating pump is fitted. Marine installation:

Conte Rosso	1935	130 atmos./475° C. (one boiler only fitted in place of a cylindrical boiler).

(*c*) *Sulzer Monotube Boiler.*—The second "once-through" boiler, without steam drums. Marine installation:

Kertosomo	1935	60 atmos./375° C.

(*d*) *La Mont Boiler.*—A forced-circulation boiler having a more or less conventional steam drum.

The first marine installation was dated 1933. In this country the only marine installations up to 1946 were in the destroyer *Ilex* (1937) and the S.G.B.'s (1942). In 1946 it was stated that 4 main and 6 auxiliary boilers were under construction for marine use. A much larger number of marine units were built abroad.

(*e*) *Velox Boiler.*—The first of the pressure-combustion boilers. An early marine installation was:

Athos (French)	1936	50 atmos./450° C.

In France, post-war vessels such as *Ville de Tunis* and *Cambodge* have also been fitted with Velox boilers.

Post-War Developments

The post-war period saw the virtual completion of the trend to replace the cylindrical boiler by the water-tube boiler in ships of moderate as well as high power.

Steam conditions advanced from the typical 450 lb./in.^{2}/750° F. of the immediate post-war period to 650 lb./in.^{2}/850° F., with somewhat higher pressures and temperatures in a considerable number of installations.

In a small number of installations there has been provision for reheating in the boilers. (*Examiner*, 1942, *Beaver* class vessels, 1946, *Empress* vessels, 1956–7.)

With these water-tube boilers, the time which can elapse between normal water level and the dangerously low level in the drum is becoming very much less. Taking the double-ended Scotch boiler, 17 ft. 6 in. diameter by 22 ft. 6 in. long, the evaporation would be about 40,000 lb. of steam per hour. For a 9 in. change in water level which would lower the water through mid-glass to the top of the combustion chamber crowns, it would take more than 23 min. In some recent boilers of the highly rated water-tube type, the corresponding figure has come down from 30 sec. to 13. This shows that with higher pressure boilers working at greater forcing rates, more and more automatic control will be required. Automatic control has already been developed to a stage in which it can take charge of the very great fluctuations in boiler output which occur during manœuvring from ahead to astern with a number of stops in between.

Fig. 18 shows a Babcock & Wilcox 1,200° F. boiler plant which has been installed during 1959 at Pametrada Research Station. The boiler basically is a de-rated version of the naval boiler produced for YEAD.1 machinery, with greater access space for maintenance, and would be suitable for use in a merchant ship machinery installation. The capacity is 264,000 lb. of steam per hour (total) at 1,200 lb./in.^{2} and 950° F. at the main superheater outlet. A separately-fired superheater in the same boiler casing raises the steam to a total temperature of 1,200° F. at outlet from the superheater stop valve. Temperatures down to 750° F. can also be obtained. This boiler is chosen as it represents a marine type boiler with the most advanced conditions yet used in the marine field. To give an idea of scale, the main steam drum is 4·5 ft. external diameter by 21·5 ft. long and weighs 20 tons.

The tables for turbine machinery, merchant, naval, and turbo-electric give some indication of boiler types and the steam conditions employed over the whole period under review.

Change from Coal to Oil-Firing of Boilers

One of the great changes in operating boilers at sea was the change from coal to oil. The general switch took place first in the Navy in large ships about 1903 when large ships were equipped to burn some oil as well as coal. The use of oil was facilitated by research work at Woolwich Dockyard in 1867–70 on steam atomization. Mechanical atomization was developed at Haslar 1898–1902.

The first naval ships to burn oil exclusively were torpedo boats 1–32 and *Tribal* class destroyers in 1905. After 1912 all new battleships and cruisers were designed to burn oil alone. The advantages were speedier fuelling with greater radius of action, steadier steaming and greater flexibility in relation to changes in power.

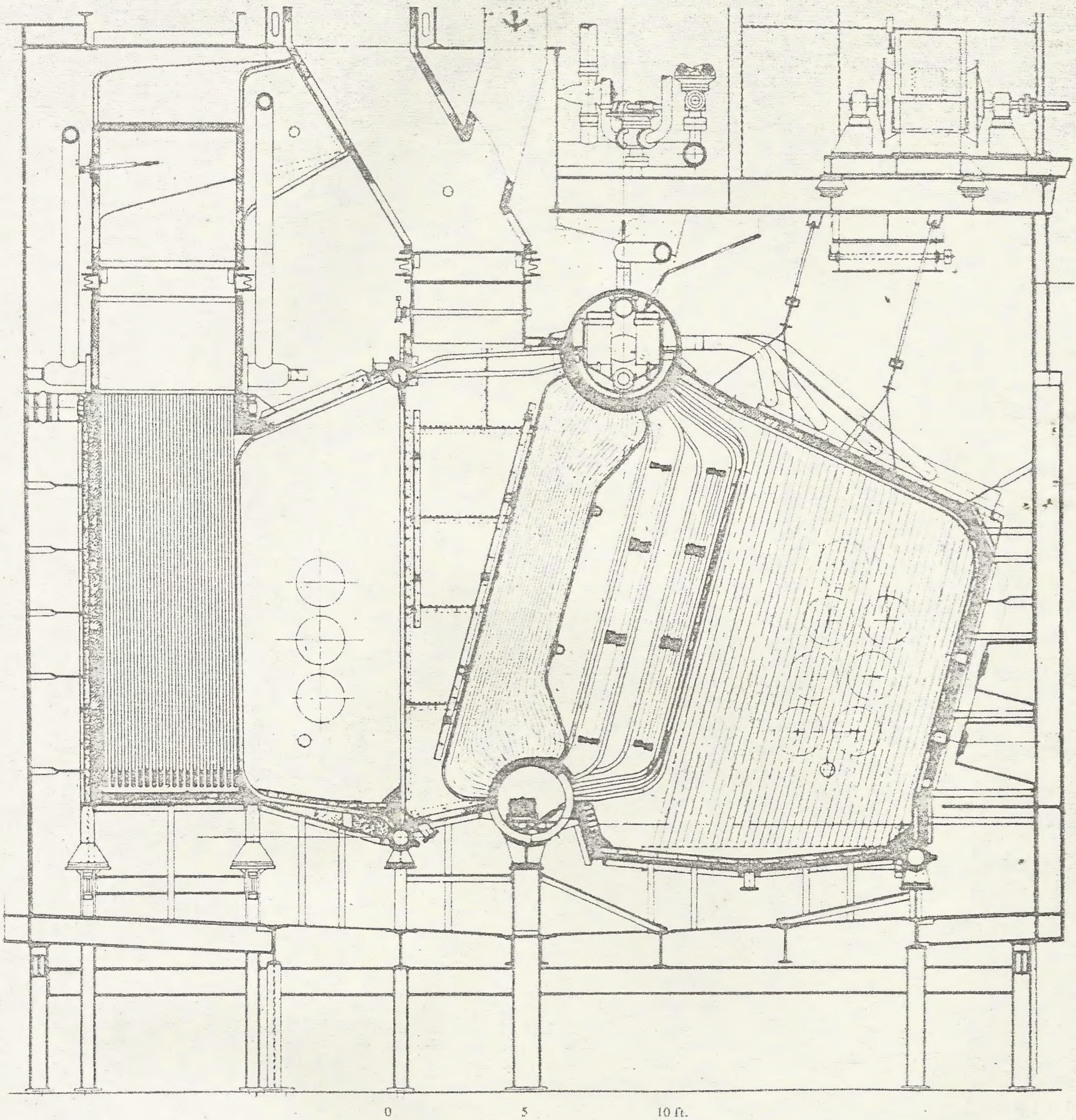


FIG. 18.—MODERN BABCOCK & WILCOX HIGH-PRESSURE, HIGH-TEMPERATURE WATER-TUBE BOILER WITH SEPARATELY-FIRED SECOND-STAGE SUPERHEATER

The change in the work in the boiler rooms and bunkers can be seen in reduced engine-room complements, for example:

Coal-burning—*Lion*	70,000 shp	E.R. complement 608
Oil-burning—*Hood*	144,000 shp	E.R. complement 306

Oil burning was introduced to merchant ships in s.s. *Oretzia*, 1881, by The Wallsend Slipway & Engineering Co. Ltd. The change was at first very slow, but greatly accelerated after World War I, until from oil-burning being used first in large passenger liners, it gradually spread to being used in the majority of machinery installations. At present 64,000,000 gross tons of shipping burn oil fuel compared with 8,000,000 gross tons burning coal (Lloyd's Register of Shipping Statistics).

The trouble in the Clyde turbine-engined ship, *King George V*, with high-temperature, high-pressure machinery, arose out of her boilers being coal fired. A burst tube, when the fire doors were open, scalded the stokers by blowing out the burning coal and steam. An oil-fired boiler under similar circumstances vents readily up the uptake and funnel and no other damage occurs providing the fuel is cut off immediately and extra feed-water supplied.

Pressurized Boilers

In the future, pressurized boilers for merchant-ship turbine installations will be considered due to the large savings in weight and space effected compared with conventional designs. In this the Navy will probably show the way as these requirements are pre-eminently necessary in warships.

The similar boiler in merchant ship use will, however, allow it to be mounted close to the steam-turbine machinery and will greatly facilitate short main steam lines, reheating, and the like at high temperatures. Pressure combustion boiler design has been facilitated by improvement in gas-turbine and axial-compressor efficiencies consequent on the development of gas turbines since the war.

Fig. 19 shows a design of pressure combustion boiler produced by Pametrada, with an optimum value of pressure ratio, while maintaining reliability, ease of maintenance, efficiency, and ability to burn residual fuel in a merchant ship application. The weight and bulk are between 16 to 33 per cent of that for typical unpressurized merchant-ship designs.

By using a 2-stage gas turbine with an economizer in between the stages, an efficiency of 90 per cent could be attained, and this represents a development for the future.

Table XI compares the pressure combustion boiler, Fig. 19, with two typical conventional boilers and shows the very large increase in heat release rate possible with the increase in density in the furnace, and the consequent large reduction in specific weight and specific volume. It should also be remarked that the efficiencies of these boilers are of the order of 87–8 per cent, based on gross calorific value of the fuel, and that if anything, the pressure combustion boiler can be more efficient by about 1 per cent with further development in the gas turbine and compressor.

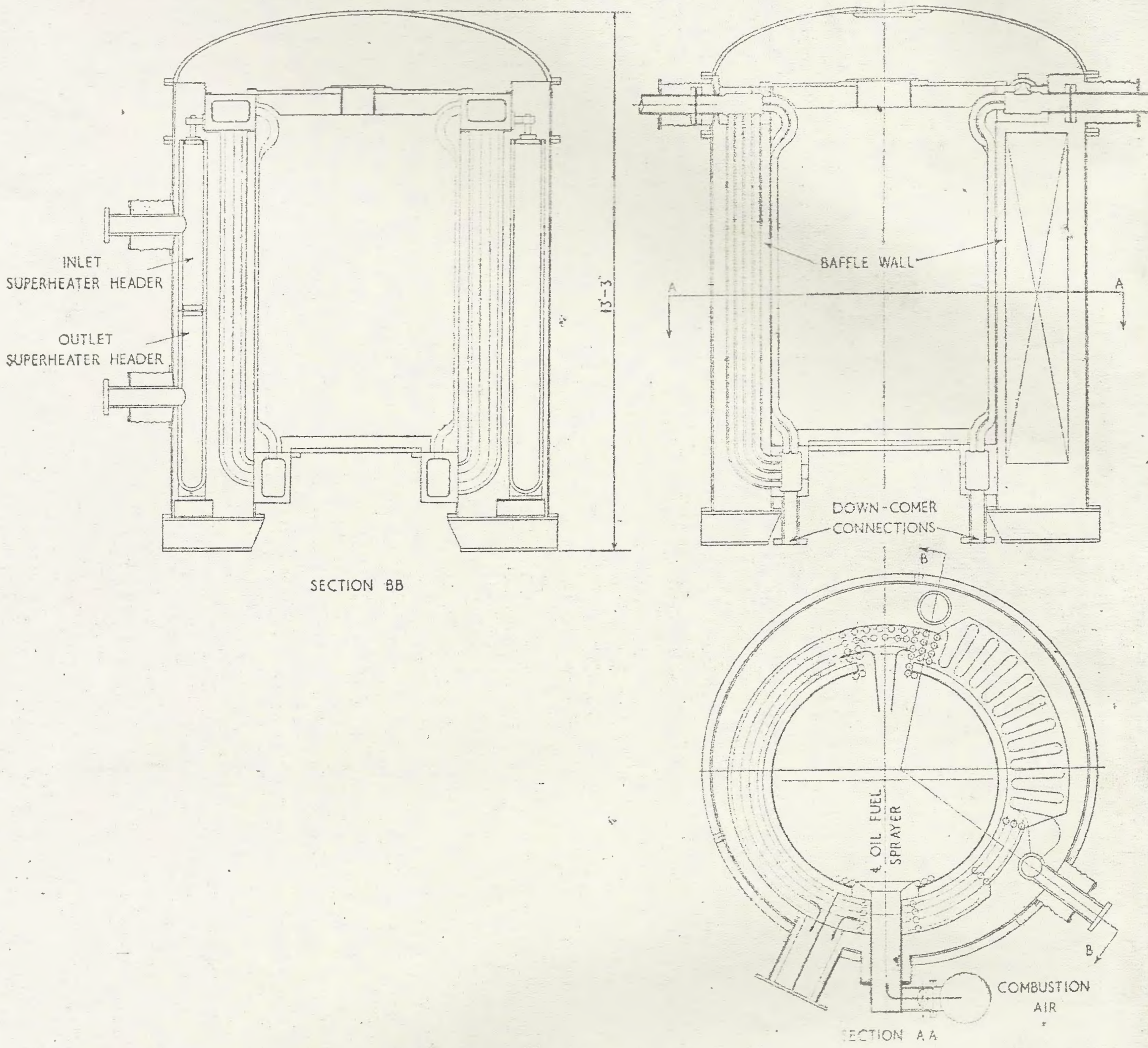


Fig. 19.—Pressure combustion boiler

The Heavy Oil Engine

Since the last war the heavy oil engine has been the major type of propulsion machinery used in merchant ships. Its history in relation to marine propulsion begins in 1910 with the first ocean-going tanker (*Vulcanus*) fitted with a Werkspoor engine.

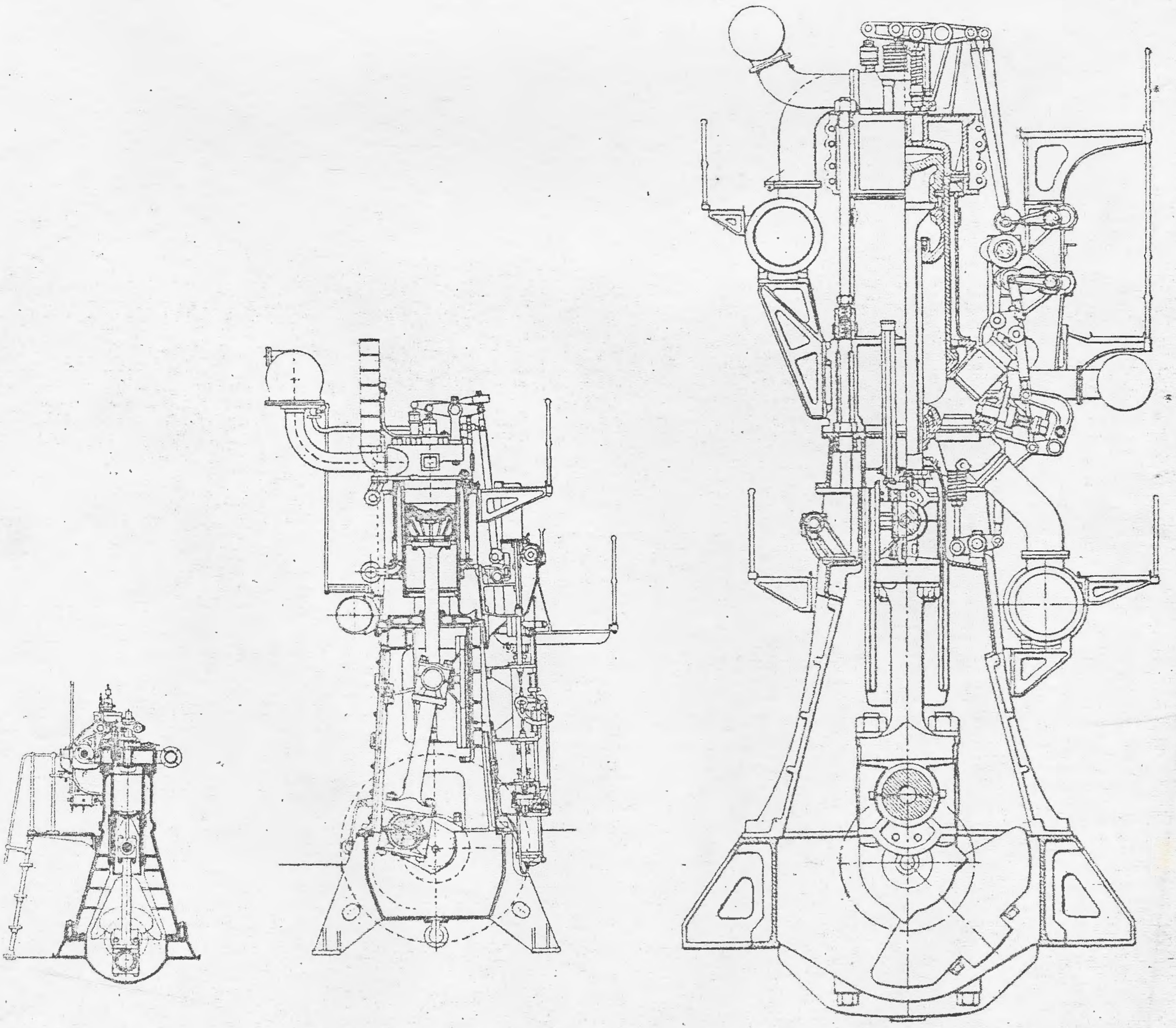
Launch engines using petrol and paraffin date from about

1888 and the start of "hot bulb" engines dates from 1890, when they were introduced by Mr. Akroyd Stuart. Dr. Diesel introduced his engine in 1893 using much higher compression than used previously, 13·5 to 1 compared with between 3 and 5 to 1. The resultant temperature produced rapid combustion of the fuel which was injected by blast air shortly before the end of the compression stroke. Some oil tankers for river use (*Wandal*, *Delo*, and *Emmanuel Nobel*) were produced by Nobel Brothers at St. Petersburg (now Leningrad) in 1904, using diesel engines of non-reversible type driving electric generators, the total power of the three engines fitted per ship being 400 hp.

