Post-war Developments and Future Trends of Steam Turbine Tanker Machinery

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The paper describes the post-war experience of the shipping department of a major oil company in the design and operation of steam turbine engined tankers. The circumstances leading to the adoption of steam turbines to provide 12,500 s.h.p. in the first post-war class of tankers of 28,000 d.w.t. are explained. After dealing briefly with the machinery design for these ships and its service performance, the evolution of this design to 14,000 and 16,000 s.h.p. is then sketched in outline.

The 50,000-d.w.t. class provided an opportunity for a new approach, and the design study leading to a simplified 16,000 s.h.p. design with higher steam conditions for this class is reviewed. Attention is drawn to the lessons learnt in the course of this study on the importance of considering the economics of the whole installation during the early stages of a design.

There follows the principal section of the paper which describes the development of 20,000 s.h.p. machinery for 68,000-d.w.t. ships. Before embarking upon this design opportunity was available for experience with earlier classes to be reviewed in detail, and salient features of this review, including descriptions of defects experienced, are highlighted. The installation design study for this machinery is described, including comparison of heat cycles, alternative proposals for turbines, gearing and condensers, and a paragraph on boiler design features and trends.

The paper concludes with a section on trends for large tanker turbine machinery installation during the next twelve years covering higher powers, higher steam conditions, sophisticated feed systems, packaged machinery, complexity and its influence on reliability, and automatic controls.

INTRODUCTION

Before the war most petroleum markets outside the U.S.A. were small and had oil product demand patterns that were difficult to match, in themselves, from conventional refinery operations. In order to gain the advantages of large scale with a balanced refinery operation, the pre-war tendency was to locate refineries close to the source of crude oil production and to serve a number of markets from one refinery. The greatest flexibility was achieved by tankers delivering directly to the port nearest the market, and for this purpose the typical pre-war tankers was of about 12,000 d.w.t. and was propelled by 3,600 b.h.p. Diesel engines at a speed of about 12 knots on a fuel consumption of about 15 tons/day[7]. Those refineries which were located in consuming areas were generally small and best served by a similar sized ship carrying crude oil. It is perhaps worth noting that while such a tanker is considered small today, it was and still is a relatively large cargo ship.

After the war the only fuel that could be developed quickly enough to meet the tremendous growth in energy needs, especially in Europe, was oil. A rapid expansion in refinery capacity was needed. American aid (Marshall Aid) was available to European governments and the new capacity was provided mainly in Western Europe by expanding existing refineries or building new ones. Concurrent with this development was the concept of large crude oil carriers whose more economic unit operation counterbalanced the cost of carrying "fuel and loss" to refineries in the consuming areas. The continued rapid increase in European demand and refining capacity, the development of large refineries in Japan and Australia, and the growth in U.S.A. crude oil import demand, together with improved shipbuilding production techniques and suitable port development, are factors which have stimulated the further growth in tanker size.

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EARLY POST WAR DEVELOPMENTS

28,000-d.w.t. Class

To meet the initial phase of this new pattern, a class of six vessels of 28,000 d.w.t. and a speed of 15 knots was programmed, the power required being 12,500 s.h.p.

In the early 20's the company had built a number of successful steam turbine vessels, the last of which remained in service until 1953. However, in 1923 these designs were displaced by Diesel engined vessels, because of their superior fuel economy. The use of high powered Diesel engines had however been confined to passenger vessels, usually twin-screw, and it was therefore judged at the time that there was no fully developed Diesel engine, capable of producing 12,500 s.h.p. on a single screw, suitable for the company's requirements.

In contrast, the post-war development of steam turbines appeared to offer reliable machinery with reasonable, if not competitive, economy. These factors, coupled with the satisfactory experience gained with steam machinery in war time T.2's led to the selection of steam turbines.

The machinery was designed during the years 1949 to

1951 and was first installed in *British Adventure,* the same machinery being repeated in a later class of six 32,000-d.w.t. vessels.

The turbines selected were of Pametrada design, comprising an all-impulse single-casing H.P. ahead turbine with overhung H.P. astern element (Fig. 1) and an all-reaction twin-flow L.P. ahead turbine with the L.P. astern element incorporated in the L.P. casing (Fig. 2).

The selected steam conditions were 450lb./sq. in. gauge, 750 deg. F. at the superheater outlet and the designed vacuum was 28·5 inches at 75 deg. F. sea temperature. A relatively simple heat balance was employed (Fig. 3) which gave a designed fuel rate of 0·615lb./s.h.p. hr.

The gearing was of double reduction articulated design employing K factors of 67 in the primary train and 60 in the secondary. The gearing materials used were forged steel 34-38 tons/sq. in. for the wheel rims and $3\frac{1}{2}$ per cent nickel steel, 40 tons/sq. in. for the pinions.

Babcock and Wilcox integral furnace boilers were fitted in certain ships of this class and Foster Wheeler D type in others. The Foster Wheeler boilers were fitted with economizers and smoke tube air heaters and the Babcock and Wilcox boilers were fitted with smoke tube air heaters only.

Service Performance

The performance in service of these ships was disappointing. Fuel consumptions in service proved greatly in excess of designed rate, and various operational difficulties arose.

