Recent Developments in British Naval Main Propulsion Steam Turbines

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The purpose of this paper is to show the considerable advances that have been made in British Naval main propulsion steam turbines since the end of the 1939/45 war. These advances include improvements in design and manufacture which have enabled higher steam conditions to be utilized, the overall result being improved performance which, together with weight and space reduction, results in increased endurance.

The initial impetus resulted from war time experiences and led very quickly to the three designs of *Daring* turbines, one of which is discussed in detail.

The next step was an investigation into all the factors affecting performance, endurance, etc., and so far as the main propulsion turbines are concerned, the various problems and resulting improvements are discussed in that section of the paper dealing with Y.E. 47A.

The Y.E.A.D. I turbine design which is described in detail is a further development of the *Daring III* design and is a logical step forward following the Y.E. 47A investigations.

The special requirements of the anti-submarine frigates required turbines which incorporate some rather unusual design features and these are described under the heading of Y.100, together with the reasons and shore and sea trials results.

INTRODUCTION

Security requirements have hitherto prevented the disclosure of the advances made in the design and construction of main propulsion steam turbines for naval purposes both during and since the Second World War, with the result that even the marine industry itself has little specific knowledge of these developments. It will be appreciated, however, that in a paper of this nature certain information, such as that relating to ships' performance, speed, etc., is still subject to security restrictions.

During the wars in the Pacific where for tactical and strategic reasons it was necessary for British Naval craft to operate with comparable United States vessels, it became obvious that the steaming performance of the Royal Navy compared unfavourably with their American counterparts. This was accounted for partly by the lower steam conditions and partly by lack of development in aspects affecting turbine efficiency between the wars which had not kept pace with land steam practice. This state of affairs was caused no doubt by the shortage of financial backing for development work generally in the marine industry.

Investigations were immediately instituted by the Royal Navy, and a Propulsion Committee, consisting of representatives of the warship building firms and the Department of the Engineer-in-Chief of the Fleet, was set up with suitable terms of reference. The several important differences in design and the basis of design between American and British ships quickly emerged.

As a result of its early deliberations, the Propulsion Committee decided to set up a number of special sub-committees, the one relevant to this paper being the Turbine Sub-Committee, under the chairmanship of Cdr.(E) I. G. Maclean (now Rear-Admiral I. G. Maclean, C.B., O.B.E.).

The purpose of the Turbine Sub-Committee was to co-ordinate marine and land turbine design techniques and make recommendations concerning future turbine designs for Naval purposes.

The first meeting of the Turbine Sub-Committee took place in London on the 1st December 1943, when representatives of the following organizations were present:—

The Admiralty.
John Brown and Co. (Clydebank), Ltd.
The Parsons Marine Steam Turbine Company.
C. A. Parsons and Co., Ltd.
The English Electric Co., Ltd.

Representatives of Metropolitan-Vickers Electrical Co., Ltd., and the British Thomson-Houston Co., Ltd., attended subsequent meetings.

In the interest of brevity, the detailed work of this Committee cannot be described but it resulted in the submission of five designs for turbines of 27,000 s.h.p. to meet an outline specification laid down by the Chairman on behalf of the Engineer-in-Chief's Department. These designs were openly tabled and adjudicated upon by the full Sub-Committee which unanimously decided that of the five designs the two submitted by the English Electric Company best met the Admiralty requirements.

The first of these designs, referred to as S.T.N.1, was a two-cylinder reaction design preceded by a two-row Curtis wheel based broadly on a design of which the U.S. Navy had supplied details.

The second design, S.T.N.2, was of impulse construction making use of alloy steel rotors and originally referred to as a "limit" design, the total weight being some 30 per cent lower than that of any other design submitted.

The *Daring* Class of destroyers was at this time (1944)

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under construction and the Admiralty decided to develop both S.T.N.1 and S.T.N.2 for installation in these ships after full shore tests carried out on prototype sets of machinery.

In order to speed development, S.T.N.1, with the English Electric Company's agreement, was entrusted jointly to Pametrada and C. A. Parsons and Co., Ltd., whilst the impulse design, S.T.N.2, was entrusted to the English Electric Company.

It will be remembered that at this time Pametrada was a new organization formed by the principal shipbuilders as a design, development, research and test station for modern machinery, capable of testing at full load sets of machinery up to 60,000 h.p. Lack of such facilities in this country had been a severe handicap to the Navy.

Whilst the above developments were proceeding, a third alternative design was introduced consisting of a British Thomson-Houston designed h.p. cylinder and a Pametrada reaction l.p. cylinder. This was sponsored by John Brown and Company, eventually constructed by them and also shore tested at Pametrada following the tests on S.T.N.1.

The three designs, S.T.N.1, S.T.N.2, and the B.T.-H./ Pametrada, as a result of the association with the *Daring* Class, were referred to as:—

Daring I reaction design, first proposed by the English Electric Company and developed by Pametrada and C. A. Parsons.

Daring II impulse reaction design H.P. cylinder by B.T.-H., L.P. cylinder as on *Daring I*.

Daring III advanced all-impulse type designed and developed by the English Electric Company.

The first section of the paper will be devoted to the *Daring III* design.

Daring CLASS TURBINES

The *Daring* Class twin-screw ships have main engines designed to meet the Admiralty requirements issued in 1944. Each set of turbines supplies 27,000 s.h.p. + 10 per cent overload with steam at 565lb. per sq. in. absolute and 825 deg. F. The steam pressure varies with the load, rising to 655lb. per sq. in. absolute at 10 per cent power. The steam temperature remains constant. The vacuum at full power is 27·5-in. Hg. (Bar. 30·0in.). An astern power of 9,000 s.h.p. is specified when supplied with steam at 515lb. per sq. in. absolute, 825 deg. F.

The turbines were to be designed to attain a high efficiency at all powers with special emphasis on the range between 20 per cent and 40 per cent full power. During the life of the ship full power would not be required for more than 2,000 hours. The Admiralty stressed that interchangeability was particularly important, both for individual items and complete assemblies, irrespective of several manufacturers being involved.

Daring III

General Arrangement

Preliminary investigations showed that the best turbine arrangement from the point of view of weight, space, and performance, etc., was a two-cylinder cross-compound arrangement with the power split more or less equally between cylinders at full load to ease the design of the double reduction locked train gear case. This arrangement (Fig. 1) was consequently adopted.

H.P. Cylinder

In 1944 current practice in the Royal Navy was to install impulse-reaction h.p. turbines and all-reaction l.p. turbines, although some Brown-Curtis all-impulse turbines were still in service. This latter type was very much out of favour because of the serious troubles constantly arising from blade vibration, wheel flap and wheel loosening. It was felt, however, by the English Electric Company that for the requirements as stated above, an all-impulse design (Fig. 2) would prove advantageous, perhaps the major attraction being that the larger axial clearances permissible with impulse blading would considerably ease the problems of differential expansion between fixed and moving parts. In the case of the impulse h.p.

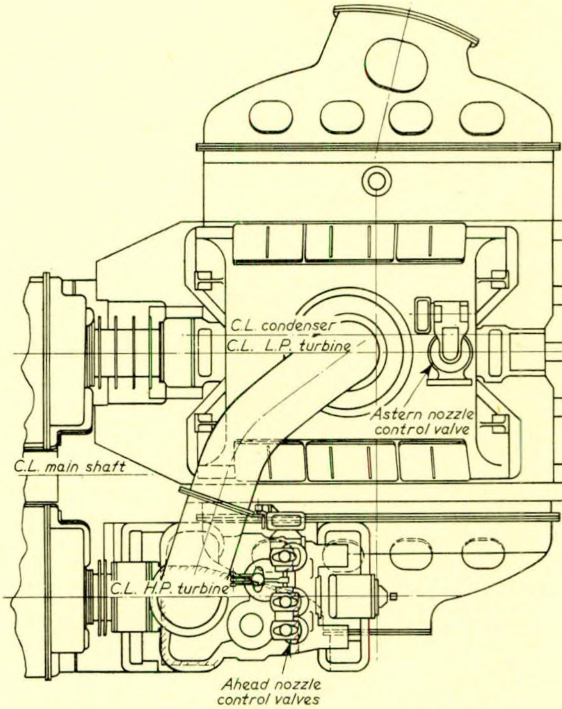


FIG. 1—*Turbine arrangement,* Daring III

cylinder, these clearances were at least five times greater than for the *Daring I* reaction turbine. In addition, the rates of heating up and cooling down of a solid gashed type rotor approached much nearer that of the cylinder than did a drum type; this eased problems of differential expansion and eliminated wheel loosening. These points were of major importance in the case of naval machinery where speed of manœuvre and consequent rapid power change were essential. The wisdom of this decision was adequately proved during shore trials.

Steam Chest

It was decided to use a steam chest cast integrally with the h.p. cylinder. This removed the complicated steam piping resulting from separate nozzle boxes and so improved habitability of the engine room.

Valve Gear

The valve gear fitted to all the turbines installed in the *Daring* Class was manufactured by J. Blakeborough and Sons, Ltd., to a design approved by the Admiralty. The individual valves were cam operated when opening and spring closing. The cams, of course, were designed by each turbine manufacturer to suit his own requirements.

In the interest of economy over the required power range, nozzle control valves were provided for loads of 20 per cent, 40 per cent and 60 per cent, each valve supplying steam to its individual group of nozzles. Above 60 per cent power a bypass valve admitted steam to a belt situated between stages 3 and 4 in the h.p. turbine. All valves opened sequentially. The astern valve was mounted on the l.p. cylinder.

Two handwheels, one for ahead and the other for astern operation, were connected to the valve gear operating mechanism by means of bevel gears and shafting. No other control was required, thus operating duties were kept to a minimum and the correct number of nozzle groups for the power required always obtained.

L.P. Cylinder

A double-flow l.p. cylinder (Fig. 3), again incorporating a solid gashed type rotor, was decided upon, as this had numerous advantages over a single flow design when dealing with powers of the order required. Blade heights were halved,

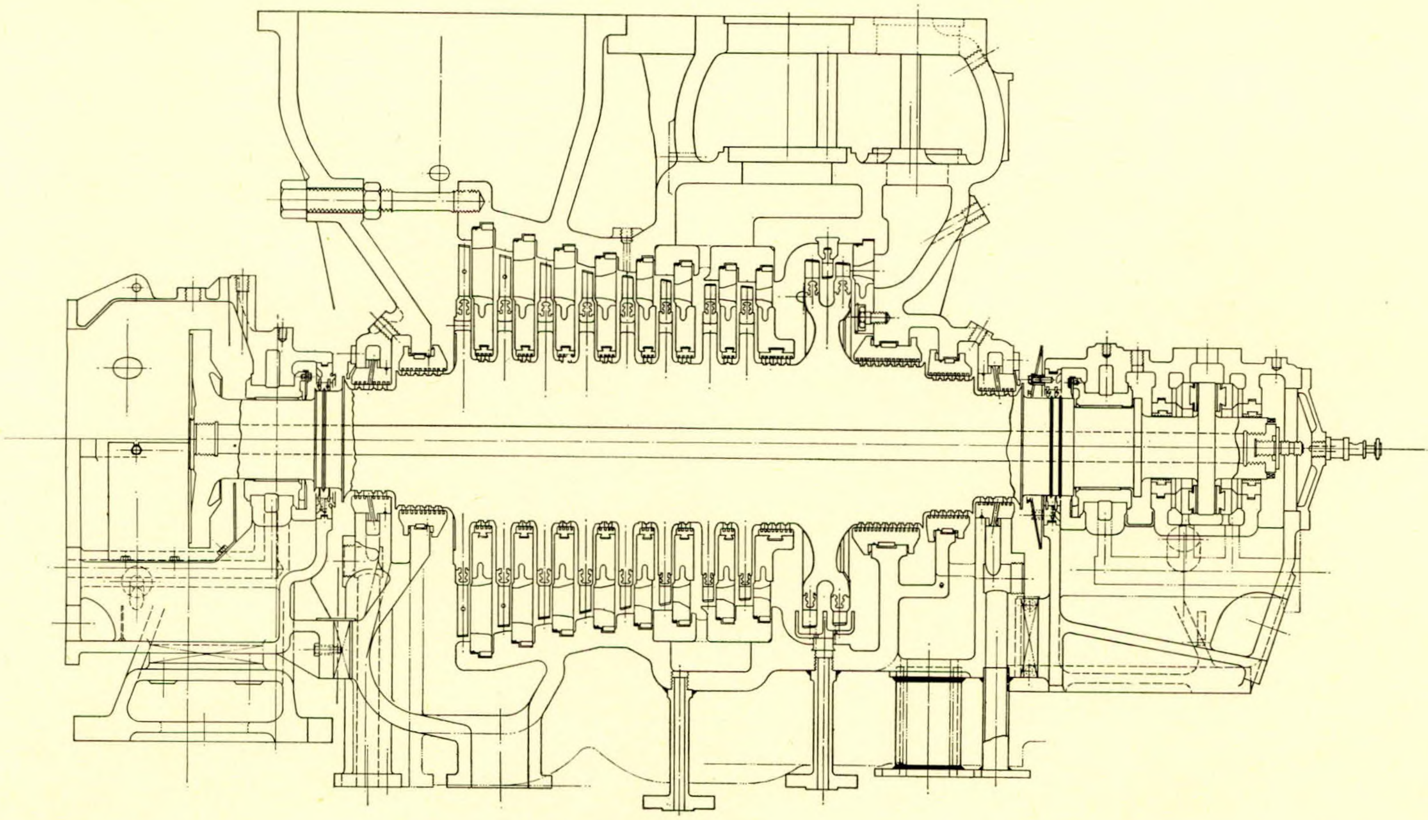


FIG. 2—*H.P. longitudinal section,* Daring III

thus reducing very considerably the overall diameter. The shorter blades and smaller diameter rotor enabled the design to be based upon higher rotational speeds without any increase in stress levels; in addition, it permitted an astern Curtis wheel to be fitted at each end of the cylinder. This minimized rotor and casing distortion in comparison with single-flow construction and so favoured rapid manœuvring.

Materials

Rotors

The abandonment of built-up rotors and the use instead of solid forged alloy steel rotors had permitted the employment of higher design stresses in the American designs. Although there was considerable controversy about the wisdom of using alloy steel for marine turbine rotors, the Admiralty, in collaboration with the forgemasters and the Turbine Sub-Committee, had put in hand an extensive programme of trial forgings for sectioning and full analysis, as a result of which a 3 per cent chrome molybdenum steel was chosen for the rotors to an Admiralty specification now known as No. 3F.1.

Considerable difficulty was experienced by some steel makers in producing sound forgings in this material. But

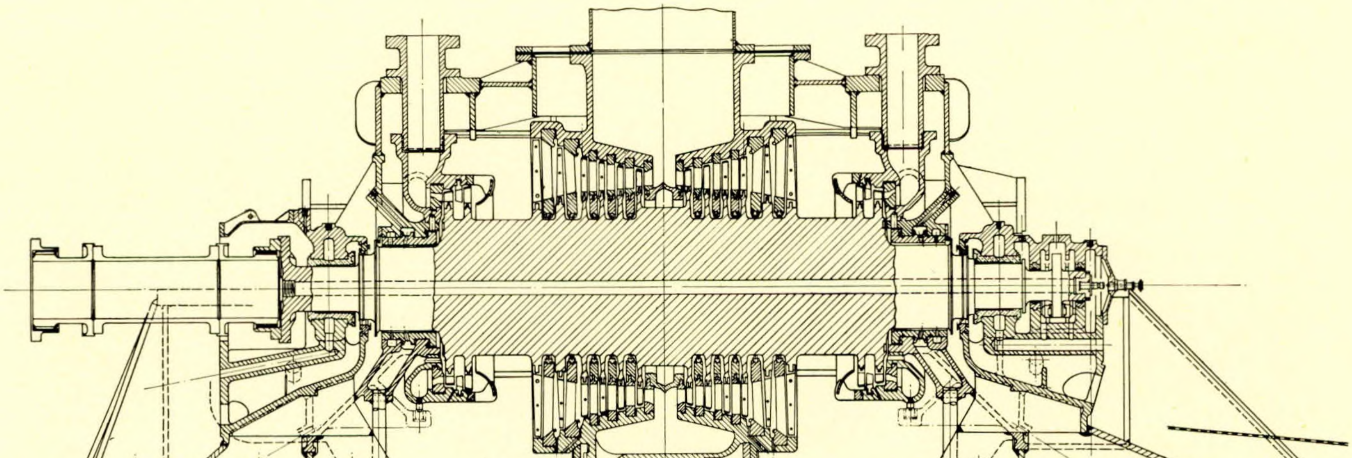


FIG. 3—*L.P. longitudinal section,* Daring III

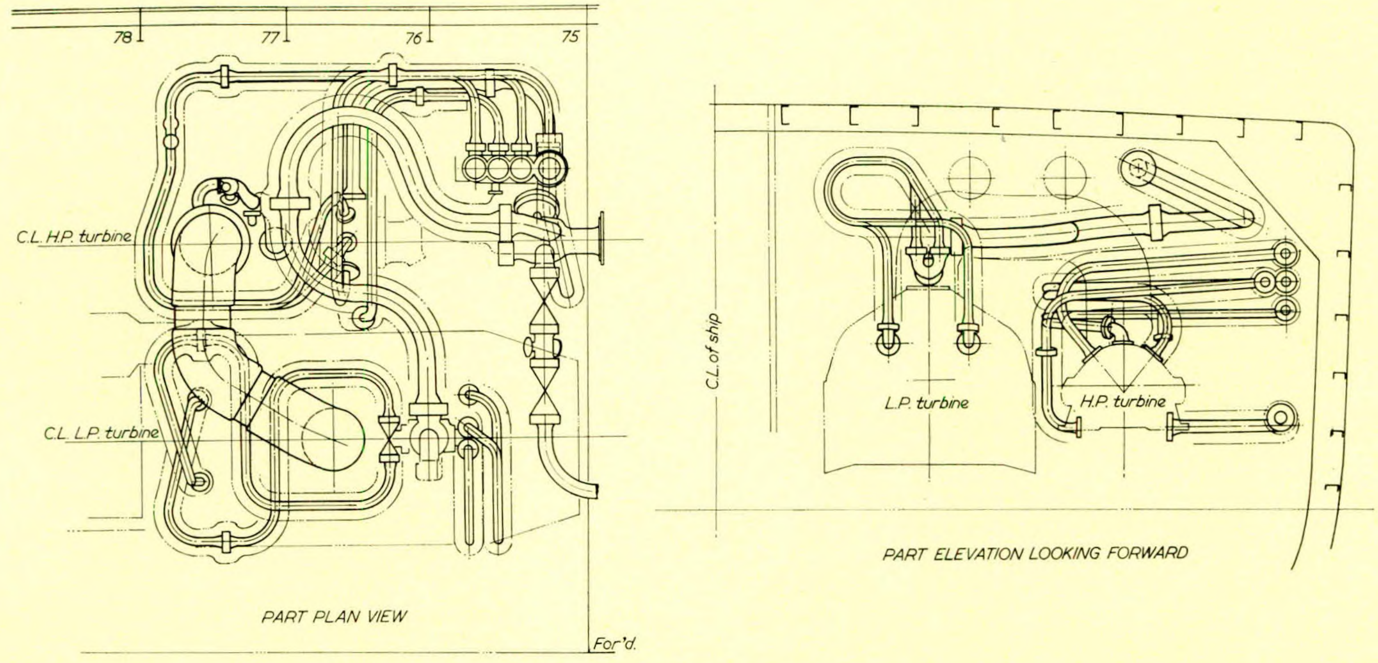


FIG. 4—*Steam chest and piping,* Daring I

forging technique has now reached a level where this material is accepted as reliable by both land and marine turbine industries.

It was only by the adoption of this material that the high rotational speeds of the *Daring* machinery were made possible.

Blading

Chloride attack had been a frequent cause of trouble in naval turbines during the war; the Admiralty was concerned to fit blading resistant to this form of attack. Three choices of material presented themselves: Monel metal, Hecla A.T.V. steel, and stainless iron.

Monel metal was avoided due to its high specific weight, adding appreciably to the blade and wheel stresses. Stainless iron, of course, was little more resistant to chloride attack than mild steel and necessitated particular attention to feed water purity if it were to be used in marine turbines, both in ordinary operation and under action conditions, especially if damage occurs to the machinery.

A.T.V. steel was eventually chosen for *Daring III* with the exception of the last two l.p. rows of blading, which were made of a special manganese-nickel steel and chromium plated in line with the maker's established land practice.

Nozzles

All the fixed blading took the form of nozzles, and here, with the exception of the last two stages, a built-up construction was employed consisting of A.T.V. steel nozzle segments

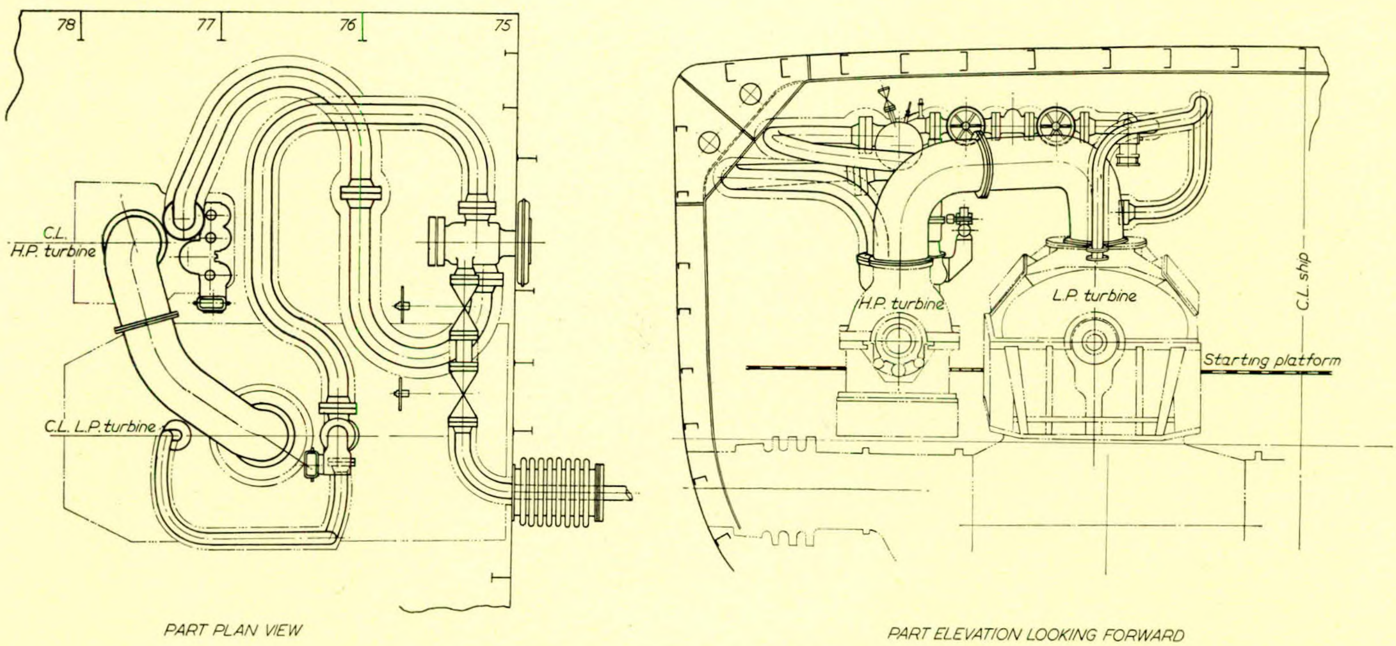


FIG. 5—*Steam chest and piping,* Daring III

fixed to mild steel centres. The last two l.p. diaphragms were made of cast steel with A.T.V. steel nozzle division plates cast in.