Even after 1906 development was slow and the first motor cargo vessel propelled by diesel engines was the *Selandia* built in 1912, followed by the *Jutlandia*. In 1914 there were 27 classed ships with diesel machinery when Lloyd's issued their first rules for diesel engines and their auxiliaries. The Great War intervened and marine diesel development ceased until about 1920. The great controversy then was whether the future would remain with the 4-stroke or 2-stroke cycle engine.

Practically all diesel engines at this date still used air injection of fuel which involved driving an air compressor, usually of 3 stages, with intercooling to produce blast air at pressures up to 1,000 lb./in.^{2}. This was a source of considerable trouble as the compressor was designed to be greatly oversize not only to provide blast air, but also to charge up high pressure starting air bottles which were frequently fitted. When the starting air-bottles were full the compressor suction was throttled until only the injection air was pumped. Many 4-stroke engines were produced, of which the most numerous and successful were those produced by Messrs. Burmeister & Wain and Messrs. Harland & Wolff.

Messrs. Vickers-Armstrongs of Barrow introduced the airless injection of fuel in their submarine engines during the 1914–18 War, and adopted it in the post-war engines for ship propulsion.

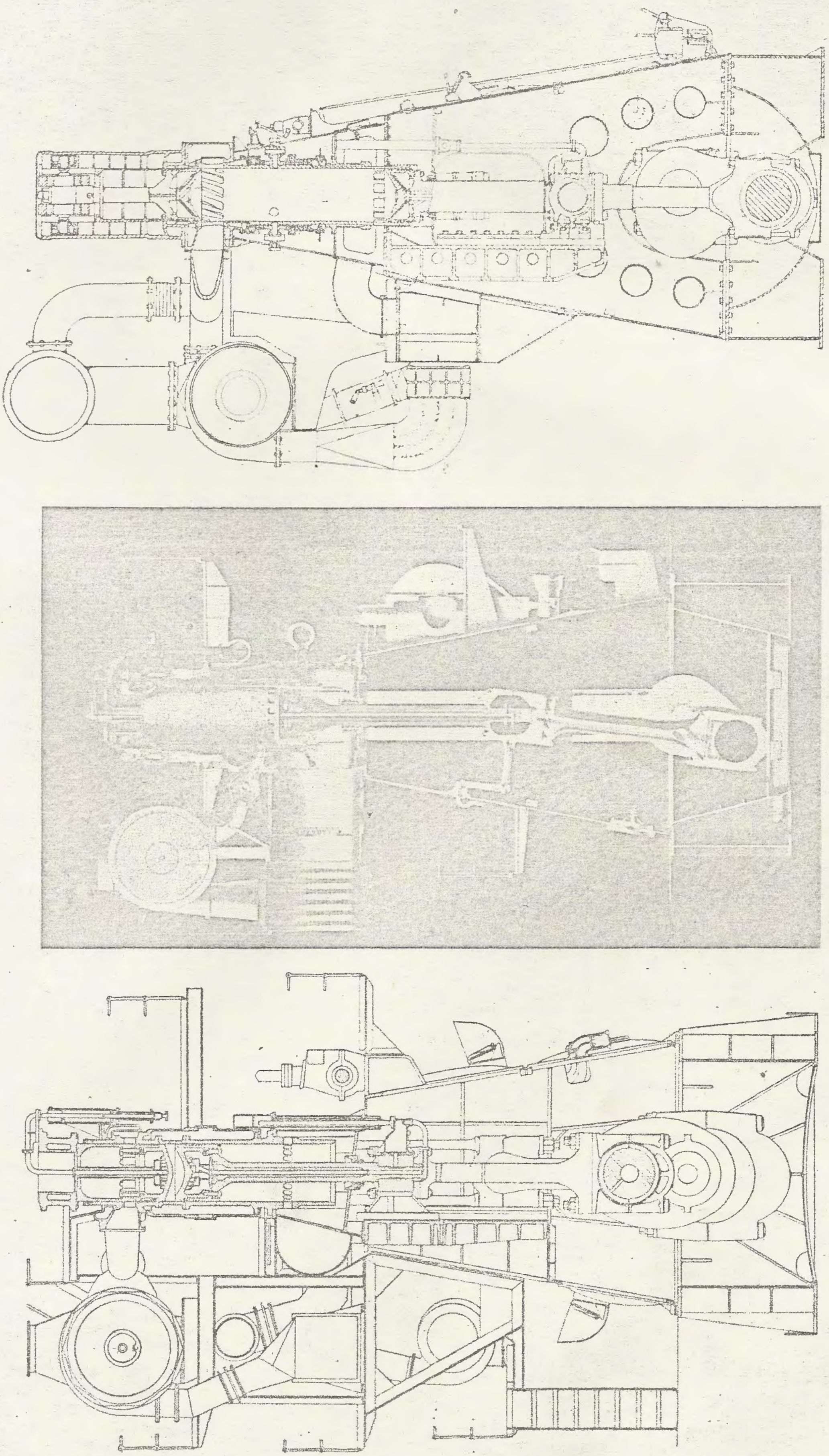


(*a*) Early oil engine
4 S.C.S.A. trunk piston
35 bhp per cyl. at 200 *rpm*

(*b*) M.V. *Selandia*, 1912
4 S.C.S.A. crosshead
125 bph per cyl. at 140 *rpm*

(*c*) M.V. *Gripsholm*, 1925
4 S.C.D.A. crosshead
1,125 bhp per cyl. at 125 *rpm*

FIG. 20.—SOME STAGES IN FOUR-STROKE OIL-ENGINE DEVELOPMENT

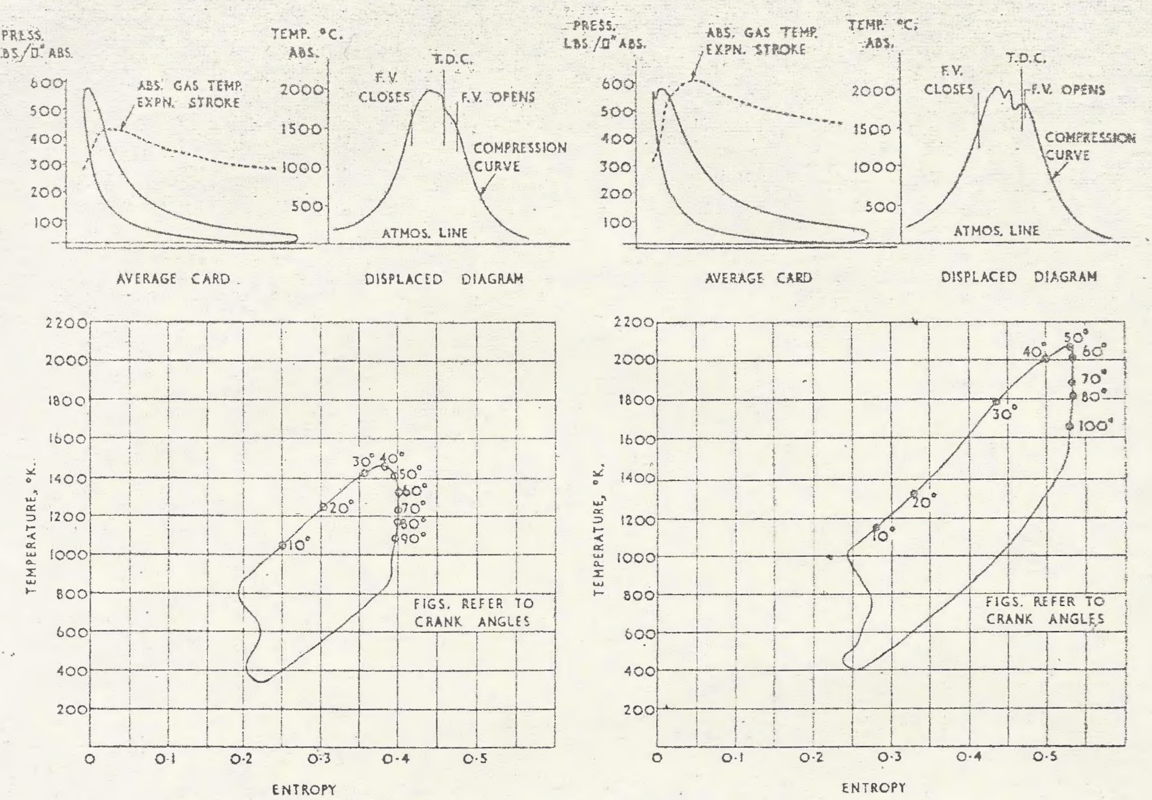


(*a*) Harland & Wolff

(*b*) Sulzer

(*c*) Doxford

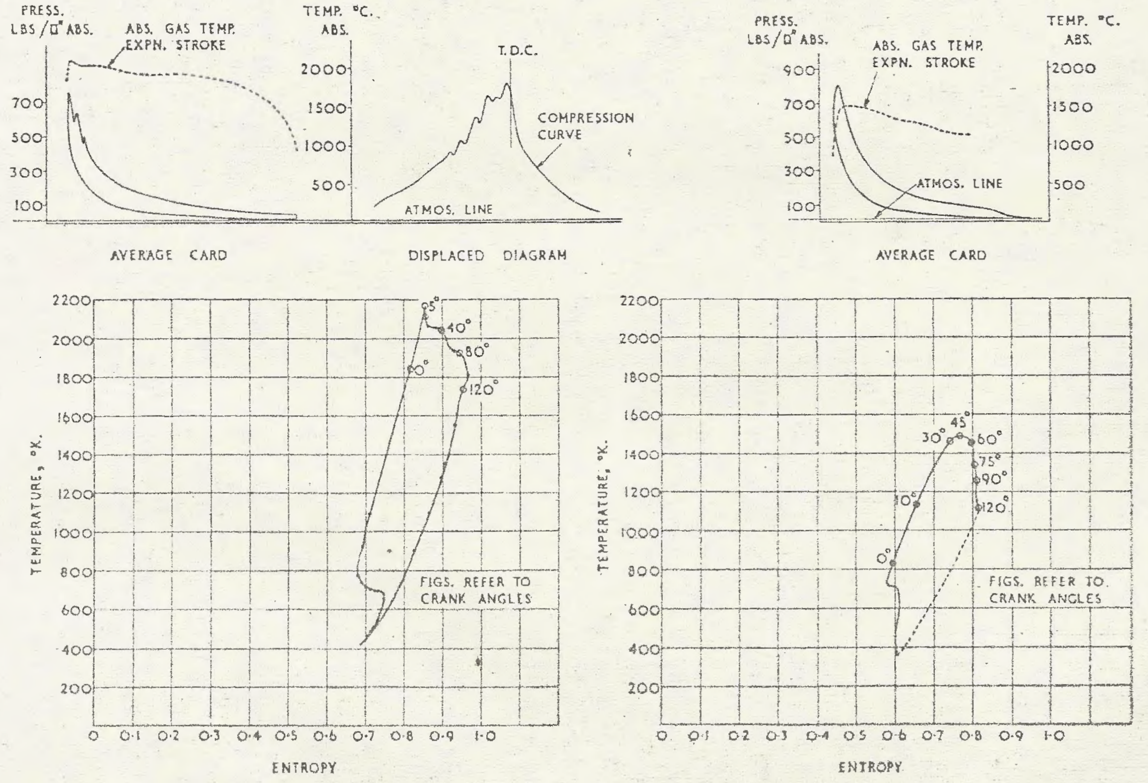
FIG. 21.—MODERN TWO-STROKE MARINE OIL ENGINES



(i) M.I.P. 94·8 P.S.I.

(ii) M.I.P. 134·2 P.S.I.

(A) 4 STROKE AIR-INJECTION DIESEL ENGINE RUNNING AT 200 R.P.M.



(B) HIGH SPEED 4-STROKE AIRLESS INJECTION ENGINE AT 1000 R.P.M.

(C) SLOW SPEED 2-STROKE PRESSURE-CHARGED AIRLESS INJECTION ENGINE AT 115 R.P.M.

FIG. 22.—OIL ENGINE INDICATOR DIAGRAMS WITH CORRESPONDING TEMPERATURE-ENTROPY DIAGRAMS

Their first ship was the tanker *Narragansett*, 1920, with twin-screw, 4-stroke, single-acting engines each having 6 cylinders 24·5 in. bore by 39 in. stroke developing a total of 2,500 bhp at 118 *rpm*.

Next year Messrs. Doxfords put their first oil engine of the opposed piston type into the single-screw ship *Yngaren*. The engine had only 4 cylinders and developed 2,610 bhp at 77 *rpm*. In 1923 the writer went to sea as a junior engineer in the *Dalgoma*, built for the British India Company by Messrs. Alexander Stephen & Sons. The twin-screw engines developing originally a total of 3,200 bhp (latterly 4,000 bhp) had at that time the largest cylinder bore of Sulzer marine designs (680 mm.) and was built by Messrs. Stephen under licence. The machinery was noteworthy as the scavenge air was supplied by motor-driven blowers instead of by the more usual reciprocating scavenge pumps. This ship survived the war and was lost on the Australian Barrier Reef some years afterwards.

