STEAM IN THE ROYAL NAVY

BY

CAPTAIN R. G. RAPER, R.N., M.I.MECH.E., M.I.MAR.E.

This paper was first presented by the Author at the Institution of Mechanical Engineers' conference at Harrogate in May, 1963, and has gained for the Author a Thomas Lowe Gray prize. It is reproduced by permission of the Author and the Institution.

INTRODUCTION

The responsibilities which the Royal Navy has to fulfil with its ships are very diverse, both in function and geographically. With a restricted budget it is therefore important to produce as large a number of ships as possible with as much versatility in each one as can be packed into a minimum displacement.

Twenty years ago, of a destroyer's total displacement about 40 per cent was absorbed by the hull and general ship's structure and fittings and 45 per cent by the machinery and fuel stowed, so that only 15 per cent was left for weapons, ammunition, stores, and personnel. If the weight of the machinery could be reduced and its efficiency increased so that it used less fuel to travel the same distance and the weight of fuel and machinery was thus reduced to 30 per cent of the displacement, the weight of weapons, radar, and communications equipment could be doubled. Alternatively, of course, more fuel could be stowed and the ship could go farther, faster. There is therefore no escape from a desire to reduce the size and weight of naval machinery and to increase its efficiency. Furthermore, if ships of the same speed are considered, the smaller the ship the greater is the proportion of the displacement absorbed by the weight of the machinery and the fuel and it is therefore to the small ships in particular that the greatest attention has been paid since 1946.

From practical and economic considerations, the speeds at which the highest efficiency is required for the longest periods are in the region of half the full speed of the ship; this means at less than 20 per cent of full power and often in the region of 5–10 per cent. At the same time it is clearly important to be able to go as far as possible at high speed without refuelling, especially at a time when bases are getting fewer and the distances between them greater.

It is these two features, the emphasis on reduction in weight and space and the need for high efficiency at a low proportion of full power, which cause the main differences between naval and merchant-ship machinery and have led to the development of various features which are peculiarly naval.

Two other factors influence the selection of machinery and its design: the need to operate at any speed from almost stopped to full power for prolonged periods and at reasonable efficiency and to manœuvre up and down within this range without any pause for sustained periods; and the need to be able to withstand very high shock accelerations, due to underwater explosions, vertically and horizontally.

Added to these known factors there are unknown factors for the future, because the tactical requirements for ships are bound to alter as new weapons and new methods of defence are developed. If a set of machinery has some particular speed or power which it is hoped to avoid because of a critical stress or vibration of some sort, it is almost certain that, in due course, this will be the very speed required for some tactical purpose. Blind spots are therefore regarded with great disfavour.

A study of these requirements and of the ways in which they might be met has resulted in two main types of steam plant for small ships using between 15,000 and 60,000 s.h.p. The first, chronologically, fitted in the *Whitby* class frigates, conists of two boilers supplying steam at 550 lb/sq in. and 850 degrees F. to two single-cylinder 15,000 h.p. turbines, each driving a propeller shaft through double-reduction gearing. In order to obtain high efficiency at low speeds a cruising turbine which can be clutched in and out as speed falls below or rises above a certain speed was also incorporated in the original design. The steam temperature can be controlled to maintain 850 degrees F. at any power down to about 10 per cent of full power or dropped to 750 degrees F. at any power for manœuvring. A feed-heater deaerator is incorporated and an economizer but no air heating. The wide variation of operating powers precludes bleed feed-heating from the main turbines and the feed-heating has been kept as simple as possible, consistent with reasonable heat recovery, in order to reduce weight and space.

A vacuum of only 23 in. at full power was selected in order to reduce the weight and size of turbine and condenser. This involves a loss of efficiency at full power but the difference at powers below half power is negligible. The effect on boiler evaporation at full power and thus the penalty in weight for the boilers, feed system, steam pipe sizes, pump sizes, etc., was assessed to obtain the optimum vacuum for minimum total weight of machinery and fuel required to travel a given distance at various specified speeds.

The second type of installation was made possible by the development of a naval gas turbine suitable for ship propulsion. This offered the possibility of obtaining a power of about 7,500 h.p. though at rather lower efficiency in a small weight and space. It was considered reasonable to rely upon one boiler in a frigate if an alternative means of propulsion to steam turbines was provided and it was hoped that by providing 30 to 50 per cent of the full power required on the shaft by a gas turbine the steam plant could be designed for a power nearer to that at which it would normally be employed. Hence a higher efficiency would be obtained at low proportions of full power. Added to this it was expected that the elimination of a boiler would substantially reduce maintenance. Such a plant also offered the possibility of reducing the height of the machinery spaces and hence the height of the decks above them.

Two classes of ship have therefore been fitted with a combination of steam and gas turbines. In a *Tribal* class frigate one gas turbine is geared to the same shaft as a steam turbine and provides 37 per cent of the total power. In a *County* class ship two gas turbines are geared to each shaft with cross-compounded steam turbines and provide 50 per cent of full power.

The steam plant in these ships is a logical development from the *Whitby* class machinery, but it differs in several important respects as a result of further requirements.

In the *Tribal* class frigates the steam conditions are again 550 lb/sq in. and 850 degrees F. In the *County* class ships they are 700 lb/sq in. and 950 degrees F. at the boiler, the increases being to improve cycle efficiency. In both these new classes automatic and remote control has been used so that the machinery can be controlled from a separate control room.

The need for a separate control room arose principally from the need to provide protection for watchkeeping personnel against the results of nuclear explosions. It also has the advantage of providing better conditions for watch-keeping in the tropics where the compression of machinery into smaller spaces makes it increasingly difficult to keep down the temperature.

In the achievement of complete remote control, probably the most difficult single problem was to develop a satisfactory system of oil burning that would operate over a range from between 5 and 10 per cent boiler output to 100 per cent, using a constant number of oil burners.

BOILERS

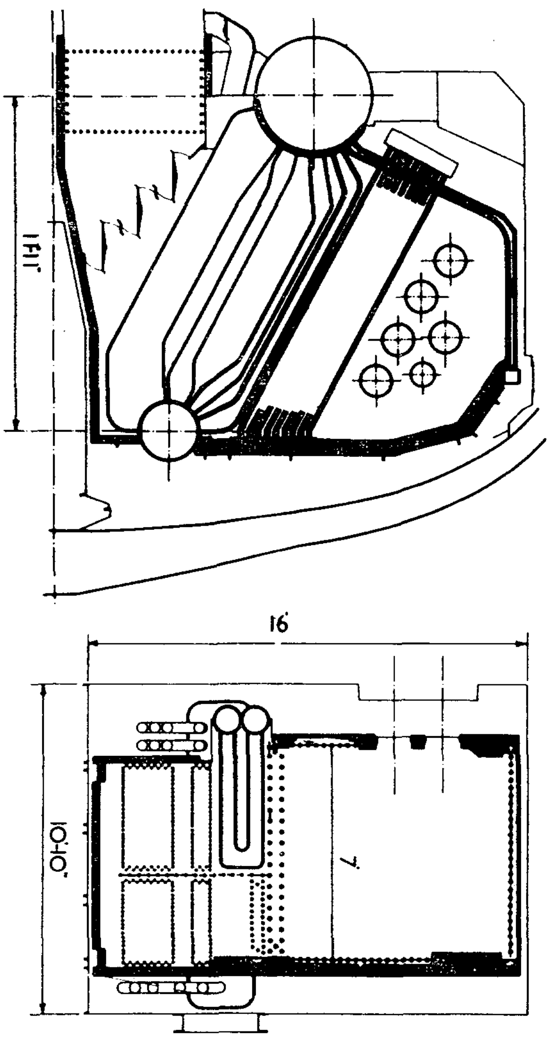

FIG. 1—BOILER OF 'WHITBY' CLASS FRIGATE

Because naval oil-burning requirements are rather specialized, development of combustion equipment has been carried out for many years at the Admiralty Fuel Experiment Station, Haslar, and naval boiler designs by industrial firms have usually been based upon the use of the equipment developed there.