Other Materials

Shock conditions due to underwater explosion precluded the use of cast iron on Naval vessels and restricted the materials for casings, pedestals, etc., to cast steel or fabricated mild steel.

Weight and Space

Due to the increasing demand on modern Naval ships for more and more space for armaments and other fighting equipment there is greater need for lighter and more compact machinery. Reduced space and weight of machinery means that the ship can carry more fuel, and thus improve its radius of action.

The advantages of the integral steam chest can be seen from Figs. 4 and 5, while the following table throws additional light on the topic of weight and space.

TABLE I.

	Daring III	*Daring II*	*Daring I*
2 h.p. turbines, including valves, pipes, etc., tons	21·2	45·3	38·0
2 l.p. turbines, including valves, pipes, etc., tons	37·8	44·1	44·1
Total	59·0	89·4	82·1
H.P. rotor bearing centres	4ft. 9¼in.	5ft. 10¼in.	6ft. 7⅜in.
L.P. rotor bearing centres	7ft. 10in.	8ft. 6⅝in.	8ft. 6⅝in.

Interchangeability

This could only be achieved by the suitable tolerancing of all dimensions, involving in many cases considerably tighter tolerances than those previously worked to for marine purposes. That interchangeability to the degree required was achieved has been proved many times. A notable example was provided when a complete h.p. rotor made by one manufacturer was fitted in a finished casing made by another, no hand fitting or adjustment being required.

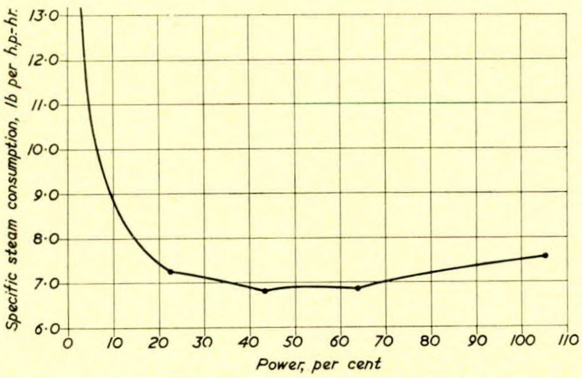


FIG. 6—*Consumption rate,* Daring III

Brief Summary of Shore Trials

Fig. 6 shows the steam consumption rate including gear losses for *Daring III* turbines over the power range based on the shore trial results. There was but little difference between the specific steam consumptions for *Daring I, II* and *III* as no design proved superior to any other over the whole power range. It can be seen how closely the specified requirements have been achieved, the curve being practically flat from 20 per cent to 100 per cent power, the lowest point being at about 40 per cent power.

All three designs of machinery proved satisfactory in their ability to manœuvre, particularly so for the all-impulse design h.p. cylinder of *Daring II* and *III.*

Resulting from the full-scale shore trials, it was found that all three designs could be improved upon in one way or another, as was to be expected when testing machinery of such an advanced character for the first time. In nearly all cases the modifications were of a comparatively minor nature.

In the case of *Daring III* vibration trouble was revealed upon opening up the l.p. cylinder after the trials. This concerned the lacing wires and blading in the later ahead stages and was overcome by providing a more efficient type of steam deflector between the ahead and astern blading. The modification consisted in substituting a deflector bolted to the casing in place of the one designed integral with the rotor.

Examination of the blading in the h.p. cylinder revealed failures in one or two segments of the Hecla A.T.V. lacing wires where brazed to the blade. Resulting investigations revealed that Hecla A.T.V. steel was subject to intergranular penetration by copper during brazing operations, but was otherwise quite suitable if brazing was avoided. This difficulty was overcome by replacing all laced rows of blading and substituting high tensile stainless iron for both blades and lacing wires.

It was also established during the trials that the measured steam flow was greater than the first-stage nozzle area would theoretically permit; steam tests carried out at Rugby Works revealed that there was steam leakage between the built-up nozzle segments. This permitted steam to enter the casing *via* the horizontal joint of the nozzle plate. This trouble has been overcome by substituting a solid type of nozzle plate in which the segments are integral with the centre and machined from the solid.

Conclusions

The *Daring III* turbines produced an economy combined with weight and space figures superior to any other main propulsion unit for Naval purposes previously built in this country. H.M.S. *Decoy* and *Diana,* fitted with this design, have both been in service for some three or four years and have given every satisfaction to date. H.M.S. *Decoy* is installed with the prototype port set of turbines built by the English Electric Company; the starboard set and both sets in H.M.S. *Diana* were built by Yarrow and Co., Ltd., under licence from the English Electric Company.

Turbines of the *Daring III* design are also being manufactured in Australia at Cockatoo Dockyard for installation in the Australian-built *Daring* Class ships, the first of which successfully completed sea trials last summer.

Table II summarizes the salient features of the *Battle* Class, *Weapons* Class and the *Daring III* propulsion machinery for purposes of comparison:—

TABLE II.

	Daring III	Weapon Class	Battle Class
Full power s.h.p. per set	27,000	20,000	25,000
Steam conditions at full power, lb. per sq. in.	550	350	350
deg. F.	825	750	650
Steam consumption at 20 per cent power, lb. per s.h.p. hr.	7·25	8·8	9·8
Steam consumption at 100 per cent power, lb. per s.h.p. hr.	7·52	7·7	8·4
Specific weight of turbine, lb. per s.h.p.	2·45	4·74	4·83
H.P. rotor, bearing centres	4ft. 9¼in.	9ft. 11in.	9ft. 4¼in.
L.P. rotor, bearing centres	7ft. 10in.	11ft. 8⅞in.	12ft. 6¼in.

Y.E. 47A

In 1947, following the advances that were made in the design of the machinery for the *Daring* Class, the Admiralty invited the English Electric Company to co-operate with Yarrow and Co., Ltd., in order to investigate the further gains that could be achieved by elevation of steam conditions and to determine the characteristics of the machinery which should be developed to utilize these conditions in the most efficient manner. The various reports produced and recommendations made covered all aspects of the whole of the machinery suitable for a Destroyer of *Daring* hull form.

The investigation assumed the title of Y.E. 47A, and, so far as the main propulsion machinery was concerned, proceeded concurrently with the final design and manufacture of *Daring III.*

Specific Requirements

The requirements were to establish the most advantageous arrangement, design and steam conditions for a set of main engines to develop a maximum power of 30,000 s.h.p., due emphasis being given to the following factors:—

(a) Reliability.
(b) Minimum weight and space.
(c) Maximum economy between 20 per cent and 40 per cent of full power.
(d) Ease of maintenance.
(e) Ease of production.
(f) Interchangeability.
(g) Operation by personnel with limited training.

Performance

Due to the fact that turbines for Naval purposes have to operate over a very wide speed range, it is essential that the design be such that the steam consumption at cruising powers does not unduly prejudice the consumption at full power as the steam rate of the latter determines the boiler capacity. On the other hand, if the turbines be designed to give a maximum efficiency at full power, the efficiency at cruising speeds is adversely affected, which seriously reduces the ship's endurance at those speeds.

Steam Conditions

It is well established that, within wide limits and for a fixed back pressure, the higher the initial steam pressure and temperature the lower will be the fuel rate of the cycle, provided that the efficiency of the various components of the cycle can be reasonably maintained.

In the case of a Naval vessel, however, fuel rate is not the only criterion. The space occupied by machinery and its overall weight are all equally important. A useful standard adopted during this investigation was made by fixing a maximum permissible weight of machinery plus fuel, and comparing the ship's radius of action with the various designs of machinery to be described below.

Early in the investigation it became clear that a very definite limit would be set to any increase of initial pressure and temperature by the increased weight of machinery necessary to withstand it. The predominant component to this limitation was the boiler plant and main steam piping.

It will also be appreciated that such a yardstick depends upon the operational duties for which the ship is intended and may vary widely from one class of ship to another.

Cruising Turbines

The basic law of ship propulsion (Fig. 7) is roughly cubic, i.e., power required varies as speed^3. (An actual power/speed curve used during the design of a set of Naval propulsion machinery is shown in Fig. 7.)

Most warships, however, only require to develop full power for a very limited part of their lives, in some cases as little as 5 per cent, the remainder of the time being spent cruising at reduced speeds, as when acting as escorts to merchant convoys.

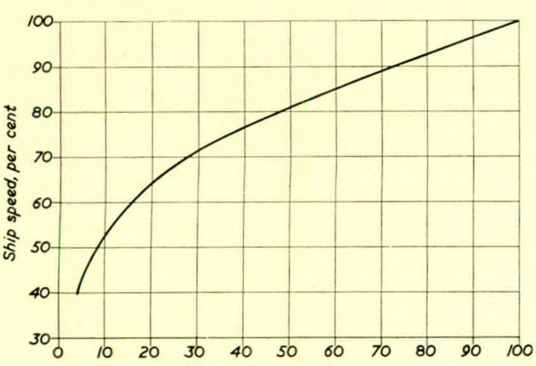


FIG. 7—*Typical power/speed curve*

It thus becomes a criterion of design in many cases that they require economy at two widely separate powers. Firstly, full power, since this will affect the total weight of machinery, and, secondly, at some cruising power which may be as low as 5 per cent. The relation between these two powers is only fixed by the operational requirements for which the ship is to be designed. This is best illustrated by an example.

Take two similar ships, one with machinery designed to give a full speed of 40 knots, and the other capable of only 30 knots. If the required cruising speed of both is, say, 20 knots, then the 40-knots ship will require only $8\frac{1}{2}$ per cent full power, while the engines for the 30-knot vessel must produce $22\frac{1}{2}$ per cent full power. Fig. 8 illustrates very clearly the differences in the power requirements of two such vessels at various cruising speeds.

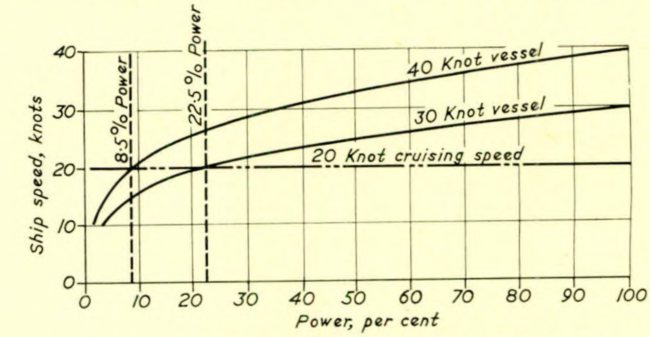


FIG. 8—*Power/speed curve (40- and 30-knot vessels)*

Now the general law connecting power developed and steam rate, ignoring irregularities due to throttling, for any variable speed turbine as used for marine propulsion is shown in curve A in Fig. 9, from which it will be appreciated that the lowest steam rate is only obtained at one particular power dependent upon the requirement for which the turbine is designed. Performance falls off at higher or lower powers, particularly for the latter.

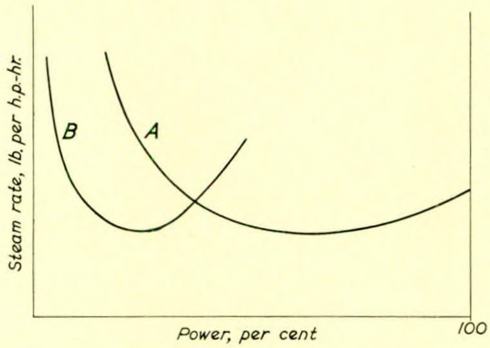


FIG. 9—*Power/steam rate*

The only way in which the relative performance at low and high powers can be changed is by moving the point at which the minimum steam rate is obtained either to the right or the left when for practical purposes the curve will move bodily in the same direction.

Where, however, it is required to improve low power performance without prejudicing full power performance, it is possible to introduce a second or "cruising turbine" having a characteristic shown by curve B which, being designed for the smaller steam flow at reduced power, will give a much improved performance at such powers than curve A, as shown. It is then possible to use turbine B for low powers and to change to turbine A for higher and full power.

The effect of such a cruising turbine is to superpose additional stages in the steam path. Table III, based on the *Daring III* h.p. turbine, shows the large variation in the Curtis stage heat drop that exists over the power range.

TABLE III.

Power percentage	100	60	40	20	10
Total heat drop, B.t.u. per lb.	490	521	532	539	539
Heat drop, Curtis stage, B.t.u. per lb.	65	92	142	210	255
U Curtis stage, ft. per sec.	824	676	596	484	386
U/Co Curtis stage	0·455	0·315	0·224	0·149	0·108

It can be seen that the ratio of heat available for useful work in the Curtis stage compared with that for the whole machine varies from only 13 per cent at full power to 50 per cent at 10 per cent power; as the efficiency falls with both increased heat drop and decreased speed, it is readily seen that at low cruising powers the effect of superposing additional stages and thus restoring the efficiency of such a large percentage of the total available heat drop has a very considerable effect on the steam rate of the machinery as a whole.

The various methods of incorporating cruising turbines will be dealt with later.

The advantages or disadvantages of employing cruising turbines depend entirely on the operational duties required of the ship.

Boiler Characteristics

Formerly normal Naval boilers gave a steam temperature rising with increasing output. This was known as "normal characteristic" and was inconvenient as high temperatures were produced when rotational stresses were highest.

This problem can be eased by incorporating a form of superheat control designed to give the following:—

Flat characteristic: constant temperature at all outputs.
Inverse characteristic: fall in temperature with increasing output.

The "inverse temperature characteristic" has the desirable feature of producing the high temperatures in conjunction with low powers: thus the maximum rotational stresses are only encountered with the reduced temperatures.

Types of Main Engines Analysed

Range of Types

Nine different designs of main engines (as listed below) were analysed as accurately as possible without producing final and detailed designs. The estimated steam and heat consumptions are given in Figs. 10 and 11 respectively, but it must be emphasized that the results were of the first order of accuracy only. This did not, however, prevent the drawing of very definite conclusions regarding the relative merits of the various arrangements.

(a) *Daring* Class with a "flat" temperature characteristic.
(b) *Daring* Class with an "inverse" temperature characteristic.
(c) High pressure, straight three-cylinder turbine for nominal boiler steam conditions of 1,400lb. per sq. in. gauge, 1,050 deg. F., with "normal" characteristic.
(d) High pressure, straight three-cylinder turbine with nominally "flat" temperature characteristic at 1,025/1,050 deg. F.
(e) High pressure, straight three-cylinder turbine with "inverse" temperature characteristic to 1,100 deg. F.
(f) High pressure, two-cylinder turbine, plus hydraulically coupled high speed cruising turbine, "normal" temperature characteristic.
(g) High pressure, two-cylinder turbine, plus hydraulically coupled low speed cruising turbine, "normal" temperature characteristic.
(h) High pressure, two-cylinder turbine, plus permanently coupled, overhung, high speed cruising turbine.

The reasons for examining these alternatives, together with their main design features, are given in the following paragraphs:—

(a) Daring *Class with "flat" temperature characteristic*

To obtain a datum upon which to base comparisons, the estimated consumption figures for *Daring* Class were used.

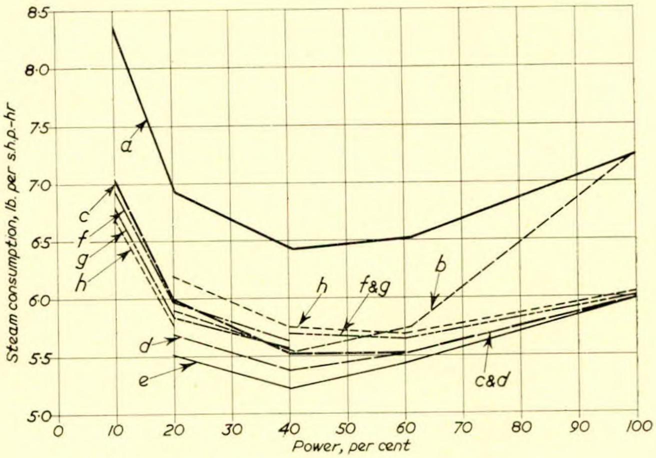


FIG. 10—*Steam consumptions*

(a)———— *Daring* Class with "flat" steam temperature characteristic
(b)— — — — *Daring* Class with "inverse" steam temperature characteristic
(c)—— —— Straight three-cylinder machine at advanced steam conditions
(d)—— —·— Straight three-cylinder machine; 1,050 deg. F. at 20 per cent and 40 per cent powers
(e)———— Straight three-cylinder machine; 1,100 deg. F. at 20 per cent and 40 per cent, 1,050 deg. F. at 60 per cent powers
(f) – – – – – Two-cylinder machine plus hydraulically coupled high speed cruising turbine
(g)— - - — - - Two-cylinder machine plus hydraulically coupled low speed cruising turbine
(f & g) — - - - Two-cylinder machine with cruising turbine uncoupled
(h)- - - - - - - - Two-cylinder machine plus permanently coupled overhung high speed cruising turbine

Daring Class

Power, per cent	Steam pressure, lb. per sq. in. abs.	Vacuum, in.Hg.	Steam temperature characteristic, deg.F. Flat (curve *a*)	Inverse (curve *b*)
100	565	27·5	825	825
61·125	625	28·53	825	1,000
40·75	641	28·8	825	1,050
20·375	651	28·95	825	1,050

Advanced steam conditions (curves *c* to *h* inclusive)

Power, per cent	Steam pressure, lb. per sq. in. abs.	Vacuum, in.Hg.	Steam temperature, deg.F. *c, f, g, h*	*d*	*e*
100	1,315	27·5	1,025	1,025	1,025
60	1,405	28·6	1,025	1,025	1,050
40	1,427	28·9	1,009	1,050	1,100
20	1,445	29·1	965	1,050	1,100
10	1,450	29·1	950		

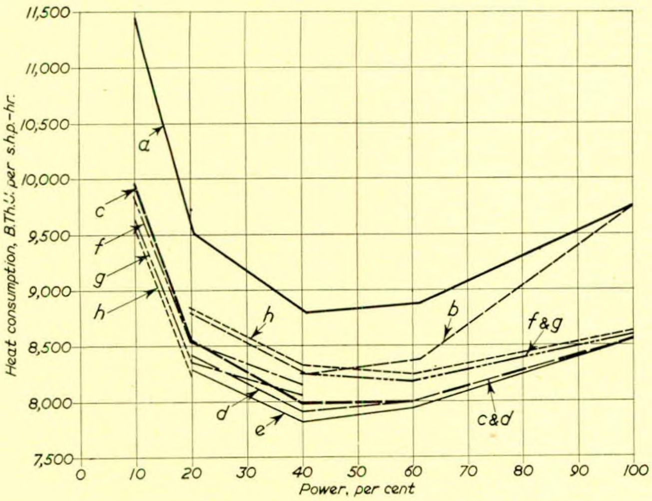


FIG. 11—*Heat consumptions*

(a)————*Daring* Class with "flat" steam temperature characteristic
(b)— — — — *Daring* Class with "inverse" steam temperature characteristic
(c)— — —Straight three-cylinder machine at advanced steam conditions
(d)— — —Straight three-cylinder machine; 1,050 deg. F. at 20 per cent and 40 per cent powers
(e)————Straight three-cylinder machine; 1,100 deg. F. at 20 per cent and 40 per cent, 1,050 deg. F. at 60 per cent powers
(f) – – – – – Two-cylinder machine plus hydraulically coupled high speed cruising turbine
(g)— - - - - - Two-cylinder machine plus hydraulically coupled low speed cruising turbine
(f & g) — - - - Two-cylinder machine with cruising turbine uncoupled
(h)- - - - - - - - Two-cylinder machine plus permanently coupled overhung high speed cruising turbine

Daring Class

Power, per cent	Steam pressure, lb. per sq. in. abs.	Vacuum, in.Hg.	Steam temperature characteristic, deg.F.	
			Flat (curve *a*)	Inverse (curve *b*)
100	565	27·5	825	825
61·125	625	28·53	825	1,000
40·75	641	28·8	825	1,050
20·375	651	28·95	825	1,050

Advanced steam conditions (curves *c* to *h* inclusive)

Power, per cent	Steam pressure, lb. per sq. in. abs.	Vacuum, in.Hg.	Steam temperature, deg.F.		
			c, f, g, h	*d*	*e*
100	1,315	27·5	1,025	1,025	1,025
60	1,405	28·6	1,025	1,025	1,050
40	1,427	28·9	1,009	1,050	1,100
20	1,445	29·1	965	1,050	1,1C0
10	1,450	29·1	950		

In addition to the general design details already described, the maximum stress figures were as shown in Table IV.

(b) Daring *Class with "inverse" temperature characteristic*

The application of an "inverse" temperature characteristic produced a reduction in consumption at cruising powers without prejudicing in any way the full power performance. The gain was due mainly to the increase in the available adiabatic heat drop resulting from the increased steam temperature provided at part powers. As the steam temperature at higher powers, 60 per cent and above, was the same as scheme (a), the combination of maximum stress and temperature conditions of moving parts also remained the same; this meant that only stationary items such as steam piping, valves, h.p. turbine inlet end, etc., required the introduction of special materials suitable for the higher temperatures. For convenience of rapid manœuvring it was considered advisable to limit steam temperatures to 800 deg. F., although the design of the machinery was suitable for a "crash stop" under any set of operating conditions.