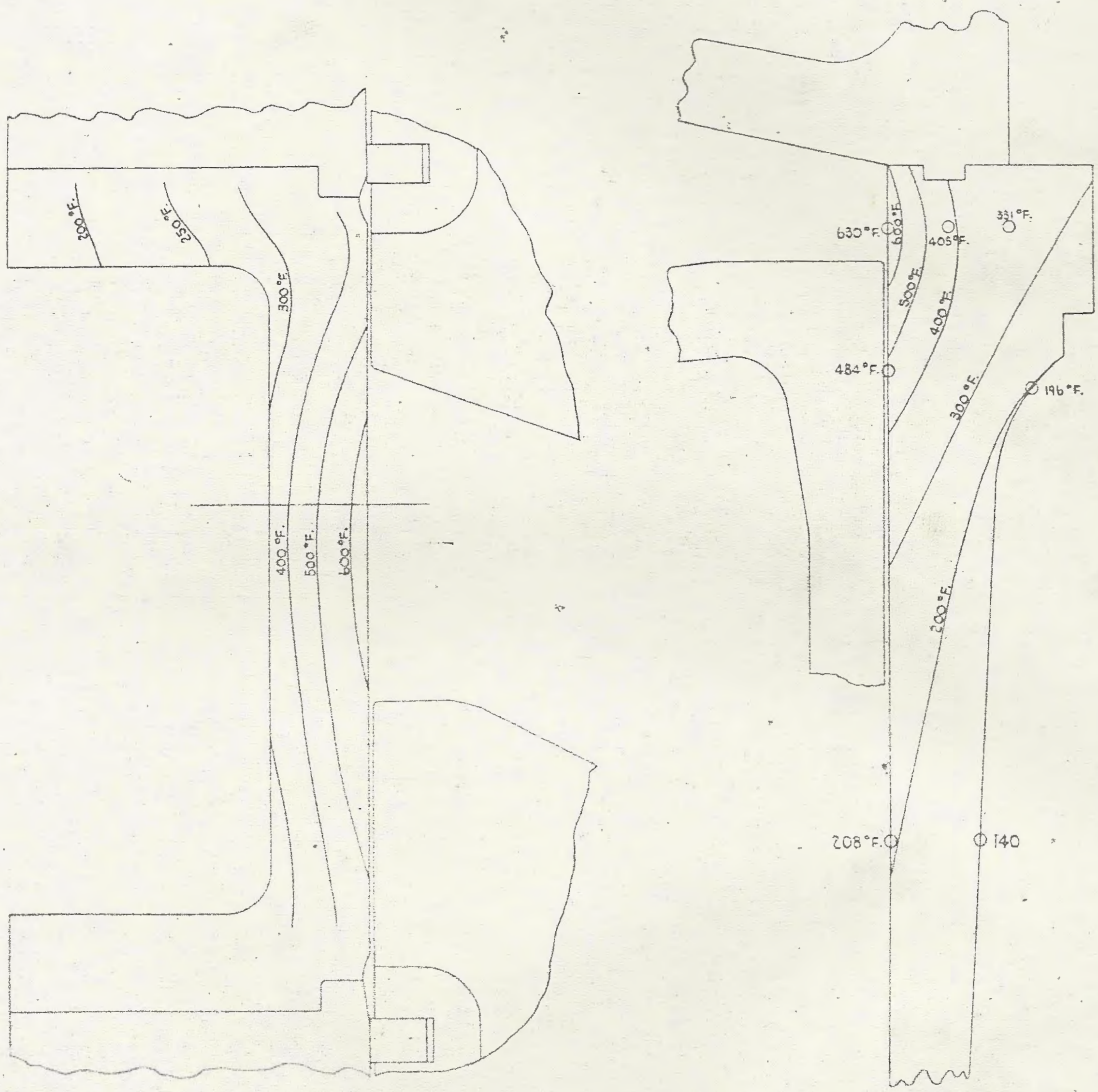
In Table XII the *Dolius* is mentioned. She was a ship engined by a Scott-Still engine which is included as it represented a very skilful but costly attempt to recover and utilize the waste heat in cylinder jackets and pistons by generating steam used on the underside of the oil-engine pistons. Her fuel consumption is scarcely bettered at the present day, but even allowing for turbo charging, etc., the complexity is less.

The year 1924 also saw the high-powered Sulzer machinery manufactured by the Fairfield Shipbuilding & Engineering Company for the M.V. *Aorangi*. She had quadruple screws each driven by a 6-cylinder engine with bores 27·5 in. and 39 in. stroke developing a total of 13,000 bhp at 125 *rpm*.

The 4-stroke cycle engine was produced in the double-acting form in 1925 by Burmeister & Wain, and used to propel the *Gripsholm*. The ship had twin-screw machinery, each engine having 6 cylinders 840 mm. diameter and 1,500 mm. stroke, giving a total output of 13,500 bhp at 125 *rpm*.

In 1927 Doxford introduced a balanced opposed piston engine and four sets of these engines developing 13,500 bhp total were fitted in the liner *Bermuda*. Each engine had 4 cylinders of 22·75 in. bore and 71 in. combined stroke.

Some of the highest powered diesel-engined ships in the world were produced by Messrs. Harland & Wolff and particulars are



(*A*) Harland & Wolff pressure-charged engine
750 mm. bore, 2,000 mm. combined stroke, B.M.E.P. 92 psi
(Estimated temperatures)

(*B*) Early Sulzer S.A. 2-stroke engine
600 mm. bore, 1,060 mm. stroke. M.I.P. 100 psi
Measured temperatures (Eichelberg, 1939)

FIG. 23.—HEAT FLOW IN DIESEL CYLINDERS

given for *Britannic*, *Reina del Pacifico*, *Stirling Castle*, and *Ionic* in Table XII.

One of the great improvements in diesel machinery about the year 1930 was the general adoption of solid injection of fuel. This removed the blast air compressor from the engine and thus improved its mechanical efficiency. It also changed the diesel method of air injection of the fuel and moved towards the original patents of Akroyd Stuart of direct injection. After this date such engines should properly be described as heavy oil engines.

The *Dominion Monarch*, built in 1939, was engined with four Doxford main engines developing a total of 32,000 bhp.

Since the last war, powers developed by single-screw machinery have increased owing to the demand for higher powered machinery in the new super tankers now under construction, and some types produced during 1959 are shown in the table already referred to.

Fig. 20 shows a trunk piston Mirrlees diesel of about 1906, with a cross-section of a crosshead engine as fitted to *Selandia* and the double-acting 4-stroke engines for *Gripsholm*. They have been photographed to the same scale and are all 4-stroke types of diesel engines.

Fig. 21, showing modern Harland & Wolff, Sulzer and Doxford engines, demonstrates that they are all single-acting 2-stroke type engines, all supercharged by turbo blowers, but having three solutions to the scavenge problem.

The Harland & Wolff engine is of the opposed-piston type, the stroke of the upper piston, uncovering the exhaust ports, being one-third of that of the main piston. No mechanical blowers are fitted, the scavenging air being produced by Napier exhaust gas superchargers supplemented by auxiliary fans at low engine speeds or when manœuvring.

The highest powered engine (1959) with cylinder dimensions 750 mm. by 2,000 mm. combined stroke, is a 10-cylinder engine developing 15,000 bhp.

The Sulzer engine is cross-scavenged through inlet and exhaust ports in the lower part of the cylinder walls. The exhaust-gas-driven turbo blowers discharge through a cooler and bank of non-return valves to a separate receiver for each cylinder. As this receiver is in communication with the underside of the working piston, some additional compression is provided by the downstroke of this piston. A rotary valve is fitted in the duct leading from the exhaust ports. When I went to sea with Sulzer machinery in 1923 the rotary valve was fitted to the upper scavenging ports and was quite successful.

A 900 mm. bore engine was announced by Sulzers in 1958 and would produce 22,000 bhp in one engine at 118 *rpm*, using 12 cylinders.

The Doxford engine shown is also turbo charged. The scavenging system works on the "uniflow" principle. The lower end of the combustion cylinder carries a complete circular row of scavenging ports while the upper end has similar but longer exhaust ports both being controlled by the movement of the lower and upper pistons. The mechanical efficiency of the engine is about 92 per cent.

The modern figure for fuel consumption of diesel engines has to some extent been improved by changes in mechanical efficiency. In 1921, with a specific fuel consumption of 0·42 lb. oil/bhp hour, the mechanical efficiency of a 2-stroke engine driving scavenge blower and air compressors was 72·5 per cent. This corresponds to a specific fuel consumption in relation to the indicator cards of 0·304 lb./ihp hour.

A modern engine has a specific fuel consumption of 0·34 with a mechanical efficiency of 89·5 per cent = 0·305 lb./ihp hour.

Fig. 22 gives some typical indicator cards:

A(1) shows a typical 4-stroke blast injection card from an engine running at 94·8 M.I.P. and 200 *rpm*. The lowest left-hand point in the $T - \phi$ diagram represents the beginning of the compression stroke. Where the line is vertical, heat is neither given out by nor lost to the walls. The numbers on the diagram refer to crank angles after T.D.C. It will be seen that combustion is complete by 60 deg. after T.D.C. The maximum temperature is 1,460° K., with 100 per cent excess air.

A(2) shows a diagram for the same engine at maximum overload without pressure-charging. Combustion is complete at 60 deg., but due to the small quantity of excess air (about 15 per cent), the temperatures have reached a maximum of 2,060° K.

The third card B was obtained on a high-speed diesel (1,000 *rpm*) and shows after burning to 110 deg. of crank angle with a maximum cycle temperature of 2,150° K.

The last card C applies to a modern direct-coupled turbo-charged 2-stroke-cycle single-acting engine provided by Mr. Pounder of Messrs. Harland & Wolff. The maximum cycle temperature shown is 1,500° K. Comparing this with the first card, it will be seen in relation to A(2) what a large overload capacity is available and how easy the heat flow conditions are in relation to cylinders and top and bottom pistons.

Mr. C. C. Pounder very kindly provided not only the indicator card, Fig. 22(*C*), but also the following data from a 6-cylinder Harland & Wolff 2-stroke cycle, single-acting, turbo-charged engine at full service power:

Bore	750 mm.
Combined stroke	2,000 mm.
rpm	114·9
bhp	8,625
Fuel consumption	3,089 lb./hr.
Air pressure	6·5 lb./in.^{2}
Air temp.	90° F.
Cooling water inlet temp.	132° F.
Cooling water outlet temp.	140° F.

Using these figures heat flow in the cylinders was calculated and shown in Fig. 23(*A*). For comparison an early Sulzer design is shown in Fig. 23(*B*), and it can be seen how regular the temperature gradients are in the Harland & Wolff engine and hence how low the thermal stresses are in this centre portion carrying the fuel valves and supporting the top and bottom cylinder liners. The diagram gives an idea of the greater output which can be obtained from such engines by greater supercharge.

The most efficient method of pressure-charging is by means of an exhaust-turbine-driven blower, but although this system has been used satisfactorily with 4-stroke engines (Buchi system) for some thirty years, its application to 2-stroke engines was more difficult since:

(*a*) the greater quantity of scavenging air used in 2-stroke engines lowers the exhaust temperature and hence the turbine heat drop;

(*b*) an adequate pressure differential is necessary between blower discharge and turbine inlet, to promote efficient scavenging.

For these reasons high compressor and turbine efficiencies are essential if the turbo-blower unit is to be self-driving, and it is only within recent years that the system has become practicable.

The reliability of the pressure-charged 2-stroke engine is now being demonstrated in service, and it seems probable that the normally-aspirated engine will be largely superseded in the future.

The burning of heavy fuel oil (although not the heaviest grades) in marine oil engines is now accepted practice. The rising price differential between diesel oil and heavy fuel in the post-war years made it worth while accepting higher maintenance costs in order to burn the cheaper heavy oil. Since then the introduction by the oil companies of special lubricating oils containing an alkaline additive has done a good deal to alleviate the difficulties associated with heavy oil burning.

The turbo-charged slow-speed oil engine is a highly efficient engine. The only way further to improve the fuel consumption, apart from utilizing the waste heat in the engine, is to use more constant volume combustion, i.e. earlier fuel injection. This, however, produces higher pressures with some detonation in the cylinder and is unlikely to be adopted to a marked degree.

Future development will probably be in the direction of further reducing size and weight. The trend to still higher degrees of supercharge will no doubt continue until a limit is reached, determined presumably by the materials and maintenance problems associated with the higher specific heat release rates in the cylinders and the higher pressure loadings on piston rings.

When this limit is reached, will we then see a re-emergence of the double-acting 2-stroke engine, this time in pressure-charged form?

A further possibility arises from the fact that as the degree of supercharge is increased the power developed by the exhaust turbine and absorbed by the blower increases until eventually it is of the same order as the useful power developed by the engine. It then becomes possible, if desired, to drive the blower from the engine and take the useful power from the exhaust turbine, so arriving at an arrangement similar (so far as the heat cycle is concerned) to free-piston machinery.

Free-Piston Gas-Turbine Machinery

This naturally leads to a consideration of free-piston gas producers in conjunction with an output turbine to form marine propulsion machinery. Most of the schemes revolve round a G.S.34 gas producer of which leading technical particulars are given below:

Engine cylinder bore	340 mm.
Compressor cylinder bore	900 mm.
Maximum stroke	550 mm.
Stroke at maximum continuous rating	455 mm.
Mean piston speed	8·4 m./sec. 1,650 ft./min.
Cycles per minute	570
Gas pressure at maximum continuous rating	3·1 Kg./cm.^{2} 44·5 lb./in.^{2}
Gas temperature at maximum continuous rating	430° C. 802° F.
Engine rated gas horsepower at maximum continuous rating	1,250
equivalent to	1,000 shp
Thermal efficiency to gas horsepower	43 per cent
Weight	8 tons

The sets so far in service are summarized in Table XIII.

The fuel consumption is attractive, and up to powers of, say, 6,000–8,000 shp a good layout should be obtained.

The gas generators represent fairly highly rated diesel engines in relation to maximum pressures and piston speeds and, in general, the only change for the future is the use of after-burning in the exhaust from the gas generators to give a much higher inlet temperature to the propulsion gas turbine. The hydraulic transmission system could well be used in this application of drive turbines to obviate the windage of ahead blading causing overheating when being driven astern and the loss in the astern wheel at a density of 1 atmosphere when the turbine is driven ahead.

Mixed Cycles

Gasifiers and output gas turbines could be used in conjunction with steam-turbine machinery to their mutual benefit. Such mixed cycles have been examined in the past, but the appropriate designs were not available and without exact figures from test, matching of components is nearly impossible.

The free piston units including an extra one to provide gas for an auxiliary generator would pass the exhaust gas from the gas turbines to the boiler front and further oil would be burnt in this gas (which has at least 100 per cent excess air) to generate steam in the boiler for the steam turbine part of the combined cycle.

For example, machinery to develop a total of 22,000 shp with steam conditions 600 psig, 950° F. would have a steam turbine developing 17,900 shp and 4 G.S.34 gasifiers would supply power to a 4,100 shp gas turbine. Its exhaust would go to the boiler. With such a scheme the fuel rate (propulsion only) would be 0·440 lb./shp/hour.

Similar cycles could be considered using the exhaust from a high temperature gas turbine to obtain even greater gains.

Many other factors such as astern power controls, weight and size of economizers, heat exchangers, and the like would all, however, require intensive study.

Gas Turbines

Gas turbines in which the whole of the compression and expansion of gas takes place in rotating machinery are naturally compared with the well-established and reliable geared steam-turbine machinery. In special applications such as cross-Channel ships where high power in small space and low weight are required, gas turbines would be very suitable, especially if fuel consumption were not important (usage 10–20 per cent), allowing a very simple cycle to be employed. In a similar application as gas boost units in the propulsion of naval ships added to a base load steam plant, they are particularly useful, and for small high-speed naval craft of the motor gunboat type they are, for high powers, the only type of machinery which can be employed. A very recent example of this type of boost gas-turbine unit is seen in Fig. 24. This is an A.E.I. G.6 7,500 shp turbine unit with a free output turbine coupled to the rest of the machinery through gearing. Particulars are given in Table XV under the ship name *Ashanti*. The fuel consumption (gas oil) is 0·754 lb./shp hour, but the specific weight is 5·5 lb./shp hour.

For the merchant ship designed to operate efficiently on long voyages, a much more complex gas-turbine plant is required. It will be seen that the only gas turbine listed in the merchant ship types with a fuel consumption below 0·50 lb./shp hour in Table XV is the Pametrada set which was built as a marine set and consisted of an H.P. gas turbine driving two compressors, L.P. and H.P., with intercooling between. The H.P. compressed air went by way of a heat exchanger to the primary C.C., where the gas was raised to a temperature of 1,250° F. (675° C.). The exhaust from the H.P. turbine drove the output turbine, coupled to the double reduction gear by the Pametrada hydraulic transmission system already referred to in an earlier section. The fuel consumption was 0·491 lb./shp hour, all propulsion purposes.

As fuel consumption of practically 0·500 lb./shp hour (propulsion only) with steam-turbine machinery have already been measured at sea, the gas turbine has to show substantial gains if it is to be used in ocean-going merchant ships.

In order to provide long-life parts, the turbine inlet temperature cannot exceed, say, 1,250° F. (675° C.) with present-day materials unless cooling is used. Cooling allows higher gas temperatures and hence even allowing for cooling losses, a substantial net increase in efficiency. At 1,250° F. and a complex cycle, thermal efficiencies of 28 per cent corresponding to a fuel consumption of 0·495 lb./shp hour, all propulsion purposes, have already been demonstrated. At an output of about 10,000 shp the gas turbine set consisting of supercharger (L.P.