The reduction of boiler size and weight clearly depends on the ability to reduce furnace sizes and, after the 1939-45 war, effort was concentrated on producing a high heat-release rate register with a short flame.

This necessitated a new basic approach to combustion equipment, since the old impingement type of register in which oil was bounced on the edge of brickwork quarl was incapable of being developed to the extent necessary. War-time experience had also shown that stablilizing a flame on a refractory edge brought with it deterioration of the refractory which became a serious maintenance problem.

The new combustion equipment used the gas-turbine technique of fixing a stabilizer to the burner, and resulted in what has become generally known as a 'suspended' flame.

The development of the new register enabled furnace heat-release rates to be more than doubled and to reach about 500,000 BTU/cu ft/hr, but this involved raising the oil pressure at the atomizer to about 600 lb/sq in., and a drop in air pressure through the register of about 18 in. water gauge at full output was necessary to produce reliable mixing at low outputs.

Higher rates were attained on test, but involved increased air and oil pressures and led to disproportionate penalties on blower size. A comppromise therefore had to be worked out to give the best over-all weights and efficiencies.

Boiler designs were developed to obtain minimum weight and space together with simplicity of control and resulted in the adoption of boilers with the following characteristics, shown in FIG. 1.

A two-drum boiler with water-cooled furnace; bricks confined to the floor and round the oil-burner registers.

The superheater tubes extend half the depth of the furnace from front to back. The gas path is divided into two by a transverse baffle wall running across the end of the superheater tubes. By controllable dampers the gases can be regulated between the superheater or saturated passes and the steam temperature controlled.

TABLE I—*Comparison of practice in boilers before and immediately after the 1939-45 war. (Performance for full power except where otherwise stated)*

Type	*Pre-war Admiralty 3-drum*	*Post-war*
Evaporation, lb/hr	170 000	147 000
Steam release rate per cubic foot of steam space, lb/hr	1380	2130
Pressure at drum, lb/ sq in.	400	550
Temperature at superheater outlet, °F	710	850
Number of burners	7 (18 in.)	5 full-size (12 in.) 1 half-size ($8\frac{1}{2}$ in.)
Heat release rate per cubic foot of furnace volume, BTU/hr	246 000	590 000
Heat release rate per square foot of projected radiant heating surface, BTU/hr	$1{\cdot}15 \times 10^6$	$1{\cdot}1 \times 10^6$*
Furnace length	10 ft $1\frac{5}{8}$ in.	6 ft 6 in.
Register pitch	2 ft $5\frac{3}{8}$ in.	1 ft 8 in.
Minimum clearance between burner centres and furnace boundary	1 ft 7 in.	2 ft 0 in.
Drop in air pressure across registers, in. w.g.	4	15
Drop in air pressure across tube bank, in. w.g.	1·5	13
Blower discharge pressure at full power, in w.g.	9·5	36
Casing design pressure, in. w.g.	N.A.	80
Steam drum diameter, in.	56	48
Tube O.D. gauge		
Generator	$1\frac{3}{4}$ in., 10 $1\frac{1}{8}$ in., 11 1 in., 12	2 in., 10 $1\frac{1}{2}$ in., 11 1 in., 12
Superheater	$1\frac{3}{8}$ in., 10	$1\frac{1}{4}$ in., 11
Downcomer	6 in. bore	$4\frac{1}{2}$ in. bore
Generator heating surface, sq ft	7688	3525
Superheater heating surface	1800	705
Economizer heating surface	7720 (plain)	2730 (studded)
Efficiency/percentage/maximum	88·75	86
at full power	85	82
Boiler height over-all	21 ft 7 in.	16 ft 9 in. †
Boiler width	18 ft 10 in.	17 ft †
Boiler length	13 ft 3 in.	12 ft 6 in. †
Boiler weight, dry, tons	72	43
Boiler weight	80	46·5
Materials and manufacture		
Steam drum	28/32T	34/38T C/Mn/Si steel welded
Water drum	28/32T	34/38T C/Mn/Si steel forged
Superheater header	28/32T	31/37T ½ per cent Mo steel
	mild steel S.D.	1. Cr/½ per cent Mo steel S.D. (Spec. D.G.S./6140 Type 2)
Other tubes	mild steel S.D.	Mild steel S.D. (Spec. D.G.S./6140 Type 1)

* Boilers have water walls.

† This includes the pressurized air casing, the pre-war design had no such casing.

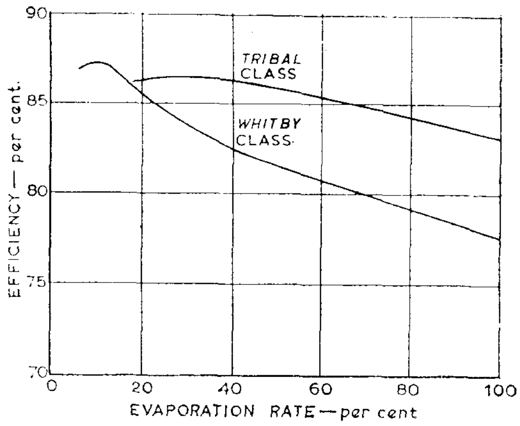

FIG. 2—BOILER EFFICIENCY CURVES FOR MODERN FRIGATES

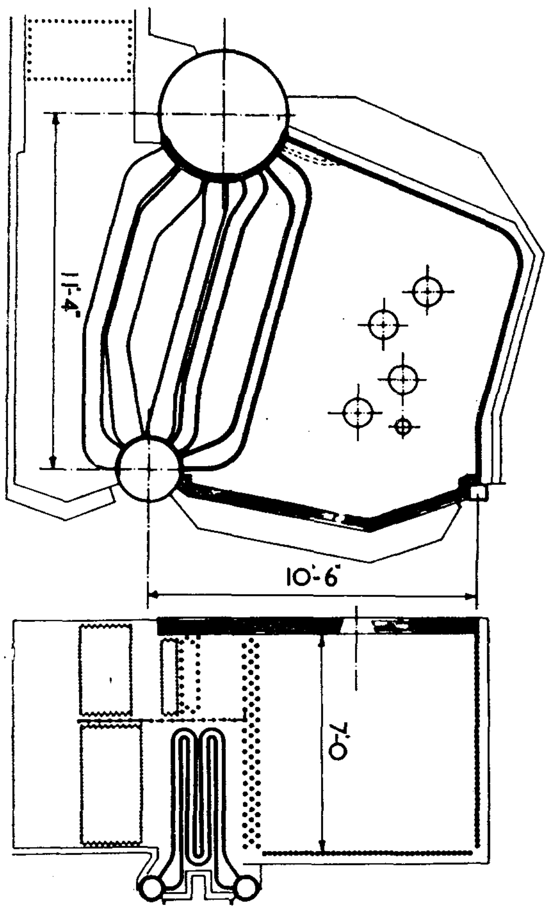

FIG. 3—BOILER OF 'TRIBAL' CLASS FRIGATE

Large-bore unheated downcomers between the drums carry all the downward water flow.

The generator tube and economizer surfaces are much reduced and extended-surface (studded) tubes are used in the economizer.

The need to ensure a good flow through the superheater at low outputs so that full temperature can be maintained makes it necessary to have several passes in the steam side of the superheater.

Tube pitching is such that high gas velocities are used to raise heat transfer coefficients and thus compensate for reduced surfaces.

Steam drums contain steam-separating equipment. Drum diameters are smaller as a result and low-alloy steel is used.

The high gas velocity through tubes, the high air velocities through the registers, and the loss through the air casings to make them compact, all conspire to increase the head of air required and blowers are required to deliver air at about 40 in. of water gauge pressure or more.

Air is trunked from the blower to air casings round the boiler. Originally these casings were made up of bolted stainless-steel panels and were somewhat difficult to keep properly air-tight. The use of welded casings has now solved this problem and the modern stainless-steel thin-gauge welded casings are very satisfactory. Bolted panels are provided for access.