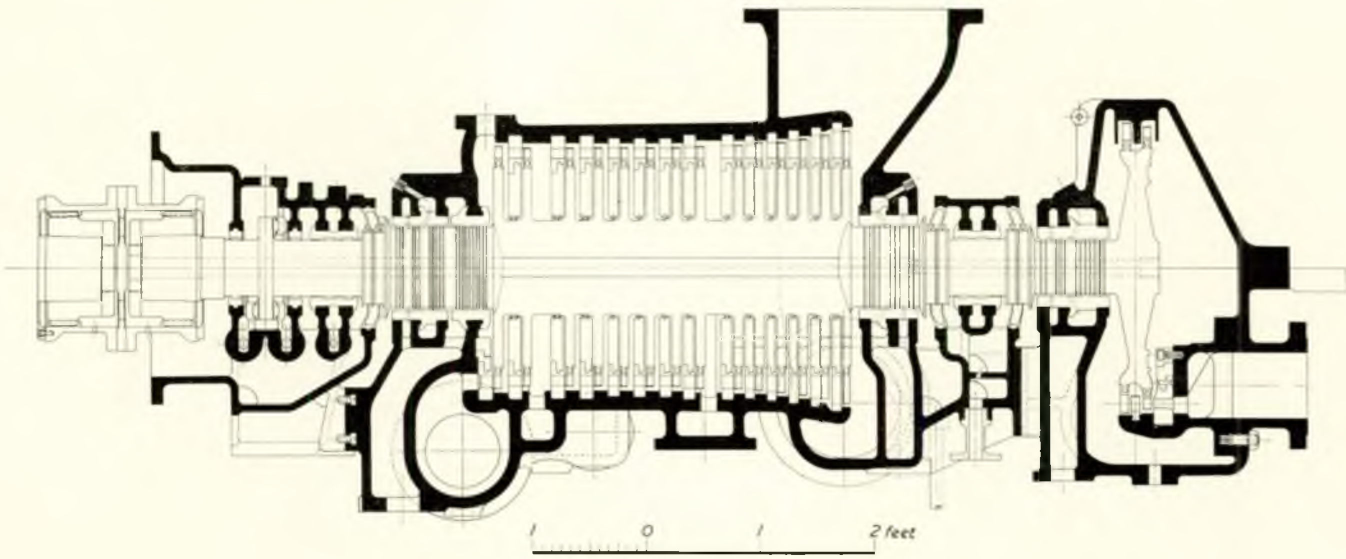


FIG. 1—*H.P. Turbine basis plan—12,500 s.h.p.*

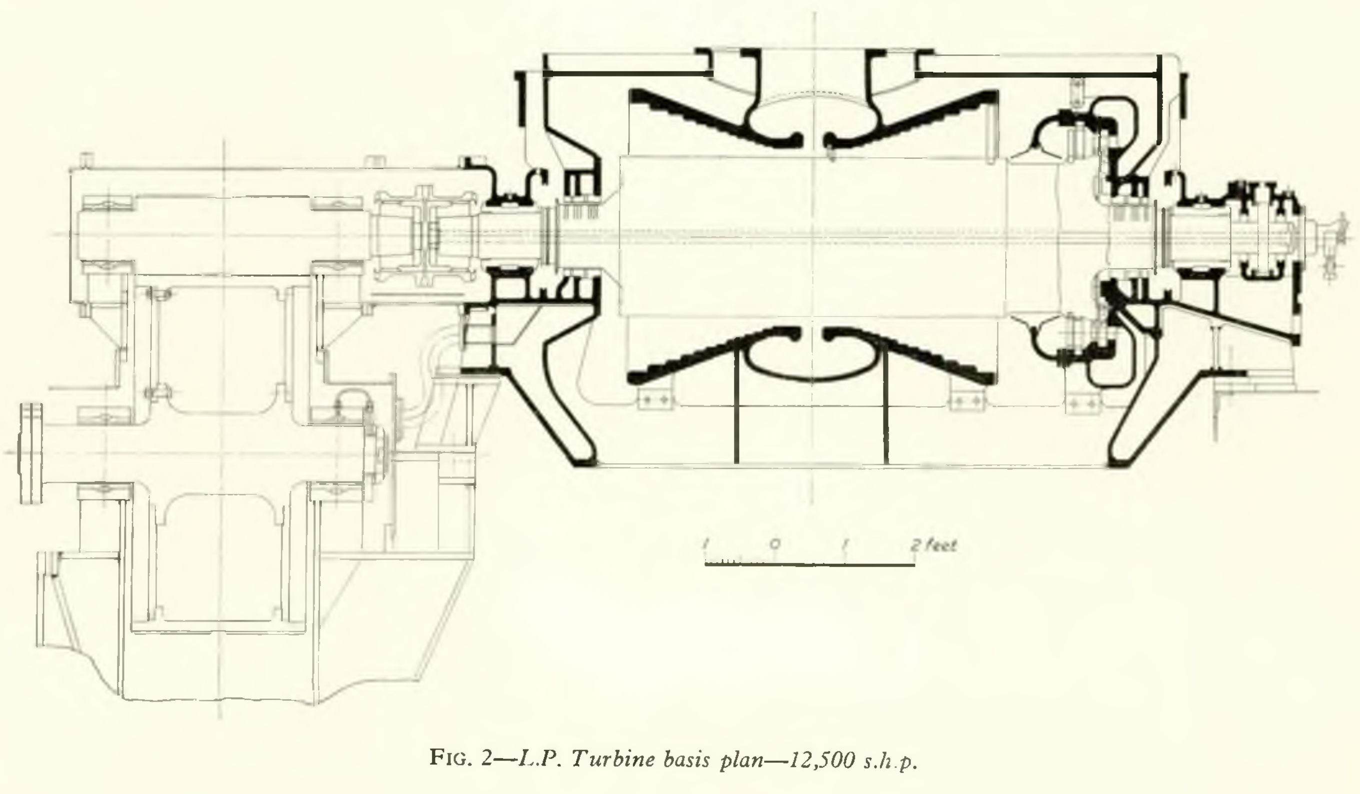
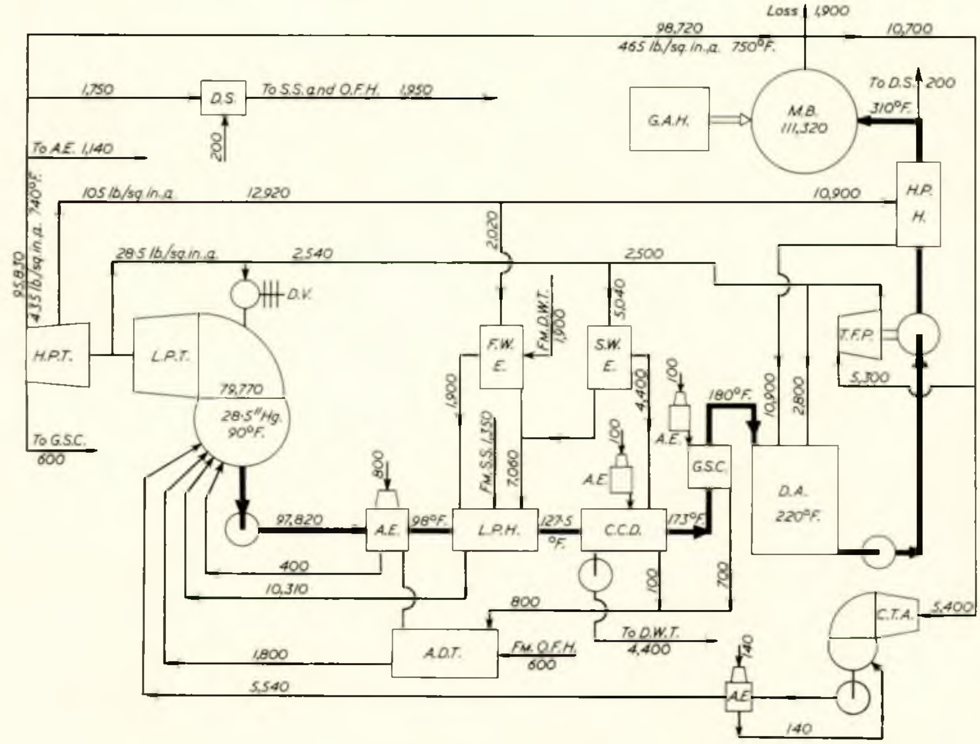


FIG. 2—*L.P. Turbine basis plan—12,500 s.h.p.*



KEY TO HEAT BALANCE DIAGRAMS
A.E.—Air ejector
A.D.T.—Atmospheric drain tank
B.T.A.—Back pressure turbo-alternator
C.C.D.—Condensate cooled distiller
C.I.E.—Cast iron economizer
C.T.A.—Condensing turbo-alternator
D.A.—De-aerator
D.C.—Drain cooler
D.S.—Desuperheater
F.W.E.—Fresh water evaporator
G.A.H.—Gas/air heater
G.S.C.—Gland steam condenser
H.P.H.—H.P. heater
H.P.S.H.—H.P. steam air heater
H.P.T.—H.P. turbine
I.P.H.—I.P. heater
L.P.H.—L.P. heater
L.P.S.H.—L.P. steam air heater
L.P.T.—L.P. turbine
M.B.—Main boilers
P.C.—Pressure controller
S.D.A.—Shunt de-aerator
S.G.E.—Steel gilled economizer
S.S.G.—Steam/steam generator
S.W.D.—Salt water cooled distiller
S.W.E.—Salt water evaporator
T.F.P.—Turbo feed pump

FIG. 3—*Heat balance—12,500 s.h.p., 450lb./sq. in. gauge, 750 deg. F.*

Air Heater Chokage

Early in the history of these vessels smoke tube air heater chokage was experienced. The cause and cure of this is now well known[1] but under the operational pressure of the time it could only be tackled by trial and error methods. Tanker tonnage was in extremely short supply and the delays resulting from this air heater chokage brought heavy financial loss which was unacceptable during a protracted period of investigation. It was decided therefore to remove the small air heaters in vessels fitted with Foster Wheeler boilers and accept the consequent 3 per cent increase in fuel rate.

In vessels fitted with Babcock and Wilcox boilers, however, where economizers were not fitted, air heaters supplied with live steam were fitted to pre-heat the air to the smoke tube air heaters to 170 deg. F. This was effective in preventing air heater deposits at the expense of increasing the fuel rate from 0·615 to 0·634lb./s.h.p. hr. The increase could have been halved had the steam air heaters been supplied with bled steam.

Closed Feed System

Difficulty was experienced with the operation of the closed feed system. As will be seen from Fig. 3, the de-aerator was designed to operate at 220 deg. F. corresponding to 2·5lb./sq. in. gauge.

With the somewhat primitive control system, based on unrelated reducing valves, it was found to be impossible to maintain this pressure in the de-aerator shell during transient conditions, which often led to "flashing" at the suction of the feed pump, with consequent tripping out. Basically the fault lay in inadequate attention to the control system in the design stage but at the time this was not fully appreciated.