TABLE IV.

	H.P. rotor	L.P. rotor
Maximum speed, light draught, r.p.m.	8,250	6,357
Maximum stress at rotor bore, tons per sq. in.	15·3	18
Temperature of rotor under maximum stress, deg. F.	800	355

Boiler designers expressed their ability to produce a load/temperature curve as shown in Fig. 12 by the introduction of a suitable form of temperature control without undue complication, increase in weight or loss in efficiency.

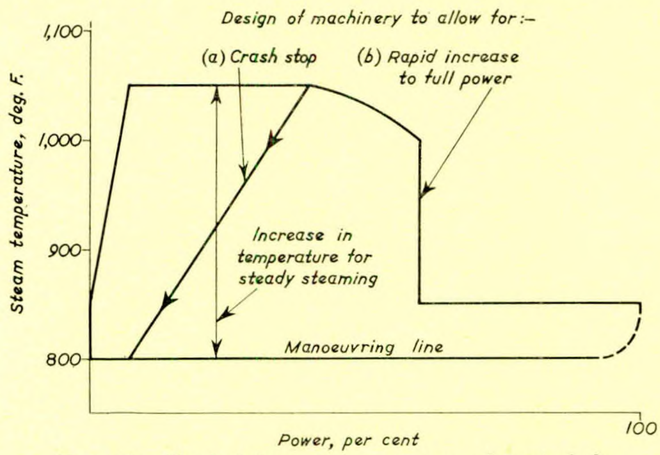


FIG. 12—*Typical inverse temperature characteristic*

The general arrangement and technical details, stresses, etc., were the same as for alternative (a).

(c) *High pressure, straight three-cylinder turbine for "normal" temperature characteristic of 1,025 deg. F.*

In order to establish what economies could be obtained by advancing steam conditions, the design of a turbine suitable for the maximum conditions which the boilermakers were prepared to adopt with known materials was investigated. With the conditions used in this alternative, Fig. 10, the steam temperature rose with an increase in boiler output.

With these increased steam conditions it was first established that, owing to the additional heat drop to be handled, and the limitations of critical speeds and shock stresses on the turbine rotors, it was not feasible to design a satisfactory two-cylinder turbine without unduly prejudicing the steam consumption by adopting inadequate $^{U}/Co$ ratios. It appeared that under the specified conditions there was every justification for adopting a three-cylinder design. The design favoured

TABLE V.

	H.P. cylinder	I.P. cylinder	L.P. cylinder
Maximum speed, light draught, r.p.m.	9,000	9,000	6,357
Maximum stress at rotor bore, tons per sq. in.	18	15	18
Temperature of rotor under maximum stress, deg. F.	984	865	355

consisted of co-axial h.p. and i.p. cylinders (Figs. 13 and 14) driving one gear pinion, with a double flow l.p. cylinder the same as *Daring III* arranged in parallel and driving a separate pinion. Other major details are shown in Table V.

Examination of the stress/temperature characteristics given for the h.p. and i.p. rotors indicated that special consideration would need to be given to the materials used. A preliminary evaluation of material requirements is discussed later.

(d) *High pressure, straight three-cylinder turbine with "flat" temperature characteristics at 1,025/1,050 deg. F.*

(e) *High pressure, straight three-cylinder turbine with "inverse" temperature characteristic, 1,100 deg. F.*

The construction of this machinery was identical with the three-cylinder machine, alternative (c), and the same remarks applied regarding the stress/temperature characteristics for the h.p. and i.p. rotors at maximum power and speeds. But was required to cater for stress/temperature characteristics.

The cruising turbine and h.p. turbine details are given in Table VI.

TABLE VI.

	Cruising cylinder	H.P. cylinder
Maximum speed, light draught, r.p.m.	15,000	8,250
Maximum stress at rotor bore, tons per sq. in.	Over 16	16
Temperature of rotor under maximum stress, deg. F.	775	920

(g) *High pressure, two-cylinder turbine, plus hydraulically coupled, low speed, cruising turbine, "normal" temperature characteristic 1,025 deg. F.*

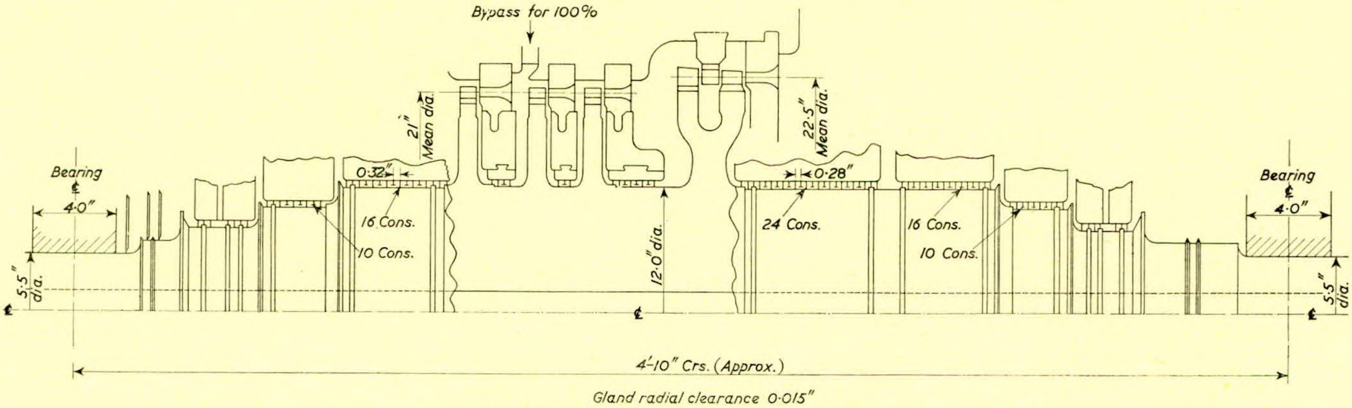


FIG. 13—*H.P. basis plan, scheme (c)*

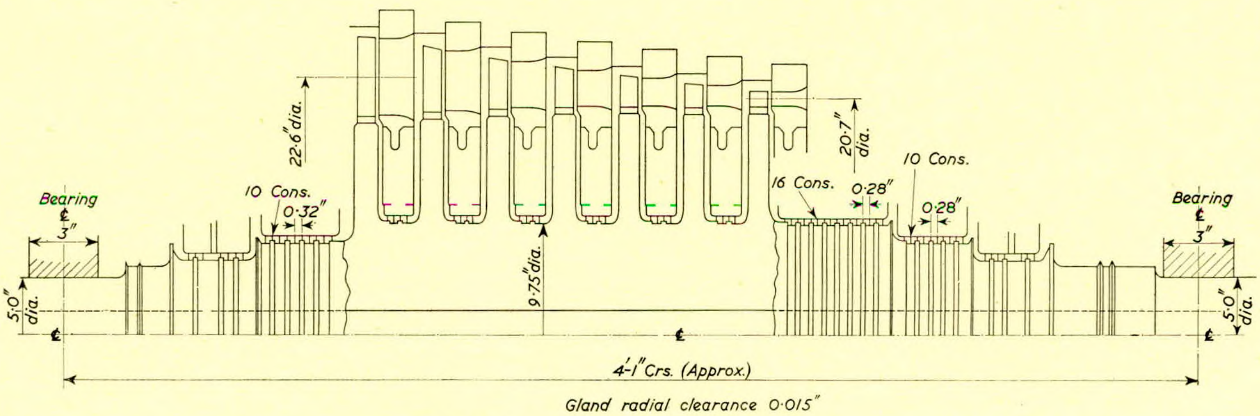


FIG. 14—*I.P. basis plan, scheme (c)*

increased consideration had to be given to the life factor when operating at higher temperatures and reduced powers.

(f) *High pressure, two-cylinder turbine, plus hydraulically coupled high speed cruising turbine, "normal" temperature characteristic 1,025 deg. F.*

This machinery comprised a small, high speed cruising turbine connected by a hydraulic coupling to the h.p. cylinder through a single reduction gear, the h.p. and l.p. rotors driving separate pinions of the main double reduction gearing.

At all loads over 40 per cent the cruising turbine and its single reduction gear was declutched; the machine then operated as a normal two-cylinder turbine. It was arranged to declutch at 40 per cent power as it was decided that this operation should occur well above the ship's normal cruising speed; it was not chosen for any reasons of turbine design.

As on the other turbine arrangements designed for operation with high initial steam conditions, due consideration

This arrangement was the same as for scheme (f) except that the cruising turbine was hydraulically coupled direct to the h.p. turbine, extra stages being required to maintain the efficiency at the lower running speed, producing a longer cruising turbine.

As in scheme (f), the cruising turbine was declutched at all loads above 40 per cent by means of the hydraulic coupling.

All remarks relating to scheme (f) applied equally to this arrangement. Details of the h.p. turbine were the same as

TABLE VII.

Maximum speed (declutched at 40 per cent load), r.p.m.	5,660
Maximum stress at rotor bore, tons per sq. in.	7·6
Temperature of rotor under maximum stress, deg. F.	925

for scheme (f) while the l.p. turbine remained the same as for scheme (c). The cruising turbine speeds and stresses were as given in Table VII.

(h) *High pressure, two-cylinder turbine, plus permanently coupled, overhung, high speed cruising turbine, " normal " temperature characteristic, 1,025 deg. F.*

This scheme was investigated with a view to reducing the size of the cruising turbine. Apart from any constructional difficulties entailed, it eliminated the long h.p. gland with its steam leakage and also one bearing on the cruising turbine. The complication of the hydraulic coupling did not arise, thus any difficulties involved in changing from the appropriate loads corresponding to cruising turbine " in " and " out " were absent.

The cruising turbine was bypassed at all loads above 20 per cent, a small quantity of steam being required for cooling purposes only.

The stress problems were somewhat greater in this type of cruising turbine as, being permanently coupled, the condition at 100 per cent power light draught had to be taken into account.

Particulars of the h.p. turbine and l.p. turbine remained the same as for scheme (f) and scheme (c) respectively, maximum speed, etc., of the cruising turbine being as follows: —

TABLE VIII

Maximum speed, light draught, r.p.m.	21,800
Maximum stress at rotor bore, tons per sq. in.	17·5
Temperature of rotor under maximum stress	Dependent on cooling steam

(j) *High pressure two-cylinder turbine, plus permanently coupled two-bearing cruising turbine*

This arrangement was the same as for scheme (g) except for the omission of the hydraulic coupling, the cruising turbine being permanently coupled to the h.p. turbine, which drove one pinion of the main double reduction gearing, the l.p. turbine driving the other. The cruising turbine was bypassed at all loads over 20 per cent except for the small steam quantity required for cooling purposes.

The weight and space occupied by this arrangement was found to be no less than that required for the straight three-cylinder machine, scheme (c), the steam consumption being generally worse with no easing of stress/temperature difficuties. It was therefore decided to proceed no further.

It was considered possible that there might be some advantage regarding efficiency if the cruising turbine were to be so designed that the critical speed of the rotor was well below any operational ship's speed (say, 10 knots maximum). Investigation showed this to be impracticable as the shaft diameter had to be reduced to such an extent that the shock-stress requirements could not be met.

Comparison of Types Analysed

The variation in steam consumption with change of load for all the alternative schemes is shown in Fig. 10 and the heat consumption in Fig. 11, in which the steam conditions and vacua at the various loads are tabulated.

Table IX has been prepared to show percentage differences in main engine heat consumptions of the various schemes. *Daring* Class estimated consumptions with steam conditions as designed, i.e. flat temperature characteristic, have been used as the datum for comparison.

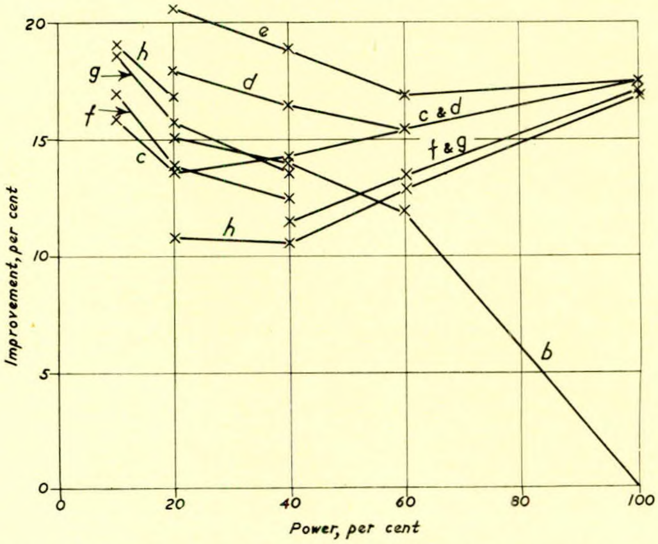


FIG. 15—*Percentage improvement in steam consumption, Y.E. 47A*

Fig. 15 shows the percentage improvement in steam consumption for all the alternative schemes at the various percentage powers. The improvements shown are only correct at the points for which the calculations were made, viz., 10, 20, 40, 60 and 100 per cent. Improvements at intermediate powers between these points cannot be obtained from the curves.

All estimates of heat consumptions have assumed the condensate to be at the saturation temperature corresponding to the vacuum obtained.

Rotor Material Requirements

At the time of the Y.E. 47A investigation the combination of high steam temperatures and high rotational stresses encountered on some of the alternative schemes made it essential that a basis should be prepared on which material properties could be discussed with the steel makers.

The turbine arrangement under consideration was scheme (e), the straight three-cylinder machine with initial steam conditions of 1,400lb. per sq. in. and 1,050 deg. F. at maximum power, with " inverse " temperature characteristic. It was

TABLE IX.

Scheme	Load, per cent 10	20 Cruising " In "	20 Cruising " Out "	40 Cruising " In "	40 Cruising " Out "	60	100
(*a*) Datum, B.t.u. per s.h.p. hr.	11,455	—	9,515	—	8,800	8,878	9,750
(*b*) Improvement, per cent	Not calculated	—	7·5	—	6·3	5·7	Nil
(*c*) Improvement, per cent	13·1	—	10·1	—	9·3	9·9	12·2
(*d*) Improvement, per cent	Not calculated	—	11·6	—	10·1	9·9	12·2
(*e*) Improvement, per cent	Not calculated	—	13·0	—	11·2	10·5	12·2
(*f*) Improvement, per cent	14·2	10·4	—	7·3	6·3	7·8	11·7
(*g*) Improvement, per cent	15·9	12·2	—	8·5	6·3	7·8	11·7
(*h*) Improvement, per cent	16·4	13·4	7·1	—	5·3	7·1	11·4

TABLE X.

	100 per cent power, light draught		40 per cent power, deep draught		20 per cent power, deep draught	
	H.P.	I.P.	H.P.	I.P.	H.P.	I.P.
Nozzle box pressure, lb. per sq. in.	1,300	—	1,410	—	1,430	—
Nozzle box temperature, deg. F.	1,025	—	1,100	—	1,100	—
Wheel case steam temperature, deg. F.	984	865	877	660	815	500
Maximum bore stress, tons per sq. in.	18	15	8·5	7·1	5·5	4·6
Average stress, tons per sq. in.	12	10	5·6	4·7	3·7	3·1

assumed that at 20 per cent power the maximum permissible steam inlet temperature would be limited by boiler difficulties to 1,100 deg. F.

Under those conditions special properties would be required in the materials for the h.p. and i.p. rotors, but no difficulty would be experienced with the l.p. rotor for which the 3 per cent chrome-molybdenum steel as used in the *Daring* Class would be quite suitable.

It was assumed that the maximum life requirements at full power need not exceed 500 hours but that at 20 per cent and 40 per cent power long life would be required.

Stresses and Temperatures

Table X gives the maximum stresses and temperatures to which the h.p. and i.p. rotors would be subjected at 100, 40 and 20 per cent powers on the assumption that the rotor material would eventually reach the ambient steam temperature, and stresses would follow the usual elastic theory.

The above temperatures can only be approximate as there is evidence which suggests that a rotor does not reach the full steam temperature even after prolonged runs, and a difference of as little as 50 deg. F. between metal and steam temperatures could considerably influence the suitability of a particular material. The main difficulty in a calculation of this nature is the determination of the temperature at the surface of the rotor and the gradient from there inwards.

Another factor which requires further reliable experimental data is the effect of stress resulting from temperature gradient between the bore and rim of the rotor. These gradients are likely to be very severe during rapid manœuvring.

Short Term Tests

General practice tends to impose a suitable safety factor based on either the ratio of the ultimate tensile stress to the average stress or the ratio of proof stress to average stress, all at working temperature.

The foregoing considerations indicate the following properties as being desirable for the rotor material:—

TABLE XI.

	H.P. rotor	I.P. rotor
Ultimate tensile strength	Not less than 36 tons per sq. in. at 984 deg. F.	Not less than 30 tons per sq. in. at 865 deg. F.
0·2 per cent proof stress	Not less than 27 tons per sq. in. at 984 deg. F.	Not less than 23 tons per sq. in. at 865 deg. F.

Creep Properties

The maximum rotational stress and temperature conditions upon the rotors occur under full power at which, however, the life required is only 500 hours.

Under long life conditions (say 100,000 hours) both temperatures and stresses are materially reduced, the worst conditions being at 40 per cent power. Since, however, there is no reliable method of correlating short term and long term creep properties, it is preferable to specify the requirements under both conditions.

The criterion is the permissible deformation, and bearing in mind that the total deformation due to creep is additional to the deformation due to elastic stress, which may be of the same order, it is not considered that the total creep deformation can be allowed to exceed 0·1 per cent.

It is also essential to guard against failure due to rupture.

On the above basis the following creep properties are required:—

TABLE XII. FULL POWER CONDITIONS

	H.P. rotor	I.P. rotor
Total deformation in 500 hours— Not greater than 0·1 per cent (2×10^{-6}in./in./hr.)	At 984 deg. F. and 12 tons per sq. in.	At 865 deg. F. and 10 tons per sq. in.
Deformation at rupture— Not less than 5 per cent.	At 984 deg. F. and 12 tons per sq. in.	At 865 deg. F. and 10 tons per sq. in.

40 PER CENT POWER CONDITIONS

	H.P. rotor	I.P. rotor
Total deformation in 100,000 hours— Not greater than 0·1 per cent.	At 877 deg. F. and 5·6 tons per sq. in.	At 660 deg. F. and 4·7 tons per sq. in.
Deformation at rupture— To be stated	At 877 deg. F. and 5·6 tons per sq. in.	At 660 deg. F. and 4·7 tons per sq. in.

In addition to the foregoing, there are many other material properties which are desirable in a material for turbine rotors. These are outlined below, no attempt having been made to place them in order of importance.

Fatigue Endurance Limit

As this may be of vital importance on components subject to vibratory stresses, it is desirable that the endurance limits should be known under the following three conditions:—

(a) At a temperature of 984 deg. F., corresponding to full power conditions.

(b) At a temperature of 877 deg. F., corresponding to 40 per cent power conditions.

(c) At room temperature.

Notch Sensitivity

Tests should be carried out to simulate as nearly as possible the effect of stress concentration due to unavoidable rapid changes in section and the effect of shock conditions at:—

(a) A temperature of 984 deg. F.

(b) A temperature of 877 deg. F.

(c) Room temperature.

Damping Capacity

The inhibition of resonant vibratory stresses is best dealt with by selecting material having suitable damping capacity. It seems essential that damping properties should be known under the following conditions of operation representing long life:—

(a) At a temperature of 877 deg. F.

(b) At room temperature.

Metallurgical Stability

It is essential that the material selected should have no undesirable metallurgical instability after prolonged heating in high temperature steam, i.e. that no chemical or permanent change in the metallurgical structure should occur under such conditions.

Thermal Stress Cycles

It is essential that the material should be capable of withstanding the stress cycles induced by rapid variations in the temperature gradients without failure or material alterations to its physical properties.

Other Physical Characteristics

Scaling, machinability, coefficient of expansion, variation of modulus of elasticity with the temperature up to 1,000 deg. F. are all properties which must be proved to be satisfactory before commencing the manufacture of rotors in the new material.

Tests

It was agreed that an experimental programme was necessary in order to check the properties mentioned under the foregoing headings and the Admiralty instituted a suitable programme in conjunction with the forgemasters, which included, amongst other essential requirements, destructive tests enabling properties to be established throughout the rotor.

General Conclusions

An analysis of the results of the Y.E. 47A investigation dealing with main engines, showed that the maximum temperature of the cycle had the most profound effect upon the cycle efficiency at all powers.

The effect of an "inverse" temperature characteristic on the heat consumption at part loads resulted in very considerable gains over the required cruising range, and this improved performance was obtained without having to use materials in the turbine any different from those required to meet similar boiler conditions, with either "normal" or "flat" temperature characteristics.

It was seen by comparing the appropriate curves, or referring to the table showing "differences in heat consumption", that increased steam pressure had relatively little effect upon main engine performance, approximately 4 per cent only being gained by increasing the pressure level from 600lb. to 1,400lb.; the optimum boiler will undoubtedly lie somewhere between the two for the type of machinery designed to meet the specified requirements.

Economy

There appeared little doubt that, for the stipulated requirements, where maximum economy was required between 20 per cent and 40 per cent load, the most economical turbine arrangement was the high pressure, straight three-cylinder machine with "inverse" temperature characteristic, scheme (e). This alternative also showed a considerable gain at 100 per cent power on the existing *Daring* design. Between 20 per cent and 40 per cent power *Daring* Class, with an "inverse" temperature characteristic scheme (b), approached the performance of the high pressure, straight, three-cylinder design with "normal" characteristic, scheme (c). The difference between the two was approximately 3 per cent but the former had the disadvantage that it entailed additional boiler capacity owing to increase in the water rate at 100 per cent power.

Unless operational requirements were such that particularly low water rates were required at powers appreciably below 20 per cent, there did not appear to be any justification for considering the incorporation of cruising turbines. Even then, a detailed analysis of the machinery installation as a whole would be necessary to establish whether the inclusion of a cruising turbine would be justified.

It was noted that *Daring* Class with the "inverse" temperature characteristic scheme (b) was 7½ per cent better than *Daring* with a "flat" characteristic at 20 per cent power, but was inferior to all the high pressure schemes by approximately 3 to 6 per cent at that load.

Summing up, therefore, from the aspect of the main turbines it appeared that very little, if any, advantage could be obtained from the use of cruising turbines, unless the ship's design was such that the required cruising speeds would be obtained with very low percentage powers, say, 10 per cent or below.