Present naval boiler weights are about 0·7 lb per pound of steam generated and the volume is about 0·24 cu ft per pound of steam.

TABLE I shows the general characteristics of a pre-war Admiralty three-drum boiler, and the post-war *Whitby* class design.

Efficiency characteristics are shown in FIG. 2. After six years of experience at sea the designs have generally proved themselves a success. In particular, they are exceedingly flexible in operation. At the A.F.E.S., Haslar, a *Whitby* class boiler has been tested over a period of a year to establish the long-term effects of raising steam very rapidly from cold. It was lit up and full pressure raised within 20 minutes more than 50 times during the year. The normal rate of change of output is 5 per cent per second between 10 and 90 per cent of full output. This time is determined more by the inertia of blowers, fuel pumps,

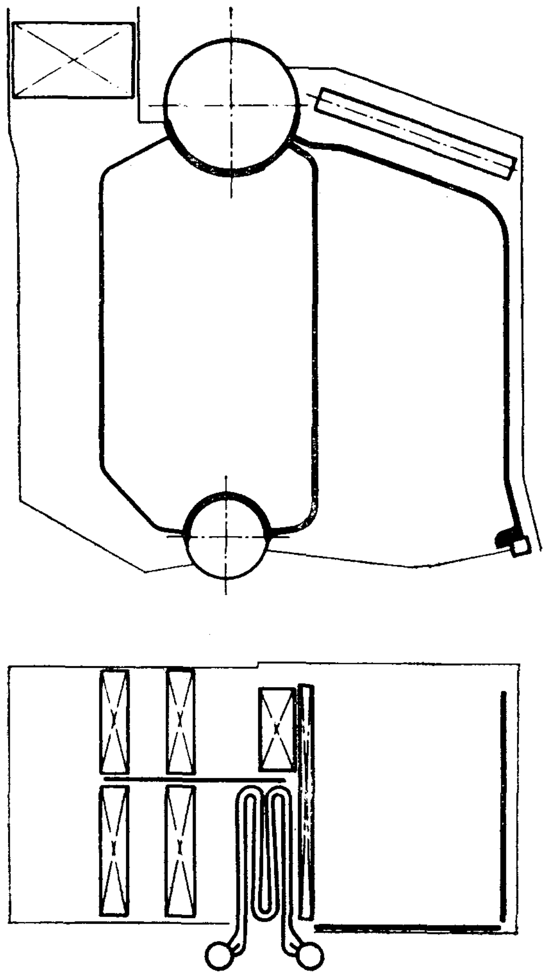

FIG. 4—'COUNTY' CLASS BOILER

etc., than the need to protect the boiler by a set rate of change of output. Careful readings of stresses and temperatures taken at points where trouble could be expected have nowhere reached limiting values.

The emphasis on reducing the space occupied originally resulted in difficult access for purposes of external cleaning and tube renewal. Later designs have therefore been modified to give better access, so that the build-up of deposits, particularly in the vicinity of the superheater tubes, can be more easily prevented (FIGS. 3 and 4).

The next step in the development of oil-burning systems was to provide a range of output of 20 to 1 on each burner so that automatic control could be adopted. To make sure this was simple and reliable it was decided that no register's part should have to move in order to maintain flame stability.

On the atomizer side this was achieved by departing from the plain pressure jet principle whose maximum practical range of output is about $2\frac{1}{2}$: 1.

For the very much higher 'turn down', atomizers were investigated with two fuel flow paths (duplex) and with return flow (spill). In both the aim was to maintain high flow velocity, giving good atomization, at low output rates. An atomizer with a fuel supply pressure of 300 lb/sq in., whose output was varied by a piston moving in a ported sleeve, has also been used for applications in older ships where existing low-head blowers and low-pressure fuel pumps had to be retained. The stable range of this last atomizer is about 4 to 1, that of the duplex 8 to 1, and that of the spill between 15 and 25 to 1, depending on various factors.

Since the spill system is the only one which approached the output range required, its development has been taken furthest. It has performed very satisfactorily, its main drawbacks being the high fuel pressure required (about 900 lb/sq in.), high air register pressure drop (about 18 in. water gauge at full output) and the need to control output on two pressures.

An increase in blower head is necessary to allow air velocities to be kept up even at minimum flow rates.

The first full-scale application of this system was in a prototype set of machinery of 30,000 s.h.p. which was set up at PAMETRADA for development testing. The boiler tests were somewhat curtailed by the incidence of boiler tube vibration of a novel and extremely damaging kind which appears to be a complex function of geometry and air flow. However, the test of combustion equipment was very successful and the many lessons learnt have been applied in the latest boiler designs now going to sea in *Tribal* Class frigates and *County* Class ships which have the same general configuration as that described above. (FIGS. 3 and 4.)

For the *County* class ships a boiler and its associated auxiliaries and controls were erected and tried out at the Admiralty Fuel Experiment Station. As a result it has been possible to establish response characteristics under automatic controls and the margins that exist under various operating conditions. Trials established that in fact the most arduous conditions could be produced while 'lighting up' and this has enabled improved operating instructions to be issued.

There is no doubt that at 950 degrees F. the margins on superheater tube temperatures are none too large under very rapid changes of output and with the ferritic steels at present in use.

For the time being, therefore, the gains available by increasing steam temperatures above 950 degrees F. do not appear to offset the difficulties they introduce in a naval application. The introduction of austenitic steels is not at present considered justified and they may not tolerate the rapid changes of temperature imposed on naval boilers. The need to be able to use a wide range of fuels without introducing corrosion or the build-up of deposits on the gas side of tubes is one which has to be taken very seriously. Work on boilers, particularly external cleaning, forms a very high proportion of the total machinery maintenance load in ships and the need to reduce this is one of the most serious requirements in steam plant at the present time.

MAIN TURBINES AND GEARING

The part of the steam installation which gives the least trouble to operators and maintainers is the main turbines.

The ship design requirement for lightness and small size is reinforced in this case by considerations of reliability. Probably the most difficult conditions met with in operating naval ships occur when it is necessary to be ready to go up to full power at any time almost immediately but the ship is actually stopped or in harbour. This is known as 'standing by' and is calculated to distort almost any large turbine casing. It was a frequent requirement in war-time, and was the cause of many anxious hours for engineer officers during the 1939–45 war. The adoption of double-reduction gearing has enabled turbine speeds to be increased to 6,000 rev/min. The adoption of solid forged rotors of low-alloy steel and modern blading techniques has made possible the use of high blade speeds and the reduction of turbine sizes and has thus given to turbines the ability to withstand the most severe operational requirements. The large drum type of rotor with reaction blading has also been discarded in favour of impulse blading on wheels machined out of the solid forging. Avoidance of distortion does, however, depend on keeping the length between bearings short and there is a fruitful field for controversy in each new design on whether to incorporate more stages to raise efficiency at low powers, on the need for 'stiff' rotors, and on whether to run above a critical speed.

With steam conditions of 500 lb/sq in. and 825 degrees F. at inlet it has been found possible to obtain a satisfactory consumption within a single-cylinder turbine down to about 10 per cent of full power. A large number of ships fitted with such turbines are at sea now and they have an irreproachable record of reliability. TABLE II and FIG. 6 give the general characteristics and dimensions, and here again the machinery is notable for its operational flexibility.

In order to provide very good efficiency at lower proportions of full power the original design included a cruising turbine which operated up to 30 per cent of full power and was then de-clutched.