Boilers

Fire row tube failures were fairly frequent in this class of ship, and in all cases reported, metallic oxide deposits were found to be present. These deposits were attributed to oxygen/carbon dioxide corrosion of certain steam lines, particularly cargo heating lines, which are normally used intermittently and may stand for long periods, full of aerated water. Further reference to this subject is made later in this paper.

Turbines and Gearing

Other failures which occurred during this period were:

a) Failure of an H.P. primary gear train.
b) Fracture of an L.P. turbine blade with consequential damage to condenser tubes.
c) A bent H.P. turbine rotor following a boiler prime.

At the time these failures were regarded as sporadic and attention was mainly directed to overcoming the difficulties with air heaters, closed feed systems and boiler tube failures. However, the L.P. turbine blade failure proved to be the forerunner of others, as will be described later in this paper.

Having overcome these early troubles this design of machinery continues to give satisfactory service in most respects except fuel consumption. The double flow L.P. turbines are however prone to distortion and further blade failures have been experienced as described later. Nevertheless the design can be regarded as a good starting point for the future developments now described.

THE 14,000 S.H.P. INSTALLATIONS

32,000 and 35,000-d.w.t. Classes

Such was the demand for tanker tonnage that a new class of ships ranging from 32,000 to 35,000 tons was in process of design while the earlier class of ships was as yet entering service. These vessels were to be propelled by 14,000 s.h.p. engines and economic considerations indicated that an increase in steam conditions to 600lb./sq. in. gauge, 850 deg. F. was desirable. As the design of this latter class was developing, experience with the earlier class of ships became available and led to the following changes.

1) It was decided that, despite the overall thermodynamic advantage offered by the smoke tube air heater cycle, delays due to chokage were such as to completely offset such gains in fuel economy. As a result this new class of ship was designed with a final feed temperature of 240 deg. F. with a cast iron economizer designed to reduce the funnel gases to 320 deg. F. Air heating was provided by means of bled steam.

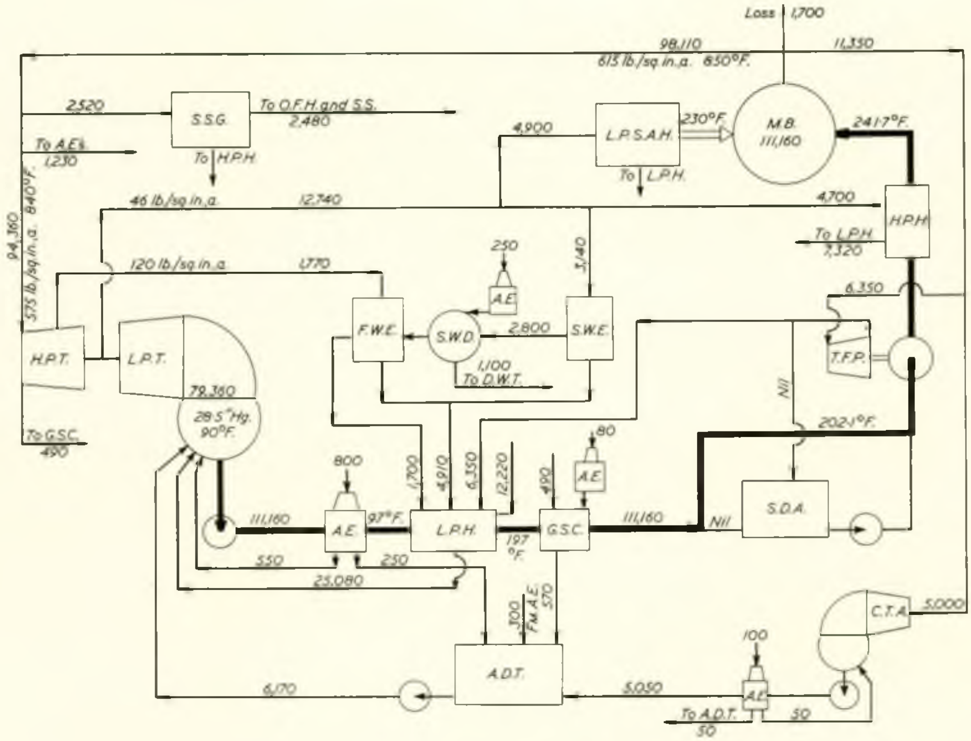


FIG. 4—*Heat Balance—14,000 s.h.p., 600lb./sq. in. gauge, 850 deg. F.*

This has been retained as a basic design feature of all subsequent classes.

2) It was decided to adopt the "shunt" de-aeration system whereby the de-aerator is only employed under port conditions when the main condenser vacuum is low and de-aeration in the main condenser is therefore incomplete. With this system the de-aerator is shut down during normal steaming at sea and the previously experienced difficulties on sudden reduction of load were avoided.

3) Steam heated L.P. steam generators were fitted to serve all oil heating and other "contaminated" services, the intention being to avoid the accumulation of metallic oxide deposits in the main boilers, and also to avoid all possibility of oil contamination.

These modifications entailed the development of a new heat balance (Fig. 4) which indicated a designed fuel rate of 0·577lb./s.h.p. hr. Unfortunately, the shunt de-aeration system became extremely complicated as design developed to cover all the varying conditions of operation of an oil tanker, and this

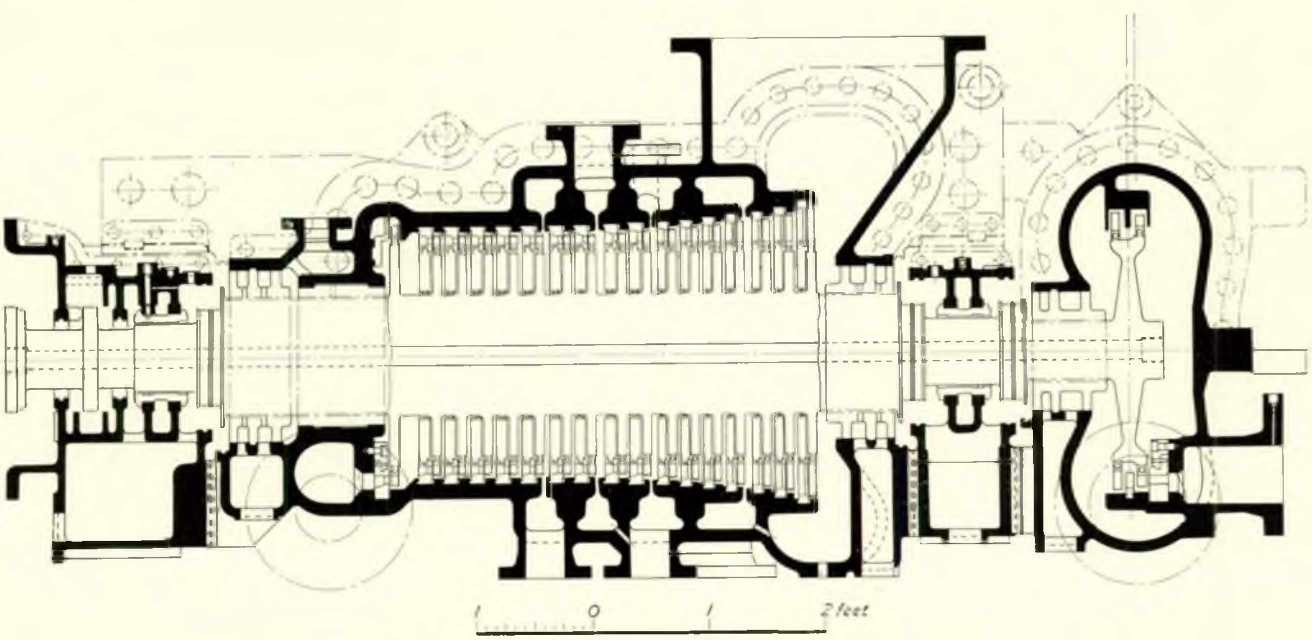


FIG. 5—*H.P. Turbine basis plan—"Fast Tankers"*

complexity increased the initial cost. De-aeration in the main condenser had not been as good as anticipated, due mainly to under-cooling of condensate when the sea temperature is below design. Despite the segregation of auxiliary services therefore, it was found that metallic oxides still continued to be deposited in the main boilers. The service record of these ships has shown some improvement on that of the earlier class of vessel, but a variation still exists in fuel rates achieved on trials between 0·585 and 0·625lb./s.h.p. hr. This variation was more than should have been experienced and has been attributed to:

a) Variation in first stage nozzle area
b) Variations in blade path
c) Heavy water losses due to joint and gland leakages.