While destroyer machinery was used to establish the above general principle, it would, of course, apply to all classes of ships.

Minimum Weight and Space

If economy were required at powers appreciably below 20 per cent, resulting in the adoption of cruising turbines, then the additional weight and space should be considered, including couplings, etc.

Similarly, adoption of a three-cylinder design, as compared with a two-cylinder design, would involve some additional weight, but it was not considered that the space occupied by the machinery would be appreciably affected as the l.p. turbine and condenser were the limiting factors.

Reliability

In view of the limited experience with steam temperatures of 1,050 deg. F., the designs utilizing these high steam conditions compared unfavourably from the point of view of reliability with those such as the *Daring* employing more conservative steam temperatures. The former, therefore, were regarded as long term projects rather than for immediate adoption. Th "inverse" temperature characteristic, applied to the *Daring* design as shown in scheme (b), could, however, be applied without any risk whatsoever, as, should an emergency arise before the high temperature components had been fully proved at sea, the maximum operating temperature could be reduced. This would increase the fuel consumption at part loads but would not in any way prejudice the development of full power.

Y.E.A.D. I

As a result of the Y.E. 47A investigations, the Admiralty decided to put in hand work connected with a research and development programme on two alternative schemes. The first of these involved immediate design, manufacture and shore testing of a set of main machinery based on the preliminary design prepared for Y.E. 47A, scheme (b). The second was in the nature of a long term project as it was based on Y.E. 47A, scheme (e), which necessitated the use of a rotor material with all the properties as previously described. As the first essential was to make such a material available, the Admiralty, in conjunction with the forgemasters, instituted the necessary research and development programme. These two schemes, known as "Yarrow, English Electric, Advanced Designs", were shortened to Y.E.A.D. I and Y.E.A.D. II respectively.

In 1950 the English Electric Company were entrusted with the design and production of a prototype set of Y.E.A.D. I turbines and condenser, the details of which are now described.

Specific Requirements

The turbines were to produce 30,000 s.h.p. when supplied with steam at 550lb. per sq. in. abs. and 825 deg. F., the vacuum at exhaust being 26·5-in. Hg. (Bar. 30in.) with a cooling water temperature of 55 deg. F. The inlet steam temperature between 5 per cent and 60 per cent power was 925 deg. F. but was reduced to 725 deg. F. over the whole power range for purpose of manœuvring. The turbines, however, must be capable of manœuvring for short periods with steam at 925 deg. F. Above 60 per cent power the inlet steam temperature is 825 deg. F.

The astern turbine was to be designed to produce 7,000 s.h.p. when supplied with steam at 530lb. per sq. in. gauge and 725 deg. F. and to be capable of operation with a steam temperature of 925 deg. F. for periods not exceeding two minutes' duration; with steam at 825 deg. F. full power astern power duration was specified as fifteen minutes.

The machinery was to be designed for a total working life of 40,000 hours, the life at full power being 1,000 hours.

A good performance was very important between 20 per cent and 40 per cent power.

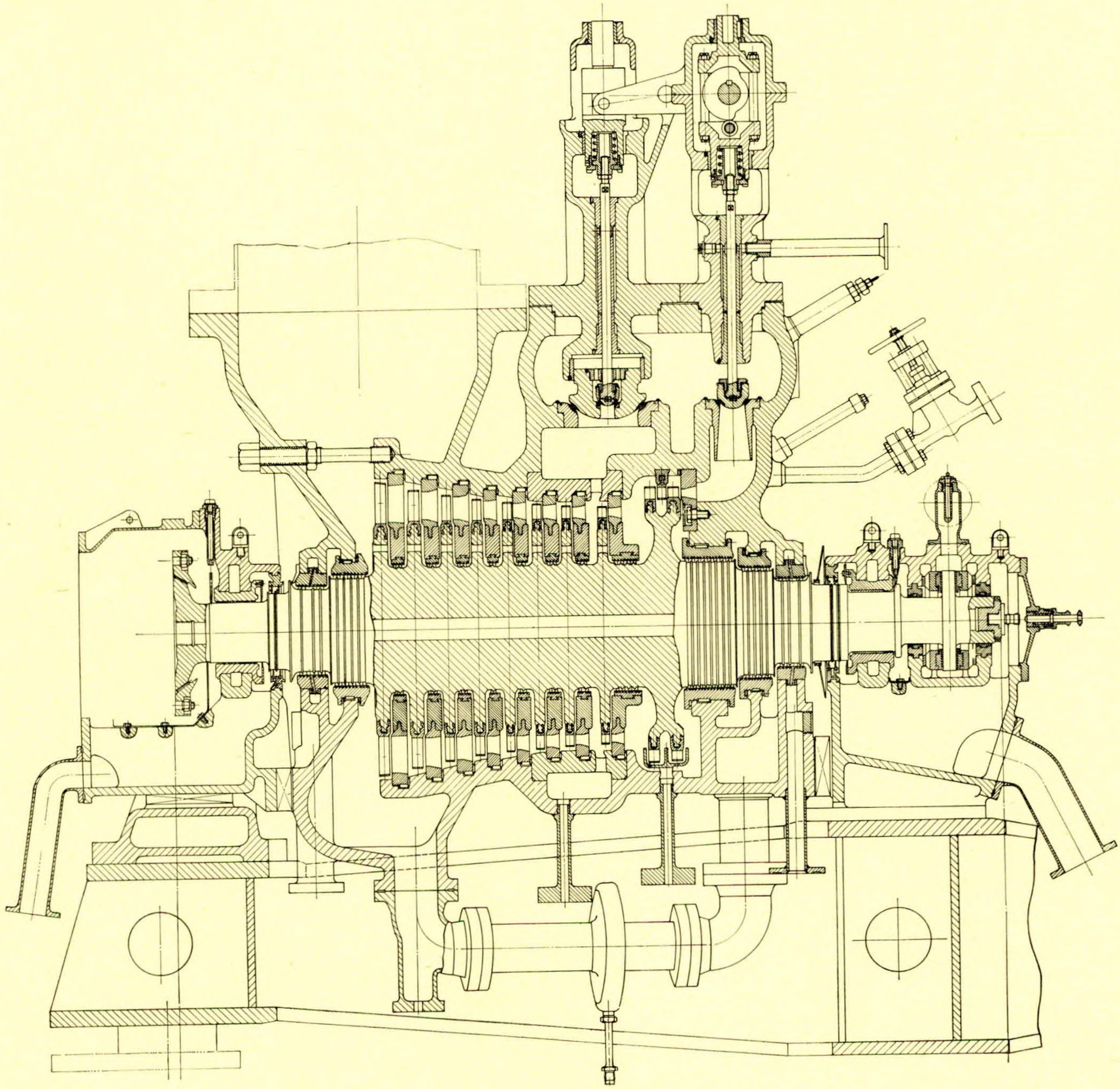


FIG. 16—*Longitudinal h.p. section, Y.E.A.D. I*

The turbines had to be designed to provide simple and safe control and manœuvring from approved positions by unskilled personnel. It was therefore desirable that the ahead turbines should be operated by a single hand wheel at each control position, with another single hand wheel to operate the astern turbine in a similar manner.

The maximum degree of interchangeability was required both in respect of components and of complete units.

H.P. and L.P. Cylinders

Some five years' service experience proved the principles upon which the *Daring III* design was based to be sound and completely satisfactory, and as the performance requirements of Y.E.A.D. I were so very similar, it was logical to adhere generally to the same design, viz., a two-cylinder cross compound arrangement comprising an all-impulse design of h.p. cylinder (Fig. 16) with a solid gashed type rotor, and a double flow l.p. cylinder (Fig. 17) with an astern Curtis wheel at each end also incorporating a solid gashed type rotor.

Steam Chest

The steam chest is cast integrally with the h.p. cylinder, the reasons again being those advanced for the *Daring III*.

Valve Gear

The valve gear is different from that for *Daring III*, being cam operated for both opening and closing of the three nozzle control valves and the overload bypass valve. Each of the control valves supplies steam to individual nozzle groups designed to provide powers of 20 per cent, 40 per cent and 60 per cent. All valves, including that for control of the astern turbine, are of the spherically seated venturi type. These valves are designed for a high steam velocity at the valve seat and throat, with a diffuser exit from the throat. The diffuser serves as a velocity to pressure conversion tube, making it

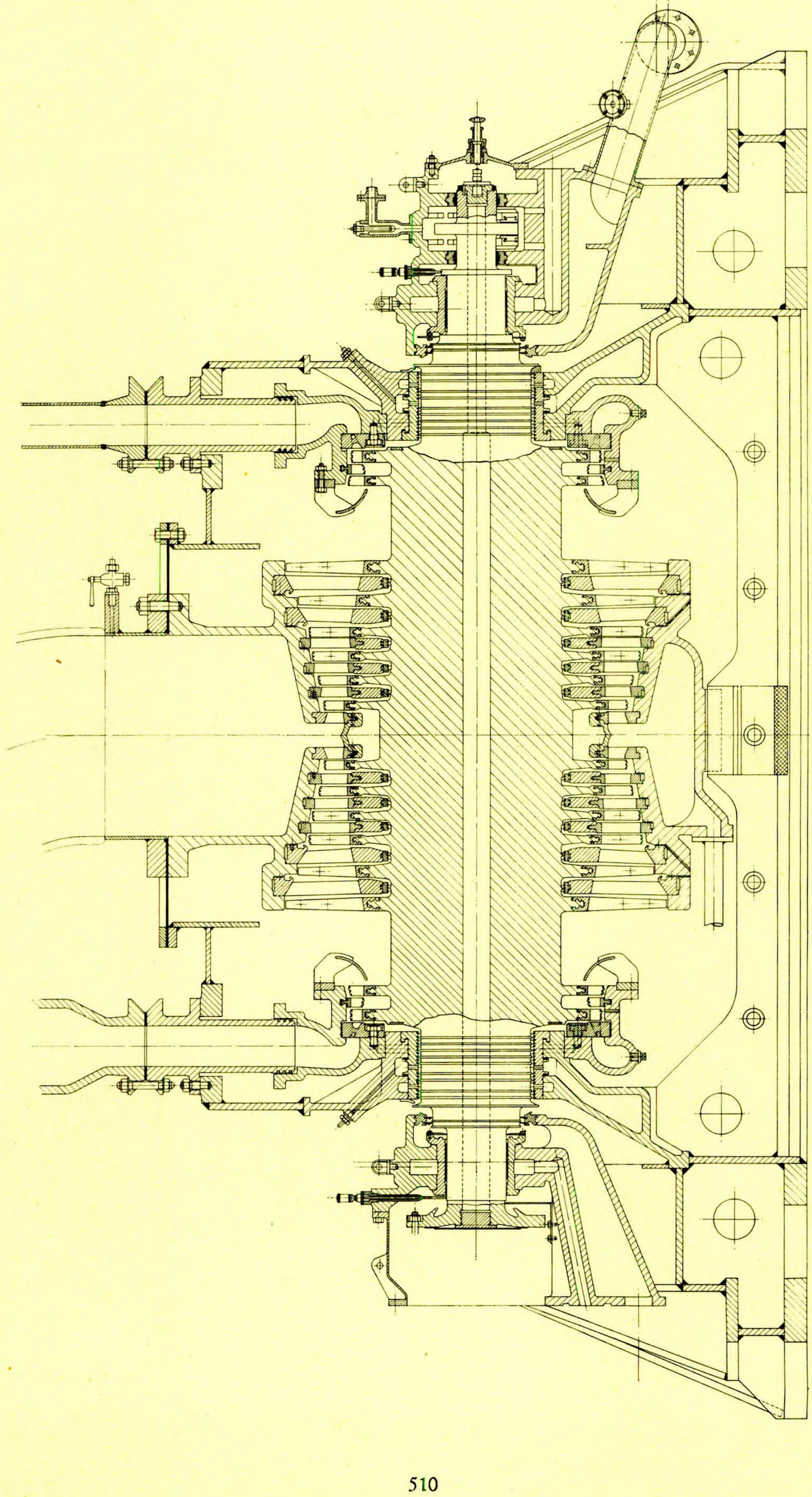


FIG. 17—*Longitudinal l.p. section, Y.E.A.D. I*

possible to use a relatively small valve whilst maintaining a small pressure drop through the valve.

Two handwheels are connected to the valve gear operating mechanism by means of bevel gears and shafting, one for ahead and the other for astern operation.

From a section through the steam chest and two of the control valves (Fig. 16) it will be noted that the overload bypass valve incorporates a pilot valve to facilitate opening of the valve by reducing the unbalanced steam forces. A similar arrangement is fitted to the astern control valve.

Materials

Rotors

Both h.p. and l.p. rotors are of 3 per cent chrome molybdenum steel to Admiralty Specification No. 3 F.1.

Blading

Experience had shown that effective measures had been taken to prevent contamination of the steam and the possibility of chloride attack on blading was now considered to be very slight. It was also felt that if, as a result of damage, the turbines were flooded with sea water, the effect on the blading due to chloride attack would be insignificant compared with that of the ship as a whole.

In view of this it was decided to revert to stainless iron which possesses the desirable qualities of resistance to corrosion, ease of machining, high internal damping factor and a similar coefficient of expansion to that of low alloy steels; consequently all moving blades are made of high tensile stainless iron and all shrouding is made of the same material.

Nozzles

All fixed blading is of stainless iron and with the exception of the last two stages takes the form of nozzles machined from the solid, a built-up construction being employed whereby the segments are attached to mild steel centres or centres of ½ per cent molybdenum steel where high steam temperatures make the latter desirable.

The h.p. turbine first-stage nozzle plate is machined from the solid, the nozzle segments being integral with the centre.

The last two l.p. diaphragms are made of cast steel with stainless iron nozzle division plates cast in.

Steel Castings

The h.p. cylinder and astern inlet end castings are made of ½ per cent molybdenum-vanadium steel, but other castings, such as pedestals, bearing caps, etc., which are not subjected to high steam temperatures, are made of plain carbon steel.

Other Materials

Shock stress requirements necessitate the use of fabricated mild steel plate of items which could otherwise be made of cast iron.

Condenser

The condenser shell is of fabricated steel construction, underslung from the bottom half l.p. cylinder, and is carefully shaped in conjunction with the tube arrangement and so designed to ensure an efficient distribution of steam over the whole condensing surface, whilst offering the minimum resistance to its passage.

The tube plates are made of rolled naval brass in 1-in. minimum thickness.

The tubes, ⅝-in. outside diameter × 19 S.W.G., are of 70/30 cupro nickel secured by Crane Wilkie cone type packings at the inlet ends and Crane full box type packing at the return ends without the use of ferrules.

The combined water boxes and end covers are made of gunmetal, the surfaces in contact with sea water being coated with steel applied by spraying; this treatment dispenses with the need for steel corrosion plates or rods which would otherwise be fitted to the tube plates.

Careful attention has been given to the design of the condenser in order to minimize weight and space; one way in which this has been achieved is by designing for full power water velocity through the tubes of 10ft. per sec., which is a considerable increase on normal practice. The higher heat transmission rate so obtained reduces the surface area of the condenser which would otherwise be required.

Performance

Y.E.A.D. I machinery is about to undergo full-scale shore trials at Pametrada, and Table XIII summarizes the estimated performance figures when operating in temperate waters.

TABLE XIII.

Power s.h.p.	6,000	11,540	17,100	30,000
Inlet steam pressure, lb. per sq. in. abs.	710	698	680	565
Inlet steam temperature, deg. F.	925	925	925	825
Circulating water quantity, g.p.m.	13,250	27,300	27,400	27,200
Vacuum in Hg. (Bar. 30in.) C.W.T. 55 deg. F.	28·57	28·69	28·28	26·5
Gear efficiency, per cent	95·4	96·4	96·9	97·2
Steam consumption, lb. per s.h.p. hr.	6·684	6·273	6·360	7·63

General Details

The following figures summarize other salient features of the Y.E.A.D. I turbines, and enable comparisons of weight and space to be made with the designs described earlier in this paper.

TABLE XIV.

Weight of h.p. turbine complete with valves, pipes, etc., tons		10·5
Weight of l.p. turbine complete with valves, pipes, etc., tons		18·5
Wet weight of condenser, tons		21·0
Specific weight of turbines in lb. per s.h.p.		2·16
	H.P. cylinder	L.P. cylinder
Rotor bearing centres	4ft. 9·9in.	7ft. 9·4in.
Maximum speed light draught, r.p.m.	8,020	6,220
Maximum stress at rotor bore, tons per sq. in.	15·9	18·7
Temperature of rotor under maximum stress, deg. F.	560	260

Conclusions

As already mentioned, the work described has occupied a considerable time for what may be thought rather minor gains, and during this period several systems of propulsion competitive to steam have been suggested. Gas turbines, oil engines, a combination of steam turbines and gas turbines, and finally nuclear power, have all achieved some degree of prominence and some are still under consideration and development. There is, however, no indication at the time of writing that the steam turbine will be ousted from its traditional pre-eminence for the propulsion of high powered ships for a long time to come.

Y.100

Following the comprehensive investigation into the possibilities of effecting improvements in Naval propulsion machinery which has already been described under the heading of Y.E. 47A, the Admiralty became interested in the possibilities of an anti-submarine frigate of some 2,000 tons displacement, to be installed with main machinery of 30,000 s.h.p. on two shafts. A speed sufficient to deal with modern submarines was required, together with a very long endurance at speeds in the region of 15 knots.

The Director of Naval Construction had stated that such a ship could only be achieved if the combined weight of propulsion machinery and fuel did not appreciably exceed 660 tons. By comparison with existing designs of Naval propulsion machinery this requirement demanded a very substantial advance in weight, space and efficiency standards. Based on the *Dido* Class cruiser machinery of 15,000 s.h.p. per shaft the reduction in weight would have to be approximately 33 per cent with a corresponding reduction in space if the requirements were to be fulfilled and it was clear that the target figure could only be achieved by incorporating many of the advances already suggested in connexion with Y.E. 47A.

The Admiralty decided that it was essential to seek the most suitable machinery that the country could produce by engaging the various specialist firms in competitive design studies for both main and auxiliary machinery. A preliminary specification for the main propulsion machinery was issued to several firms early in 1949, and from the resulting tenders that submitted by the English Electric Company was considered by the Admiralty to be the most suitable; the contract was therefore placed accordingly for the design and manufacture of one set of prototype machinery to undergo full-scale shore trials at Pametrada.

The following description outlines the essential design features of the Y.100 turbines as fitted to the *Whitby* and *Blackwood* Class A/S frigates, together with appropriate comments respecting performances and operation.

Specific Requirements

(a) The full power of 15,000 h.p. per shaft to be obtained when supplied with steam to nozzles at 450lb. per sq. in. gauge and 825 deg. F., the vacuum at exhaust being 23-in. Hg. with a cooling water temperature of 85 deg. F.

(b) The ahead turbines to attain a high efficiency at all powers between 5 per cent and 100 per cent; a good performance over the range of 5 per cent to 20 per cent is particularly important.

(c) The astern turbine to produce 5,000 s.h.p. under both tropical and temperate conditions.

(d) Weight and space to be kept to a minimum.

(e) Simplicity of manufacture for quantity production.

(f) Avoidance of relatively scarce materials.

Effects on Design of These Requirements

(a) The low vacuum enables a comparatively short last blade to be used thus easing problems of root stresses, therefore permitting higher rotational speeds than would be possible with a higher vacuum.

(b) Makes it essential to provide some form of arrangement incorporating a cruising turbine, in order to obtain the highest possible overall efficiency at low powers without prejudicing the performance at full power. A requisite number of nozzle control valves minimizes losses due to throttling at intermediate and low powers.

(d) The comparatively poor vacuum reduces considerably the weight and size of the main machinery, particularly the condenser, without unduly affecting part load performance.

(e) Number of different items kept to a minimum, e.g. blade sections, blade roots, nozzles, etc., with machining tolerances as large as possible consistent with interchangeability requirements and requisite fits of the various components.

(f) High percentage alloy steel to be avoided where possible.

Factors Affecting Cruising Turbines

The Y.E. 47A investigation has already shown that when a good performance at both low percentage power and full power is required, a machinery arrangement incorporating a cruising turbine is desirable. This was corroborated beyond doubt by the preliminary investigation work on Y.100 involving estimates of steam consumption over the power range combined with weight and space requirements of several alternative schemes. It became obvious also that a declutchable cruising turbine possesses considerable advantages over one that is permanently coupled. The former can be declutched at some appropriate point along the power range when its efficiency falls below that of the main turbine, thus at higher powers all rotational losses that would otherwise be increased are eliminated. A declutchable cruising turbine also affects an appreciable saving in weight and space due to the design being based on a maximum speed corresponding to some predetermined part load, whereas if permanently coupled the turbine must be designed to withstand the stresses encountered at the higher speed corresponding to the full power.

In view of the above comments it was decided to proceed with the design of a machine incorporating an automatic mechanical clutch as any additional complication entailed was considered to be more than offset by improved performance combined with reduced weight and space.

General Arrangement

A diagrammatic arrangement of the port set of main machinery is shown in Fig. 18 and the photograph (Fig. 19) was taken in a shipbuilder's erection shop and shows a set of machinery assembled and ready for shop trials.

Referring to Fig. 18, the turbine arrangement comprises a cruising turbine (1) situated outboard of the main turbine (5) connected through a clutch (2) and a single reduction gear (3) to the primary train of the double reduction gear box (4). The main and astern turbines are combined in the same casing and coupled to the primary train of the double reduction gears. The main turbine and condenser are integral, the bottom half of the turbine casing forming the top portion of the condenser shell.