To avoid any hand operation of clutches it was specified that engagement and disengagement of the clutch should be automatic. The need to provide

TABLE II

	Whitby *main turbine*	Tribal *Class steam turbine*
Steam inlet pressure, lb/sq in. . .	550	550/475
Steam inlet temperature, °F . . .	825	825
Shaft speed, rev/min	227–232	275–283
Gear ratio	25·2:1	18·4:1
Rotor material	3 per cent Cr–Mo steel	3 per cent Cr–Mo steel
Rotor diameter, stage 1	2 ft 8·27 in.	2 ft 8·437 in. 1st Curtis
P.C.D. blading, stage 1	2 ft 10·02 in.	2 ft 10 in. 1st Curtis
Diameter of last-row blading . . . tip	3 ft 4·032 in.	3 ft 11·4 in.
root	2 ft 3·32 in.	2 ft 3·75 in.
Number of stages	8	Curtis + 11
Diameter of astern blading, stage 2 tip	2 ft 5·906 in.	3 ft 4·958 in.
root	1 ft 11·49 in.	2 ft 6·5 in.
Maximum radial stress in rotor, ton/sq in.	16	16
Maximum radial stress in blading, ton/sq in.	14	12·7
Length between bearing centres . .	6 ft 3 in.	8 ft 8·242 in.
Diameter of journals	6 in.	8·986 in.
Length of bearings (white metal) . .	4·5 in.	5·125 in.
Nominal load on bearings, lb/sq in. .	85	138 (aft) 105 (for'd)
Vacuum (55°F cooling-water temperature).	23 inHg	27 inHg

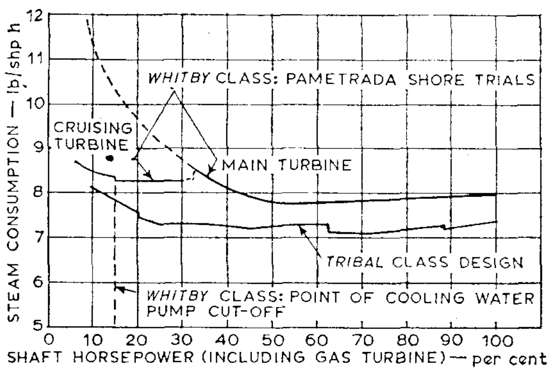

FIG. 5—STEAM CONSUMPTION CURVES FOR MAIN TURBINES OF MODERN FRIGATES

automatic engagement when the main turbine was slowing down and the cruising turbine was accelerating free up to a synchronous speed proved to be an extremely difficult one to meet. A single hand-wheel was also specified for operating both turbines so that the transition had to be achieved by cam-operated nozzle-control valves on both turbines, operated by one hand-wheel.

After very satisfactory operation of the clutch during shore tests on a brake, it was found that conditions at sea, which included the effect of the ship's inertia and a trailing propeller, were considerably more difficult, and much extra development work has been required to solve the problem. However, a satisfactory clutch has now been developed and the cruising turbine, running at 8,500 rev/min, transmitting through another reduction gear to the main double-reduction gears, can now be operated to give added efficiency in applications where this is needed. The effect of this on the steam-consumption curve for the main engine is shown in FIG. 5.