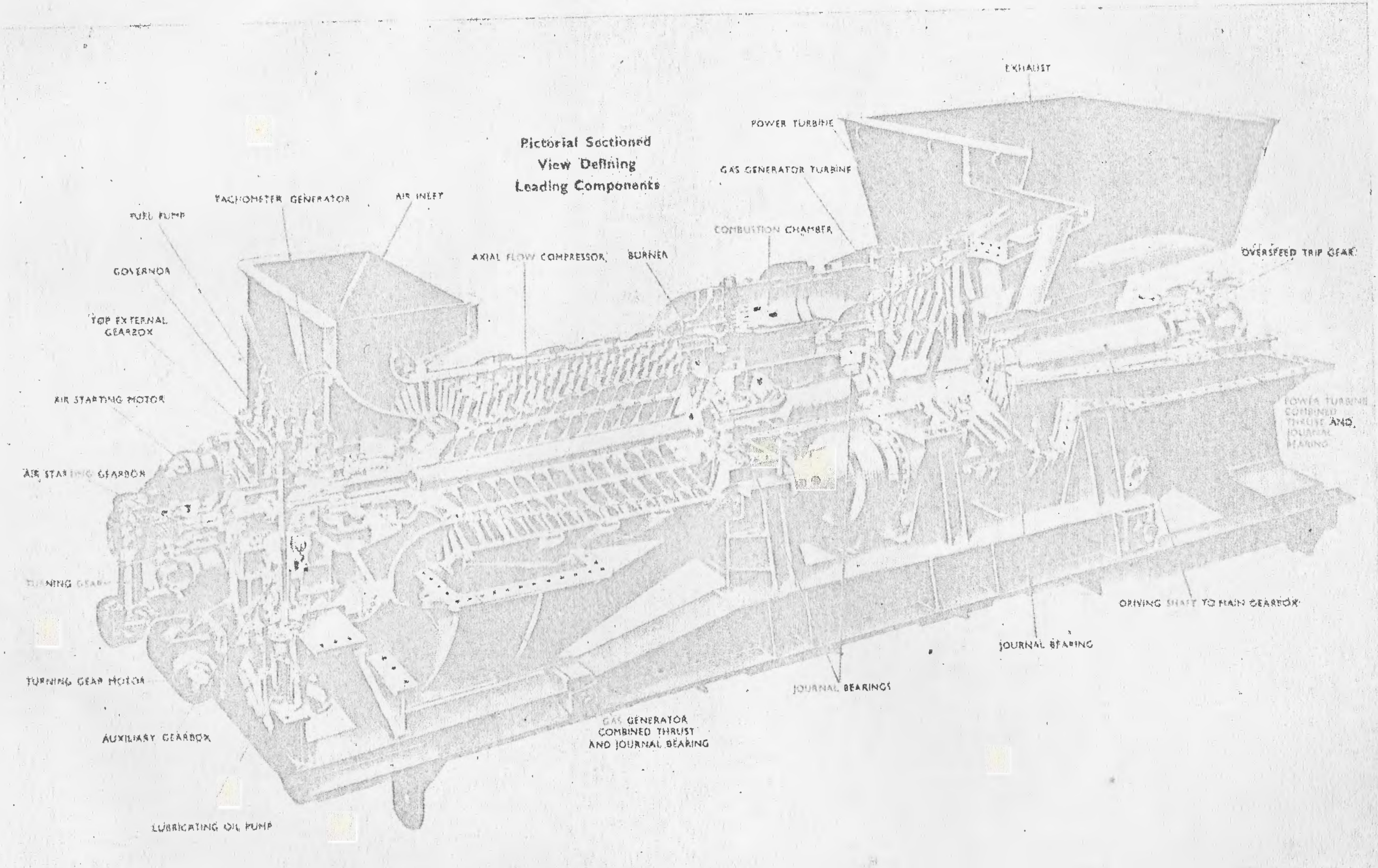


FIG. 24.—A.E.I. G.6 7,500 SHP BOOST GAS TURBINE

compressor driven by L.P. turbine), 2 compressors with intercoolers driven by H.P. turbine, primary and reheat combustion chambers, heat exchangers, and I.P. output turbine, would have a complete machinery weight of 610 tons (about 140 tons lighter than a corresponding steam-turbine installation). The engine-room would have 8 per cent less floor area and about 12 per cent less cubic capacity than that required for the corresponding geared steam-turbine unit. These savings coupled with the hoped-for reliability and low maintenance costs would be the reasons why gas-turbine machinery should be considered at this date. In the meantime, however, the steam-turbine consumption improves and the gains for the gas turbine become less unless the inlet temperature to the turbines is raised. The other important factor may be the ability to burn residual fuel and many experiments have been made leading to particular solutions for certain residual heavy oils. The problem of reversal is the same as that in the output gas turbine operating in conjunction with the free-piston gas generators or high-temperature steam turbines.

A single-stage liquid-cooled turbine has operated at Pametrada at 2.200° F. (1,200° C.), but a great deal of development work is still required in relation to mechanical details, ability to burn residual fuels and layout of the component parts, particularly combustion chambers. However, it holds great promise in the future as already it has been shown that the mean metal temperature in the rotor blading is 700 to 800° F., the maximum local temperatures being 1,000° F. when operating in a gas stream at a temperature of 2,200° F. The blade metal temperatures are actually lower than those in an uncooled gas turbine with an inlet temperature of 1,250° F. The temperature at the rotor where the blades are supported is that of the cooling water, i.e. not exceeding 140° F. (60° C.). Full strength can consequently be developed by the alloys employed in blading, rotor, and other parts and no question of creep or life is involved at the parts of greatest stress. The blades are spark-machined from solid bar and end caps welded on after sodium potassium eutectic has been inserted in the hollow blades in a helium atmosphere. Two water channels in the rotor are at each side of the blades. and heat is brought to the cooling water by the high-speed convection currents in the liquid metal induced by the high gravitational field arising from rotation. A tip speed of 1,000 ft./sec. for the blading causes an acceleration in excess of 10,000 "g".

Table XVI shows a comparison between a present-day steam

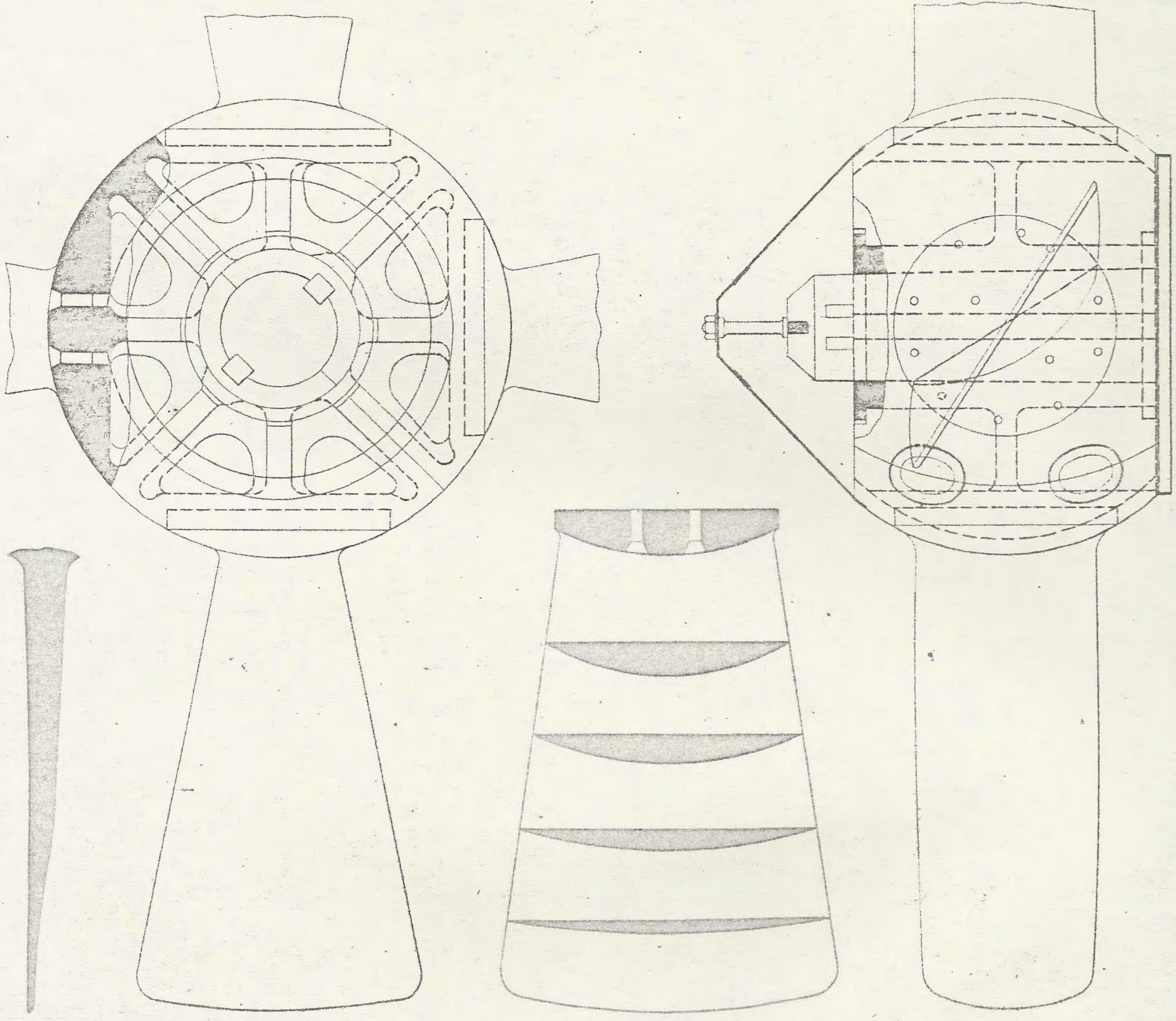


FIG. 25.—SCREW PROPELLER OF S.S. "GREAT EASTERN"

turbine, the high-temperature prototype steam turbine, the liquid-cooled gas turbine, and a heavy oil engine.

This high-temperature liquid-cooled gas turbine will also be of especial use in high-temperature gas-cooled reactors either in a straight cycle or with a free-running gas-turbine circulator. The liquid-cooled turbine will possibly be the only type of machine available, to develop power when reactor gas coolant temperatures reach 1.200° C. or above. The drive to use such high temperatures is the pressing need in nuclear work to reduce the enormous running costs by producing work from nuclear heat as efficiently as possible.

Propellers

In 1860 paddle wheels were freely used. The *Great Eastern* had paddles for part of her propulsion and the *Scotia* worked on the Atlantic for many years propelled by paddles. Paddle wheels have indeed survived to this date for special vessels used in shallow water or with special manœuvring requirements.

The screw had, however, in 1860, reached quite a modern form. Fig. 25 shows the screw for the *Great Eastern* with four separate cast-iron blades riveted to a cast boss to produce a propeller of 24 ft. diameter by 36 ft. pitch. The power absorbed was 4,890 ihp at 38·8 *rpm*. The fairness of the blades on the propeller boss is much better than those using separate bronze blades fitted to bosses with studs and nuts in the 1920's, the whole being faired off with cement.

A modern solid bronze propeller for a large tanker is shown in Fig. 26. This propeller is 23 ft. diameter and weighs 34·5 tons.

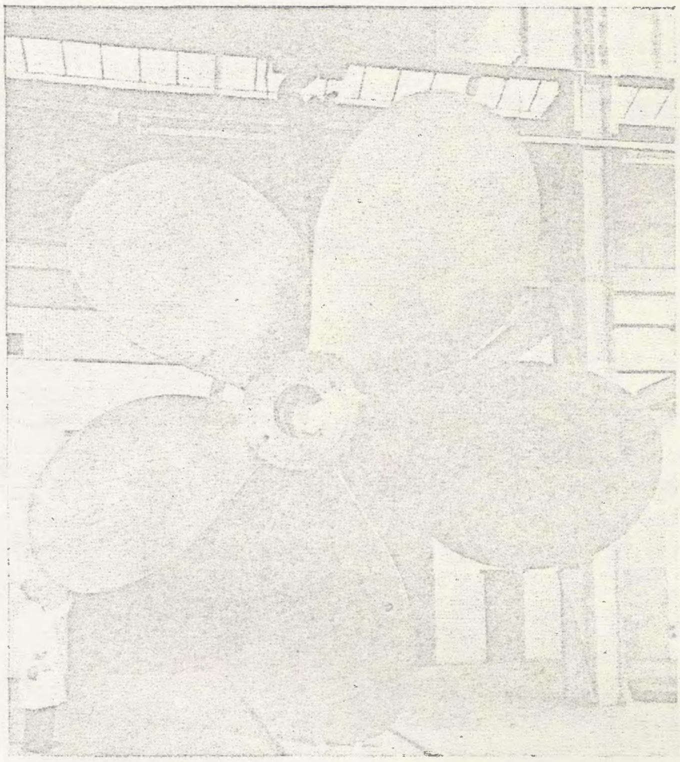


FIG. 26.—MODERN FIVE-BLADED PROPELLER

It absorbs 20,625 shp at 109 *rpm*. The five-bladed propeller is particularly suited to a single screw after end installation as the excitation is small as two blades cannot pass the stern frame at the same time, and any critical frequency of the shafting, gearing, turbine system divided by 5 is likely to be well down in the range of revolutions per minute, and hence well clear of any likely running speed.

Supercavitating propellers have been suggested, but on comparing sizes, the supercavitating propeller would have a somewhat larger diameter, and run at a correspondingly lower *rpm* than those produced by current design methods.

Correctly used, a supercavitating propeller would have design conditions roughly comparable to those of a conventional propeller designed for the same application, but would eliminate the cavitation loss associated with high powers and speeds. These propellers operate with cavitation over the whole of the suction surface of the blades, the cavity commencing at the leading edge and extending beyond the trailing edge. Erosion is avoided and by suitable designs efficiencies comparable with those of a conventional propeller operating under non-cavitating propellers can be obtained. Cavitation is promoted by introducing air through channels in blades leading to the suction surfaces.

Nuclear Propulsion for Ships

Utilizing the strategic advantages of an almost unlimited range of action and the fact that nuclear reactors do not consume oxygen, the United States has embarked on a large nuclear submarine building programme. Both *Nautilus* (1954), 13.400 shp, 2 shafts, and *Skate*, single-shaft, are world-famous for having passed under the North Polar Icecap successfully on August 3, 1958, and August 9, 1958, respectively. Experience has been so successful and the strategic results so great that the largest single reactor programme in any country in the world is devoted to the propulsion of ships. Last year (1959) in the United States Navy:

5 submarines were in commission
1 submarine (*Sea Wolf*) de-commissioned to put in a new reactor
4 submarines were launched and fitting out
12 submarines were under construction
11 submarines were assigned but still to be laid down.

The cruiser *Long Beach* (2 reactors) is launched and the carrier *Enterprise* is fitting out (horsepower 300,000—4 shafts, 8 reactors). There is also a large destroyer *Bainbridge* (2 reactors) under construction. The total is 36 ships and 46 reactors. There are also 6 land-based prototypes.

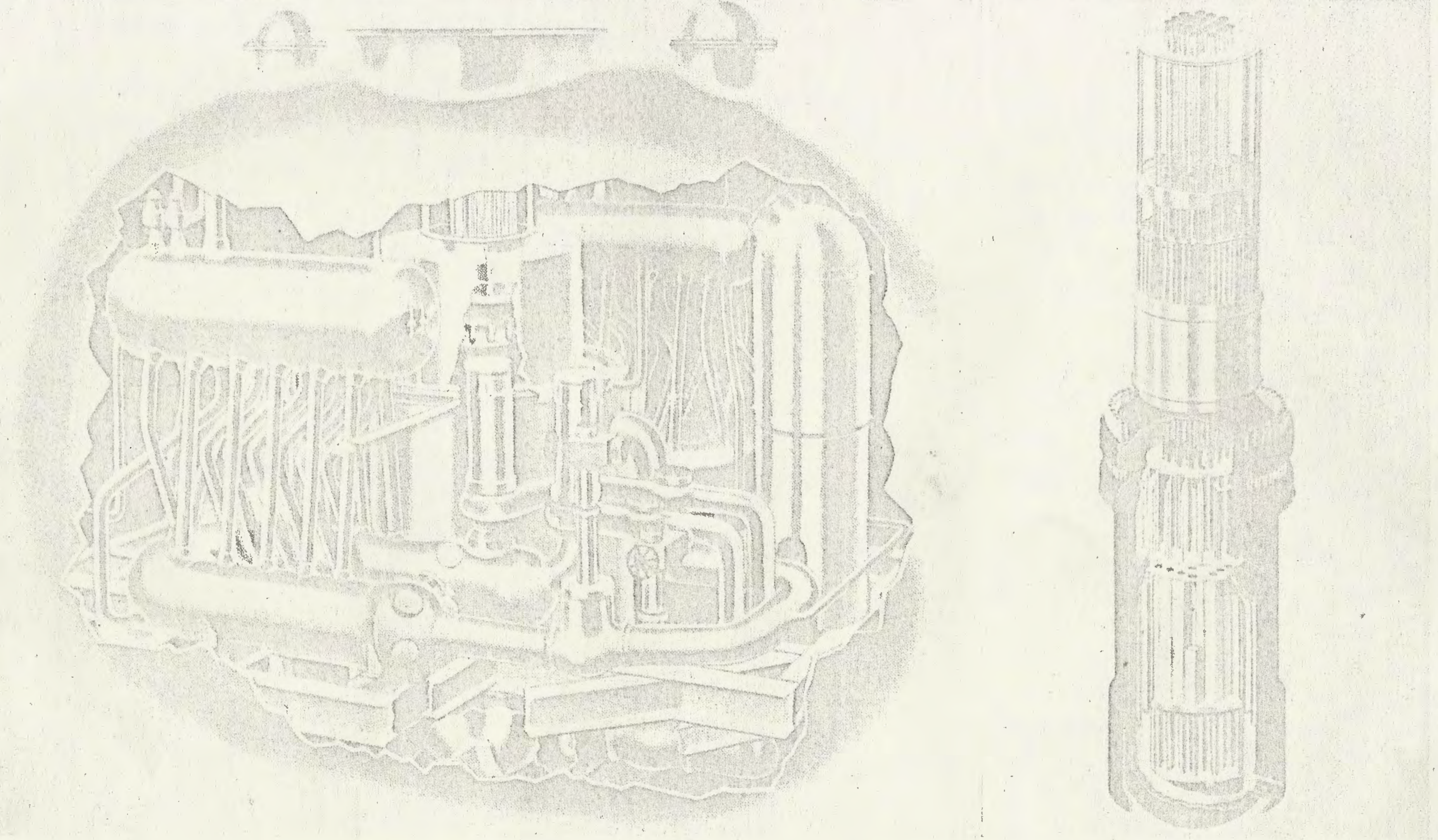
In Great Britain one submarine is under construction with a reactor and machinery bought from Westinghouse in the United States of America. There is a naval reactor test facility at Dounreay and various attempts here and there to start marine reactor work.