The "Fast Tanker" Design

Two noteworthy 32,000-d.w.t. ships were completed at this time, designed to have an appreciably higher maximum speed necessitating a maximum service power of 22,500 s.h.p. The turbines were designed to develop this power but were also required to be capable of operating as economically at 14,000 s.h.p. as the standard ships of the class. These were, in effect "two-speed" ships.

The H.P. and L.P. turbines are shown in Figs. 5 and 6 and a number of differences will be noted between this design and that shown in Figs. 1 and 2.

The H.P. turbine was arranged to provide 14,000 s.h.p. on the normal nozzle sector, with a small bore bypass valve giving 15,500 s.h.p. An additional large bore bypass valve was fitted for the 22,500 s.h.p. condition. The L.P. turbine was supported on the condenser instead of the condenser being underhung from the turbine and a gashed L.P. rotor was employed.

It will be noted that three bleed points were provided on the H.P. turbine and two on each flow of the L.P. turbine.

The necessity for this arrangement will be apparent from the heat balance shown in Fig. 7. When operating at the lower power level, the bleeds were changed over to a higher stage of the turbine to preserve the heat balance at both power levels.

The circuit included a split economizer scheme with H.P. feed heater between the two sections of the economizer and a shunt de-aerator. The designed fuel rate of 0·571lb./s.h.p. hr. at 14,000 s.h.p. was attained on trials and has since been maintained in service. Despite the apparent complexity of the feed system, these two vessels have given excellent results. The operational facility of this comparatively complex feed system has influenced future thinking considerably, as will be seen in the development of the 68,000-d.w.t. class 20,000 s.h.p. design described later in this paper.

14,000 s.h.p. Italian-built Vessels

During this period a total of six 35,000-d.w.t. vessels were ordered in Italy. These ships are propelled by 14,000 s.h.p. turbines of U.S. De Laval design (Figs. 8 and 9) manufactured by the shipbuilders. Steam conditions were 600lb./sq. in. gauge, 850 deg. F. at the superheater outlet, as in the corresponding British-built ships, and the designed fuel rate was 0·56lb./s.h.p. hr. as shown in the heat balance (Fig. 10).

Automatic controls were extensively applied to these vessels, particularly to the high level de-aerator control, and these have performed extremely well in service. Fuel rates achieved on trials have varied between 0·538 and 0·565lb./s.h.p. hr. and averaged 0·54lb./s.h.p. hr. All these vessels have so far maintained the trial trip performance in service.