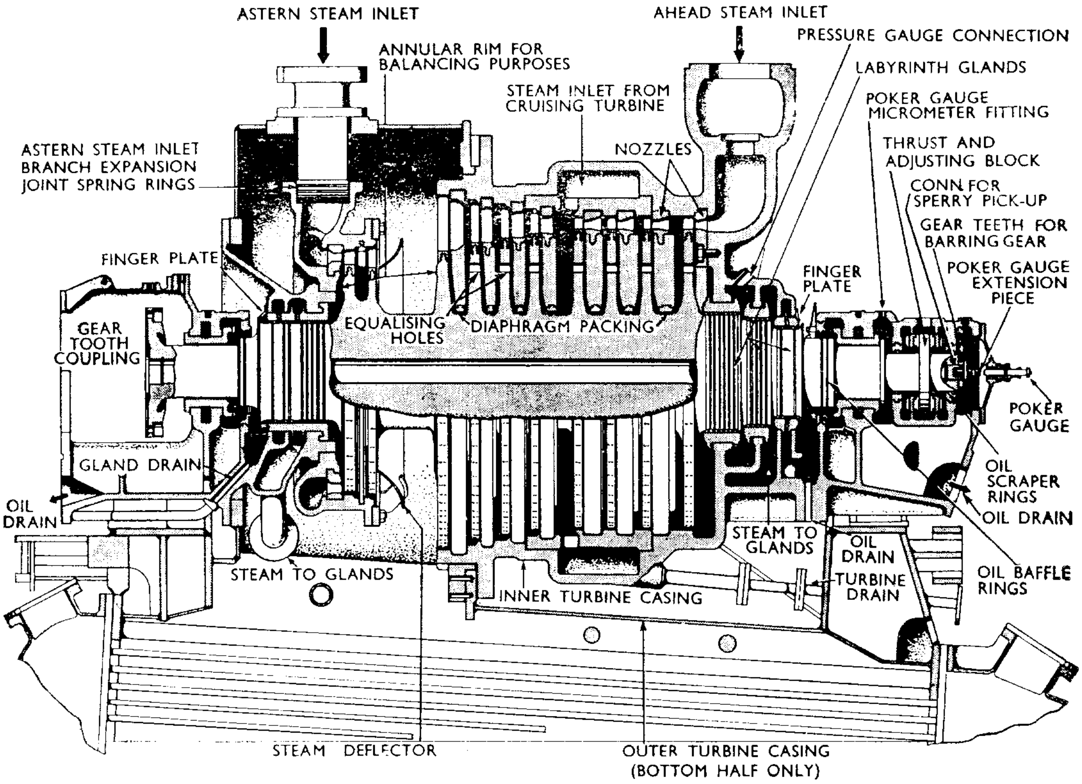

FIG. 6—MAIN TURBINE OF 'WHITBY' CLASS FRIGATE

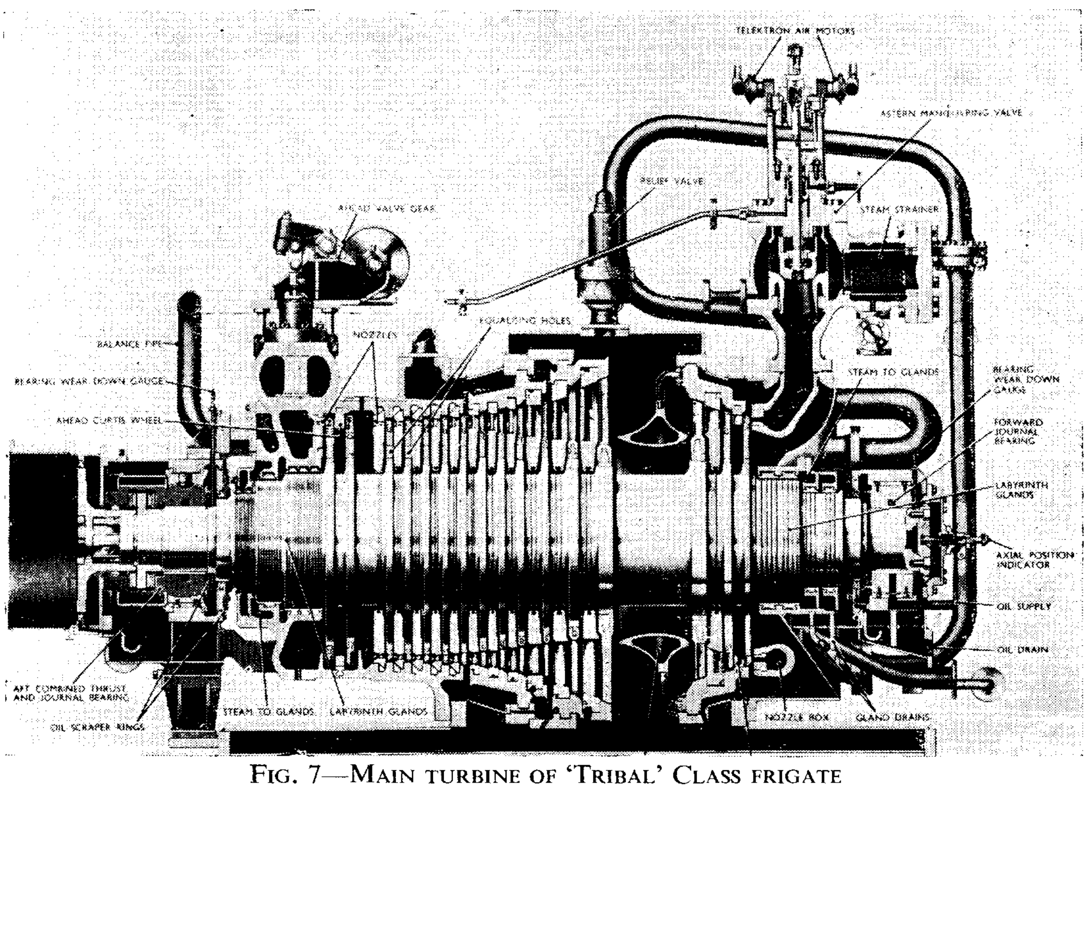

FIG. 7—MAIN TURBINE OF 'TRIBAL' CLASS FRIGATE

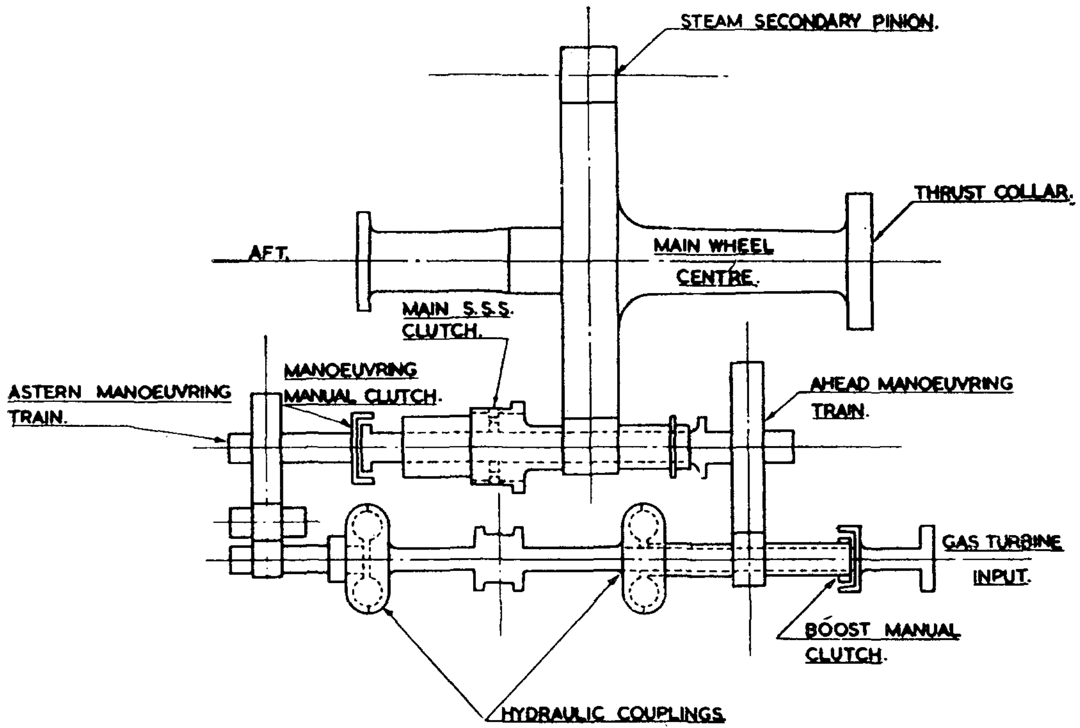

FIG. 8—GAS TURBINE MANOEUVRING GEAR TRAINS IN 'TRIBAL' CLASS FRIGATE

Reliable clutches were, of course, fundamental to the design of a set of gas turbines and steam turbines geared to a single shaft. The operating conditions for engaging the clutches are rather less severe than in the previous case of the cruising turbine so that the self-synchronizing clutch used finally for cruising turbines was adopted with confidence in its reliability. This had been amply justified by experience in extensive tests both ashore and in ships. A typical arrangement of clutches is shown in FIG. 8 as it applied to the gas-turbine manœuvring train in a *Tribal* class frigate.

With the single-cylinder steam turbine geared with one gas turbine the point of maximum efficiency of the steam turbine can be chosen to give very good low-power performance and this is shown in FIG. 5. The difference in the shapes of the two curves shown is also governed by changes of inlet steam conditions over the power range and their values are also affected by differences in gearing efficiencies and vacuum between the two classes.

With steam conditions (700 lb/sq in. and 950 degrees F.) chosen for the steam installation in the *County* class, where two gas turbines are used in combination with the steam turbines, two-cylinder cross-compound steam turbines were used to improve performance.

In order to get good low-power consumption the blading is designed to operate at almost optimum velocity ratio at 35 per cent of full propeller power. At over 50 per cent of total power this results in a slight reduction in efficiency as the turbine speed is increased by gas-turbine boost.

The selection of vacuum at full power is a balance between considerations of weight and space and of efficiency at low as well as at full powers.

In an all-steam plant the selection of a low vacuum for full turbine output will not have a very severe effect on the efficiency at low proportions of propeller power since the condenser vacuum goes up fairly steeply at first as the load is reduced. Where the full steam-turbine output corresponds to only 50 per cent of full propeller power, however, trying to save space by designing for a lower vacuum at full turbine output has a much more serious effect on efficiency at the same proportions of full propeller power. The last-row turbine blading is thus designed for higher vacuum in the boost than in the plain steam plant.

One further feature should be mentioned: the requirement for power to go astern. The need for efficiency is to a large extent ignored, the real criterion being to enable sufficient power to be extracted from the full boiler output of steam to get the required full-astern power without overheating the rest of the turbine or condenser, if this is required for sustained periods.

This can normally be achieved in a two-row Curtis wheel with suitable baffles to prevent direct impingement on the ahead exhaust blading. In the boost plant, however, where the boiler output only corresponds to a fraction of the total ship's power, two or more stages are necessary.

The need to reduce steam temperature when going astern may be a problem and this has been one of the reasons for retaining control of boiler steam temperature, rather than using a fixed temperature characteristic.

GEARING

The transmission of power produced by turbines revolving at high speeds to a propeller whose efficiency depends upon its revolving at low speeds, has been a subject on which much development work has been done under Admiralty auspices.

It having been accepted that double-reduction gearing was essential if turbines were to be made small, the need to reduce the size and weight of the gearing itself became of great importance if it was to match similar reductions in the rest of the machinery. This in turn depended on much improved standards of production for both the gears themselves and their alignment in the gearboxes. The Admiralty formed an association known as The Admiralty Vickers Gearing Research Association in which the makers and users of gear hobbing, shaving and grinding machines were represented with the object of improving methods of production.

The capacity for the grinding of large diameter gears has been limited and with the possibility of a large production programme in mind much attention has been given to raising the loading on through-hardened, hobbed, and shaved gears. Originally the hardest materials workable by these processes were chosen (En 26 pinions, En 30B wheels) and helix correction was applied by shaving to compensate for loading deflection of the teeth. A loading of $275K$* was used on the secondary gears. Experience with this combination of materials and processes on test caused the practice to be revised.

Trouble with scuffing of gear-tooth flanks was in fact cured by using lubricating oil with extreme-pressure additives. Gearing bearings were shortened to give length/diameter ratios of 0·37–0·7 with loading of 400–500 lb/sq in. and these have been entirely satisfactory on service.

Slight pitting has occurred on service in most of the sets of the original materials, particularly on the uncorrected portions of helices towards the gap.

Full-scale gear tests have proved that the materials are far less resistant to pitting than the ultimate tensile strength and hardness would lead one to expect. Investigations into pinions of different hardnesses have again demonstrated the importance of the combination, as such, of pinion and gear materials although the reasons are not yet fully understood.