On the merchant-ship propulsion side, the Russian icebreaker *Lenin* was on her maiden voyage in September 1959. She has three pressurized water reactors and associated generating plant with a total capacity 1·5 times the power of her propulsion equipment. Any two reactors can therefore satisfy her full-load propulsion requirements, leaving one in reserve or to supply heat to water pumped for ice-melting purposes.

Before the end of this year (1960) the United States merchant ship *Savannah* should enter British ports. This ship also is propelled by a pressurized water reactor. Fig. 27 shows the containment vessel in *Savannah* with the reactor, two heat exchangers, and the coolant circulating pumps in position. The reactor vessel on the right is cut away to show the core and control rods in their correct relative positions.

Table XVIII gives some data for the machinery installations of the *Lenin* and *Savannah*, together with particulars of a number of published design studies for marine nuclear propulsion installations.

To obtain some guidance on reactors for merchant ships, an Admiralty Committee in 1959 called for tenders, and the results of these tenders as far as published are marked with an asterisk in Table XVIII. In the table "indirect" means that a heat



(*a*) Containment vessel showing reactor and heat exchangers in place.

(*b*) Reactor.

FIG. 27.—NUCLEAR POWER-PLANT FOR N.S. "SAVANNAH"

exchanger is interposed between the primary coolant circuit and the working fluid in the turbine to prevent the turbine system becoming radioactive should fission products render the primary circuit dangerous.

It is noteworthy that all the naval reactors in the United States Navy and that of U.S.S.R., together with the ships *Lenin* and *Savannah*, use pressure water reactors. This type was used first in *Nautilus* and has produced a technology using highly-enriched uranium which has led to the merchant ship reactors for *Lenin* and *Savannah*, using enriched uranium, but at a much lower enrichment than that used for naval reactors. In *Savannah*, for example, the enrichment is 4·7 per cent U_{235} (about six times the concentration found in natural uranium).

For size to allow shielding and containment in a merchant reactor, an enriched core is necessary for any arrangement put forward. The hope is to keep the enrichment low to assist in progressing towards an economic reactor which has to pay in relation to present-day orthodox machinery and fuel costs.

It is clear from published figures that the capital cost including the provision of the first core is the main factor which prevents nuclear power from being competitive with machinery using oil fuel at this stage. Capital cost, however, can only be brought down by good engineering improving on an installation already constructed, and no amount of paper work or studies in committee will advance the matter beyond making a wise choice.

Actual construction is essential, otherwise the necessary techniques and craft skills are not learned, and a country becomes dependent on those who have proceeded to build. On building, great knowledge is gained; for example, Babcock & Wilcox can offer now a marine reactor with 2·5 times the output of that in *Savannah* requiring only 20 per cent more space. Techniques of fuelling, de-fuelling, disposal of radio-active wastes, monitoring at sea, reactor effects caused by rolling and pitching, must all have service experience.

Table XVII summarizes all the types of power reactors in the world as far as known to the author, but excludes all research reactors and critical facilities. The variety of reactors will be seen to be large. Nevertheless, from the Second United Nations Conference on the Peaceful Uses of Atomic Energy, Geneva, 1958, it is quite clear that no new reactor systems additional to those propounded in 1955 were produced. Of the graphite-moderated gas-cooled reactors with natural uranium fuel from Calder to Hinkley, the electrical power has gone from 26 MW/reactor (35,000 shp) to 313 MW, taking the output clear of any foreseeable marine requirement.

The pressurized water reactor, which is the only type yet at sea, gives steam conditions at the turbine which are typically 400-500 psig. saturated. These conditions were well known to turbine designers more than thirty years ago. The only difference now in steam-turbine design is that blade speeds are much higher to reduce size and increase efficiency. Moisture in the low-pressure stages has consequently to be removed or serious erosion damage might ensue. Blade shields assist, but removal is a better solution.

Boiling water reactors, if operating on an indirect cycle, give steam conditions at the turbine of the same order as pressurized water reactors. If a direct cycle is used, much higher pressures become practicable, and superheating, although difficult, becomes possible. Large experimental boiling water reactors have been in operation for some years in the United States, and have shown themselves to be safe and reliable.

In the organic moderated and cooled reactor the highest temperature allowable in the coolant stream is 370° C. (692° F.) as above this the polymerization rate would be far too high, which with attendant fire risks and disposal problem of liquid wastes may rule it out as a long-term project, although it is probably the cheapest reactor offering at present for marine use. Steam conditions would therefore be of the same order as with the pressurized water reactor.

With reactors which are either gas (or steam) cooled or liquid-metal cooled, there is no fundamental bar to the attainment of much higher turbine steam conditions, provided the materials problems associated with the core design can be solved. The highest steam temperature arising from a reactor is 1,000° F. from the Sodium Reactor Experiment in the United States of America. Above such temperatures gas turbines would probably be used in association with gas-cooled reactors.

In Table XVIII some longer term projects requiring the use of gas turbines to develop the propulsion power are given. A number also are grouped in the succeeding Table XIX, particularly to show the gas conditions which are postulated to be used by the gas turbines to propel ships.

The coolant fluids which could be used include helium, nitrogen, and carbon dioxide. Neon has also been mentioned, but supply would be difficult. Air is generally ruled out because of the Argon content, which would become radio-active.

There is general agreement that the optimum cycle conditions for these coolant fluids in the reactor (the working fluids in the gas turbine if no heat exchanger is used) occur at about the same temperature ratio (rather than pressure ratio) and that the differences between the peak efficiencies are small. Since comparisons should be based on constant maximum cycle pressure, a fluid having a small optimum pressure ratio such as helium has an initial advantage in that the L.P. section of the circuit can be at a higher pressure.

The relatively good heat transfer properties of helium reduce the size of the heat transfer equipment relative to nitrogen and carbon dioxide.

Comparing the sizes of the rotating machinery if the blade speeds were maintained constant, the number of stages required would be proportional to the specific heat at constant pressure C_p which puts helium at a disadvantage, since its specific heat is about five times that of nitrogen or carbon dioxide. On the other hand, in the case of the compressor the blade speed for helium can be increased since the velocity of sound in helium is very high. The blade speed would then be controlled by blade stress and not by Mach number. In the case of the turbine, stress would control blade speeds for all fluids.

Table XX summarizes the effect of the different working fluids on the design of the gas turbine.

The natural course of reactor development will be towards higher operating temperatures, and as mentioned under the question of the liquid-cooled turbine, turbines capable of working at very high temperatures will be required at least to circulate the coolant through the reactor.

In any longer term survey breeder reactors would need to be considered. These include fast breeder reactors of the general type of Dounreay. In fact, at this stage the best use for plutonium which will arise from the operation of the present civil power stations in Great Britain will be in fast breeder reactors which will form phase 2 of the power-station programme. In this connection it should be noted that the Dounreay fast breeder reactor produced power on November 14, 1959. There are also the thermal breeder reactors designed to make full use of the latent energy in thorium. In the initial loading of a reactor working on a thorium uranium cycle, thorium is inserted with U_{235} and converted to U_{233}, which replaces the fissionable material utilized during reactor operation.

Power from Controlled Thermonuclear Fusion

Considerable research effort is being expended in many countries on finding methods of developing useful power from nuclear fusion. The reactions of most interest are those involving deuterium, tritium, and possibly also helium and lithium. For useful power to be developed temperatures as high as 10^7 to 10^8 °K. will be required, and temperatures of this order will necessitate keeping the hot gases out of contact with the walls of solid containers, both to avoid unacceptable heat losses and to prevent vaporization of the containers.

All thermonuclear devices so far proposed make use of magnetic fields to contain the gas, which is possible since at such temperatures the gases are completely ionized. The heat loss then consists entirely of radiation to the walls, and this is not unacceptably large provided impurities are kept to very small amounts.

At the present time experiments are in progress on a wide variety of systems. The earliest and still the most widely used makes use of the "pinch effect," in which a high current is passed through a tube containing low-pressure gas, when the magnetic field produced by the current causes the ionized gas to constrict towards the axis of the tube, so isolating it from the walls. The gas is heated by the current passing through it and also by the compression associated with the contraction. The tube may be in the form of a torus (as in Zeta, Sceptre, Perhapsatron, etc.), which eliminates the losses at the ends of the tube, but linear tubes are sometimes used for experimental purposes.

Other important devices include the Stellarator and the various "magnetic mirror" systems. In the Stellarator the gas is contained in an endless tube, but the required magnetic field is produced by external coils instead of by a current passed through the gas. The magnetic mirror devices have a straight tube containing the gas and the magnetic field is produced by external coils, end leakage being reduced by making the field strength greater at the ends of the tube than in the middle, giving a so-called "magnetic bottle."

With regard to systems using the pinch effect, the results of different experimenters are in broad agreement. Electron "temperatures" of 10^5 to 10^{6}° K. and ion "temperatures" of $1 - 5 \times 10^{6}$° K. have been obtained, but there is a large unaccountable energy loss. True thermonuclear fusion has not yet been achieved.

Results from other types of system have as yet been equally unsuccessful, with the possible exception of "Scylla," a small magnetic mirror device built at Los Alamos. This has given results which are consistent with thermonuclear fusion having taken place, but no definite claim has been made as yet by the experimenters.

Ware points out: ". . . it must be remembered that the achievement of a detectable reaction is only the first step on a long road to much higher temperatures and densities before a useful amount of energy is released."

Summing up on thermonuclear research generally, Bickerton says: "The general conclusion from the 1958 Geneva Conference is that the behaviour of fully ionized gases is more complex than that accounted for by present theories, and that until a more basic understanding of their properties is obtained, no major break-through in thermonuclear research can be expected."

The energy releases for some fusion reactions are given in Table XXI.

The percentage of heavy hydrogen in ordinary hydrogen is 0·015 per cent. The amount of heavy water in the world is of the order of $1 \cdot 9 \times 10^{14}$ tons and the weight of deuterium is $3 \cdot 8 \times 10^{13}$ tons.

If the third reaction in Table XXI were feasible, the available energy would be 50 million times that of all known hydrocarbon fuel reserves and 3 million times that available from the fission of nuclear fuel. The availability of heavy hydrogen in the waters of the world would prevent any group controlling this source of power. A ship floating anywhere in the oceans would have much more power available than if she floated on pure octane petrol. The argument is as follows:

1 gallon of sea water weighs 10·26 lb. and contains 1·14 lb. of hydrogen.

The corresponding weight of deuterium is 0·000171 lb.

Taking Reaction 3, the heat release is $2 \cdot 48 \times 10^7$ B.Th.U./gal.

1 gallon of petrol weighs 7·4 pounds and energy content = $1 \cdot 5 \times 10^5$ B.Th.U.

The ratio of energy contents is 165 to 1.

Similarly if Reaction 1 were taken the ratio of energy would still be 39 to 1.

So that independently of the efficiency of the heat source, the potential energy available in each gallon of sea water is from 39 to 165 times greater than if high octane petrol were substituted for it.

It is because of such a promise that research effort throughout the world is on the scale shown in Table XXII.

The economics of a fusion reactor plant for marine propulsion are beyond calculation. It can, however, be stated that capital charges will far outweigh fuel costs. Although there will be no fission products in fusion reactions a reaction between deuterons will produce much larger quantities of neutrons than those arising from fission which may make coolants and reactor structure more radio-active. There will therefore still be shielding and containment problems.

Considerations Influencing the Future

The paper has summarized marine propulsion machinery developments up to the present and has given some guidance as to the trends of changes in a relatively short future, certainly within the next hundred years. It must be remembered, however, that despite all efforts to leave our planet or to use ballistic missiles as passenger carriers between land and land, 72 per cent of the globe is covered with water (130×10^6 square miles) and that ships will be required. At present it costs between £400 and £800, depending on the type of goods, to carry 1 ton of freight by air from London to New York, and about £10 by sea. The corresponding heat units required are from 1·0 to $1 \cdot 6 \times 10^8$ B.Th.U./ton for various types of aeroplane; by sea $2 \cdot 0 \times 10^6$ B.Th.U./ton using steam-turbine machinery and $1 \cdot 4 \times 10^6$ B.Th.U./ton using heavy oil engines. Therefore on a utilization of energy basis, carriage of goods by sea only uses between 2 and 1 per cent of the energy required to carry the same weight by air, quite apart from the differential in cost caused by burning distilled fuel instead of residual heavy oil.

Most of the earth mass under the sea is unexplored. Drilling for oil, coal, mineral deposits, and similar projects are in their infancy. The mining of manganese nodules and other metallic ores on the sea bed will also require the use of ships or seaborne platforms.

Although progress will begin by improvements to relatively well-known types of propulsion machinery (including nuclear propulsion), the possibility of new forms of propulsion must not be forgotten. Apart from ship forms, such as hydrofoil craft, hovercraft, hydrocones, and even submarine tankers or ore carriers (which would become practical politics if their voyage to carry oil or ore were materially shortened by passage under the Polar Ice cap), many forms of power producers are being developed in their primitive form in the laboratory. Fuel cells, thermoelectric generation, and electromagnetic generation of power are not at present far advanced, but they may be the first shadows of what will be the method of power generation in the future.

Conclusion

In promoting progress in the propulsion of ships, many powerful bodies in Great Britain such as the Admiralty, Ministry of Transport, Lloyd's Register of Shipping, and the great technical societies such as the Institution of Mechanical Engineers, our own Royal Institution of Naval Architects, the Institute of Marine Engineers, the North-East Coast Institution of Engineers and Shipbuilders, the Institution of Engineers and Shipbuilders in Scotland, and many others abroad have played a dominant part.

In a large measure, however, outstanding achievements are associated with the names of famous inventors such as Watt, Elder, Maudsley, Rankine, Napier, Penn, Kirk, Smith, Ericson in the development of the reciprocating engine and screw, Parsons, De Laval, Zoelly, Curtis, Rateau in the development of the steam turbine, Akroyd Stuart, Diesel, Blache, Kellar, Sulzer, Pounder in the development of the diesel or heavy oil engine, and a host of others. It is clear that in the ultimate all progress depends on the power of the human mind to design, improve, build, run, and repair all these creatures of its dreams.

I began with a quotation from Robert Murray's paper given to this Institution in 1860, and I should like to conclude with a quotation from his second paper to the Institution given in 1865:

> "The engineers of our modern screw steamers, 'combining all the latest improvements,' groan under the endless variety of work now thrown upon them and regret the good old times when surface condensers, superheaters, hydraulic apparatus, centrifugal pumps, auxiliary boilers and steam winches were unknown."

He also remarked that:

> "if a good engine falls into the hands of an uneducated, ignorant, careless or unqualified man, the greatest improvements possible only become tools of injury to the character of the engineer and of loss to the owners."

These statements have equal relevance today, and emphasize the importance of the men who actually build and run the machinery, and we remember with pride tales of the skill, endurance and devotion to duty of the engineers who have served in naval and merchant ships during the hundred years that this Royal Institution has flourished.

Acknowledgments

The author thanks the Council of Pametrada for permission to give this paper. He has been specially aided by Mr. C. C. Pounder, who has taken a great deal of trouble to give information both on important ships built by his company and to provide completely up-to-date technical information on the Harland & Wolff diesel engines.

Drawings of the *Aquitania* were supplied by the kindness of Mr. T. McLaren, Chief Superintendent of the Cunard Company, who also supplied valuable information on some of the early ships belonging to his company.