Oblique reference has been made, both before this Institute

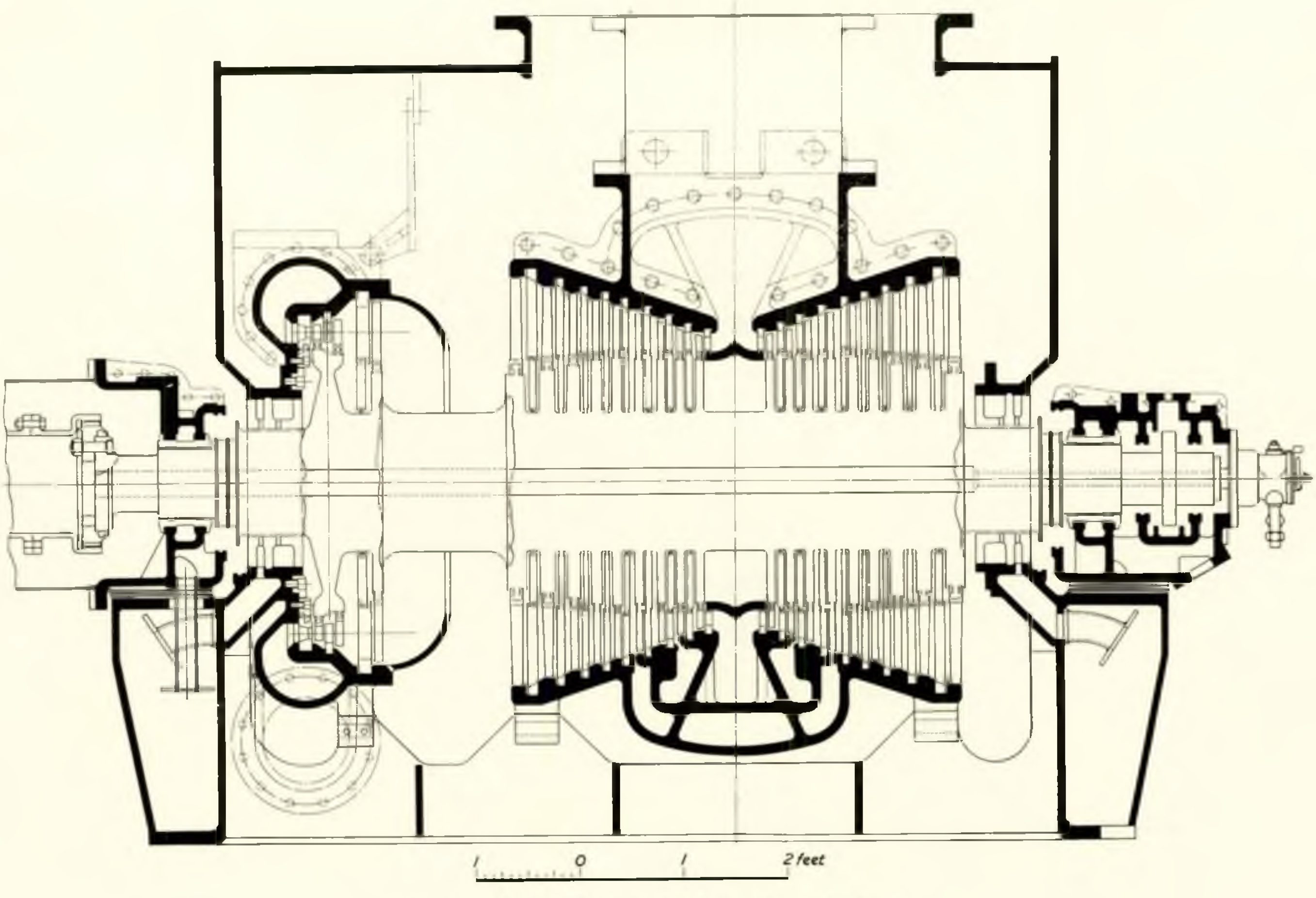


FIG. 6—*L.P. Turbine basis plan—"Fast Tankers"*

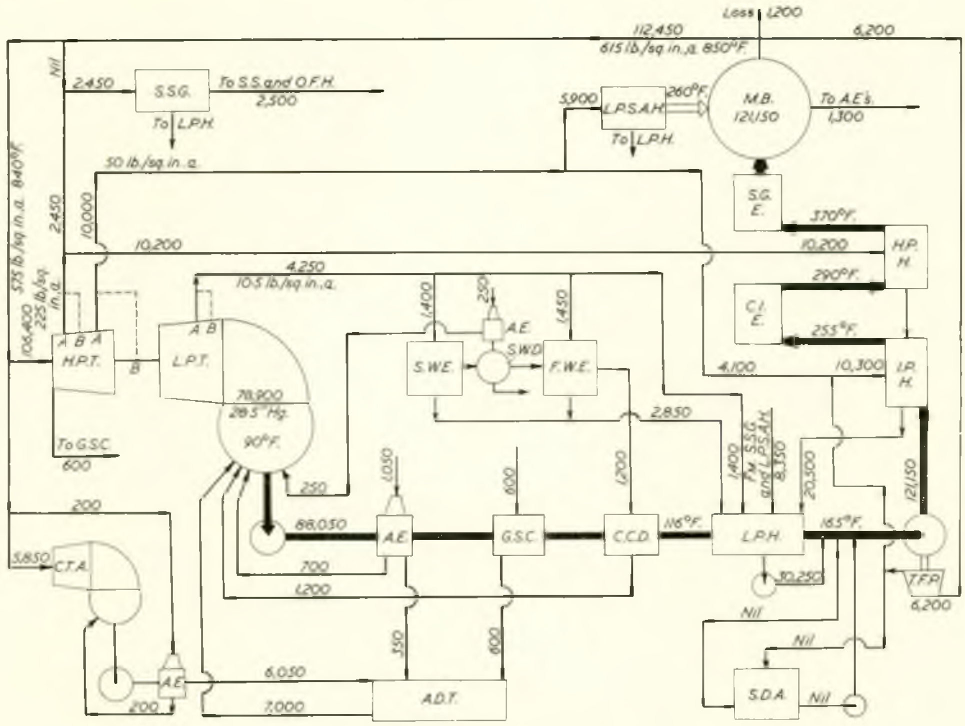


FIG. 7—*Heat balance—"Fast Tankers"*

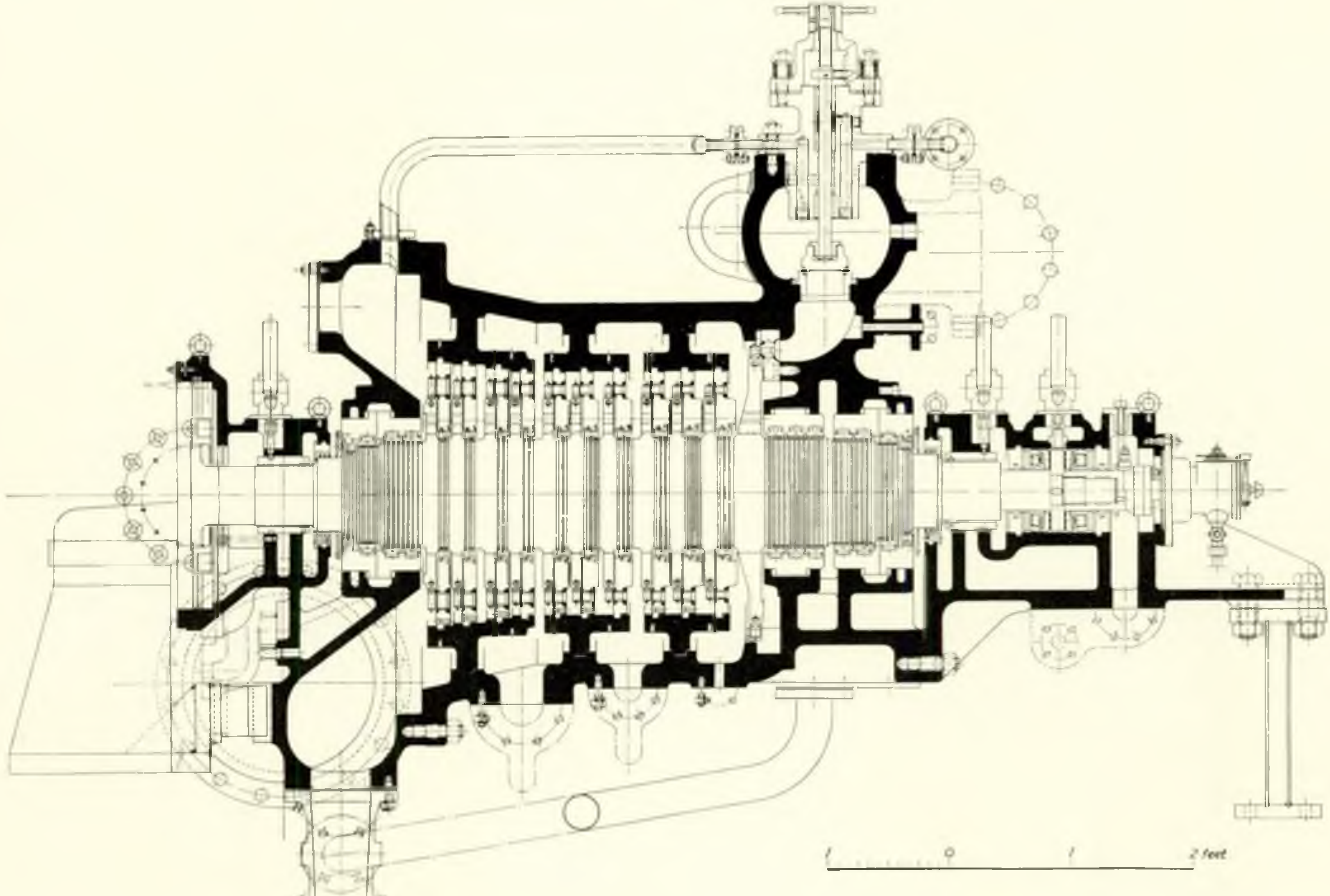


FIG. 8—*H.P. Turbine basis plan—Italian-built tankers*

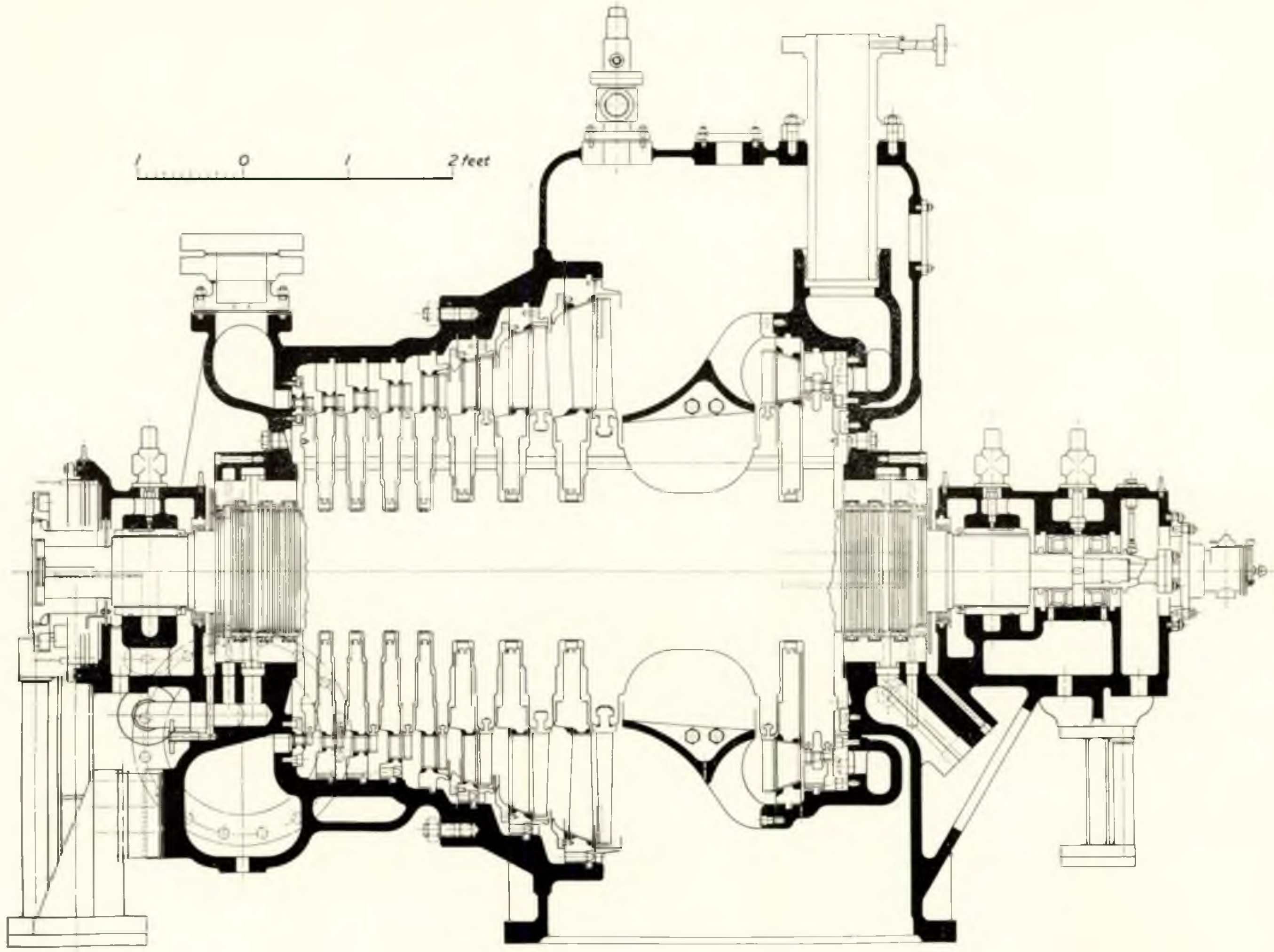


FIG. 9—*L.P. Turbine basis plan—Italian-built tankers*

and elsewhere, to the comparison between the performance of these Italian-built vessels and their U.K. counterparts and it has been suggested that an unfair comparison had been drawn between the British and the Italian/American turbines of differing dates of origin.

The facts are as follows:

	Designed steam rate	*Designed fuel rate*	*Fuel rate, all purposes trials Minimum*	*Maximum*
British ships	6·2	0·59	0·567	0·634
Italian ships	6·2	0·56	0·538	0·565

It will be noticed in the first place that the water rates and hence the internal efficiencies, of both turbines are identical. This serves merely to illustrate how hard won are the marginal gains in the internal efficiency of the turbines themselves, although there are many other factors affecting reliability and robustness which have been gained through the years by the embodiment of features which are generally common to development in all countries.

The features in the Italian-built installations which lead to their superiority in performance may be summarized as follows:

a) A well designed heat balance employing the maximum possible use of L.P. bled steam.