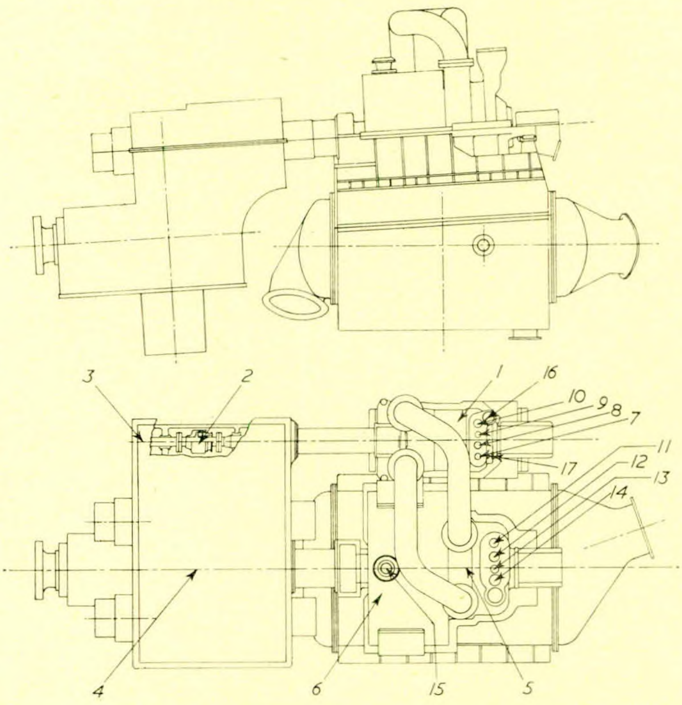
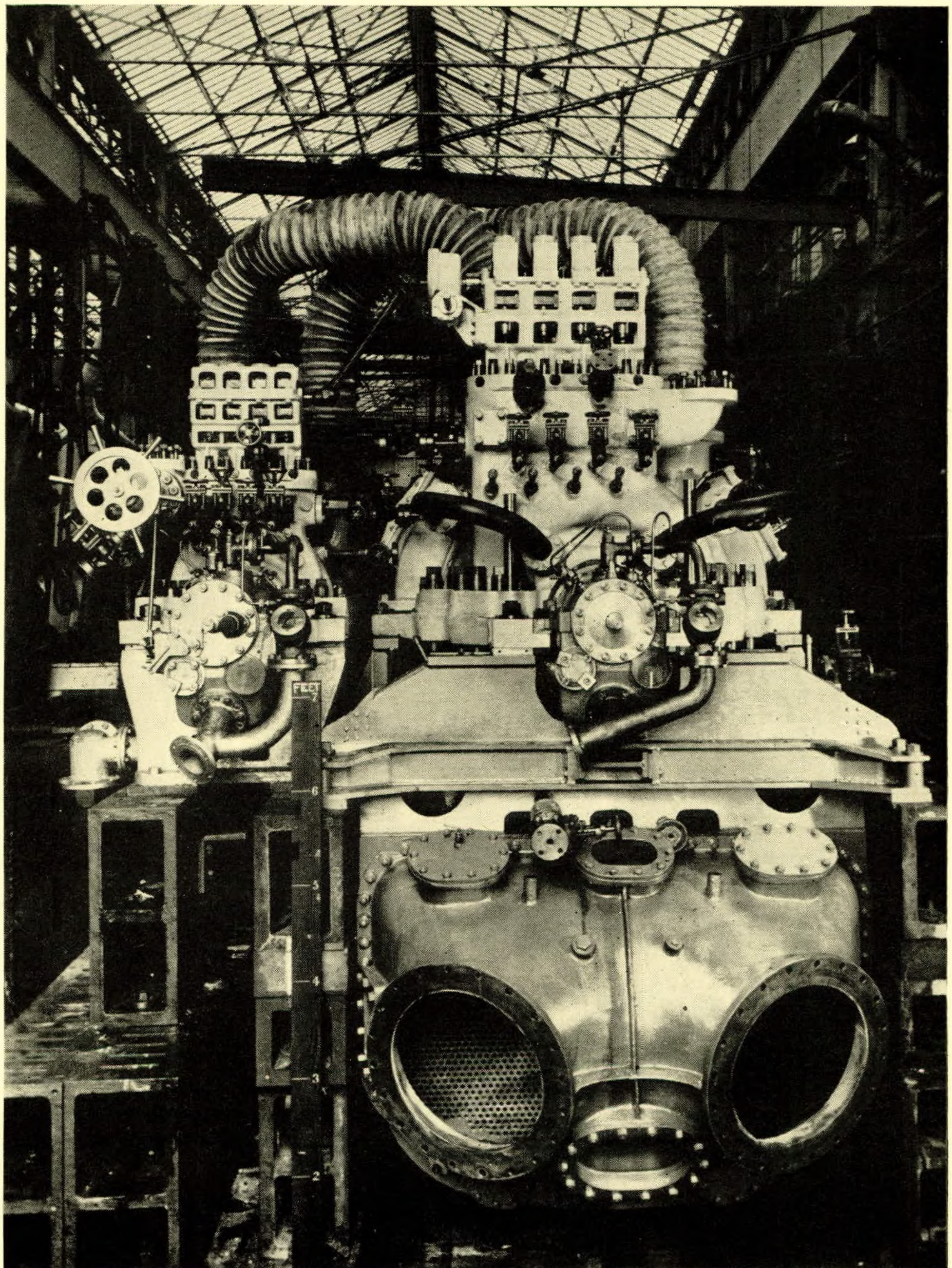


FIG. 18—*Diagrammatic arrangement of port set main machinery, Y.100*

Special Features

Economy in operation at cruising speeds is achieved by the use of the cruising turbine (exhausting to the main turbine bypass belt) which is engaged for all powers up to approximately 33 per cent and by the transference of all load to the main turbine at powers above that point. The change of load from one turbine to the other is carried out by means of the manually operated nozzle control valve mechanism in conjunction with the clutch, which automatically engages and disengages the cruising turbine at some predetermined point in the power range established during design as being that where the efficiency of the cruising turbine falls below that of the bypass stages in the main turbine. The main and cruising turbines are controlled throughout the entire power range by means of a single hand wheel.

A rolling steam supply sufficient to turn the cruising rotor at about 400 r.p.m. is incorporated in the first nozzle control valve which prevents rotor hogging when the cruising turbine is disengaged.



By courtesy of the Parsons Marine Turbine Co., Ltd.

FIG. 19—*Photograph taken during shore trials, Y.100*

Overspeeding of the cruising turbine is prevented by the emergency stop valve which automatically shuts when tripped by the shock proof emergency overspeed governor or by the hand trip lever.

Cruising Turbine

The cruising turbine, an all-impulse design, comprises a two-row Curtis wheel followed by eight single impulse stages.

A gashed type rotor with discs machined from the solid forging carries the moving blades.

The steam chest is cast as an integral part of the top half casing and contains the emergency stop valve and four mechanically operated nozzle control valves, each of which supplies steam to individual nozzle groups.

The exhaust steam is led through two cross-over pipes into the fifth stage of the main turbine.

All diaphragms are of built-up construction, the nozzle segments being machined from the solid and fixed to mild steel centres into which the gland sleeves are dovetailed.

The first-stage nozzle plate with its four groups of nozzles, all in the top half, is machined from the solid, the segments being integral with the centre, thus minimizing steam leakage.

A longitudinal section through the cruising turbine is shown in Fig. 20.

Main Turbine

The main turbine comprises a single-flow cylinder with a gashed type of rotor of eight single-impulse stages and an astern two-row Curtis wheel. The ahead and astern cylinders together with their common rotor are incorporated in a fabricated casing which in turn is built integrally with the condenser shell.

The astern cylinder is flexibly mounted on radial pins and supported on an extension of the aft end gland housing.

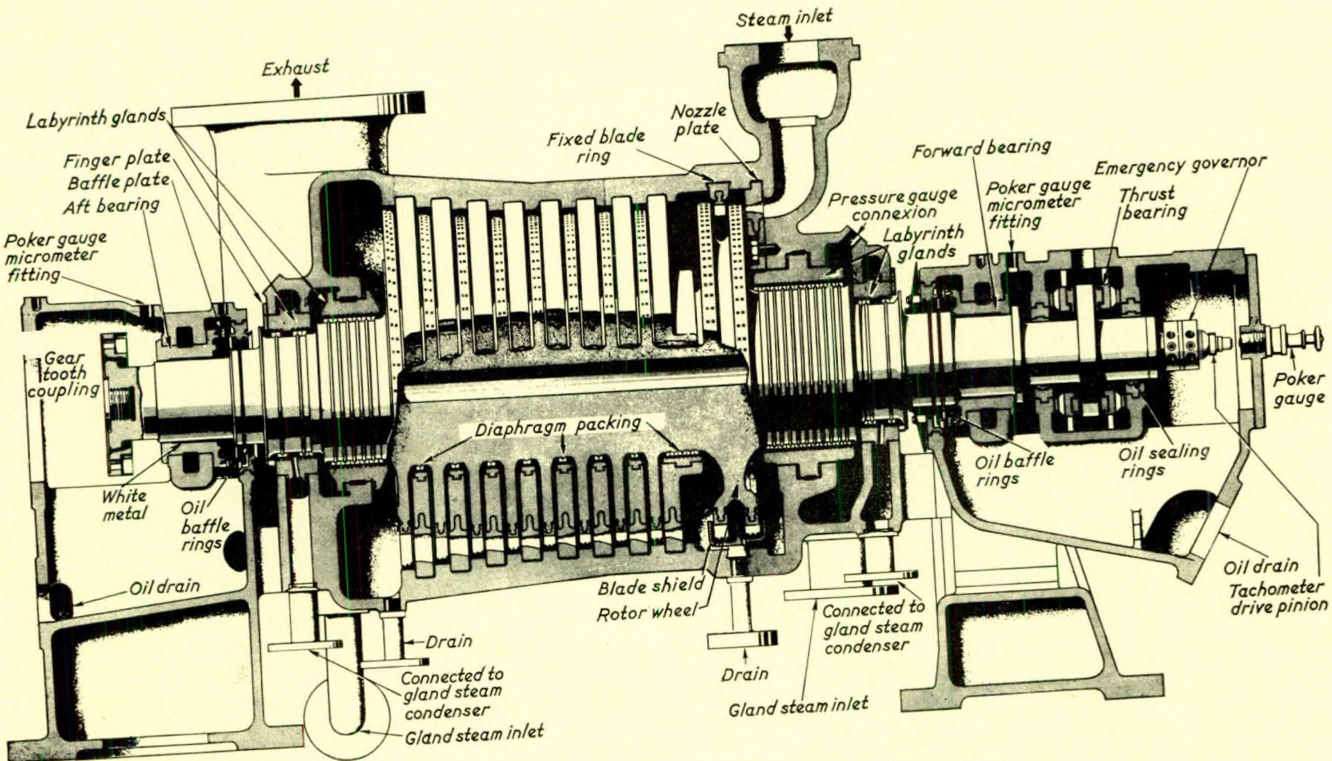


FIG. 20—*Longitudinal section of cruising turbine, Y.100*

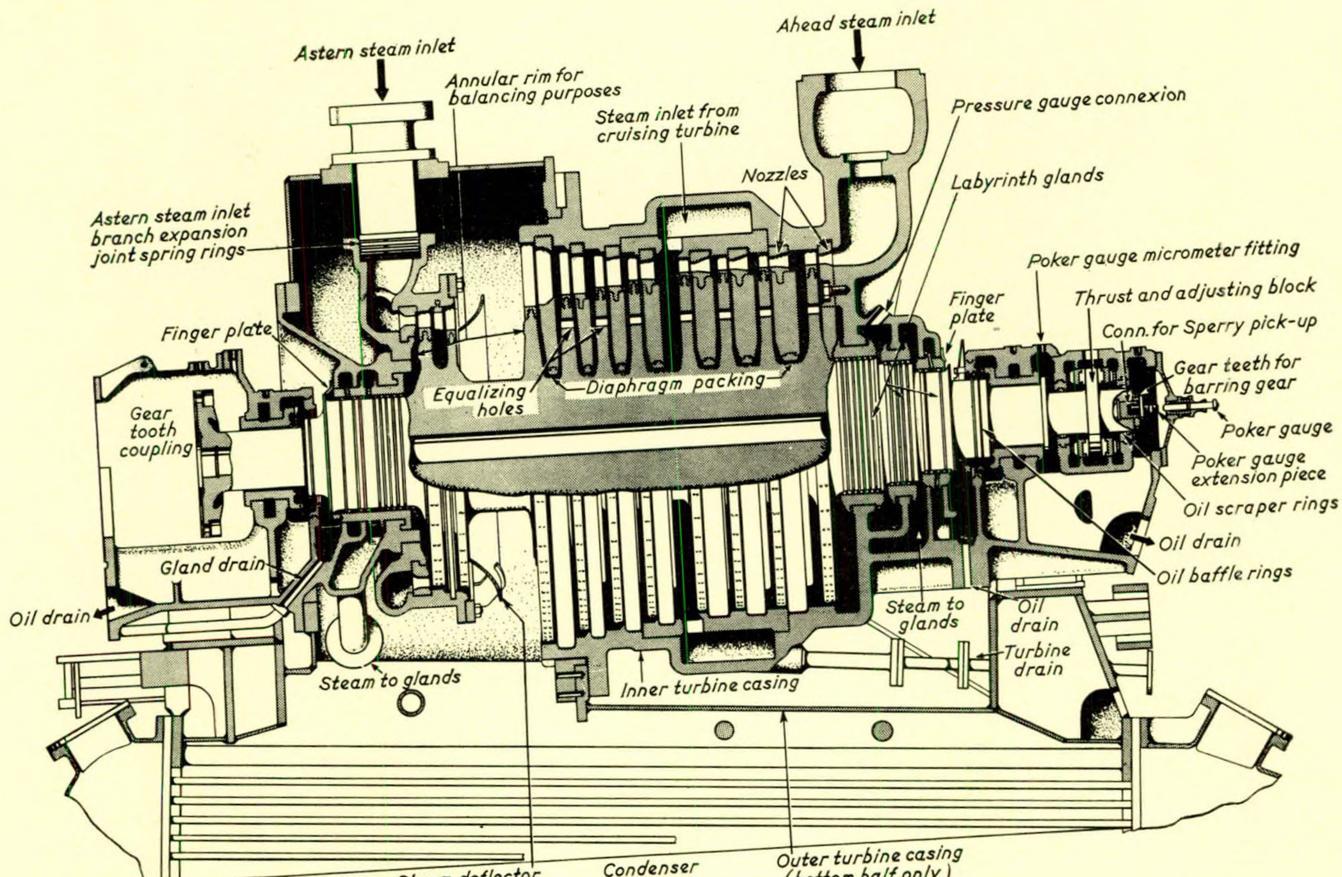


FIG. 21—*Longitudinal section of main turbine, Y.100*

A deflector ring fastened to the astern cylinder reduces losses and minimizes blade vibration by ensuring that exhaust steam from either cylinder does not impinge on the last blade of the other.

The steam chest is cast integrally with the top half-cylinder and houses the four nozzle control valves, each of which supplies steam to the individual groups of nozzles.

The astern control valve chest is mounted on the exhaust end and is bolted to the flange of the astern steam inlet branch.

The diaphragm of the first seven stages are of built-up construction similar to those of the cruising turbine, whereas the last diaphragm consists of a cast steel centre with preformed division plates cast in.

The first stage nozzle plate with its four groups of nozzles is of built-up construction and bolted to the top half cylinder, steam admission being to the top half only.

The full admission astern nozzle plate is machined from the solid.

A longitudinal section through the main turbine is shown in Fig. 21.

Materials

Rotors

Both cruising and main rotors are of 3 per cent chrome molybdenum steel to Admiralty Specification No. 3 F.1.

Blading

All moving blades, shrouding and lacing wires, where fitted, are of stainless iron.

Nozzles

All nozzle segments are of stainless iron attached to mild steel centres; the cruising turbine nozzle plate is machined from a forging of stainless iron.

Steel Castings

The ahead and astern cylinders are cast from 0·5 per cent molybdenum steel. Pedestals, gland housings, bearing blocks, etc., are machined from plain carbon steel castings.

The main fabricated casing and exhaust hood is made of mild steel plate.

Condenser

A single flow underslung condenser is incorporated in each main unit assembly. Considerable reduction in weight and space has been achieved in the overall design by adopting a circulating water velocity through the turbines of 10ft. per sec.; also by unit construction of the main turbine lower half casing and condenser shell, and by the acceptance of a vacuum of 23-in. Hg. at full power in tropical waters (cooling water temperature 85 deg. F.). A suitable margin on design has been allowed to enable the full vacuum to be obtained with the condenser in a fouled condition.

The cooling water supply for normal ahead running is provided by scoop, boosted as necessary by the main turbo-driven circulating pump.

The cooling water inlet pipes and water boxes are designed to give a smooth uniform flow entering the tube plate and to minimize the possibility of erosion caused by turbulence, eddies and friction.

The shell, together with its external ribs, internal gussets and washplates, is of welded construction, fabricated from steel plates and bars.

The tube plates are of rolled Naval brass.

The tubes, $\frac{5}{8}$in. outside diameter × 19 SWG., are of 70/30 cupro-nickel secured by Crane Wilkie cone type packings at the inlet ends and Crane full-box type packing at the return ends without use of ferrules. Two sagging plates are fitted between the tube plates.

Unrestricted flow of the steam over the entire length of the tubes is facilitated by the unit construction of turbine exhaust casing with the wide boat shaped shell.

Approximately 18 per cent of the tube surface is separated from the main bank of tubes by baffles of steel plate construction running the full length of the condenser and forms the air cooling section.

The water boxes and end covers are combined and made of gunmetal, although on later single screw ships installed with Y.100 Mk. II machinery these are fabricated from $\frac{1}{4}$-in. thick mild steel plate, the surfaces in contact with sea water being rubber coated. This modification also embodies a reduction in size of the water boxes, the overall effect being a considerable saving in both weight and space. Steel corrosion plates are secured to the tube plates.

Valve Gear

In order to minimize losses due to throttling, both cruising and main turbines are each provided with four valves for the supply of steam to individual nozzle groups. The nozzle groups are designed to provide the following nominal power outputs:—

TABLE XV.

Group numbers	Cruising turbine percentage of full power	Main turbine percentage of full power
1	5	30
1 and 2	10	45
1, 2 and 3	17	65
1, 2, 3 and 4	25	100

Referring to Fig. 18, the control valves for the cruising turbine and main turbine are numbered (7) to (10) and (11) to (14) respectively. Item (15) indicates the astern control valve and item (16) the cruising turbine stop and emergency valve.

The nozzle control valves are entirely cam operated and are all of the spherically seated Venturi type design for high steam velocity at the throat, with a diffuser exit, which serves as a velocity to pressure conversion tube. This design makes it possible to use a relatively small valve whilst maintaining a small pressure drop across the valve.

Fig. 22 illustrates the design of valve and control gear for the cruising turbine; that for the main turbine is essentially the same.

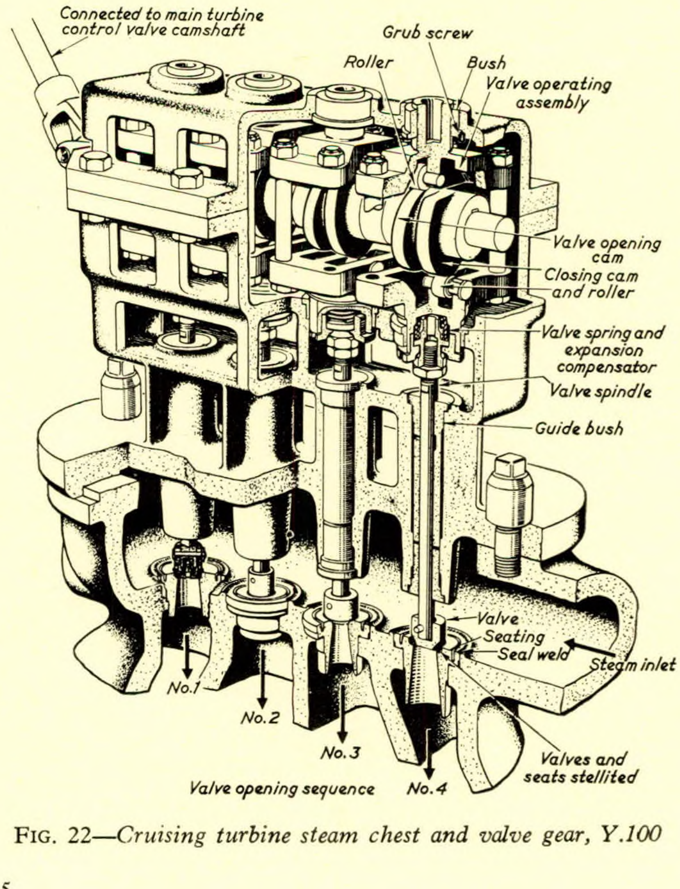


FIG. 22—*Cruising turbine steam chest and valve gear, Y.100*

It will be noted that cruising turbine no. 1 valve incorporates a pilot valve which supplies the rolling steam mentioned earlier. A balance valve is incorporated in no. 4 main turbine and astern turbine control valves in order to reduce the effort required to open them, as, being relatively large, the unbalanced forces due to steam pressure would otherwise be excessive.

The main and cruising turbine control camshafts are connected by Hardy Spicer couplings and are rotated simultaneously by the ahead manœuvring wheel through suitable shafting and a worm drive.

Operational Characteristics

The curves shown in Fig. 23 illustrate the nozzle control valve motion relative to the angular movement of the camshaft over the power range.

Referring again to Fig. 18, speed is increased by turning the ahead hand wheel in the opening direction, which opens the four control valves (7), (8), (9) and (10) sequentially until at approximately 25 per cent power all four are fully open. Until this point is reached, the steam from the cruising turbine exhausts into the main turbine, bypassing the first four stages. Therefore all the power is produced jointly by the cruising turbine and the last four stages of the main turbine.

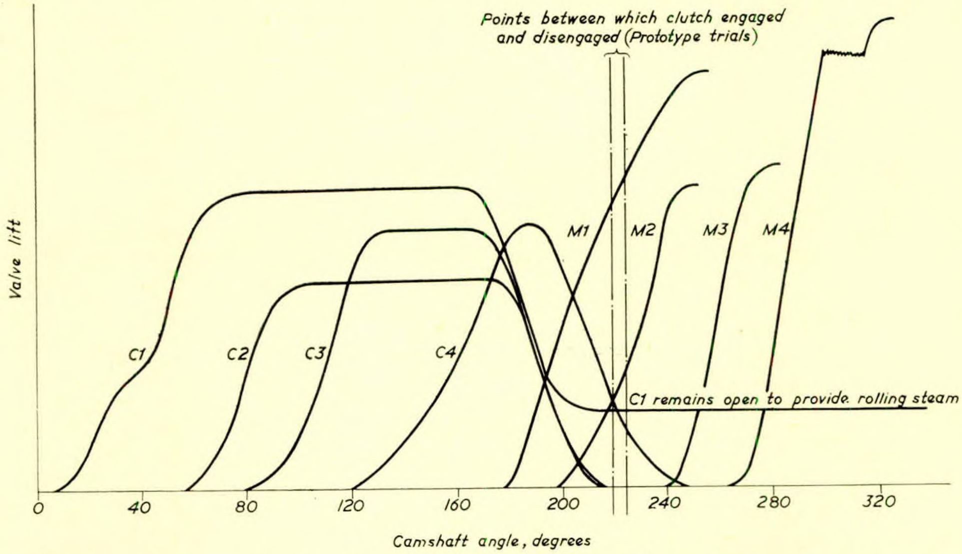


FIG. 23—*Curve showing nozzle control valve motion relative to the angular movement of the camshaft, Y.100*

Further opening of the hand wheel rapidly closes valves (7), (8), (9) and (10), simultaneously opening the first main turbine control valve (11), followed sequentially by valves (12), (13) and (14); thus at all powers above approximately 33 per cent all steam is admitted to the first-stage nozzles of the main turbine except for a small quantity admitted through the small pilot valve incorporated in no. 1 cruising turbine control valve. Between approximately 25 per cent and 33 per cent power, steam is admitted simultaneously to the nozzles of both cruising and main turbine in order to produce a smooth change when transferring power from the cruising turbine to the main turbine and *vice versa* when decreasing power.