Mr. W. J. Ferguson of Lloyd's Register of Shipping gave the writer very valuable information on the Inman liners and the *General Admiral.*

I would also like to thank Mr. G. Strachan for information about the *City of Paris* and *City of New York*, built by Messrs. J. & G. Thomson, the forerunners of Messrs. John Brown & Company (Clydebank) Ltd.

The following are thanked for supplying the illustrations mentioned:

Rear-Admiral J. G. C. Given, C.B., C.B.E. The Parsons Marine Turbine Company.	Fig. 14, showing H.M.S. *Grafton* machinery.
Mr. E. R. Cameron, Messrs. Yarrow & Co. Ltd.	Fig. 17, showing an Early Yarrow Water-Tube Boiler.
Messrs. Babcock & Wilcox Ltd.	Fig. 18, showing a High-Pressure, High-Temperature Water-Tube Boiler.
The Manganese Bronze and Brass Company.	Fig. 26, showing a Modern Five-Bladed Propeller.
D. G. Ogilvie, Esq., Messrs. Hawthorn Leslie (Engineers) Ltd.	Fig. 21(*b*), showing a Modern Two-Stroke Sulzer Engine.
G. F. Oliver, Esq., Messrs. Wm. Doxford & Sons (Engineers) Ltd.	Fig. 21(*c*), showing a Modern Two-Stroke Doxford Engine.

Figs. 3 and 24 are taken respectively from *Engineering* and the *Oil Engine and Gas Turbine*.

I have also been helped by many of my colleagues, but particular mention must be made of the untiring efforts of Mr. H. C. Wilkinson.

Statements of facts given in the tables have the references stated and the booklets published by the Science Museum on ships and marine engines have been particularly useful. Other references would have been too voluminous to give individually.

LIST OF TABLES

TABLE I

PROPORTIONS OF SAILING AND STEAM-PROPELLED SHIPS

Date	Tonnage of British ships in service		Percentage of steamships in relation to total tonnage	Iron ships as percentage of all ships launched	Iron ships as percentage of all steamships launched
	Steamships	Sailing ships			
			Per cent	Per cent	Per cent
1849	170,000	3,000,000	5·6	10	20
1866	747,000	4,705,000	13·7	33	80
1871	1,290,000	4,343,000	22·8	50	90

TABLE II

TOTAL GROSS TONNAGE OF VESSELS IN EXISTENCE WITH VARIOUS TYPES OF PROPULSION MACHINERY

Type of machinery	1948		1958	
	Gross tons	Per cent	Gross tons	Per cent
Reciprocating steam engines	42,190,003	52·6	35,430,398	30·0
Reciprocating steam engines with L.P. turbine	1,650,152	2·1	2,144,112	1·8
Turbine	13,595,629	16·9	28,863,219	24·5
Turbo-electric	5,664,959	7·0	5,286,081	4·5
Total steam	63,100,743	78·6	71,723,810	60·8
Diesel	17,037,487	21·2	45,974,649	38·9
Diesel-electric	153,363	0·2	335,272	0·3
Total motor	17,190,850	21·4	46,309,921	39·2
Total steam and motor	80,291,593	100·0	118,033,731	100·0

From Lloyd's Register of Shipping Annual Report, 1958.

TABLE III

PARTICULARS OF "GREAT EASTERN," 1859, CONTRASTED WITH "OCEANIC," 1899

	Great Eastern		*Oceanic*
Length *BP*	680 ft.		685 ft.
Breadth extreme	83 ft., less sponsons 120 ft., including sponsons		68 ft.–5 in.
Depth of side	58 ft.		49 ft.–6 in.
Gross tonnage	18,914 tons		17,274 tons
Displacement	27,384 tons		28,500 tons
Bunkers	10,000 tons		3,700 tons
Speed	14½ knots		21 knots maximum 19 knots service
	Screw engines	Paddle engines	Twin screw engines
Builder	J. Watt & Co.	J. Scott Russell	Harland & Wolff
Type	Horizontal direct-acting	Oscillating cylinder	Vertical 4-crank triple
Number of cylinders	4	4	4 each engine
Diameter of cylinders	84 in.	74 in.	H.P. 47½ in. I.P. 79 in. L.P. 2 at 93 in.
Stroke	4 ft.	14 ft.	72 in.
rpm Normal	40	11	— —
rpm Maximum	53	14	77
ihp	4,886	3,411	
	Total: 8,297 ihp		28,000
Boilers:			
Description	Tubular of rectangular (Box) pattern, double-ended type		Double-ended cylindrical
Dimensions	18 ft.–6 in. long, 17 ft.–6 in. wide, 14 ft. high		16 ft.–6 in. diameter × 18 ft.–6 in. long
Heating surface/boiler	4830 sq. ft.		4,970 sq. ft.
Number of furnaces per boiler	12		8
Steam pressure	25 lb./in.^{2}		192 lb./in.^{2}
Number of boilers	6 to screw engines, 4 to paddle engines		15

Year	1859		1862
Ship	*Great Eastern*		*Scotia*
Shipbuilder	Scott Russell & Co.		Robert Napier & Sons
Engine-builder	Scott Russell & Co.	James Watt & Co.	
Propulsion	Paddles	Screw	Paddles
Number of screws	—	1	—
Type of engine	Simple expansion oscillating	Simple expansion horizontal	Simple expansion side lever
Engine size	4 cylinders, 74 in. dia., 14 ft. stroke	4 cylinders, 84 in. dia., 48 in. stroke	2 cylinders, 100 in. dia., 12 ft. stroke
ihp (total)	3,411	4,886	4,632
rpm	10·75	38·8	15
Mean piston speed (ft./min.)	301	310	360
M.E.P. refered to L.P.-cylinder (lb./in.^{2})	21·8	23·4	27·0
Steam pressure at boilers (psig)	24	25	25
Steam temperature (° F.)			
Vacuum (in.Hg)			
Number of boilers	4 D.E.	6 D.E.	8
Size of boilers	17·8 ft. × 17·5 ft. × 13·8 ft high	17·5 ft. × 18·5 ft. × 14 ft. high	
Total heating surface (sq. ft.)	19,200	30,000	27,600
Total grate area (sq. ft.)	960	1,368	860
Diameter of propeller or paddles	(1) 56 ft. (2) 50 ft.	24 ft.	39 ft. 10 in.
Pitch of propeller	—	37 ft.	—
Coal consumption (lb./ihp/hour)	3·7 (based on 330 ton/day at above powers)		3·4
References	1		1, 2

* Converted to oil burning 1920.—All-purposes fuel rate at 58,000 ihp ÷ shp, 0·97 lb./shp/hour.

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1902 *Kaiser Wilhhelm II* Vulcan Co., Stettin	1910 *Olympic* *Titanic* Harland & Wolff	1931 *Corrales* Alexander Stephen & Sons	1953 *Baron Ardrossan* Wm. Pickersgill & Sons North-Eastern Marine Engineering Co.
2	3	1	1
3-crank quadruple expansion (2 engines per shaft)	4-crank triple expansion engines on wing shafts. Exhaust turbine on centre shaft	Triple Expansion	Triple expansion with Bauer–Wach turbine
37 in. × 49 in. × 75 in. × 112 in., 70·8 in. stroke	54 in. × 84 in. × 97 in. (2), 75 in. stroke	27½ × 46½ in. × 78 in., 54 in. stroke	24 in. × 39 in. × 68 in., 48 in. stroke
40,000	*Recip.* 38,390 *Turbine* 19,270 shp	3,579	2,603
80	*Recip.* 79·2 *Turbine* 169	79·6	78·0
945	990	715	600
35·3	43·2	34·5	38·0
225	215	210	200
		38° F. superheat	600° F. at engine, reheated to 600° F. at I.P. cylinder inlet
		25·5	28·6
12 D.E. + 7 S.E.	24 D.E. + 5 S.E.	4 S.E.	2 S.E.
9 ft. 5 in. dia. × 20 ft. 10 in. (D.E.) 12 ft. (S.E.)	15 ft. 9 in. dia. × 21 ft. (D.E.) 11 ft. 9 in. (S.E.)	16 ft. dia. × 11 ft. 9 in.	17 ft. dia. × 11 ft. 6 in.
107,643	137,000	11,512	6,650
3,121		274	
22 ft. 9 in.	*Wing* 23 ft. 6 in. *Centre* 16 ft. 6 in.	17 ft. 6 in.	17 ft. 9 in.
33 ft. 9½ in.	*Wing* 33 ft. *Centre* 14 ft. 6 in.	19 ft.	16 ft. 7½ in.
1·4	1·24 (propulsion only) 1·40 (all purposes)*	1·37	0·68 (oil–coal equivalent 0·875)
6	1, 9	7	8

TABLE V

LARGE GERMAN LINERS FITTED WITH RECIPROCATING ENGINES

Date	Name	Type of engine	Cylinder size	Stroke in.	lb./sq.in. wp	Total ihp
1897	*Kaiser Wilhelm der Grosse*	4-crank triple	2 H.P. 52 in.; 2 I.P. 89 in.; 4 L.P. 96·4 in.	68·8	178	30,000
1900	*Deutschland*	4-crank quadruple	4 H.P. 36·61 in.; 2 1st I.P. 73·6 in.; 2 2nd I.P. 103·9 in.; 4 L.P. 106·3 in.	72·8	220	36,940
1901	*Kronprinz Wilhelm*	6-crank quadruple	4 H.P. 34·2 in.; 2 1st. I.P. 68·8 in.; 2 2nd I.P. 98·4 in.; 4 L.P. 102·3 in.	70·8	213	36,000
1903	*Kaiser Wilhelm II*	6-crank quadruple	4 H.P. 37·4 in.; 4 1st I.P. 49·2 in.; 4 2nd I.P. 74·8 in.; 4 L.P. 112·2 in.	70·86	225	38,000

TABLE IV

MARINE STEAM RECIPROCATING ENGINES (MERCHANT)

1873 *City of Richmond* Tod & Macgregor, Glasgow	1874 *Britannic* *Germanic* Harland & Wolff Maudsley Sons & Field	1884 *Umbria* *Etruria* Fairfield Shipbuilding & Engineering Co.	1889 *City of Paris* & G. Thomson	1892 *Campania* *Lucania* Fairfield Shipbuilding & Engineering Co.
				Screw
1	1	1	2	2
Compound	Tandem compound	Compound	[illegible] expansion	5-cylinder 3-crank triple expansion
76 in. × 120 in., 60 in. stroke	2 pairs, 48 in. × 83 in., 60 in. stroke	71 in. × 105 in. (2), 72 in. stroke	71 in. × 113 in., [illegible] in. stroke	37 in. (2) × 79 in. × 98 in. 69 in. stroke
3,430	4,970	14,321	20,117	31,050
55	52	69·91	90·8	84
550	520	840	908	967
18·2	29·2	32·5	36·4	35·1
60	70	110	150	165
	Saturated			
26	27			
10 S.E. oval ft. 8½ in. × 12 ft. 6 in. × 10 ft. $4\frac{3}{16}$ in. long	8 D.E. Oval 9 ft. wide × 14 ft. 4½ in. high × 19 ft. 10 in. long	9 D.E. 16 ft. 6 in. dia. × 17 ft. long	9 D.E. 15 ft. 6 in. dia. × 19 ft. long	12 D.E. + 2 aux. D.E. 18 ft. dia. × 17 ft. long
18,216	19,500	38,817	50,265	64,700
585	650	1,606	1,293	2,625
—	23·5 ft.	24 ft. 6 in.	19 ft. 6 in.	23 ft. 0 in.
—	28 to 31·5 ft.	33 ft. 0 in.	28 ft. 0 in.	35 ft. 0 in.
—	3	2·1	1·93 (at 16,000 ihp)	1·6
11	1, 9	2, 3	4, 10	2, 5

Year		1897	1901
Ship		*Turbinia*	*King Edward*
Type of ship		Yacht	River Steamer
Shipbuilder		P.M.S.T. Co.	Wm. Denny & Bros.
Turbine builder			P.M.S.T. Co.
Type of machinery		Direct drive H.P. turbine on starboard shaft I.P. turbine on port shaft L.P. turbine on centre shaft	Direct drive H.P. turbine on centre shaft 2 L.P. turbines on wing shaft
Number of propeller shafts		3	3
shp (total)		2,100	3,500 approximately
rpm		Wing shafts 2,230 Centre shaft 2,000	Wing shafts 755 Centre shaft 505
Steam pressure	lb./in.^{2}	210	150
Steam temperature	° F.		
Vacuum	"Hg	28	26½
Number of main boilers		1 D.E.	1 D.E.
Type of boiler		Yarrow	Scotch
Total heating surface	sq. ft.	11,000	4,909
Steam consumption	lb./shp./hour	14·5 (estimated)	17·1 main turbines (estimated
Fuel consumption (lb./shp/hour)	Coal	Not measured at full power	1·6—1·7
	Oil	—	—
Reference		1	2

* Data for the 48-hour trials of the sister ship, *Lusitania*, built by John Brown Co. (Clydebank) Ltd.

† Measured consumptions at 114,000 shp.

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1904 *Virginian*	1907 *Mauretania*	1887, re-engined 1910 *Vespasian*	1914 *Aquitania*
Liner	Liner	Cargo ship	Liner
Alex. Stephen P.M.S.T. Co.	Swan, Hunter & Co. Wallsend slipway	Short Bros. Re-engined by P.M.S.T. Co.	John Brown
Direct drive .P. turbine on centre shaft L.P. turbines on wing shafts	Direct drive 2 H.P. turbines on wing shafts 2 L.P. turbines on inner shafts	2-cylinder S.R. geared	Direct drive H.P. turbine on port shaft I.P. turbine on starboard shaft 2 L.P. turbines on inner shafts
3		1	4
12,700	70,000	1,095	60,000
250	180	73	180
	195	145	195
Saturated			
	28	28·5	28½
9 S.E. Scotch	23 D.E., 2 S.E. Scotch	2 S.E. Scotch	21 D.E. Scotch
	159,000	3,430	138,596
	12·77* (main turbines) 14·46* (all purposes)	14·3 (main turbines) 14·95 (all purposes)	11·15 (main turbines) (estimated)
1·30	1·43* (all purposes)		1·38 (all purposes)
—	—	—	1·12 (all purposes)
10	3, 4	5	6, 7

. 168.

lation

Year		1904
Ship		*Amethyst*
Type of ship		Cruiser
Shipbuilder		Armstrong Whitworth
Turbine builder		P.M.S.T. Co.
Type of machinery		Direct drive H.P. turbine on centre shaft 2 L.P. turbines on wing shafts
Number of propeller shafts		3
shp (total)		14,000
rpm		Port 499, Starboard 484 Centre 449
Steam pressure	lb./in.^{2}	260
Steam temperature	°F	Saturated
Vacuum	"Hg	Port 26·5 Starboard 27·4
Number of boilers		
Type of boiler		Yarrow
Total heating surface	sq. ft.	25,968
Steam consumption	lb./shp/hour	13·6 (all purposes)
Fuel consumption, lb./shp/hour	Coal	1·74 (all purposes)
	Oil	—
Reference		1

BLE VI

URBINES (MERCHANT)

1928	1929	1931	1936	1952
Duchess of Bedford	*Bremen* *Europa*	*Empress of Britain*	*Queen Mary*	*Normannia*
Liner	Liners	Liner	Liner	Cross-Channel Steam
John Brown	Deschimag	John Brown	John Brown	Wm. Denny & Bros
3-cylinder S.R. geared	3-cylinder S.R. geared	3-cylinder S.R. geared	4-cylinder S.R. geared	Single-cylinder, D.R. geared
2	4	4	4	2
18,000	90,000	60,000	158,000	8,000
115	182·5	Inner 150, outer 200	180	270
350	327	425	400	350
686	680	725	700	650
29	28½	29	29	28½
6	11 D.E., 9 S.E.	8 Yarrow	24	2
Yarrow	Water-tube, 3-drum	1 Johnson	Yarrow	Foster-Wheeler
	183,500		234,000	10,790
	11·1† (all purposes)		7·2 (non-bleed, estimated)	8·16 (non-bleed, estima
—	—	—	—	—
0·57 (propulsion only)	0·735† (all purposes)	0·543 (propulsion only)	0·56 (propulsion only)	
8	12	8	8	9

TABLE VII

MARINE STEAM TURBINES (NAVAL)