b) The well designed main steam system resulting in a low pressure loss (15lb./sq. in. gauge) between the superheater outlet and the H.P. turbine nozzle chest.
c) The use of cavitating extraction pumps in the closed feed system to obtain self-regulating control.
d) Careful attention to "margins" on auxiliary units by all parties concerned in the design.
e) High manufacturing standards, particularly in respect of turbine nozzle plates and blade paths. In practice a tolerance of ± 1 per cent was allowed during manufacture.

The lesson to be learnt is the importance of applying equal attention to all the details of the installation, so that the gains made in one feature are not offset by lack of attention to another. Considerable progress had been made in collaboration with the British industry on these lines before the Italian installations went into service; the success of this design merely served to illustrate, in advance of the application of this work, how valuable it was.

THE 16,000 S.H.P. INSTALLATIONS

42,000-d.w.t. Class

The next step in size of ship was to 42,000 d.w.t. for which 16,000 s.h.p. engines were required. A class of eight of these ships was laid down, one, *British Queen,* being later increased to 49,000 d.w.t., without change in the machinery.

The turbines and gearing were of similar overall design to those in the earlier classes of vessels and steam conditions of 600lb./sq. in. gauge, 850 deg. F. were retained.

As the shunt de-aeration system had proved to be expensive and less satisfactory than had been hoped it was decided in this class to adopt full feed de-aeration, using a low level de-aerator, taking advantage of more sophisticated methods for