---

*where K is the Hertzian stress $= \frac{p}{d}(r + 1/r)$

p = load per inch length of tooth
d = pinion diameter
r = ratio of diameters of wheel and pinion.

TABLE III

	Shaft horse-power	*Reduction ratio*	*Weight, tons*	*P.C.D. main gear wheels, in.*
Daring I	27 000	17·7	44	102
Y.E.A.D.I.	30 000	34·5	27½	83

Sets of hobbed and shaved gears now being fitted use a combination of En 26 pinions and En 9 wheels. The design uses a slightly lower *K* factor for both primary and secondary gears.

Thirty sets of a later design of gear using En 36 carburized, hardened, and ground secondary pinions have been fitted, the design remaining unchanged in other respects. These have been entirely satisfactory and have also withstood appreciable overloads due to misalignment as a result of inferior standards of manufacture and assembly.

The development of surface-hardened and ground gears has led to the fitting of these in the latest ships. The permissible tooth loadings are now over 50 per cent higher for carburized, nitrided, and induction-hardened gears, and at such loadings full-scale tests indicate that these gears have large factors of safety from failure by pitting or tooth breakage.

The use of hardened and ground gears and shorter bearings has not only reduced the size and weight of naval gears, without sacrifice of reliability, but has also permitted the use of larger reduction ratios. Comparative figures for one of the *Daring* Class hobbed and shaved gear designs fitted immediately after the 1939–45 war and a carburized, hardened, and ground design built in 1956 and tested ashore are given in TABLE 3.

MAIN CONDENSERS

Main condensers have a twofold influence on machinery spaces. Together with the main gearing they are an important factor in determining the length of the engine room, which must usually be kept to a minimum; their height, together with the head to be provided over the extraction-pump suction, the height of the main turbine plus the space for lifting its top half-casing, is often a critical factor in determining the height of the engine room.

A study of the possible ways of reducing size and weight of condensers in the *Whitby* class resulted in the selection of a vacuum at full power of 23 in. at tropical sea-water temperatures. In order to keep the weight to a minimum it was decided to design the last-row blades in the main turbine for these conditions without adding a large margin to take advantage of the higher vacuum which would obtain in temperate sea-water conditions. At the same time the speed of the circulating water which had previously been limited to 7 ft/sec was raised to 10 ft/sec at full power. Analysis of optimum weight and space resulted in the retention of the previous Admiralty practice of using tubes of $\frac{5}{8}$ in. outside diameter. With cupro-nickel tubes no trouble has been experienced in service. Aluminium-bronze tube plates are now fitted in all new designs and tubes are normally rolled into the inlet end tube plate to avoid setting up turbulence. Packing is fitted at the outlet end.

The effect of raising water speed through the condenser tubes to 10 ft/sec was to increase the resistance head of the circulating water. At 7 ft/sec a shaped nozzle inlet from the sea into the ship could be designed to give enough head to overcome the condenser resistance. At 10 ft/sec, however, a circulating pump had to be included in the system. The pump used was a turbo-driven, epicyclic-geared, axial-flow pump. The pump's highest duty is required when the engines are going full speed astern but with two or three controllable nozzle valves on the circulating pump turbine the combination of sea-water inlet head and pump head can be made to match the condenser circulating requirements very closely without imposing any substantial undercooling at low powers.

FEED SYSTEM

Of the various problems to the solution of which naval effort has been directed that of the feed system has so far yielded the least. In the *Daring* class, designed at the end of the 1939–45 war, a boiler pressure of 650 lb/sq in. was used for the first time in the Royal Navy and for the sake of simplicity no deaerator was fitted. There were many warnings about the danger of corrosion in economizers, but experience for a period of over 12 years has revealed no trouble.

In the next class of frigates a feed system as shown in FIG. 9 was fitted, including a direct-contact deaerator feedheater intended to work on a shunt principle. In its original form during shore trials of the installation its water level was extremely unstable. FIG. 9 shows the modifications made, and this system now works satisfactorily. The deaerator is of a very large size, however, dominating the engine room arrangement. Trouble was also experienced because of the high temperature of the water at feed-pump suctions if an extraction pump failure occurred. After modification the deaerator contained additional complications, with level controls added to automatic trip gear for excess level or loss of pressure. Later classes have therefore reverted to a full feed pressure heat exchanger to absorb auxiliary exhaust, instead of a deaerator.

With constantly changing ship speed and hence boiler output, however, it was thought that the surge tank (known as main feed tank) in the system, with vents to the atmosphere, must be a source of oxygen absorption into the feedwater although make-up feed was sprayed into the condenser. In the *Tribal* Class, therefore, an attempt was made to eliminate this by the feed system shown in FIG. 10. This shows a novel feed-tank arrangement and also a pressure feed-heater. The latest system, in the *County* Class (FIG. 11) where boiler pressure is up to 700 lb/sq in., includes a direct-contact deaerator feed heater designed to carry out full deaeration at 30 per cent of full power with a decreasing performance above this. This was prompted once more by the belief that the pressure might now have reached a stage where deaeration was required beyond that achieved in the main condenser.

From the corrosion aspect the results are negative so far, no corrosion attributable to oxygen during normal steaming having been experienced in any of these classes. On the rare occasions that economizers have suffered it the trouble has been traceable to the state of the boiler under shut-down conditions.

One returns then to reliability, reduction of maintenance, and minimum weight and space as the criteria. On these grounds the system in the *Tribal* Class frigates is probably to be preferred.

AUXILIARY MACHINERY

Past experience has shown electric supplies to be vulnerable to heavy shock or near-miss damage from enemy action and although it is hoped that the causes of this have been overcome the policy of being able to continue steaming without being dependent on electric power has continued.

In an attempt to make auxiliary turbines as small and efficient as possible these have been developed to use blade speeds of up to 800 ft/sec. Auxiliary turbine speeds of up to 17,000 rev/min are being used on geared auxiliaries and these give very compact and light units.