1906	1916	1927	1931	1946
Dreadnought	*Repulse*	*Nelson*	*Acheron*	*Vanguard*
Battleship	Battle Cruiser	Battleship	Destroyer	Battleship
ckers, Sons & Maxim P.M.S.T. Co.	John Brown	Armstrong Whitworth Wallsend Slipway	Thornycroft P.M.S.T. Co.	John Brown
Direct drive 2 H.P. turbines on wing shafts 2 L.P. turbines on inner shafts	Direct drive 2 H.P. turbines on wing shafts 2 L.P. turbines on inner shafts	2-cylinder S.R. geared	3-cylinder S.R. geared	2-cylinder S.R. geared
4	4	2	2	4
23,000	112,000	45,000	34,000	130,000
320	275	160		
250	235	250	500	400
Saturated	Saturated	540	750	750
		8	3	8
Babcock & Wilcox	Babcock & Wilcox	Admiralty type	Thornycroft	Admiralty 3-drum
55,400	157,206		26,000	
13·5	11·5 (propulsion only) 12·75 (all purposes)		7·77 (all purposes)	
1·52	—	—	—	
—	1·28	0·789	0·608 (all purposes)	
2, 3	2, 4	8	5	6

1956 *Empress of Britain*	1956 *Caltex Rotterdam*	1958 *Pendennis Castle*	1960 Pametrada Prototype I Machinery for advanced steam conditions
Liner	Tanker	Liner	
Fairfield	Wilton, Fijenoord	Harland & Wolff	Hawthorn-Leslie
3-cylinder, .R. geared, with reheat between H.P. and I.P. cylinders	2-cylinder D.R. geared	2-cylinder D.R. geared	2-cylinder D.R. geared
2	1	2	1
27,000	12,500	42,000	22,000
123	105	130	108
680	600	600	850
850, reheat to 850	950	850	1,050
29	28½	29	28½
3	2	3	—
Foster-Wheeler	Foster-Wheeler	Babcock & Wilcox	—
			—
5·31 (non-bleed)	5·70 (non-bleed, estimated)	6·02 (non-bleed, estimated)	5·16 (non-bleed, estimated)
—	—	—	—
0·495 (propulsion only) 0·537 (all purposes)	0·508 (all purposes)	0·573 (all purposes, at 60 per cent power)	0·477 (propulsion only, estimated)
11			

1951 YEAD I Machinery
—
English Electric (gears by Vickers-Armstrong)
2-cylinder D.R. geared
1 set built for testing
30,000
200
600
950 (60 per cent power) 850 (full power)
26·5
1 for test installation
Babcock & Wilcox
14,793
7·63 (design)
—
—
7

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r steel/carbon st
25/EN. 8

26/EN. 9

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TABLE IX

SOME TURBO-ELECTRIC INST

Year		1927	1929
Ship		*Lexington* *Saratoga*	*Viceroy of India*
Type of ship		Aircraft carrier (U.S.N.)	Liner
Shipbuilder		Fore River S. Corp.	Alex. Stephens
Turbine builder		G.E. (U.S.A.)	B.T.H.
Type of machinery		4 single-cylinder turbo-alternators 8-propulsion motors	2 single-cylinder turbo-alternators 2 propulsion motor
Number of propeller shafts		4	2
shp (total) service		180,000	17,000
rpm		317	109
Steam pressure	lb./in.^{2}	265	375
Steam temperature	° F.	460	700
Vacuum	"Hg	28·5	28·15
Number of boilers		16	6
Type of boiler		Yarrow	Yarrow
Total heating surface (including superheater)	sq. ft.	192,752	42,880
Steam consumption	lb./shp/hour	14·52*	
Fuel consumption (oil)	lb./shp/hour	1·365*	0·65 (all purposes in servi
Reference		1	2

* Measured at 140,000 shp on

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TABLE VIII

SOME HIGH-POWERED NAVAL SHIPS WITH GEARED STEAM-TURBINE MACHINERY

Ship	Type	Date commissioned	hp	Number of shafts
Hood	Battle cruiser	1920	144,000	4
Ark Royal	Aircraft carrier	1938	102,000	3
Victorious	Aircraft carrier	1939	110,000	3
King George V	Battleship	1940	152,000	4
Manxman	Fast minelayer	1940	72,000	2
Vanguard	Battleship	1946	130,000	4
Ark Royal	Aircraft carrier	1955	152,000	4
Forrestal Class (U.S. Navy)	Aircraft carrier	1955	260,000	4
Enterprise (U.S. Navy)	Aircraft carrier	Under construction	300,000	4

IONS

1933	1935	1935
Queen of Bermuda	*Normandie*	*Scharnhorst*
Liner	Liner	Liner
Vickers-Armstrong G.E.C.	Soc. des Ch. et At. St. Nazaire Als.-Thom. Co.	Deschimag A.E.G. Berlin
2 single-cylinder turbo-alternators 4 propulsion motors	4 two-cylinder turbo-alternators 4 propulsion motors	2 single-cylinder turbo-alternators 2 propulsion motors
4	4	2
19,300	120,000	[illegible]
150	[illegible]	[illegible]
350	400	[illegible]
650	[illegible]	[illegible]
28	[illegible]	[illegible]
8	29	[illegible]
Babcock & Wilcox	Penhoët	Wagner-Deschimag
57,560	312,000	[illegible]
	0·649 (all purposes in service)	0·605 (all purposes at [illegible] shp)
3	4, 5	5, 6

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s.s. *Lexington*," *Jnl. A.S.N.E.*, [illegible]

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Propulsion," *Proc. I.Mech.E.*, [illegible]

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TABLE X

Some Typical Gear Particulars

Ship	Type of gearing	Gear materials, pinions/wheel	K value	
Vespasian (1910)	S.R.	[illegible] Ni–Cr steel/carbon steel	83	
Duchess of Bedford [illegible]	S.R.	[illegible] Ni–Cr steel/carbon steel	52	
Caltex Rotterdam [illegible]	D.R. articulated	EN. 25/EN. 8	Primary 75	Secondary 71
British Courage *British Valour* (1957)	D.R. articulated	EN. 26/EN. 9	96·5	106
Naval gears (modern)	D.R. dual tandem articulated	Pinions and wheels hardened and ground	←450→	

Year		1910	1912
Ship		*Vulcanus*	*Selandia*
Shipbuilder		Nederlandsch-Scheepsbouw Maats.	Burmeister & Wain
Engine builder		Nederlandsch Fabriek	
Type of engine		Werkspoor 4-stroke S.A.	B. & W. 4-stroke S.A.
Number of cylinders		6	8
Cylinder bore	mm.	400	530
Stroke	mm.	600	730
Total shp		450	2,500 ihp
Number of screws		1	2
rpm		180	140
Mean piston speed	ft./min.	708	670
B.M.E.P.	lb./in.^{2}		
M.I.P.	lb./in.^{2}		90
Mechanical efficiency	per cent		
Type of injection		Blast	Blast
Scavenge pressure	lb./in.^{2}		
Inlet pressure	lb./in.^{2}		
Number of turbo-blowers		—	—
Type of turbo-blower		—	—
Fuel consumption (at service power on Diesel oil)	lb./bhp/hour lb./ihp/hour	0·415	0·32
Engine weight	tons		
Engine length			35 ft. approximately
Reference		1	2

* From service data in 1935.

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1921 *Yngaren*	1923 *Dalgoma*	1923 *Medon*	1924 *Dolius*	1924 *Aorangi*	C
Doxford	Alex. Stephen	Palmers S. & E. Co. Burmeister & Wain	Scotts	Fairfield	Armstr Burm
Doxford 2-stroke opp. piston	Sulzer 2-stroke S.A.	B. & W. 4-stroke S.A.	Still 2-stroke S.A. above piston Steam below piston	Sulzer 2-stroke S.A.	4-s
4	4	8	4	6	
580	680	740	22 in.	27½ in.	
1,160 + 1,160	1,100	1,500	36 in.	39 in.	
2,610	3,200	3,000 ihp	2,500	13,000 (service)	
1	2	1	2	4	
77	85	85	120	125	
587	614	836	720	812	
90	76·5		74·1	74	
105	105·5	89	Gas 78·8 } 85·4 Steam 6·6 }	90	
85·6	72·5		87·8	82·2	
Airless	Blast	Blast	Airless	Blast	
			2·35	1·8 to 2·0	
—	—	—	—	—	
—	—	—	—	—	
0·441 } Heavy fuel 0·377 }	0·41 } M.E. 0·397 } only		0·358 } All 0·319 } purposes	0·395 (M.E. only) 0·431 (all purposes)	0·395 0·326
300 approximately					
43 ft. approximately				48 ft. (approximately)	3
3	4	5	6	7	

E., 1925, Vol. I, p. 439.

TABLE XII

MARINE OIL ENGINES

1925 *ripsholm*	1929 *Rangitiki*	1930 *Britannic*	1931 *Reina del Pacifico*	1935 *Stirling Castle*	1939 *Dominion Monarc*
ong, Whitworth eister & Wain	John Brown	Harland & Wolff	Harland & Wolff	Harland & Wolff	Swan Hunter Swan Hunter Doxford
B. & W. troke D.A.	Sulzer 2-stroke S.A.	H. & W.—B. & W. 4-stroke D.A.	H. & W.—B. & W. 4-stroke S.A. trunk piston (pressure charged)	H. & W.—B. & W. 2-stroke D.A.	Doxford 2-stroke opp. pisto
6	5	10	12	10	5
840	900	840	630	660	725
1.500	1,600	1,600	1,200	1,500	1,300 + 950
13.500	10,500	20,000	22,000	24,000	32,000
2	2	2	4	2	4
125	91·5	102	145	102	133
1,233	960	1,070	1,140	1,105	1,133 (lower piston
70	73	71·5	110	74	84
84·5	97·5	92			
83	75	78			
Blast	Blast	Blast	Airless	Airless	Airless
			4·2		
—	—	—	1 per engine	—	—
—	—	—	Brown-Boveri	—	—
Propulsion only		0·467 All 0·364 purposes*			
1,030				1,000	
0 ft. 6 in.					
				72 ft. 8 in.	55 ft. approximatel
8	9	10	11	12	13

COMPARATIVE SIZE

Evaporation ..	..
Steam Pressure ,, Temperature	at s out
Heat release rate	..
Boiler efficiency	..
Air pressure to boiler	..
Specific boiler weight (we	
Specific boiler volume	..

1953 *Middlesex*	1959 *City of Melbourne*	1959 *Corhampton*	1959 *Ionic*	1959 *Kongsrang*
Alex. Stephen	Alex. Stephen	Sir James Laing	Cammell Laird	Rosenberg Mek. Werk.
Alex. Stephen		Doxford	Harland & Wolff	Burmeister & Wain
Sulzer				
Sulzer	Sulzer	Doxford	H. & W.	B. & W.
2-stroke S.A. geared	2-stroke S.A. (pressure charged)	2-stroke opp. piston (pressure charged)	2-stroke opp. piston (pressure charged)	2-stroke S.A. (pressure charged)
10	12	6	8	12
580	760	700	750	740
760	1,550	1,400 + 920	1,500 + 500	1,600
9,000	15,600	10,450	13,300	15,000
2	1	1	1	1
225 engines 101 propeller	119	120	117	115
1,120	1,210	1,100 (lower piston)	1,170 (lower piston)	1,210
65	100·5	105	104	102·5
				114·5
				89·5
Airless	Airless	Airless	Airless	Airless
		7·1		6·4
—	4	2	4	4
—	Sulzer	Brown Boveri	Napier	Rateau
0·37–0·375 (M.E. only, excluding transmission losses)	0·34 (M.E. only)	0·335 (M.E. only)		0·35 0·313 } M.E. only
197 tons per engine (excluding transmission)	720	510		600
	72 ft.	61 ft. 1⅝ in.		69 ft. 7 in.
14	15	16	17	18

TABLE XI

Weight of Pressurized Boiler, compared with Two Modern Unpressurized Types

	Boiler A, unpressurized	Boiler B, unpressurized	Pressure combustion boiler
lb./hour	90,000	115,000	87,500
psig.	525	600	600
°F.	825	850	900
B.Th.U./cu. ft. per hour	76,000 (outboard furnace) 102,000 (inboard furnace)	104,000	750,000
per cent	88	88	87·5
lb./sq. in.	—	—	51·4
atmosphere	—	—	3·5
$\frac{\text{lb.}}{\text{lb./steam/hour}}$	3·97	3·11	0·615
$\frac{\text{ft.}^3}{\text{lb./steam/hour}}$	0·123	0·0745	0·027

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28
2
2
2
2
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TABLE XIII

FREE-PISTON MACHINERY

Year	1953–57	1954	1957	1958	1959	1959
Ship	French Navy Minesweepers	*Cantenac Merignac*	*William Patterson*	*Sagitta*	*Morar*	*Goodwood*
Type of ship		Coasters	Liberty ship	Trawler	Ore carrier	Cargo ship
shp (maximum) . .	2,000	1,800–2,000	6,000	2,000	2,500	2,000
Number of screws . .	2	1	1	1	1	1
rpm	500	220	100	250	115	125
Number of free-piston units	2	2	6	2	3	2
Number of turbines . .	2	2	2	1	1	1
Turbine *rpm*	9,000	9,000	5,500	8,000	5,250	7,507
Number of turbine stages ahead		3	6		6	6
Number of turbine stages astern		1	2	—	2	1
Turbine inlet pressure (psig.)			43·5	44	48	38–40
Turbine inlet temperature (° F.)			827	815	900	752–797
Fuel consumption (lb./shp/hour)	0·43 (On test bed at 970 shp per turbine)		*Trials*:— 0·465 (gasifiers) 0·516 (all purposes) *In service*, Voyage 4:— 0·519 (gasifiers) 0·593 (all purposes)		0·49 (Improvement expected after fitting new turbine)	0·435 (At 1,700 shp)
Remarks		Service power 1,200 shp		Reversing by V.P. propeller		Service power 1,700 shp

TABLE XIV

FUTURE CONSUMPTION FIGURES FOR STEAM-TURBINE MACHINERY

Conditions			Reheat points	Heat rate		Auxiliary factor (propulsion only)	Thermal efficiency		Fuel rate (propulsion only), lb. per shp-hour
Inlet pressure, lb./in.: gauge	Inlet temperature, °F.	Vacuum, inches of mercury		B.Th.U. per shp-hour	B.Th.U. per equivalent kW.-hour		Turbines only, per cent	Overall, per cent	
650	850	28½	None	7,941	10,650	1·060	32·1	26·6	0·516
650	850	28½	1	7,596	10,180	1·067	33·5	27·6	0·498
650	850	28½	2	7,423	9.950	1·070	34·3	28·2	0·488
1,100	1,200	28½	None	6,855	9,190	1·069	37·1	30·6	0·450
1,100	1,200	28½	1	6,574	8,820	1·070	38·7	31·8	0·432
1,100	1,200	28½	2	6,383	8,560	1·079	39·9	32·5	0·423
1,500	1,500	28½	None	6,410	8,590	1·077	39·7	32·4	0·424
1,500	1,500	28½	1	6,060	8,120	1·083	42·0	34·1	0·403
1,500	1,500	28½	2	5,926	7,940	1·082	42·7	34·9	0·394

Reheat in all instances to initial temperature.

		Naval			
ear of completion		1947	1951	1951	1954
ip		*M.G.B.* 2009	*Bold Pathfinder* *Bold Pioneer*	—	*Grey Goose*
pe of ship		Motor gunboat	Fast patrol boat	—	Gun boat
ipbuilder				—	J. Samuel White
gine builder		Metropolitan Vickers	Metropolitan Vickers	English Electric	Rolls Royce
igine designation		Gatric	G2	EL60A	RM60
ity		Boost propulsion	Boost propulsion	Main propulsion	Main propulsion
p (one gas turbine)		2,500	4,500	6,000	5,400
imber of screws		1 Gas turbine 2 diesel	2 Gas turbine 2 diesel	—	2
m of screws		1,087	1,100	—	675
ta for one engine—					
Number of compressors ..		1	1	1	3
Number of intercoolers ..		—	—	—	2
Nu[illegible]er of turbines		2	2	2	3
Ou[illegible] turbine		L.P.	L.P.	In parallel with compressor turbine	I.P.
Number of reheaters		—	—	—	—
Heat exchanger thermal ratio ..		—	—	0·75	0·48 at full power bypass open
aximum gas temperature ..	° F.	1,382	1,472	1,299	1,521
cle pressure ratio		3·5	4·0	4·02	18·5
r rate	lb./shp/hour	68·4	52·5	70·8	43·1
el consumption (*a*)	lb./shp/hour	1·075	0·80	0·675	0·675
pe of fuel		Distillate	Distillate	Not quoted	Distillate
ecific weight (*e*)	lb/shp/hour.	2·77	2·28	27·2 (excluding bedplate)	5·3
ethod of reversing		Diesel engines	Diesel engines	Electrical transmission	V.P. propeller
ference		1	1	1	1

Corrected to an *LCV* of 18,000 B.Th.U./lb. Except where stated, the figures are for main engines only.
All purposes. Measured consumption on first voyage 0·51 lb./shp/hour, using diesel oil, *CV* unspecified.
Excluding transmission loss.
Gas turbine replaced one of the original four diesel alternators used for main propulsion.
Main engines, gearing and bedplates, except where stated.
C[illegible] cycle; compressor inlet pressure, 240 p.s.i.a.