The above sequence of operation is reversed when it is required to reduce speed.

The clutch designed and manufactured by D. Napier and Son, Ltd., operates automatically; when increasing power it disengages as soon as the steam quantity to the cruising turbine is insufficient to maintain it when running at the synchronous speed of the main turbine.

When reducing power the clutch engages when the steam flow to the cruising turbine is sufficient to take it up to the synchronous speed of the main turbine and it will remain engaged so long as there is a power output from the cruising turbine. The speed in r.p.m. at which the clutch engages or disengages is dependent upon various factors, namely, the valves, cam profiles and phasing of the main and cruising turbine camshafts, combined with cruising turbine efficiency, and these items are so designed as to produce disengagement and re-engagement of the cruising turbine at an appropriate speed consistent with safe stress levels and adequate safety factors on all moving parts.

Incorporated in the clutch is a device whereby the clutch can be withdrawn manually and held in the disengaged position. This action is effected by winding a hand wheel to the disengaged position, the first movement of which trips the cruising turbine emergency stop valve; this shuts off all steam to the cruising turbine and so prevents the overspeeding which would otherwise occur due to removal of the load following clutch disengagement.

The clutch is put into the manually withdrawn position when the ship is undergoing operations involving rapid manœuvring, also when steaming astern. In the latter case, as the turbines run in the reverse direction, the clutch would, if permitted, engage automatically and therefore the cruising turbine would overheat due to windage and disc friction during prolonged high speed astern operation. In the event of any damage occurring to the cruising turbine, it can be disconnected as stated above, in which case the ship is capable of all operational requirements, power being obtained from the main turbine alone, the only effect on performance being a considerable increase in the steam consumption at cruising powers due to the decrease in overall efficiency of the main engines and the throttling losses that occur below 27 per cent power when the four cruising nozzle control valves are inoperative.

The cruising turbine can be re-engaged by winding the clutch hand wheel to the "engaged" or "automatic" position at any time except when running astern, following which the emergency stop valve can be reset and steam admitted to the cruising turbine *via* the nozzle control valves.

Summary of Shore Trials

Table XVI shows the steam consumption over the power range examined during the shore trials at Pametrada, together with the appropriate steam conditions, vacuum, etc., and including the gear efficiency.

In two columns the consumption figures have been omitted as no consumption tests were carried out at those powers. However, Fig. 24 shows the steam consumption over the full power range, the two missing points being estimations.

Referring again to Fig. 24, the upper dotted line shows the steam consumption at cruising power when running with the cruising turbine disconnected and it will be noted that at

TABLE XVI.

	Cruising turbine (clutch engaged)				Main turbine (Cruising turbine declutched)			
Power, per cent	5·32	10·07	18·19	27·86	33	45	65·8	109
Power, s.h.p.	798	1,510	2,720	4,180	—	—	9,850	16,320
Steam pressure, lb. per sq. in. absolute	594	592	590	585	583	573	554	409
Steam temperature, deg. F.	827	827	827	827	827	827	827	827
Vacuum (Bar. 30 in.) Hg.	27·41	26·88	27·34	26·78	26·59	26·0	25·15	23·0
Gear efficiency, per cent	92·5	92·95	93·3	94·3	—	—	95·94	96·7
Steam consumption, lb. per s.h.p. hr.	8·93	8·57	8·24	8·24	—	—	7·85	8·03

8 per cent power the increase in consumption is about 38 per cent. Between 27 per cent and 33 per cent power with the cruising turbine in operation the curve is shown dotted, as within this range steam is admitted directly to both main and cruising turbine nozzles.

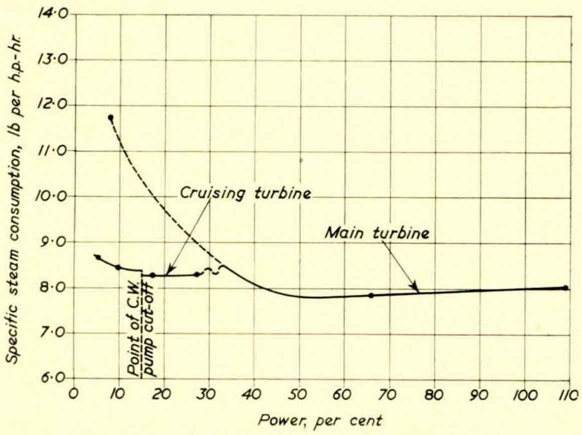


FIG. 24—*Steam consumption in Y.100*

An accurate assessment of the steam consumption, particularly at low powers, proved rather more difficult than expected, the main factor in this respect being due to the very wide divergence of opinion regarding gear efficiencies, particularly at the important point of 5 per cent power, this variation being 85 per cent to 92·5 per cent. As the accuracy of the turbine performance figures can be no better than the gear efficiencies upon which they are based, the latter obviously required some investigation. The conclusions and very considerable amount of work involved by the various authorities concerned in the investigation is outside the scope of this paper except to state that final figures of gear efficiency were agreed upon and the above table of performance incorporates the requisite correction factors.

The shore trials revealed one or two features that entailed modification, which is to be expected when testing machinery of such an advanced character for the first time. So far as the turbines are concerned, these were of a comparatively minor character and were chiefly concerned with the valve gear and its operating mechanism.

During the manœuvring trials, the curve of power *versus* valve camshaft angle exhibited undesirable characteristics in that it was found that the intended smooth changeover from the cruising turbine to the main turbine and *vice versa* was not obtained. This was overcome by suitable modification to the cams operating the valves at the appropriate portion of the power range and entailed a fair degree of overlap being provided in the valve opening, as shown in Fig. 23, where it can be seen that as the changeover point is approached, no. 2 valve on the main turbine is partially open in addition to no. 1 main and no. 4 cruising valves.

The shore trials demonstrated satisfactorily that the Y.100 turbines fulfilled their performance requirements, including that of rapid manœuvrability. The automatic clutch performed its duties satisfactorily although it was realized that actual service conditions could not be reproduced during shore trials.

Sea Trials Experience

Sea trials on two or three ships have now been successfully completed except for the fact that the automatic clutch has had to meet various conditions which could not be simulated during the shore trials, e.g. the continuously smooth power/speed characteristic, the effect of a trailing propeller, also that of pursuing a zig-zag course at a speed which causes the clutch repeatedly to engage and disengage.

In order to meet these and other conditions it has been clearly demonstrated at sea that the clutch requires some considerable modification and additional safeguards, both of which are being carried out in order to preserve the very attractive automatic features inherent in the design.

Conclusions

That the Y.100 turbines fulfil the requirements of performance, particularly at very low partial powers, was adequately proved during the shore trials and sufficient experience has been obtained at sea to show that they satisfactorily perform the duties for which they were designed. There is little doubt that the difficulties which have been encountered relating to the clutch will be overcome by the modifications which are now being undertaken.

One aspect relating to the performance of Y.100 that is perhaps of major importance, viz., operation of the nozzle control valves combined with clutch engagement and disengagement is automatic, therefore the trial consumptions will be obtained in practice, whereas previous installations of a similar nature relied on manual operation and test consumptions could only be repeated in service if conditions favoured the operation of the nozzle control valves and cruising turbines in the desired manner, which of course in service was frequently impracticable.

Various manufacturing difficulties have been encountered but the majority of these result from the fact that, due to emergency, production had to be commenced before the completion of final detailed drawings, indeed before completion of general design details. In this respect it should be pointed out that several turbines built by various manufacturers were in the final stages of their assembly before the end of the full-scale shore trials of the prototype at Pametrada.

The requirement of light weight machinery has been achieved, as shown by the following table of weights. A comparison of the Y.100 machinery with that of the *Dido* Class cruisers, both of 15,000 s.h.p. per shaft, is shown, from which it can be seen that the reduction in weight is well in excess of the stipulated 33 per cent.

TABLE XVII.

	Y.100	Dido class
Turbines complete, tons	19·93	30·41
Gearing, tons	15·25	17·65
Condenser (wet), tons	6·97	17·43
Total	42·15	65·49
Weight in lb. per s.h.p.	6·3	9·77

As in all cases of machinery of advanced design, only operational experience and time can finally provide the proof of reliability, etc., but, as shown in this paper, there appears to be little doubt that these essential requirements will be achieved; that this opinion is shared is perhaps indicated by the large number of firms manufacturing this machinery under licence in this country, Canada, and in Australia.

ACKNOWLEDGEMENTS

A paper of this type involves the assistance of many people and in this connexion thanks is expressed to:—

The Department of the Engineer-in-Chief of the Fleet for assistance given and for permission to publish the information contained in the paper.

The English Electric Co., Ltd., for providing the facilities for its preparation.

Rear-Admiral I. G. Maclean, C.B., O.B.E., for his helpful suggestions during the preparation of the paper.

Yarrow and Co., Ltd., Pametrada and the various shipbuilders concerned for their co-operation during the shore and sea trials.

Discussion

REAR-ADMIRAL I. G. MACLEAN, C.B., O.B.E., (ret.) (Member) said that it gave him great pleasure to open the discussion on Mr. Veitch's interesting and constructive paper. Obviously, the pleasure was tinged with sadness that Mr. Cowlin was unable to be present himself. The Royal Navy owed a very great deal to Mr. Cowlin and his team, and his sudden and unexpected death was a great shock to his friends.

He hoped that it might be significant on the occasion of the presentation of this first technical paper in the Memorial Building of the Institute that the subject was the development of naval turbines. Whether by accident or design, it surely emphasized the importance of turbines to the marine industry and perhaps also illustrated one of the rôles the Royal Navy could play in this field—that of the pioneer whose advances could benefit all who cared to profit by them.

As his name had been mentioned by Mr. Veitch, he ought to make it clear that he had now retired and no longer had any responsibility for naval development. It was true that at the time of the first design he was the Commander in charge of the turbine and gearing desk at the Admiralty and, as Mr. Veitch had mentioned, was the chairman of the turbine and gearing committees responsible for the genesis of the original design. But he would like to emphasize that none of the results achieved by the turbine section of the Admiralty or the committees or subsequently by Mr. Veitch's firm could have come about had there not been a close degree of co-operation—co-operation between individuals, between firms and between representatives of the United States Navy and themselves as representatives of the British Navy. In the latter connexion, he would like specially to mention the help received from Admiral Mills of the United States Navy, who went out of his way to see that the experience of the United States Navy in this whole field was put at the disposal of the Royal Navy, so that they might cut many corners and avoid some of the inevitable troubles which beset the United States Navy in its earlier work.

Mr. Veitch's straightforward account of the evolution and basic features of a series of turbine designs for the Royal Navy illustrated how much painstaking work went on in this kind of development and into the eventual choice, a choice which had—he thought he was right in saying—an increasing influence on ship design.

He did not think he could make any comment which would adorn the work that had been described, except perhaps to say that it was the production of an able team working in the closest harmony with the Admiralty, a harmony which was virtually essential to the satisfactory outcome of this kind of work. At the same time, he would like to congratulate Mr. Veitch on his very clear paper and marvel that he had been able to compress the anxious thought of so many people into such a succinct account. He would leave it to others to tease him about the many interesting points of design that had to be faced and overcome.

One feature of their working together might be of interest. They had battles over most of the critical features of the design. It was a cardinal feature, however, that the Admiralty never imposed its will on the designer. The matter in dispute was thrashed out until one side or the other was convinced. He believed there was no single point on which they failed eventually to reach agreement. It might sound a time consuming process but, he would suggest, it was well worth while.

Before sitting down, he would like to touch on one or two points. The first was the tremendous sense of urgency they were all working under at the stage of the war when the *Daring* concept was born. For one reason or another, as Captain Raper had recently reminded him, all this work was carried out under conditions of urgency.

In considering the paper, it was important to remember that the designs were essentially naval designs and that some of the special features which made them unusually difficult would not apply to commercial turbines. He referred especially to the high part-load performance ultimately required necessitating cruising turbines in the Y.100 designs.

Mr. Veitch had drawn attention to a number of other qualities in these designs—light weight, small bulk, shock resistance, reliability, and so forth. He had not, however, mentioned cost. It would be interesting to know how the cost of this very special machinery compared with its more conventional counterpart of comparable horse power—if it could be called comparable. It would also be nice to hear from those now serving how the ships and the turbines were behaving.

There was no doubt that the adoption of solid forged alloy steel rotors, which was regarded at the time as a bold if not foolhardy step could only have come about by the close collaboration given by the forgemasters. It was quite remarkable. Indeed, the history of the development of the rotor steels and so on would form an interesting paper.

Mr. Veitch referred to the sequential operation of the nozzle valves from the manœuvring platform. This was an important point. When the plans were discussed, Admiral Mills had told him that the greatest single gain the United States Navy had made compared with older ships was not the adoption of higher temperature or pressure, not even the adoption of special turbine designs or double reduction gearing or improved heat balance, but the adoption of means such as the sequential operation of nozzle valves to ensure that at all times the machinery of their warships could only be operated in the manner the designer intended, thus giving at all times a performance matching that for which it was designed. This was a very useful lesson and worth passing on.

One final point worth mentioning was that the type of turbine design described involved the adoption of double reduction gearing, a step regarded with misgiving by quite a lot of people, due no doubt to unhappy experiences earlier in the century when design and production techniques were not so well established. The stage had, he hoped, now been reached when fears about gearing could be dismissed.

Having been involved in this sort of thing for quite a long time, he could go on talking for a long time, but he thought it better now to leave the discussion to others.

MR. B. J. TERRELL, M.B.E., B.Sc., said it was a great pity that Mr. F. J. Cowlin, with whom he was once associated and towards whom he always felt the greatest personal regard, was no longer with them. In this paper, which had been so ably completed by Mr. A. F. Veitch, he had described a very real contribution to the advancement of British naval engineering.

At the same time, it might also be fitting to mention the name of the late Mr. H. G. Yates, who was so largely instrumental in the conception of the S.T.N. 2 project in the early days. He felt sure that Mr. Cowlin would never have wished him to be forgotten.

In speaking of the advanced projects for high temperature steam, the paper appeared to concern itself exclusively with the effect of these high temperatures on the turbine design. Certainly where creep strength was the criterion it was an advantage to combine the highest temperature with the lowest stress by using the lower temperature at full power when the centrifugal forces were highest, and the highest temperature at cruising power when the stresses were lowest. He felt, however, that the limit to steam temperature would not be set so much by the turbines as by the superheater. It was well known that both oxidation and vanadium-corrosion rapidly worsened with increasing temperature and even with austenitic steels which were fairly safe from the oxidation point of view vanadium-corrosion started to be serious at about 1,150 deg. F. He would suggest that with steam at 1,100 deg. F. the metal temperature in a highly rated naval type boiler would be considerably in excess of 1,150 deg. F., even under cruising conditions. With this type of design, the superheater would spend most of its time (i.e. at cruising power) at this elevated temperature, so that for the boiler it would appear that the system was the wrong way round.

It was stated there was but little difference between the specific steam consumptions for *Daring I, II* and *III*, but it was his impression that *Daring I* gave somewhat better figures. He did not say this in any derogatory sense, but he would like to couple with it the other statement in the paper that in the case of the impulse h.p. cylinder of *Daring III* the axial clearances were at least five times greater than for the *Daring I* machinery. From certain measurements that had been made, he would suggest that the importance of clearances in impulse turbines was apt to be underestimated and that had *Daring III* been built with finer clearances it might well have surpassed *Daring I* in efficiency. He would also make reference here to the published work* of Mr. J. L. Jefferson in a paper entitled "Some Practical Effects of Tip Clearance in Turbine Blading".

Fig. 24 of the paper showed the specific steam consumption of the Y.100 machinery, plotted against power, as measured during the shore trials at Pametrada, suitably corrected for the agreed gearing losses—*the agreed gearing losses*. The major proportion of so-called gearing losses were in fact due to bearing friction and it might be of interest to point out that these losses were greatly reduced by the use of short bearings of about 1/3rd L/D, as recommended by Pametrada which were, to the best of their knowledge, performing very satisfactorily in service.

The design of the clutch for this installation was very much more difficult than seemed to have been at first anticipated. Basically, all that was necessary was a bicycle free wheel, but running at very high speeds and powers. He wondered if Mr. Veitch was not being a little over-optimistic on this subject in his conclusions to this excellent—and he meant excellent—paper.

CAPTAIN R. G. RAPER, R.N., said Mr. Veitch had dealt with turbine designs which had a familiar look to them now. They were straightforward high-speed turbines, using gashed solid, low alloy forgings for rotors and driving through double reduction gears. One's first impression was that there was little more to be added to the excellent account he had given, but if one went back fourteen years to where he started, these things were new to many people in the marine field. They did indeed have the example of the United States Navy developments, but what applied to their steel production industry might well not be attainable in this country in war time when everyone was so hard pressed.

* Jefferson, J. L. 1954. "Some Practical Effects of Tip Clearance in Turbine Blading". Trans.N.E.C.Inst., Vol. 70, p. 419.

It would have been very easy to play safe. It was thought by many that it would be going a long way just to adopt double reduction gears without alloy steel rotors. Indeed, the original decision to fit reaction turbines in the *Daring* class was swayed a good deal by the desire to stick to something familiar. There had been much trouble with blade vibration in impulse blading during the war and people had to be convinced that the causes were fully understood.

But the English Electric Company "limit" design which became *Daring III* made the bold move of taking full advantage of the added strength of the 3 per cent chromium molybdenum rotor material.

Mr. Veitch had mentioned that it was between December 1943 and April 1944 that the turbine sub-committee which originally examined these matters met, considered various designs and made their recommendations. In this four or five months Mr. Cowlin had not only given the Admiralty what it had asked for; he had shown the committee what it ought to ask for and had in fact pointed the way to a new generation of naval turbines. Without belittling the great contributions made by others in the development of this machinery, it was no exaggeration to say that it was very largely Mr. Cowlin's technical ability, courage and wisdom which gave them the necessary confidence to step off into what were new fields for them at that time.

The very familiarity of this type of turbine now bore witness to the soundness of this approach.

The authors had set out the requirements which their firm were called upon to meet in each design. As usual, it was clear that the customer wanted the best of every world in each case. Having set difficult targets for weight and space, he demanded ease of operation, ease of maintenance and ease of production. The designer might or might not find that these conflicted with one another but some certainly would, and it was perhaps worth trying to weigh up their relative values. The ship as a tactical weapon was what mattered, and so the attainment of space and endurance targets was essential if she was to do her job properly. The ingenuity of operators was reasonably great but if their job was made really easy it would add greatly to the effective use of the ship on service and so this stood high in the order of needs. Ease of maintenance affected the availability of ships and so it was of great importance in wartime, but probably came level with the need that the ship should excel at its job when operating. So one came to ease of production.

This was important for two reasons; the time it took to produce the machinery, especially in an emergency, and the cost of it to the taxpayer in peacetime. His own belief was that the period immediately after a war was a time when ease of production should not be allowed to stand in the way of achieving the highest performance in other directions. What people called difficult now would be commonplace after it had been done for several years, but the ship and those who operated and maintained it would suffer for its whole life if other qualities were sacrificed to ease of production. Likewise, what one firm regarded as difficult another firm would overcome without much fuss and the organization of production would have a profound effect on what people called "easy". Y.100 turbines were being made in Canada and Australia as well as in this country. The production was centralized in both these countries and the standard of production that these turbines demanded seemed to be taken for granted as something attainable and justified.

To sum up, the small size, high stressing and performance of these turbines had demanded a fairly high standard of production, but with the general advance of engineering it did not appear to be an unreasonable level to expect in view of the great advantages it had bestowed on the performance of the ships concerned. This machinery was unquestionably easy to operate. The maintenance of main turbines was made easier by their small size but complicated by the nozzle valves. However, these did not seem to be a real problem.

He therefore believed that the balance of these qualities, which had to some extent complicated production, was of the right sort. The real test of these designs was, of course, on service. Here, as far as experience went, they had fulfilled and in some respects exceeded what was hoped of them.

MR. E. NORTON, B.E.M. (Member) said it was a privilege to take part in the discussion on this valuable paper and in particular to pay tribute to the outstanding contribution to marine turbine design made by the late F. J. Cowlin.

The paper outlined fifteen years of effort on the development and production of a completely new and modern range of naval turbine units. Among other achievements this work had resulted to date in the evolution of propulsion machinery for anti-submarine frigates of the Royal Navy, which was second to none throughout the world.

The paper was refreshingly frank concerning the difficulties which had been experienced during the course of the work. These were probably less than might reasonably have been expected in such a revolutionary development programme, and they certainly could in no way detract from the outstanding advances that had been achieved.

The turbine machinery available to the Royal Navy fifteen years ago and used throughout World War II consisted almost entirely of low speed turbines driving through single reduction gearing and with the best efficiency at or near full power. Rotor forgings were of the drum type, forged in two pieces and secured with a pinned and shrunk joint. This type of construction had clearly reached its limit from many points of view. In particular, no further advances could be contemplated beyond the maximum power of 37,500 s.h.p. per shaft as fitted in aircraft carriers, and even this limit was regarded by many as having been pushed too far. Likewise, the large bulk and weight of the rotors and turbine casings prohibited any serious consideration of advancing steam conditions much beyond the then prevailing 350 per sq. in. 700 deg. F. Operating experience was already showing that the thermal stresses and distortions associated with these large and heavy turbine structures had more than reached the practical limit.