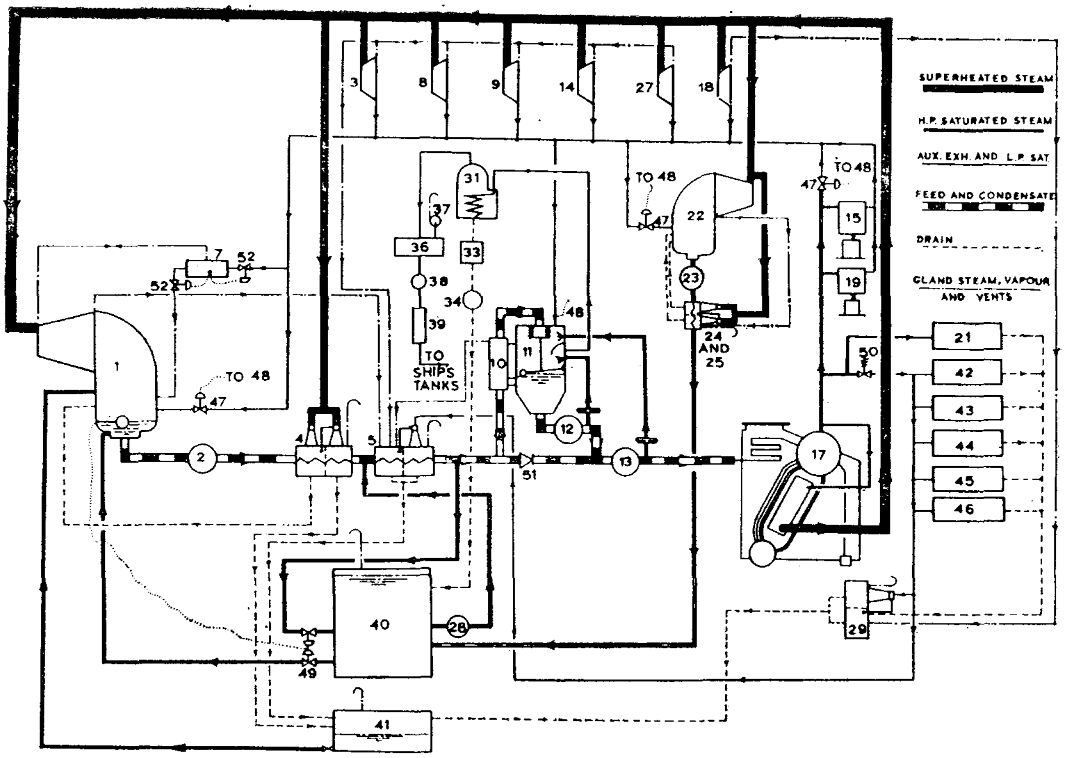

FIG. 9—FEED AND STEAM SYSTEMS OF 'WHITBY' CLASS FRIGATE

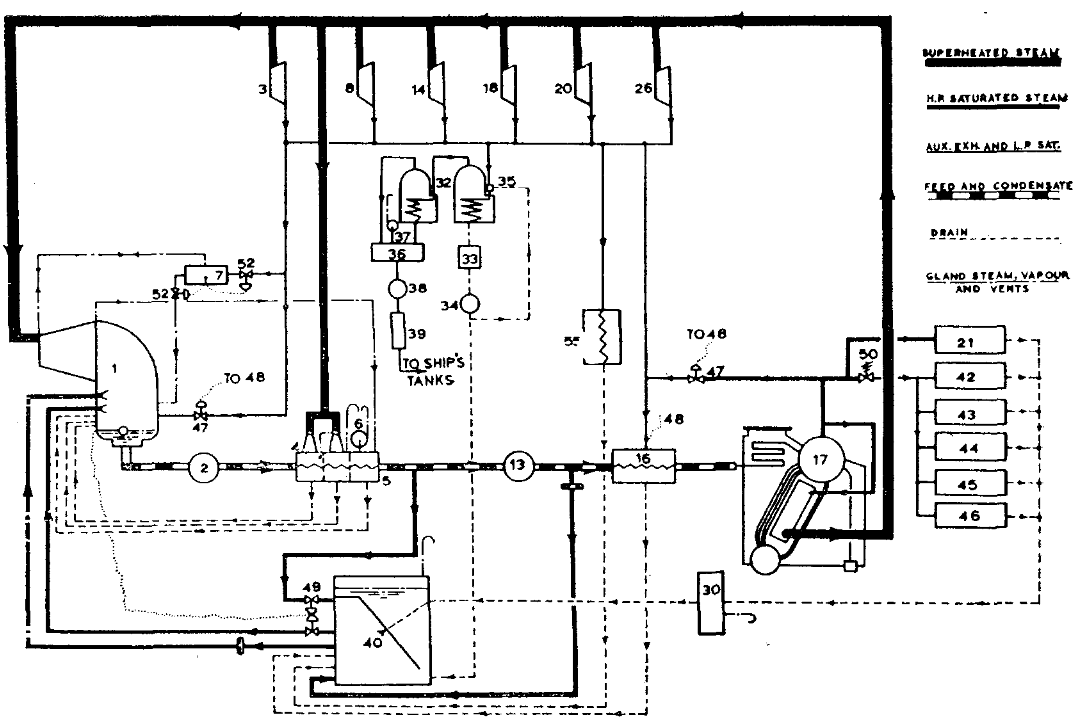

FIG. 10—FEED AND STEAM SYSTEMS OF 'TRIBAL' CLASS FRIGATE

The full steam conditions are used and an auxiliary exhaust system is maintained within 1 lb/sq in. of a pressure of 10 lb/sq in. by automatic valves which can either dump steam to the main condenser, or to the turbo-generator condenser when the ship is in harbour, or add steam from the boiler steam-drum.

The diagrams show how auxiliary exhaust steam can be used in various installations for distilling fresh water from the sea, steam-heating of the combustion air to the boilers, feed-heating downstream of the main feed pump or