1. Trewby, Cmdr. G
2. *Oil Engine and G*
3. Forsling, B. E. G
4. Lamb, J., and D
5. McMullen, J. J.
6. Lamb, J., and B
7. Sawyer, J. W.,

COMPARISON OF MAIN

Type of installation
Steam conditions .. ..
Supt. outlet .. ..
Maximum gas temperature.. .. .. ..
Fuel consumption (propulsion only, lb./shp/hour)
Tons/day .. ..
*Main machinery weight ("steam up") .. ..

* Main machinery weight comprises (*a*) turbines,
compressors, combusti

	Merchant			
1950	1951	1956	1959	Under construction
—	*Auris* (*d*)	*John Sergeant*	*Auris* (re-engined)	—
—	Tanker	Liberty ship	Tanker	—
—	Hawthorn Leslie		Hawthorn Leslie	—
Marine turbine-building firms	B.T.H.	G.E.C. (U.S.A.)	B.T.H.	Escher-Wyss
[illegible] gas turbine	—	—	—	—
		Main propulsion		
1,500	1,200	6,000	5,500	10,000
1	1	1	1	1
85	—	110	120	350
2	1	1	2	1
1	—	—	1	—
2	2	2	2	2
L.P.	L.P.	L.P.	L.P.	L.P.
1	—	—	—	—
0·8	0·5	0·8	0·65	Not quoted
1,200	1,160	1,450	1,200	Not quoted
5·5	4·0	4·9	6·1	3·8
48·5	75	50·3	Not quoted	0·5 (estd. 10 per cent
0·491	0·66 (*c*)	0·546 (estd.) (*b*)	0·538 (estd.) (*c*)	0·73 (estd. full power)
		Heavy fuel		Diesel oil
107	102 (including alternator)	45	70	32
Hydraulic reversing couplings	Electrical transmission	V.P. propeller	Hydraulic reversing couplings	V.P. propeller
2	3, 4	5	6	7

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[illegible] Gas Turbine Project," A.S.M.E. Paper No. 58—GTP—12.
[illegible] Gas Turbine Progress Report: Marine," *A.S.M.E.* Paper No. 58-A-46K.

[illegible] 10,000 shp

[illegible]	[illegible]	[illegible]	[illegible]	Heavy oil engine
600 (psig) 850° F	[illegible]	—	—	—
—	—	[illegible]	1,200° F	—
0·57	[illegible]	[illegible]	0·39	0·38
54·75	[illegible]	[illegible]	41·75	40·75
[illegible]	[illegible]	140	103	425

[illegible] compressors and boilers; for gas turbines, ducting, intercoolers, [illegible] heat exchangers and piping.

TABLE XVII—*continued*

Reactor	Station electrical output, M.W.	Status	Country
Sodium Graphite			
S.R.E.	6	Operating	U.S.A.
Hallam–S.G.R.	76	Design study	U.S.A.
Soviet S.G.R.	50	Constructing	U.S.S.R.
Graphite-Moderated, H_2O Cooled			
A.P.S.–1 Obminck	5	Operating	U.S.S.R.
Soviet–Ural (2 reactors)	200	Constructing	U.S.S.R.
Soviet–Siberia	600	Operating	U.S.S.R.
Ural Atomic Power Station	94	Constructing	U.S.S.R.
Organic Moderated			
O.M.R.E.	15 (thermal)	Operating	U.S.A.
Piqua	12	Planned	U.S.A.
Burlington	50	Planned	U.S.A.
Homogenous Aqueous			
H.R.E.–2	5 (thermal)	Operating	U.S.A.
Czech H_2O	10	Planned	Czechoslavakia
Soviet D_2O Boiling	35 (thermal)	Constructing	U.S.S.R.
Dutch H_2O	0·25 (thermal)	Constructing	Netherlands
D_2O Moderated and *Cooled*			
N.P.D.	20	Constructing	U.S.A.
C.A.N.D.U.	200	Constructing	Canada
Carolinas–Virginia	17	Design study	U.S.A.
Sulzer	30 (thermal)	Design study	Switzerland
Halden	10 (thermal)	Operating	Norway
Swedish R-3/ADAM	11	Constructing	Stockholm, Sweden
D_2O Moderated Gas Cooled			
D.M.–G.C.R.	50	Planned	U.S.A.
Czech–G.C.R.	150	Constructing	Czechoslovakia

	Actual			
	Pressurized water reactors		Pressurized wate	
Author or installation	N.S. *Savannah*	Ice-breaker *Lenin*	Babcock & Wilcox*	Japanese en
Direct or indirect cycle	←		Indirect	
Reactor heat output (max.) MW.	74	3 × 90	180	18
Core length	7 ft. 6·24 in.	5 ft. 3 in.		4 ft.
Core diameter	5 ft. 2·06 in.	3 ft. 3 in.		5 ft.
Fuel				
Type	UO_2	UO_2 (sintered)	UO_2	U
Initial enrichment, per cent U_{235}	4·7	5	4 per cent (S.S. can) 2 per cent (Zr can)	1·
Total weight in core tonnes	7·065		9 (approximately)	8·
Maximum fuel temperature ° F.	3,660			2,7
Mean rating, MW/tonne	10·45		20	21·
Average burn-up MWD/tonne	7,360			9,2
Fuel Can				
Material	Boronated S.S.	Zirconium alloys	Zirconium or S.S.	Zircal
Maximum surface temperature	610			
Moderator				
Type	Light water	Light water	Light water	Light w
Coolant				
Medium	Light water	Light water	Light water	Light w
Mean pressure (lb./in.^{2})	1,735	2,940	1,750	1,99
Outlet temperature (° F.)	521	617		538
Pressure Vessel				
Type	Vertical cylinder	Vertical cylinder		
Diameter	8 ft. 2 in.	6 ft. 7 in.		
Length	28 ft. 6 in.	16 ft. 5 in.		
Containment				
Dimensions	35 ft. dia. × 50 ft. long		42·5 ft. dia. sphere	
Main Machinery				
Type	←			
Full power shp (total)	22,000 (single screw) 110 *rpm*	44,000 (3 screws) 185 *rpm* centre 205 *rpm* wing	65,000	60,00
Turbine inlet pressure (lb./in.^{2})	435	399	410	56
Turbine inlet temperature (° F.)		590	447	482
Condenser vacuum ("Hg.)	28			
Cost				
Capital cost (production model)	$14·5 M.			
Fuel cost pence/shp-hour	0·71		0·3 to 0·6	
Weight				
Total machinery weight (tons)	4,348 (shielding and containment 2,418)	5,767 (shielding 1,963)	3,000	
Reference	1	2	3	7

TABLE XVII—*continued*

Reactor	Station electrical output, M.W.	Status	Country
Sodium Graphite			
S.R.E.	6	Operating	U.S.A.
Hallam–S.G.R.	76	Design study	U.S.A.
Soviet S.G.R.	50	Constructing	U.S.S.R.
Graphite-Moderated, H_2O Cooled			
A.P.S.–1 Obminck	5	Operating	U.S.S.R.
Soviet–Ural (2 reactors)	200	Constructing	U.S.S.R.
Soviet–Siberia	600	Operating	U.S.S.R.
Ural Atomic Power Station	94	Constructing	U.S.S.R.
Organic Moderated			
O.M.R.E.	15 (thermal)	Operating	U.S.A.
Piqua	12	Planned	U.S.A.
Burlington	50	Planned	U.S.A.
Homogenous Aqueous			
H.R.E.–2	5 (thermal)	Operating	U.S.A.
Czech H_2O	10	Planned	Czechoslavakia
Soviet D_2O Boiling	35 (thermal)	Constructing	U.S.S.R.
Dutch H_2O	0·25 (thermal)	Constructing	Netherlands
D_2O Moderated and *Cooled*			
N.P.D.	20	Constructing	U.S.A.
C.A.N.D.U.	200	Constructing	Canada
Carolinas–Virginia	17	Design study	U.S.A.
Sulzer	30 (thermal)	Design study	Switzerland
Halden	10 (thermal)	Operating	Norway
Swedish R-3/ADAM	11	Constructing	Stockholm, Sweden
D_2O Moderated Gas Cooled			
D.M.–G.C.R.	50	Planned	U.S.A.
Czech–G.C.R.	150	Constructing	Czechoslovakia

TABLE XVII

Power Reactors (Not Including Research Reactors or Critical Facilities)

Reactor	Station electrical output, MW.	Status	Country
Nonboiling H_2O			
Shippingport	60	Operating	U.S.A.
Yankee	135	Constructing	U.S.A.
Indian Point	275	Constructing	U.S.A.
A.P.P.R.—1	2	Operating	U.S.A.
Savannah Ship	20,000 shp	Constructing	U.S.A.
Voronezh—P.W.R.	420	Constructing	U.S.S.R.
Leningrad—P.W.R.	420	Planned	U.S.S.R.
Soviet P.W.R. Mobile	2	Constructing	U.S.S.R.
Icebreaker *Lenin*	66,000 shp	Operating	U.S.S.R.
Emigrant Ship	180 (thermal)	Design study	Japan
Submarine Tanker	180 (thermal)	Design study	Japan
S.M.I.	1·85	Operating	U.S.A.
Boiling H_2O			
Borax IV	3·5	Operating	U.S.A.
E.B.W.R.	5	Operating	U.S.A.
V.B.W.R.	5	Operating	U.S.A.
Dresden	192	Constructing	U.S.A.
Elk River	22	Constructing	U.S.A.
Northern States	62	Planned	U.S.A.
Pacific Gas and Electric	50	Planned	U.S.A.
Soviet B.W.R.	50	Constructing	U.S.S.R.
Belgonucleaire IC–B.W.R.	129	Design study	Belgium
Cuba B.W.R.	22	Design study	Cuba
KAHL/MAIN	15	Constructing	Germany
Graphite Moderated, Gas-Cooled			
Calder (2 reactors)	92	Operating	U.K.
Calder B (2 reactors)	92	Operating	U.K.
Chapelcross (4 reactors)	182	Operating	U.K.
Berkeley (2 reactors)	332	Constructing	U.K.
Bradwell (2 reactors)	352	Constructing	U.K.
Hunterston (2 reactors)	344	Constructing	U.K.
Hinkley Point (2 reactors)	626	Constructing	U.K.
Trawsfynydd (2 reactors)	550	Constructing	U.K.
Dungeness (2 reactors)	550	Planned	U.K.
U.K.–A.G.R.	28	Constructing	U.K.
U.K.–H.T.G.R.	10 (thermal)	Planned	U.K.
O.R.N.L.–G.C.R.	252	Design study	U.S.A.
K.A.C.F.–G.C.R.	253	Design study	U.S.A.
G. 1	5	Operating	France
G. 2	30	Operating	France
G. 3	30	Operating	France
Latina	200	Constructing	Italy
Tokyo	159	Planned	Japan
E.D.F. 1	68	Constructing	France
E.D.F. 2	195	Constructing	France
Fast Reactors			
E.B.R.	20	Constructing	U.S.A.
Enrico Fermi	100	Constructing	U.S.A.
Dounreay	15	Constructing	U.K.
B.R. 5	5 (thermal)	Operating	U.S.S.R.
B.N. 50	50	Planned	U.S.S.R.
B.N. 750	250	Planned	U.S.S.R.

Table XVII *continued on p.* 430

TABLE XIX

Gas Turbine Nuclear Power Plant Designs

Organization responsible for design	Power output	Heat output	Working fluid	Gas conditions (reactor outlet)	Fuel	Enrichment	Can	Moderator	Ref. No.
American Turbine Corporation	60 MW.	148·5 MW.	Helium	1,000 p.s.i.a./1,400° F.	← Not covered in report →				1
Aerojet-General†	Heat dumped	Classified	Nitrogen	Several hundred lb./in.^{2} >1,000° F.	UO$_2$ in stainless steel	"Highly enriched"	Stainless steel	Classified	2
General Atomics (Division of General Dynamics)	20,000 shp	55 MW.	Carbon dioxide	2,000 lb./in.^{2}/1,300° F.	UO$_2$ dispersed in stainless steel	28 per cent U$_{235}$	Stainless steel	Zirconium hydride	3
General Motors	20,000 shp	50 MW. service 55 MW. max.	Helium	1,000 lb./in.^{2}/1,300° F.	UO$_2$ dispersed in stainless steel	"Fully enriched"	Stainless steel	Graphite	4
G. G. Sharp Inc. American Turbine Corporation	20,000 shp	48·5 MW. Normal	Nitrogen	705 p.s.i.a./1,300° F.	UO$_2$ dispersed in stainless steel	20 per cent U$_{235}$	Stainless steel	Beryllium oxide	5
U.K. Atomic Energy Authority*	Heat dumped	10 MW.	Nitrogen	150 lb./in.^{2}/1,382° F.	Enriched uranium and thorium pellets	Not quoted	Graphite	Graphite	6

* Not specifically intended for use with a gas turbine.

† The Gas Turbine Test Facility (G.T.T.F.) being erected at the U.S. Army Engineer Research and Development Laboratory at Fort Belvoir is to proof-test a gas-turbine generator set for this application with the specification:—

Power output	400 kW.
Thermal efficiency	16 per cent at least.
Turbine inlet conditions	212 p.s.i.a./1,150° F.

References

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3. Gas-Cooled Reactor Concepts. W. I. Thompson (General Atomic Division of General Dynamics Corporation), PAM/B/DOC/76.
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6. Interesting Reactor Designs (I) High-Temperature Gas Cooled Reactor. *Nuclear Engineering*, Oct. 1958, p. 428.

TABLE XX

Comparisons of changes in Design arising in Gas Turbine from different Working Fluids

(At constant maximum cycle pressure)

Gas	Comparative surface areas heat exchangers	Comparative surface areas intercoolers	Comparative surface areas precooler	Comparative annulus area L.P. end of compressor	Comparative number of stages of compressor	Comparative number of stages of turbine	Turbine mean blade speed, ft./sec.
He	0·34	0·39	0·34	1·03	1·33	2·67	1,100
N_2	1	1	1	1	1	1	875
CO_2	0·86	0·92	1·10	0·92	1·55	1·67	527

TABLE XXI

Energy Releases for Some Fusion Reactions

	Mass defect, lb./lb.	Energy release, B.Th.U./lb.
Fusion of deuterium to form He^3	$8{\cdot}7 \times 10^{-4}$	$3{\cdot}4 \times 10^{10}$
Fusion of deuterium to form tritium	$10{\cdot}7 \times 10^{-4}$	$4{\cdot}1 \times 10^{10}$
Fusion of deuterium and tritium to form He^4	38×10^{-4}	$14{\cdot}5 \times 10^{10}$
Fusion of deuterium and lithium to form He^4	30×10^{-4}	$11{\cdot}6 \times 10^{10}$

TABLE XXII

Nuclear Fusion Devices described at Geneva, 1958

(from Bickerton, *Engineering*, 1958, Vol. 186, page 824)

Main classification	Sub-division	Devices	Location	Dimensions cm.	
				Torus bore	Torus dia.
Closed magnetic lines	Toroidal stabilized pinch	Uppsala torus	Sweden	28	130
		Saclay torus	France	8	78
		Alpha	U.S.S.R.	150	450
		Zeta	U.K.	100	300
		Perhapsatron S4	U.S.	14	70
		Gamma pinch	U.S.	10	60
		Sceptre	U.K.	30	110
		Moscow	U.S.S.R.	48	125
				Tube bore	Tube length
	Linear pinch	Moscow pinear pinch	U.S.S.R.	40	50
		Uppsala linear pinch	Sweden	30	60
		Munich linear pinch	Germany	20	50
		Columbus II	U.S.	10	30
		Columbus S4	U.S.	13	60
		Columbus T1	U.S.	15	600
		Maggi	U.K.	15	28
		Saclay linear pinch	France	28	100
		Tri-axial pinch	U.S.	10	100
		Screw dynamic	U.S.	10	20
	Stellarators	B1	U.S.	5	450
		B2	U.S.	5	600
		B3	U.S.	5	600
		B65	U.S.	15	500
		C	U.S.	20	1,200
	Astron	Pilot model	U.S.	100	1,000
		Final power model	U.S.	200	2,000
Open magnetic lines	Mirror or adiabatic trap devices	DCX	U.S.	50	50
		Ogra	U.S.S.R.	140	2,000
		Felix	U.S.	45	270
		High compression	U.S.	15	100
		Scylla	U.S.	7	7
	Rotating plasmas	Ixion	U.S.	24	86
		Homopolar	U.S.	25	10