The development programme covered by the authors in this paper had brought about some profound changes. There were now available high speed turbines driving through double reduction, and in some cases treble reduction, gearing. The rotors were of the solid forged gashed type in alloy steel. Reference to Table II, page 501, indicated that the specific weight of the turbines had been reduced by nearly 50 per cent, and the reduction in bulk that had been achieved could be judged by the 40 per cent reduction in turbine bearing span. This reduction in bulk and weight had enabled advanced steam conditions to be applied with safety. Similarly, the best efficiency point of the turbines had been moved down to the 20-40 per cent power zone where it was most required in naval installations; and once again, referring to Table II, it was seen that steam consumptions at 20 per cent power had been improved by up to 20 per cent.

As far as the machinery for anti-submarine frigates was concerned, the turbine developments described in this paper, together with similar developments associated with the other major components, had resulted in an outstanding overall achievement. As mentioned by the Engineer-in-Chief of the Fleet* in his Parsons Memorial Lecture last year, a development expenditure of 1 per cent of the capital cost of the anti-submarine frigates building and ordered had resulted in the overall cost of each ship being reduced by about 10 per cent. This had been achieved by the saving made on the size of the ship due to the reductions in weight and space of the machinery.

The turbine developments described in the paper added up to an impressive record of which the author, and those associated with him in his company, could be justly proud.

COMMANDER E. TYRRELL, R.N.(ret.) (Member) said Mr. Veitch was to be congratulated on the detail and wide scope of the subject which he had presented.

Before coming to the main substance of what he had to say, he would like to confirm from bitter experience when he was in the Engineer-in-Chief's Department in charge of the big ship new construction section the statement by Mr. Norton that the turbines fitted in the large aircraft carriers had, in fact, reached the limit of their design. That had caused him many sleepless nights, much travelling and other worries in trying to get them to run satisfactorily.

It was interesting that Mr. Veitch considered that the poor performance of Royal Naval vessels compared with their American counterparts during the last war was caused by the shortage of financial backing for development work generally in the marine industry. Would Mr. Veitch let them have his opinion on whether this serious shortcoming had now been corrected, observing that there was an unparalleled boom in shipbuilding at the moment and also observing that the contents of the White Paper on Scientific Manpower 1956 seemed to indicate that the number of qualified and professional engineers, technologists, scientists, etc., employed by the shipping and shipbuilding industries was considerably lower than in industries of comparable importance in this country?

On page 498 Mr. Veitch indicated that by the change to all-impulse turbines increased reliability was likely to be achieved with the *Daring III* design as opposed to the other two designs mentioned. Had the hoped for increase in reliability in fact been achieved and had the *Daring III* design been less free from troubles than the other two?

On page 501 he made mention of the leakage troubles occasioned by the use of built-up nozzle segments. Would his company in the future favour a change to all-welded diaphragm and nozzle construction, which tended to obviate these troubles and also, it would seem, had greater strength in the built-up portions, which was making shorter rotors possible? This was already standard practice in the United States and on the Continent.

He was mainly interested in Mr. Veitch's paper from the aspect of what could be learned from naval practice and applied with advantage to merchant machinery.

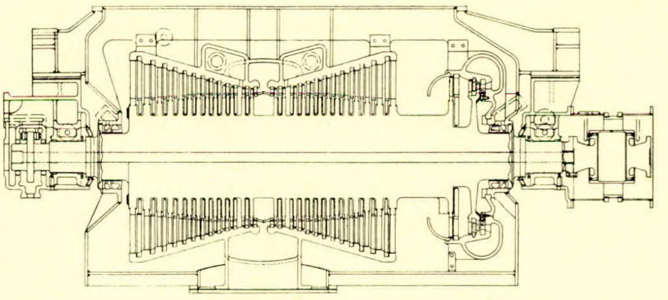


FIG. 25—*L.P. turbine basis plan*

Figs. 25 and 26 showed a conventional set of British merchant service machinery of 22,000 s.h.p. It must be noted that they compared most unfavourably in complexity with the almost comparable Y.E.A.D.I design shown by Mr. Veitch. The very large number of stages would be noticed.

However, the steam consumption of the merchant service set at 100 per cent power—which after all was all the average shipowner was interested in—was considerably lower than the figures quoted for the Y.E.A.D. turbine. Could Mr. Veitch possibly say what he considered the steam consumption of a turbine similar to Y.E.A.D.I, retaining the same number of stages and redesigned to give maximum efficiency at 100 per cent power condition, would be?

The blading of the Y.E.A.D.I turbine was of stainless iron, but the paper indicated that experience had shown the corrosion resistance properties of stainless iron and mild steel under chloride attack to be very little different. Also, it would be

* Mason, F. T. 1956. "A Review of Naval Propulsion Engineering Progress in the Last Ten Years". Trans.N.E.C.Inst., Vol. 73, p. 37.

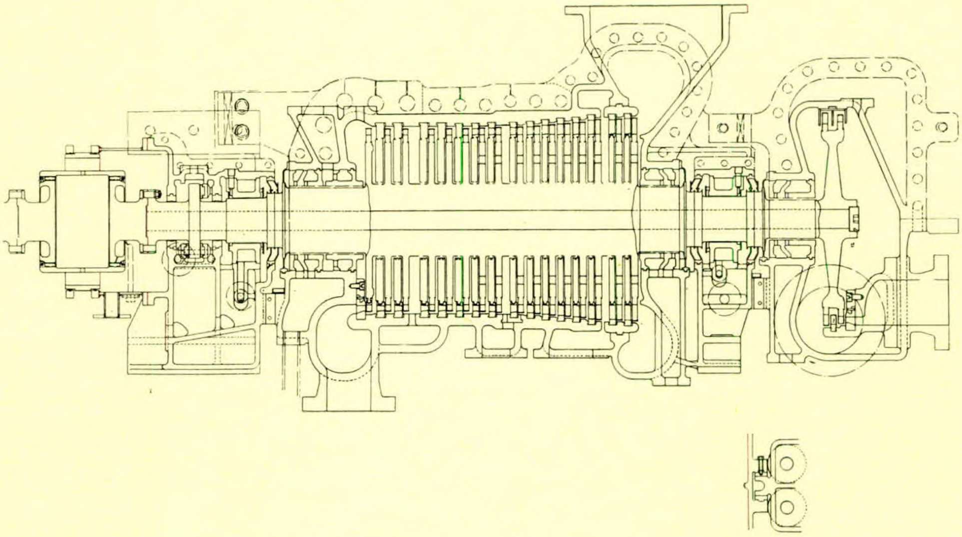


FIG. 26—*H.P. turbine basis plan*

quite untrue to say that water treatment and the quality of water used in ships today was not improving. It had improved greatly during the years. Did Mr. Veitch consider that the extra cost of stainless iron as opposed to mild steel for blading which was lowly stressed could be justified in a competitive market for merchant ship machinery?

COMMANDER L. D. DYMOKE, R.N., said he stood before the meeting as representative of the "pudding-eating department". That was to say, he now sat at the desk where the telephone rang when things blew up or did not blow up in service!

Admiral Maclean and Commander Tyrrell in their contributions had asked for information as to how these machines were operating in service. He himself was far too junior and insignificant to have any connexion whatever with the developments which took place in the design of these machines, so perhaps he came to them relatively unbiased.

Taking the *Daring* class, so far the ships had been in service for periods falling between about 3½ and 6 years. It was only honest to say that during that time there had been some trouble with *Daring I's* which could be ascribed, he thought, to retaining the reaction type blading in the h.p. turbine. There had been no trouble of any significance with *Daring III* in service and equally none with the B.T.H. design. In the case of the Y.E.A.D.I experimental set on trial at Pametrada there had been slight trouble with casing distortion but in other respects the turbines had performed excellently. Y.100 was the current subject of growing experience, experience growing rapidly now.

He thought the Engineer-in-Chief's Department would say that it was too early for Y.100 really to be declared entirely clear of trouble in the turbines, leaving the matter of the clutch at the moment. That was to say, it was too early to be utterly convinced there was not going to be trouble. It was certainly true at the moment that the turbines were operating very satisfactorily and were liked greatly by their operators.

Mr. Veitch had mentioned in his paper that work was still in hand on the clutch, owing to the unforeseen effects met with at sea as compared with shore trials, and this was a fair statement of the situation.

An interesting point from the user angle had been brought into limelight in Y.100. Whereas previously main engine maloperation (in the sense of not achieving the best performance possible) was the factor principally affecting the cruising radius of the ship, such maloperation had now been largely eliminated in the design stage and maloperation (in the same sense) of the remainder of the machinery had come leaping to the fore. In fact, it was possible, so he was informed by the section at Bath which dealt with these matters, to swamp the gains down at the 5 to 10 per cent power level (by making too much drinking water and so on) if the ship operators did not pay a lot of attention to operating their auxiliaries efficiently. Of course, there was more opportunity for doing it unwisely in Naval ships than in commercial plant.

There was one point which seemed to him important. Mr. Veitch in his paper had made the point that all this hung or very largely hung on rotor material. Furthermore, the gains due to increased temperature were very great. This question of material was still exercising their minds very much in the Admiralty. No turbine designer would claim to be able to forecast or define the stresses which occurred during rapid temperature fluctuations in manœuvring in a high-temperature turbine. For example, the temperature gradients and resulting stress patterns occurring where the discs joined the shaft were largely unknown. And this was becoming of increasing importance, because the steel mentioned, for example, for the Y.E.A.D. II (and other steels which could be considered for the purpose) perhaps had less ability to accept violent stress redistribution by plastic movement than steels which had a lower creep value but were of greater—he was tempted to say "ductility" but ductility was not really correct, and perhaps he should say greater capability to redistribute stress. It was not actually defined by any test.

In order to try to clarify this, the Admiralty had in hand at the moment at London University a research contract where, by mathematical relaxation and analogue computer methods, an attempt was being made to forecast in the design stage exactly what stress levels occurred in these areas of doubt where plastic movements could take place. It was, he thought, true to say that not only the Admiralty but any turbine designer who began to get trouble by crazy cracking or other difficulties occurring, where (say) discs joined shafts, would not know what to do about it in design. It was hoped that this method would help them to find out.

Lastly, if he might dare to be very presumptuous, personally he would like to ask Mr. Veitch, as a personal opinion, where he thought the next step in naval turbine design should lead. As a user, and sitting in the chair at Bath now the great battles of the past were over for him and it was what a turbine had to do that mattered.

He should also explain that he was responsible too for gas turbines in the Admiralty and if that coloured his thinking he hoped he would be excused. It seemed to him that even though the turbines described in the paper could be thrown about and power level flexibility brought to a degree undreamed of ten years ago, it was at least arguable that there was room for further improvement. He would like to ask Mr. Veitch whether he thought that decisive moves in perhaps casing construction, and even—dare he mention it?—possibly composite rotors, could be made to enable turbines to be manœuvred even more cavalierly.

Commander Tyrrell had referred to the large aircraft carrier turbines. He might have had a certain amount of trouble getting them to go at the start, but there was still a lot of trouble in keeping them going. It only emphasized once again that at 400lb. per sq. in., 700 deg. F., the limit with these large turbines had indeed been reached.

MR. R. J. HOOK (Associate Member) said he had a few questions to ask on a specific point in the paper.

On page 501 of the paper it was noted that the authors referred to the failure of blading in the h.p. cylinder due to intergranular penetration by copper during brazing operations. Would Mr. Veitch kindly state in which stage or stages of the turbine these failures occurred and whether both rotor and stator blading were equally affected? Or was the fault more pronounced in either the rotor or the stator blading?

Would Mr. Veitch also state the length of time for which the turbine was under test before these faults were discovered?

He would like to know what appearance these faults exhibited. Were they visible to the naked eye? Were there surface cracks, and if so, what was their magnitude? Or were special means required to detect these faults, and if so, what were the methods employed?

Finally, it would be of interest to learn the chemical composition of the Hecla A.T.V. blading material, the brazing temperature and the subsequent heat treatment, if any.

Correspondence

REAR-ADMIRAL J. G. C. GIVEN, C.B., C.B.E., (ret.) (Member) wished to pay tribute to the work of the late Mr. F. J. Cowlin in respect of progress in naval turbine design described in the paper, and also to congratulate his co-author on such an interesting paper.

Having been personally associated with the beginning of the Y.E.47A investigation, he naturally found the paper particularly interesting, though he did feel the information contained therein could, with advantage, have been published appreciably earlier.

He would like to mention some general conclusions which, in his opinion, emerged from the Y.E.47A investigation:—

(1) The investigation could, with advantage, have been carried out at least ten to fifteen years earlier, when it might have had some profound effects on the naval machinery designs used in the last war.

(2) The design of a marine steam machinery installation, as a whole, was often more important than the design of the individual components.

(3) A team of specialists in the various component designs could, if well led, work together and produce a better overall answer for any important design than was achieved by the loose correlation of components which too often occurred.

(4) The investigation revealed that the customer often did not appreciate either the most important requirements to be specified, or the true effect of the many parameters in the ultimate overall answer.

If there was any agreement in the substance of these last three conclusions, he suggested that, in the national interest as a shipowning and shipbuilding country, they should take them more seriously to heart in the future.

REAR-ADMIRAL L. A. B. PEILE, C.B., D.S.O., M.V.O., commented that this paper told the story of one of the big strides forward in naval engineering. Running through it was the thread of the method by which naval engineering designs are produced. First of all they had one of the big engineering firms of the country giving very freely of their services to the Navy. Then they had a design team, specialized in land generating station turbines, working in close co-operation with the professional officers of the Engineer-in-Chief's Department of the Admiralty. They saw a wide variety of other firms brought in for discussions, culminating in a combined operation of production and, finally, Pametrada undertaking the complete testing of a marine unit on shore.

The degree of co-operation between the designer and the Admiralty had to be experienced to be fully appreciated. It was easy to appreciate that the naval turbine problem differed from the land generating station problem; but these differences had first to be pin pointed before their full implications could be understood. Reference was made in this paper to the need for rapid manœuvring, to the problems of astern power, to the peculiar requirements for the Navy of minimum weight and space, to the short life requirement at full power, and to chloride attack on materials resulting from impure feed water. It required full confidence between all parties to accept that these peculiar problems were true and real. It required high level technical discussion to determine exactly how much influence these factors should be allowed to exert upon the design. Each discussion involved drawing the other man beyond the limits of his own experience and carrying him along into the acceptance of other people's experience. Free and frank discussion and mutual confidence were absolutely essential to success.

The designer must carry the responsibility for the safety and soundness of his design; the manufacturer for his production. But the design and the manufacturing capacity must be mutually compatible, and this further required a very high level of co-operation when manufacture was to be undertaken by others than the design firm and when new ground was being broken which was making the highest demands upon all concerned. The ultimate responsibility of satisfactory performance in the Fleet lay with the Engineer-in-Chief, and again it required full co-operation and understanding, and clear professional speaking, for these three authorities to explain to each other the risks and the limitations involved, and to reach a compromise which operating experience would subsequently prove to be sound.

Any difficult task was difficult because the task ideally required more than was practicable. If the full Statement of Requirements of a naval design could be fulfilled, it probably meant that the aim had not been high enough. That was certainly not the case in the projects described in this paper; and because the full requirements were not capable of fulfilment, compromise became necessary. In reaching this compromise, mutual trust and a high level of professional appreciation and judgement were absolute essentials.

The story of the work of the English Electric Company, Yarrow and Co., Ltd., and of the many other firms involved, of the rotor forging problems posed to the forgemasters, of the manufacturing problems encountered by the marine engineering firms and of the extensive testing at Pametrada, were a tribute to the integrity and the professional capacity of the designers, the production people, and the naval and civilian engineers in the Admiralty service, both at headquarters and in the overseeing service who were involved.

Throughout the work covered by this paper the integrity, ability, and understanding of Mr. Cowlin were quite outstanding. To work with him was a stimulating experience. The Navy lost a very good friend, and industry an outstanding engineer, when he died. He would, he was sure, have been the first to pay tribute to the support that he had from Mr. Veitch, and he would like to thank Mr. Veitch for the work he had put into the preparation and production of this paper. The story it unfolded and the subsequent success of this machinery in the Navy was a high tribute to some remarkable team work.

CAPTAIN SIR JOHN S. W. WALSHAM, Bt., O.B.E., R.N., wrote that he congratulated the authors on their interesting paper which presented to the public for the first time a concise review of naval steam turbine development over the last fifteen years. He himself found the paper of particular interest as he had been connected with these developments during a recent period of service at the Admiralty.

The adoption of higher steam conditions in the latest British naval turbine designs was undoubtedly a most important factor in their performance but the authors showed clearly how the new turbines benefited by the adoption of a design point at 40-60 per cent full power. It was chastening to reflect that steam turbines had been installed in British warships for some thirty-five years before designers took advantage of the fact that naval propulsion machinery operated for the majority of its life at a low percentage of total installed power.

The new turbine designs described in the paper were all tested ashore before installation and this itself was a major innovation of the greatest importance. It appeared that the majority of teething troubles in a new design could be eliminated by testing ashore but there would also be some snags which would only manifest themselves at sea. The Y.100 automatic clutch was a good example of this. Could the authors say whether they considered that "development" of new designs as opposed to "testing" should be carried out ashore or afloat? The need for a "Trials Ship", where a machinery installation could be evaluated under seagoing conditions, had for a long time been considered a possible requirement.

He felt that it was only fair to point out that some of the gains obtained in the new designs were the result of appreciably higher turbine r.p.m. made possible by the production of satisfactory double reduction gearing. Indeed, the full benefit of the impulse turbine could only be realized at high r.p.m.

It would be interesting to hear Mr. Veitch's views on the effect of full power condenser vacuum on the overall weight of machinery. It appeared that the choice of full power vacuum had a profound effect, particularly in a turbine designed for efficiency at low power. The striking saving in weight of the Y.100 installation appeared to confirm this point.

The paper showed clearly the spectacular advances which had been made in turbine efficiency and weight reduction. Nevertheless, did Mr. Veitch not think the many advances in detailed design just as important; for while the *Daring* and Y.100 designs were in line with normal land turbine practice, they were radically different from pre-war British marine practice. Many of the changes were necessary to cope with the higher steam conditions but were also based on sounder principles. For example, alignment of the rotor and casing whilst allowing free expansion for each was achieved by using bearing housing and pedestals separate from the main casing, but suitably located with sliding keys, the casing being separated by integral palms bolted through over-size holes to the pedestals. As well as providing more freedom for expansion, this arrangement was inherently much more shock resistant than the old integral cast casing and pedestal, and rendered the provision of separate shock mountings unnecessary. The adoption of fully floating astern nozzle chests with piston ring sealing between steam pipe and chest was another example of a change in design aimed at reducing casing distortion.

The paper points out very clearly two main design principles; firstly, the necessity for economy at cruising powers to improve endurance and, secondly, the importance of an inverse temperature characteristic on turbine design. The lowering of the "design point" of the turbine to 40 per cent affected full power consumption adversely and hence increased boiler weight. Why not therefore install a small "full power" pressure combustion boiler with gas turbine driven blowers which could exhaust into the main boiler? One of the biggest steam consumers in modern high draught loss boilers was the turboblower which produced more auxiliary exhaust at full power than could be dealt with.

A cruising turbine was fitted for economy at low powers; why not a "full power" boiler for weight saving at high powers?

The design criterion of minimum "weight of fuel plus weight of machinery" for a given endurance was a useful yardstick but fuel was not an easy thing to stow in a single hull ship and unless water compensation, with its consequent troubles, was accepted, the difference between deep and light draughts raised stability problems.

Finally, it would be interesting to have Mr. Veitch's comments on the effect of increased main turbine efficiency and reduced weight on the machinery installation as a whole. It was obvious from the paper that a great deal of effort and money had been spent in refining main engine design but the propulsion auxiliaries and the "hotel" equipment accounted for a substantial proportion of the machinery weight and total steam consumption. The latter was extremely important at cruising powers where the fuel consumption of domestic machinery could be as high as 15 per cent of the total. The balance of the installation certainly had a profound influence on the warship's endurance and such details as the percentage capacity installed and the design margin on each piece of auxiliary machinery were obviously important.

Author's Reply

Mr. Veitch said that Admiral MacLean's query regarding the cost of the type of machinery described in the paper compared with its more conventional counterpart was not easy to answer as each design probably yielded a different result and as Y.100 was of a very special nature there was no comparable counterpart. If the *Daring III* were used as an example, it was the author's opinion that the higher speeds utilized, with consequent reduction in the number of stages and therefore blades, nozzles, etc., more than offset the added cost of alloy rotors. It should also be mentioned that such items as the more intricate blade roots which were required to cater for the higher centrifugal loads imposed were but little more costly than the more usual type, once a carefully planned manufacturing technique in conjunction with suitable machine tools, etc., had been established.

Mr. Terrell pointed out that the paper appeared to be concerned exclusively with aspects of turbine design, neglecting boiler problems in particular; the reply could only be that this was the intention.

Regarding Mr. Terrell's impression that *Daring I* gave somewhat better consumption figures than *Daring II* and *III*, performance in service did not suggest this to be the case.

The effect of clearances in impulse turbines was fully appreciated and in this respect the *Daring III* incorporated clearances of the required degree in order to maintain the very desirable quality of rapid manœuvrability.

In reply to comments by Commander Tyrrell, it must be stated that serious effort has been made in post-war years, probably due to Admiralty interest to a great extent, to improve generally the performance of naval and merchant main propulsion machinery; the creation of Pametrada and the Yarrow Admiralty Research Department (Y.A.R.D.) had obviously had very considerable effect when one examined the improvement shown in published results of trials of such machinery since the formation of these organizations.

Commander Dymoke, who referred to users' experience, replied to the point regarding the reliability of the *Daring* designs.

The benefits to be derived from the use of welded diaphragm construction had not escaped the notice of the designers concerned; this was a production problem, of course, which posed more than a few difficulties.

Commander Tyrrell referred to Figs. 25 and 26 and in reply to his questions the author stated that the organization with which he was associated found that the designs described in the paper, apart from being more robust, were less costly to produce, also cheaper to maintain from the point of view of spares, etc.

With a design similar to Y.E.A.D.I, but specifically for merchant ship conditions, performance figures would compare very favourably with those obtained in current merchant service designs.

Chloride attack was not now a serious hazard, bearing in mind the water treatment currently employed, but other factors such as erosion and corrosion, improved physical properties, etc., definitely favoured the use of stainless iron rather than mild steel in the machine industry generally.

Commander Dymoke referred to changes in casing construction and rotor design and although his suggestions were not impossible they would certainly entail a complete change in manufacturing technique allied with design problems which would probably result in as yet unforeseen troubles and difficulties; it was debatable whether the resulting improvement in manœuvrability would offset the increased complication and effort.