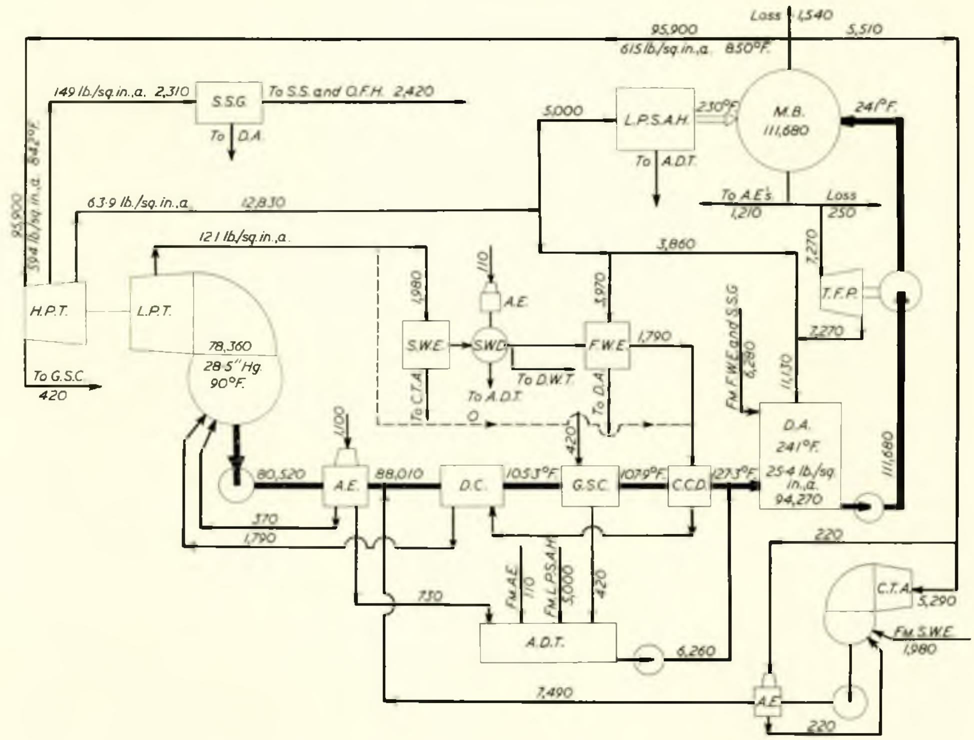


FIG. 10—*Heat balance—Italian-built tankers*

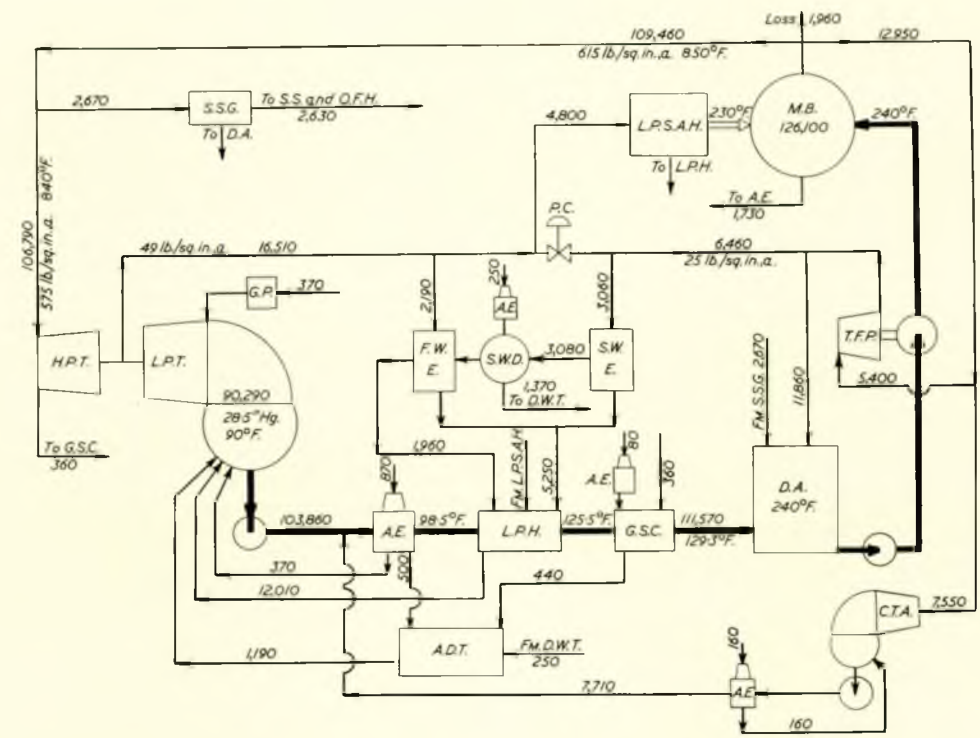
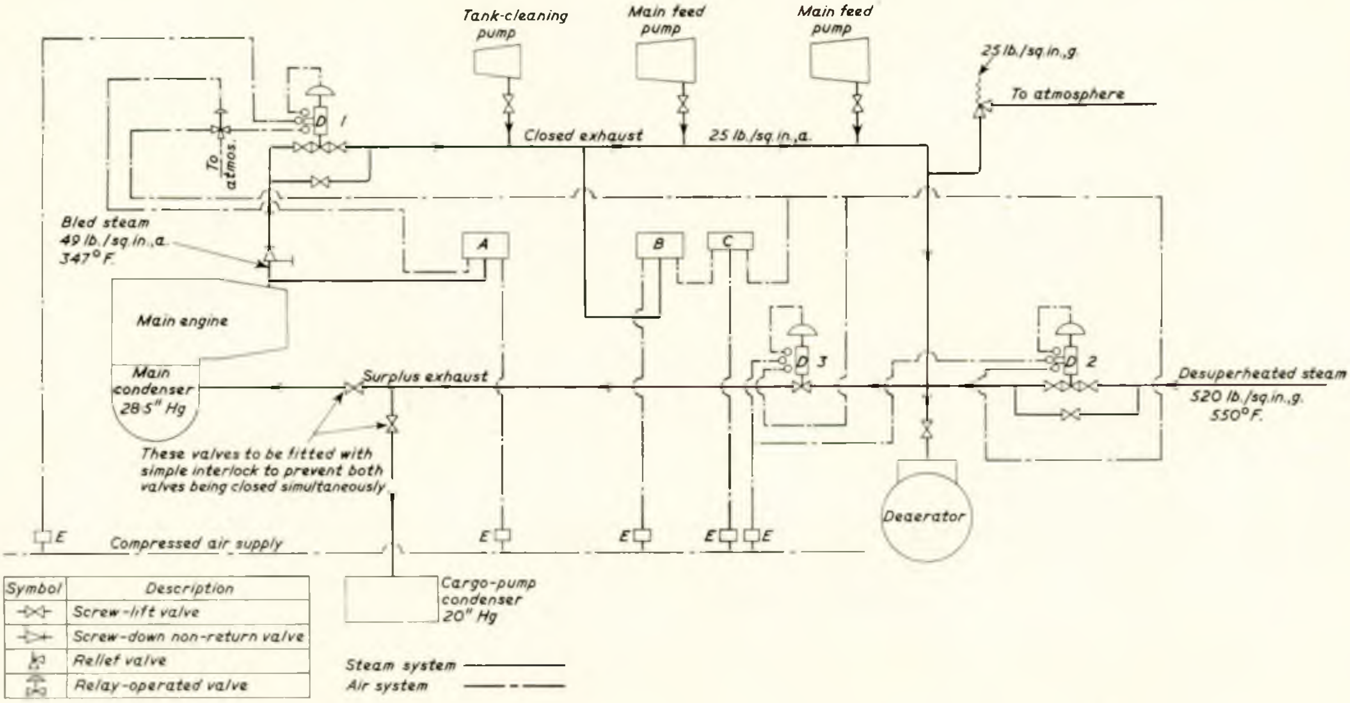


FIG. 11—*Heat balance—16,000 s.h.p., 600lb./sq. in. gauge, 850 deg. F.*



Item	*Description*	*Purpose*
1	Control valve	Bled steam make-up to closed exhaust line.
2	Control valve	Desuperheated make-up to closed exhaust line.
3	Control valve	Surplus exhaust to main or cargo-pump condenser.

Item	*Description*
A	Snap-action controller
B	Proportional controller
C	Integral action relay
D	Valve positioner
E	Combination filter regulator

FIG. 12—*Automatic control of closed exhaust system*

the control of the de-aerator in order to eliminate the difficulties previously experienced with full flow de-aeration. The final feed temperature, as indicated in the heat balance (Fig. 11) was 240 deg. F. corresponding to a de-aerator shell pressure of 10·25lb./sq. in. gauge. An integrated closed exhaust control system, employing pneumatically operated control valves was adopted (Fig. 12) and experience has shown that the de-aerator shell pressure can be maintained within close limits so that the operational difficulties associated with the 12,500 s.h.p. class have been avoided.

The performance of this class of vessel has been generally good. The designed fuel rate of 0·573lb./s.h.p. hr. has generally been achieved on trials and maintained or improved on in service.

50,000-d.w.t. Class—A new approach

When a new class of vessel of 50,000 d.w.t., propelled by 16,000 s.h.p. turbines was projected in 1957, an offer by Alexander Stephen and Sons Ltd. to collaborate in a comprehensive design study with a view to improving efficiency and reducing costs was accepted. The keynote of the design study was to be simplicity.

The manner in which this investigation was carried out, and the ground covered by it, was very similar to that described by Bonny[2].

The results of the study indicated that an increase in steam conditions to 600lb./sq. in. gauge, 900 deg. F. was economically justified. A further increase in steam temperature to 950 deg. F., while showing an apparent economic advantage of balance between first cost and fuel rate, was not adopted at this stage because:

a) It was considered desirable to obtain experience at 900 deg. F. under service conditions before meeting this further increase in temperature.
b) Experience in the design and erection of pipe systems for merchant ships at temperatures exceeding 850 deg. F. was still very limited in this country.
c) It was considered that there was insufficient evidence to justify an assumption that superheater fouling would not become a disadvantage offsetting the thermodynamic gain at the higher temperature.

At this temperature level, the design of H.P. turbine shown in Fig. 1, which has been a standard feature of the preceding classes of U.K.-built ships, was considered unacceptable and a double casing design (Fig. 13) was recommended and adopted. This design, besides being more suitable for higher inlet steam temperatures, showed promise of being simpler and cheaper to construct and offered a lower steam rate.

Examination of records showed that the overload power of 10 per cent which in all previous designs had been provided by a bypass valve in the H.P. turbine was never employed. This feature was therefore omitted from this and all future designs of turbine.

It was agreed with the builders that the nozzle area of the first stage nozzle plates should be manufactured to produce the designed 16,000 s.h.p. in service within a tolerance of +2 per cent −0.