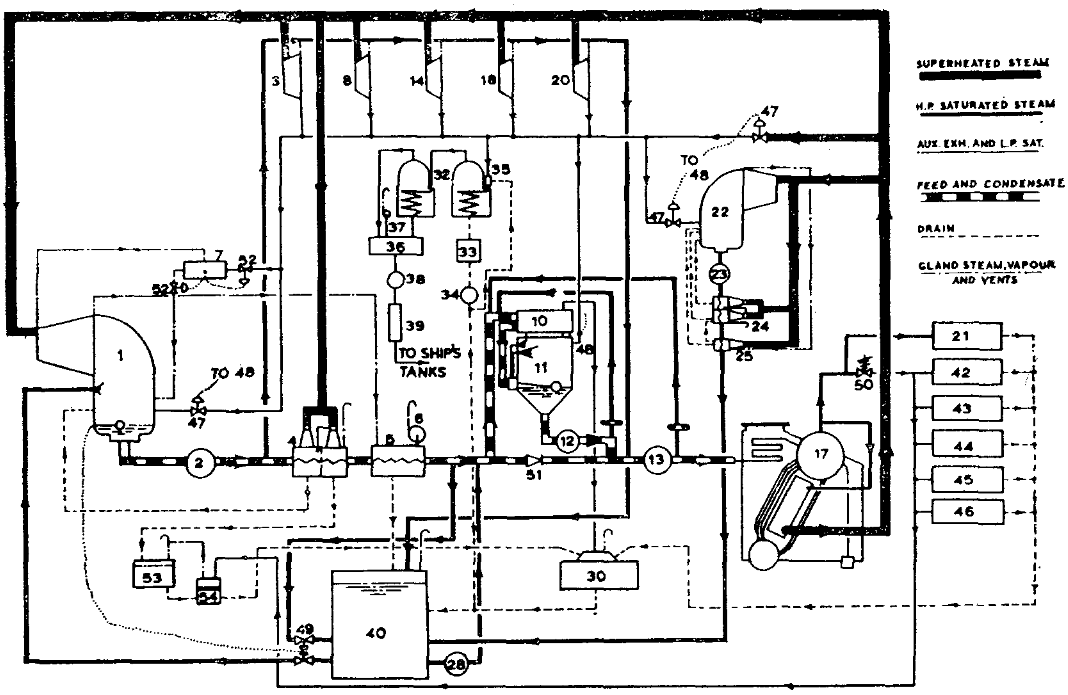

FIG. 11—'COUNTY' CLASS FEED AND STEAM SYSTEMS

1. MAIN ENGINES
2. MAIN EXTRACTION PUMP
3. MAIN EXTRACTION-PUMP TURBINE
4. MAIN AIR EJECTOR
5. MAIN GLAND VAPOUR CONDENSER
6. MAIN GLAND VAPOUR CONDENSER FAN (MOTOR DRIVEN)
7. MAIN GLAND STEAM COLLECTOR
8. MAIN CIRCULATING-PUMP TURBINE
9. FORCED-LUBRICATION-PUMP TURBINE
10. DEAERATOR VENT CONDENSER
11. DEAERATOR WITH AUTOMATIC LEVEL-CONTROL VALVE
12. DEAERATOR EXTRACTION PUMP (MOTOR DRIVEN)
13. FEED PUMP
14. FEED-PUMP TURBINE
15. DIRECT-ACTING AUXILIARY FEED PUMP
16. FEED HEATER
17. MAIN BOILER
18. FORCED-DRAUGHT BLOWER TURBINE
19. DIRECT-ACTING FUEL SERVICE PUMP
20. FUEL PUMPING UNIT TURBINE
21. FUEL HEATER
22. CONDENSING TURBO-ALTERNATOR
23. ALTERNATOR EXTRACTION PUMP (ENGINE DRIVEN)
24. ALTERNATOR AIR EJECTOR
25. ALTERNATOR GLAND VAPOUR CONDENSER
26. TURBINE FOR BACK-PRESSURE TURBO-ALTERNATOR
27. FIRE AND BILGE PUMP TURBINE
28. HARBOUR SERVICE DEAERATOR SUPPLY PUMP (MOTOR DRIVEN)
29. DRAIN COOLER AND GLAND VAPOUR CONDENSER (SEA-WATER COOLED)
30. DRAIN COOLER (SEA-WATER CIRCULATED)
31. SINGLE-EFFECT EVAPORATOR
32. DOUBLE-EFFECT EVAPORATOR
33. EVAPORATOR ELEMENT DRAIN COOLER (S.W.C.)
34. EVAPORATOR ELEMENT DRAIN PUMP (MOTOR DRIVEN)
35. EVAPORATOR STEAM SUPPLY DE-SUPERHEATER
36. DISTILLER (S.W.C.)
37. DISTILLER AIR PUMP (MOTOR DRIVEN)
38. FRESH WATER PUMP (MOTOR DRIVEN)
39. FRESH-WATER COOLER (S.W.C.)
40. MAIN FEED TANK
41. OVERFLOW FEED TANK
42. FUEL-TANK HEATING
43. LUBRICATING OIL SEPARATOR HEATER
44. GALLEY, LAUNDRY, CALORIFIERS
45. SHIP'S HEATING
46. WINTERIZATION
47. EXHAUST RANGE AUTOMATIC PRESSURE-CONTROL VALVES
48. EXHAUST RANGE PRESSURE-CONTROL POINT
49. MAIN CONDENSER AUTOMATIC WATER-LEVEL CONTROL VALVES
50. L.P. SATURATED STEAM REDUCING VALVE
51. DEAERATOR BY-PASS VALVE
52. GLAND STEAM AUTOMATIC PRESSURE-CONTROL VALVES
53. ATMOSPHERIC DRAIN TANK
54. PUMPING DRAIN TRAP
55. COMBUSTION AIR HEATER

in a contact deaerator and feed heater before the feed pump. Supplementing from the boiler or dumping to the condenser is of course inefficient and the aim is to adjust the exhaust to feed- and air-heating requirements for as wide a zone of the power range as possible.

PIPE ARRANGEMENTS

The problem of fitting the main steam system, auxiliary steam system, auxiliary exhaust, and ventilating system into the small machinery space is one which becomes more critical as the size of the main components of the installation and hence the size of the compartment is reduced. In order to allow maximum flexibility by use of small pipe diameters, steam speeds of up to 350 ft/sec are being used in main steam systems at full power and some sections of exhaust systems have near-critical velocities. Here again the balance between weight and space for machinery and fuel has to be considered as a whole. The use of computers to overcome what were hitherto impossibly laborious calculations now make possible the routine stressing of pipe systems with many anchor points and this is likely to ease pipe system problems profoundly. The present practice is to use piping of 1 per cent Cr–$\frac{1}{2}$ per cent Mo steel with joints between pipe lengths welded on site.

LAYOUT AND CONTROLS

The difficulty of fitting in all the systems required in a machinery space was further aggravated by the need to provide an enclosed control position where the watchkeepers would be protected.

As a result of experience gained from various conversions made in older ships, control can now be exercised from a remote position. In modern classes this is the normal method of operation.

Low-pressure air was chosen as the operating medium after an exhaustive examination of possible methods.

The combustion system devised to give each burner a very wide range so that the number of burners alight need not be altered has been briefly described above.

An automatic combustion control system, comprising a cascade of closed loops controlling combustion air, fuel spill, and supply pressures, was devised for the *Tribal* class frigates. The master quantity driving the cascade was steam-drum pressure, and facilities were provided for remote variation of the desired value of drum pressure and for the biasing of air pressure. With drum pressure set at normal value the output of the steam plant is completely controlled by a remotely controlled turbine throttle valve. The machinery and systems associated with the steam propulsion plant are all either self-regulating or remotely controlled. Those that are self-regulating include pneumatic automatic control of closed-exhaust system pressure and furnace fuel-oil temperature; those that are remotely controlled include superheat temperature and lubricating-oil temperature.

In the gas-turbine boost plant remote start-up and power control of the gas turbine, coupled with the arrangements described for control of the steam plant, give remote control for manœuvring.

Monitoring of the operating plant is based on the viewing of a sufficiently comprehensive range of indicating instruments in the control room, together with a manual scanning device for checking main-bearing thermocouples.

The installation described has operated very satisfactorily and the boiler control system will hold steam pressure with an accuracy of 15 lb/sq in. at 550 lb/sq in. during the severest manœuvre of the ship.

The installation in the *County* Class followed that evolved for the frigates, with the addition of pneumatic three-element control systems for boiler water-level, automatic control of steam temperature, and more refined control systems for furnace fuel-oil temperature, closed-exhaust pressure, and lubricating-oil temperature.

The instrumentation used is of current commercial type and was selected because of its ability to withstand warship conditions. The valve operators used include the diaphragm type and the ratchet air motor. There have been problems connected with the safety of the plant and in this respect the ratchet type has the advantage, for without additions, it offers 'fail set' and immediate manual take-over. With suitable additions, however, the diaphragm operator can also provide these facilities. A further safety problem, associated with all servo systems, has been that of failure of the air supply. Apart from automatic start-up of stand-by compressors, ultimate reliance is placed on air reservoirs. Recently one ship steamed under remote control (with emergency lighting) for more than 10 minutes after electric power had failed.

Experience at sea with remote-control systems has been gained so far primarily in cruisers and aircraft carriers. The consensus of sea opinion is that the controls can be relied upon implicitly in most circumstances, that remote control of certain functions can be more accurate than local control, and that the main problems relate to installation details and instrument-testing facilities.

CONCLUSION

It is hoped that, in spite of its many shortcomings, this paper will have provided some indication of the efforts made in the past 15 years to meet the stringent naval requirements for steam plant of small size, weight, and fuel consumption. The success achieved has been the result of very close co-operation between the Admiralty and industry, in which the firms concerned have been pressed to meet many new conditions of ship operation and have responded most effectively.

Often there has not been time between the decision to build a new class of ships and the actual building of them to test the new designs of steam machinery exhaustively and modifications have had to be made while manufacture was proceeding.

Largely because of this, perhaps, experience suggests that, although great attention has been paid to making the right decisions about the type of steam plant to be fitted in new classes of ships, the subsequent effort to ensure that the decision is justified by practical results has sometimes not been great enough. Ships' performances have been enormously increased and the present need in respect of steam plant is for increased attention to detail in design and manufacture so that the reliability of all components and ease of maintenance reach an equivalent standard.

The introduction of automatic and remote controls, successfully engineered, has meant a reduction in watchkeeping staff, but it has not led directly to a similar reduction in the total engineering staff because this is now governed by maintenance requirements rather than by the number of men needed to steam the ship.

It is probably only by the timely manufacture and exhaustive testing of prototypes, before production of a number of components is put in hand, that the details of designs can really be made fully reliable and the servicing and maintenance tasks reduced so that a real reduction in operating staff is possible.

The costs of modern ships can only be justified by very high usage of them. At the same time the technological race is an ever-present challenge. There is therefore no question of stopping development in order to concentrate on reducing the maintenance task. Improvement of performance and reduction of the work which has to be done to keep machines going have both to be pursued together.

This places great demands upon the resourcefulness as well as the diligence of people in a very large number of firms and organizations.

We have reason to be very grateful for the response made to these demands and it is with the full consciousness that this paper presents the results of very many people's work that the author acknowledges his debt to his colleagues in the Admiralty and in industry who have made possible both the achievements themselves and this brief record of some of them.

ACKNOWLEDGEMENTS

The help of my colleagues in the Design Sections of the Admiralty in the preparation of this paper is acknowledged with much gratitude.

Also the help and encouragement received from various firms, particularly The English Electric Company, Associated Electrical Industries, Messrs Babcock and Wilcox, and the Yarrow-Admiralty Research Department.