In reply to Mr. Hook, who raised questions regarding blading failures of Hecla A.T.V. materials, it must be stated that no such troubles occurred on the fixed blading, and it was only after opening up the turbines upon completion of the above trials that subsequent examination revealed failure of the Hecla A.T.V. lacing wires. Complete fractures were found on l.p. stages 3, 4, 5 and 6 and on h.p. stages 7 and 9; further detailed metallurgical examination showed that all rows of blading fitted with lacing wires were affected, the visible signs varying in degree from surface cracks to complete fracture, as mentioned.

The above trials took place over a period of some six months, although the turbine was not of course in operation continuously. An undue amount of record research would be entailed to establish the actual steaming time with any accuracy.

The chemical analysis of Hecla A.T.V. is Ni. 35 per cent, Cr. 11 per cent, Mn. 1·5 per cent and C. 0·2 to 0·45 per cent.

The brazing temperature of the silver solder used should be about 650 deg. C., but accurate control was not possible and the skill of the operator was relied upon; in this connexion it must be remembered that ideal conditions could not be obtained in practice due to the difference of the sections of the pieces to be silver soldered. There was no subsequent heat treatment.

Captain Sir John Walsham referred to the effect of degree of vacuum on the overall weight of the main propulsion machinery. An increase in the back pressure resulted in a shorter last blade and consequent easing of blade and root stresses, which in turn enabled higher maximum speeds to be utilized without decrease in safety factors; this in turn caused a reduction in the number of stages without loss in turbine efficiency and hence the overall result was a smaller and more compact turbine; similarly the size and weight of the condenser was considerably reduced due to the increased rate of heat transfer resulting from the increased temperature difference between steam and circulating water.

It should be realized when considering the effect of vacuum on steam consumption that although the difference in vacuum might be some 4in. Hg. at full power as in the case of Y.100, at low part powers this difference was reduced to as little as ¾in., therefore unless the emphasis was on high power requirements, it was obviously advantageous to effect major savings in weight and space at a slight expense in steam consumption, the overall result being a considerable increase in endurance.

The author concurred with Captain Sir John Walsham's statement regarding the importance of design details; also with the examples quoted and their effect.

It had always been the opinion of the designers that final proving of the machinery should be done afloat and this had been confirmed by experience.

In an exercise of this type which was aimed at giving a ship maximum endurance, reduction in weight and improvement in efficiency was just as important in ancillary equipment as in main engines.

In view of his close association with the late Mr. F. J. Cowlin, the author was deeply appreciative of the many kindly references made to his late chief, and in addition would like to express his sincere thanks to those who rendered valuable contributions to the discussion.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at The Memorial Building on Tuesday, 8th October 1957

An Ordinary Meeting was held by the Institute on Tuesday, 8th October 1957, at 5.30 p.m., when a paper entitled "Recent Developments in British Naval Main Propulsion Steam Turbines" by the late F. J. Cowlin, O.B.E., M.I.Mech.E. (Associate Member) and A. F. Veitch (Associate), was presented and discussed. Rear-Admiral F. E. Clemitson, C.B., (ret.) (Chairman of Council) was in the Chair and 103 members and visitors were present. Seven speakers took part in the discussion that followed.

A vote of thanks to Mr. Veitch, proposed by the Chairman, was accorded by acclamation. The meeting ended at 7.25 p.m.

Autumn Golf Meeting

The Autumn Meeting of the Institute's Golf Society took place at the Berkshire Golf Club on Thursday, 3rd October 1957. Thirty members played in the morning and afternoon competitions. The morning competition, a Stableford, was won by Mr. H. E. Upton, O.B.E., with a total of 35 points; Mr. C. J. Probett was second with 34 points. In the afternoon the members played in a Bogey Greensome competition which resulted in Messrs. C. Winyard and E. F. J. Baugh winning with a score of 2 up. Messrs. S. J. Jones and R. Wallace were second with a score of 1 up.

The prizes consisted of a folding garden chair and a travelling rug respectively for the winner and runner-up of the morning competition, and travelling alarm clocks for the winners and vases for the runners-up in the afternoon competition.

Mr. Stewart Hogg, the Chairman of the Social Events Committee, presented the prizes and thanked the committee, secretary and staff of the Berkshire Golf Club for the hospitality extended to the Society; he also thanked those who had contributed to the Prize Fund.

The meeting terminated with a vote of thanks to the Chairman, the members of the Social Events Committee and to the secretary, Mr. M. J. Pearce.

The leading scores in the morning and afternoon competitions were as follows:—

MORNING COMPETITION
SINGLES STABLEFORD

	Handicap	*Points*
H. E. Upton	9	35
C. J. Probett	8	34
G. M. McGavin ...	16	33
E. F. J. Baugh ...	10	33

AFTERNOON COMPETITION
BOGEY GREENSOME

C. Winyard and E. F. J. Baugh ...	+2
S. J. Jones and R. Wallace ...	+1
W. Ridley and C. J. Probett ...	−1
R. K. Craig and H. Armstrong ...	−1

Section Meetings

Scottish

A general meeting was held at the Institution of Engineers and Shipbuilders in Scotland, Glasgow, on Wednesday, 13th November 1957, at 7.30 p.m. There were 115 members and visitors present.

Mr. W. Young (Chairman of the Section) presided and introduced Mr. A. N. Savage (Member) who gave a most interesting paper entitled "Developments in Marine Electrical Installations with Particular Reference to A.C. Supply".

Mr. E. Souchotte (Member of Committee) aptly proposed the vote of thanks to Mr. Savage and this was carried enthusiastically.

The meeting terminated at 9.30 p.m.

West Midlands

At a meeting held on Thursday, 10th October 1957, at the Birmingham Exchange and Engineering Centre, Rear-Admiral W. G. Cowland, C.B., ret. (Member), presented an illustrated lecture entitled "Turbocharging for Two-stroke Oil Engines". Mr. H. E. Upton, O.B.E. (Chairman of the Section) was in the chair and there was an attendance of sixty-seven members and visitors.

Admiral Cowland opened his lecture with a description of the various methods of scavenging at present in use and discussed the merit of each method and its particular application. He concluded by commenting upon likely trends of turbocharging and the effects of this development on future applications of oil engines.

Some twenty-five members took part in the very lively and enlightening discussion which followed, the Chairman expressed the meeting's warm appreciation of Admiral Cowland's lecture and the meeting closed at 9.0 p.m.

Sydney

The annual dinner of the Sydney Section was held at the Wentworth Hotel on Thursday, 24th October 1957. There was an attendance of 110, of which sixty were members of the Institute. The official guests included Rear-Admiral C. C. Clark, O.B.E., D.S.C.; Mr. D. S. Carment (Australian Branch of the Institution of Naval Architects); Mr. H. B. Wood, O.B.E. (Chairman of the Sydney Division of the Institution of Engineers, Australia); Mr. T. M. Devitt and Mr. N. D. Pixley. The Local Vice-President, Captain G. I. D. Hutcheson, R.A.N.(ret.), presided at the dinner.

After the Loyal Toast, the toast of "The Institute of Marine Engineers" was proposed by Rear-Admiral Clark; Mr. N. A. Grieves (Honorary Secretary) responded on behalf of the Sydney Section. The toast of "The Guests" was proposed by Mr. B. P. Fielden and answered by Mr. Pixley.

Victorian

Students' Meeting

On 31st October 1957 the Victorian Section, in conjunction with the Australian Institute of Marine and Power Engineers and the Australian Steamship Owners Federation, held a students' night at the Radio School Theatre of the Royal Melbourne Technical College. The object was to encourage students of technical colleges to take up marine engineering as a career. Several technical colleges were contacted and a maximum of five students from each school was invited; there was an attendance of approximately forty students from these outside schools, together with the marine engineers' class of the Royal Melbourne Technical College. The function went very well and quite a number of questions were asked which indicated the interest of the students.

General Meeting

On Friday, 8th November 1957, the Section held a general meeting at the Kelvin Hall for the presentation of a lecture by Mr. K. Smith. The meeting was well attended and the lecture proved of great interest, there being considerable discussion following Mr. Smith's address.

Election of Members

Elected 13th November 1957

MEMBERS

Thomas Henry Allan
James Morrison Blair
Stanley Bolton
Norman Eric Dalton, Vice-Admiral, C.B., O.B.E.
Ludwig Richard Dittersdorf
Hugh Wordley Drummond, Lieut.-Cdr., R.N.
Arthur Abraham Duff
Reginald Charles Dyer, Lieut.-Cdr., R.N.
Hugh Alan Dyson
Hubert Royston Evans
Thomas George Hutchinson
Ronald Chudleigh Larcombe
William Johnstone Macaulay
Frederick Allison Macmillan
William John McRonald
Ronald James Moir
William Burgess White Morgan
K. B. Myers
Hugh Vincent O'Neill
Richard Guy Peach
Frank Edwin Poulter, Eng. Lieut., R.N.
William Robert Richards
Maurice Gilbert White
William White
Robert King Wood, B.Sc.

ASSOCIATE MEMBERS

Clive Allon
Richard Francis Bullock
William Moody Campbell
Albert John Clark, Eng. Lieut., R.N.
Ivan Dudley Conway
James Edward Cunningham
Cecil John Fitzpatrick, Lieut.(E), South African Navy
Kenneth Anthony Gonsalves, Lieut., I.N.
Frederick Laurence Greensmith
Kenneth Leslie Hammond
Henry Neville Hemsley, M.A.
Sydney Hunter
Peter Jackson
Cedric Manoel Koch
Gilchrist Goold MacAdam
William John McCash
Charles Magowan
John Charles Matthews
Ayudha Nath Mukherjee
Henry Navickas
Roger Louis Onfray
Arthur Parkes
Robert Paterson
Ronald Peacock
John Thomas Prowse
Peter Alfred Sait
Alexander George Simpson
Norman Snaith, B.Sc.(Durham)
Herbert Thein
Ronald Ayre Tindle
Michael Christopher Turner, Lieut., R.N.
John Denys Wakinshaw
James Findlay Watson, B.Sc.(Glasgow)
Robert Edward Witton
George Henry Wright
John Arthur Young

ASSOCIATES

Michael John Adams Bolton
Effiong Etim Duke
Thomas William Haycroft
Stanley Alfred Jones
Pavel Murdechai Katriel
John Winsor Noel
U Thet Tin
Robert John Tucker, Sub.-Lieut., R.N.
Saiyid Mohd. Kamaluddin Zafar

GRADUATES

Michael David Constable
Brian Ernest Culley
Adrian George Glentworth
Cyril Edwin James
Fernandes Reginald Joachim
James Derrick Judge
Parmanand Menghraj Kalvani, B.Sc.(Durham)
Sudershan Kumar Kapur, Lieut., I.N.
Azhar Hussain Khan
John Michael Kingsland, Lieut., R.N.
Peter Gavin McGregor
Lakshmi Narayan Misra
Prem Prakash, Lieut., I.N.
Zahedur Rahman
Derek Remosani Robinson
Keith Turnor

STUDENTS

William Kennedy Cameron
David Thomas Dennis
Colin Norman Godfrey
James Allen Greenwell
Keith Michael Howes
Keith King
Marcel Levesque
Robert William Middlemiss
Gordon Victor Miles
Christopher Harrison Petrie
James Anthony Rothwell
Fred Williams
Raffat Zaheer

PROBATIONER STUDENTS

Paul Richard Barnes
Kendrick Harold Findlay Barton
Ivan Bell
John Ritchie Benge
Gareth Brian Cox
Michael Frederick Crawley
David Charles Dupree
Richard Frank Gilbert
Brian Ralph Graham
Robert Groizard
Eric Wilfred Hilton
Michael Franklin Hogg
Michael John Hutson
David Robert Lockwood
John Phillips Lord
David Edward Pelton
Rodney William Sawyer
Christopher Smith

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER

Charles Edward Street

TRANSFER FROM ASSOCIATE TO MEMBER

Robert James Chapman
Douglas Graham Hayward
Sidney Joseph Jones, B.Sc.(Eng.) (London)
John Nankervis Rowe
John William Waller

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Ahmed Mohamed Abdalla
Donald William Challinor
Samuel Edwin Gay
Ewen Haddon Kennedy
Imtiaz Muhammad Qureshi, Lieut., P.N.
Harry Strachan
Velluppillai Subramaniam Thiruchelvam
Henry Wilkinson

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Satya Dev Batra
Anthreas Nicholas Charcharos, B.Sc.(Durham)
David Martin Leamon
Maurice Peters
William Swindells
Bernard James Vaughan

TRANSFER FROM PROBATIONER STUDENT TO STUDENT

Maxwell Keith Bruce
Derek Burton
George Arthur Crawford
Gareth Boycott Oddy
Brian Charles Osborne
Edward Arthur Popple
Peter Henry Ryder
Laurence Roy David Saunders
Brian Smith
David Samuel Waind
David Alan Wickham

OBITUARY

Samuel Newbiggin Kent

Appreciation by Eng. Capt. W. A. Graham, O.B.E., R.N.R.(ret.) (Honorary Member)

A great friend, a good companion with an undoubted sense of humour, Samuel Newbiggin Kent has passed away at the advanced age of eighty-two, after an illness lasting many months.

Elected to the Institute on 20th June 1916, my old friend was a Member of the Council from 1927/29, 1932/34 and again from 1936/39, being Chairman of Council in 1933. He was a Vice-President from 1939/52 and Honorary Vice-President from 1953 until his death on 21st October 1957.

He will be remembered for his work as a Member of the Papers and Transactions Committee, 1939/44, and as a Member of the Guild of Benevolence Committee from 1939 to 1950.

After serving an engineering apprenticeship with R. and W. Hawthorn, Leslie and Co., Ltd., Newcastle upon Tyne, he went to sea, where he obtained a First Class Board of Trade Certificate, afterwards joining the staff of Lloyd's Register of Shipping as a ship and engineer surveyor, being stationed at Middlesbrough for three years. On transfer to Manchester he held the rank of senior surveyor for six years until he resigned the appointment in 1917. He joined the Prince Line in February 1917 as superintendent engineer and, after two years in Newcastle upon Tyne and a period in London he went to New York early in 1920 on behalf of the company to act as superintendent engineer there for a year, afterwards resuming his duties in London.

Retiring from the Prince Line on 31st December 1939 after twenty-two years' service, he was able to recount with a certain pride his connexion with the supervision and introduction and development of Diesel engines and the conversion of steam vessels from coal to oil burning.

Incapacitated through poor health at the time of his retirement from the Prince Line, he made a good recovery and eventually returned to the City and acted in an advisory capacity to the Faros and Ceres Shipping Companies.

When he finally went into retirement he used his engineering skill in the making and perfecting of mechanical toys for poor children, believing that to give a little love to a child was to get a great deal back.

Ever a good companion, with an appealing sense of humour, Samuel Newbiggin Kent will be greatly missed, but older members will doubtless remember him for his work for, and interest in, all that appertained to the well-being and general advancement of the Institute.

Now the years will no longer weary him he can stand before kings having stemm'd the sea of life, languished after earthly joys, labour'd in the stormy strife, grieved for trifles and amused—with toys.

Stanley Dale (Member 9010) died suddenly on 2nd July 1957 while travelling by train from Newcastle upon Tyne to London on business. He was born in 1911 and served an apprenticeship with the North Eastern Marine Engineering Co., Ltd., Wallsend, from 1926/31. After sea service for the next five years with the Moor Line, the New Zealand Shipping Co., Ltd., and the Hudson Steamship Company, during which time he obtained a First Class Board of Trade Steam Certificate with Motor Endorsement, he came ashore to take an appointment as assistant superintendent with Monroe Brothers of Liverpool. In 1937 he joined his father's firm, J. W. Dale and Son, consulting marine engineers and ship surveyors of North Shields.

Mr. Dale was one of the North East Coast representatives on the council of the Society of Consulting Marine Engineers and Ship Surveyors and was a Member of the North East Coast Institution of Engineers and Shipbuilders. He had been a Member of the Institute of Marine Engineers since 1943.

William Dick (Member 7030) was killed in a road accident on 7th November 1957, aged fifty-nine. He was educated at Coopers' School, London, and the London County

Council School of Engineering and Navigation, Poplar, and served an apprenticeship with R. and H. Green and Silley Weir, Ltd., at the Royal Albert Docks. He obtained a First Class Board of Trade Steam Certificate and Motor Endorsement whilst serving in the Port and Nelson Lines. For the last twenty-two years he had been employed by the Ilford and Barking Joint Sewerage Disposal Committee and for the last ten years he held the position of chief maintenance engineer.

Mr. Dick had been a Member of the Institute since 1932.

KEITH ELWYN JONES (Member 12278), superintendent engineer to the South American Saint Line, Ltd., Cardiff, died suddenly, aged fifty-seven, while playing golf on 20th September 1957. He served an apprenticeship with the Mountstuart Dry Docks, Ltd., Cardiff, from 1916/21, and then spent six years as a seagoing engineer as seventh to chief engineer. From 1930/39 he managed an engineering business on his own account but from 1939/42 he returned to sea as a lieutenant commander in the Royal Naval Reserve and was then appointed assistant to the manager of the engineering department of H.M. Dockyard, Malta, and non-exclusive engineer surveyor to Lloyd's Register of Shipping. From 1945/46 he was assistant to the manager of the engineering department of the dockyard at Rosyth. For a year or so after this war service he was a consulting engineer with his own business but in 1947 he took his final appointment with the South American Saint Line.

Mr. Jones had been a Member of the Institute since 1949.

ARCHIBALD DUNCAN CAMPBELL LINN (Member 4643) was born in 1891. After serving an apprenticeship with Stewart, Luckie and Company, he spent six months with John Shearman and Co., Ltd., Avonmouth, before going to sea for some months as a junior engineer with Canadian Northern Steamships, Ltd. From 1915/20 he was an assistant with Alcock, Ashdown and Co., Ltd., in Bombay, being promoted assistant manager in 1920 and manager in 1922. Six years later he returned to Britain and became a partner in a small engineering business in London. Mr. Linn died on 26th October 1957, aged sixty-six. He had been a Member of the Institute since 1922.

NORMAN STANLEY ROWSTRON (Member 6630), chief superintendent engineer of Stephenson, Clarke, Ltd., died 1st August 1957 aged fifty-six, after a long illness. He served an apprenticeship with Richardsons, Westgarth and Co., Ltd., Sunderland, from 1919/24, before becoming a seagoing engineer first with the Nautilus Steamship Company and then with Messrs. Alfred Holt and Company; he obtained a First Class Board of Trade Certificate. In 1929 he was appointed junior assistant works manager to the Middle Docks and Engineering Company, South Shields, and two years later became assistant superintending and consulting engineer to Messrs. Robert Eeles and Company, Newcastle on Tyne. In 1935 Mr. Rowstron joined the technical staff of Stephenson, Clarke, Ltd., and received his final appointment as chief superintendent engineer in 1951. He had been a Member of the Institute since 1931.

MICHAEL CAMERON RUMSBY (Graduate 16065) entered the Royal Naval College, Dartmouth, by special entry on 23rd September 1948 and was appointed to the training cruiser, H.M.S. *Devonshire,* as a cadet in the following January. For the next six months he was a midshipman(E) in H.M.S. *Unicorn* and then spent three years at the Royal Naval Engineering College, Greenwich, before appointment as sub-lieutenant to H.M.S. *Glory*. He attended a specialist engineering course in H.M.S. *Thunderer,* with promotion to lieutenant, from May 1953 to May 1954 and then served in H.M.S. *Birmingham* for a year. He was appointed to H.M.S. *Raleigh* in September 1955 and served in her until he entered hospital in July 1957; he died on 5th November 1957.

Lieutenant Rumsby was elected a Graduate of the Institute in 1954.

REGINALD WATSON SMEDLEY (Member 11184) died suddenly at his home on 24th September 1956, aged forty-nine. He was apprenticed to Aire and Calder Navigation, Leeds, from 1922/28, and attended evening classes during this period at Hull Technical College. He first went to sea in the s.s. *Portsea* as fourth engineer in 1929 and then joined the Hain Steamship Co., Ltd., serving as fourth to chief engineer in their ships from 1930/42. He obtained a First Class Board of Trade Steam Certificate in 1933 and a Motor Endorsement in 1938. The steam ship in which he was serving in 1941, the *Trevilley,* was attacked by submarine and from then until the end of the second world war he was a prisoner of war, returning to the Hain Steamship Company as chief engineer in 1946 and remaining in their employment until his death; unfortunately his war experiences had undermined his health, however, and from 1951 he was only able to undertake relieving work in coastal waters.

Mr. Smedley was elected to Membership of the Institute in 1947.

WILLIAM REID WALLACE (Member 4554) served his apprenticeship from 1911/15 at the Blackwall Yard, Orchard House Works, of the Union Castle Mail Steamship Company and at the same time attended technical evening classes at the West Ham Municipal Technical Institute. When war started in 1914 he had the experience, enthralling to a boy, of making a few cross-channel trips in the engine room of the old *Comrie Castle,* which was thrown into the trooping effort at short notice. In the autumn of 1915 he made his first deep sea voyage in the old cable ship *Faraday* and on returning to London the following spring he was appointed as junior engineer in the Union Castle troopship *Briton.* He served in her and in similar vessels until the end of the war and obtained his First Class Board of Trade Steam Certificate in 1920 when he joined the Glen Line in order to gain experience of their Diesel-engined vessels, serving in the *Glenade* and making three trips to China in the *Glenamoy* before coming ashore in 1922 to take, first, his Motor Endorsement, and then an Extra First Class Certificate. He joined the Institute as a Member at this time.

Mr. Wallace then obtained an appointment as engineer surveyor with the Municipal Mutual Insurance Association and in 1924 was called to the head office to act as deputy to the chief engineer but as his greatest interest lay with ships and maritime affairs he entered in 1928 for one of the infrequent examinations for applicants for Board of Trade Surveyorships and was successful. He was appointed to the Port of Liverpool, where he remained for the rest of his life. In 1943 he was selected, with other Ministry of War Transport surveyors, to cross the channel in wake of the advance forces to staff the various Continental ports as they were recaptured but shortly before D-day he collapsed due to heart failure and he had to be relegated to the physically less arduous work of the examination room. The heart weakness which overtook him in 1944, however, was serious and intractable; he carried on until the summer of 1956 when he was compelled to stay in bed and he died on 3rd October 1957.