It is an interesting side light on the rate of progress in

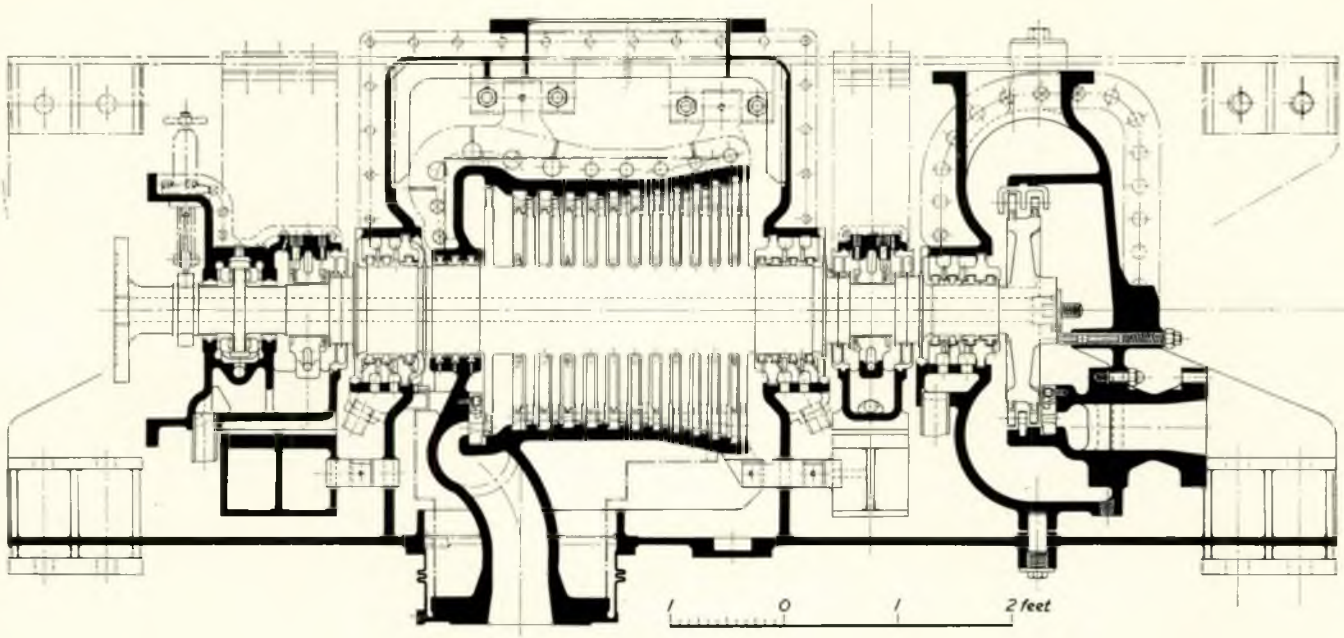


FIG. 13—*H.P. Turbine basis plan—50,000-d.w.t. vessels*

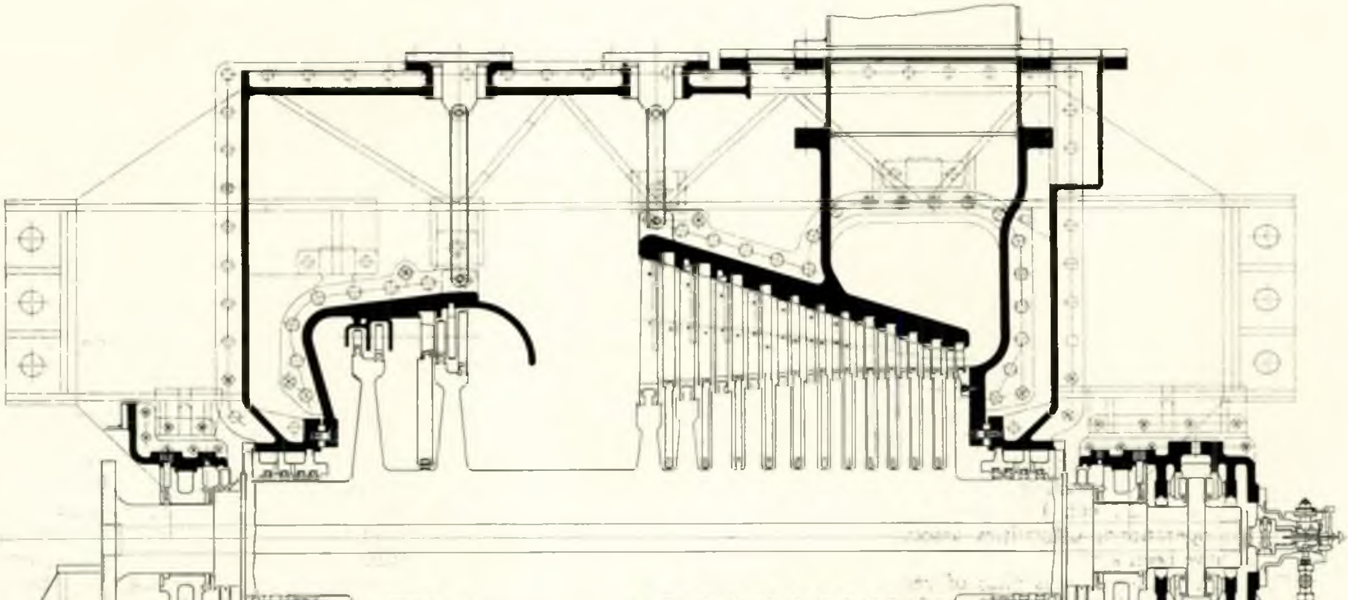


FIG. 14—*L.P. Turbine basis plan—50,000-d.w.t. vessels*

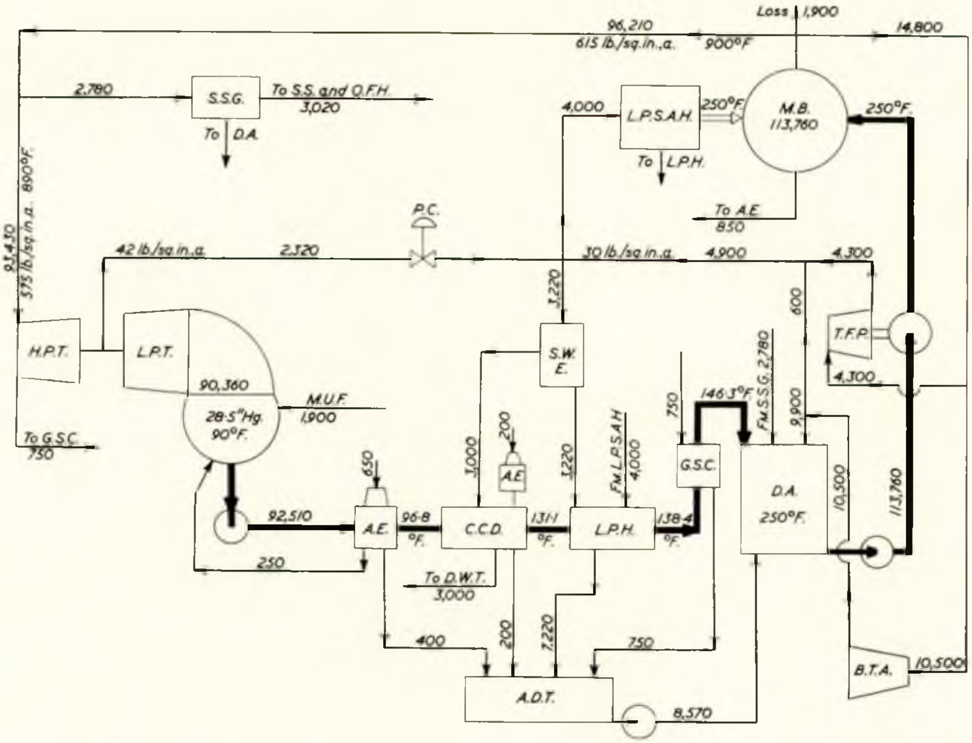


FIG. 15—*Heat balance—50,000-d.w.t. vessels*

turbine design to note that the 1950 turbine design was already regarded as obsolescent after an interval of only eight years, whereas the Parsons turbine design with end tightened blades had virtually remained a standard for more than 30 years.

The L.P. turbine selected, shown in Fig. 14, was of the single flow design, again on the grounds of higher efficiency and reduced cost.

With the accent on simplicity, a heat balance incorporating back pressure turbo-alternators and a single bleed point at crossover was selected (Fig. 15). The effect of multiple bleed points was considered but was not proceeded with since the inclusion of an H.P. bleed point was inconsistent with the turbine design selected, as its major advantages in simplicity are lost if a bleed point is introduced.

Careful consideration of comparative heat balances, in association with the operating cycle of the ships, indicated that the addition of an L.P. bleed point alone did not show any marked advantage. With back pressure alternators the addition of an L.P. bleed would result in a surplus of turbo-alternator exhaust. The possibility of including an L.P. bleed point to supply the evaporators only was considered but it was felt that the extra cost and complication were not justified. The designed fuel rate for the installation was calculated at 0·535lb./s.h.p. hr.

The opportunity was also taken during this design study to simplify as far as possible the auxiliary machinery installation and to reduce design margins to the minimum. One of the major factors affecting the fuel rates of the earlier classes of vessels was the cumulative effect of the widespread practice of adding margins to the capacity of the auxiliary units

Analysis of service results indicated that the margin of from 35 per cent to 45 per cent on boiler evaporation rate, which had been allowed for in previous classes to cater for the hot tank washing condition, was in fact only used for about four days per annum. This margin was therefore reduced to a level which allowed for cargo heating and for restricted hot washing with two tank cleaning machines only in use. As a result the boiler evaporation rate was reduced from 180,000lb./hr. (two boilers) to 150,000lb./hr., with consequent reduction in forced draught and oil fuel pumping and heating requirements. Similar attention was given to design margins on condensers and circulating pumps. As a result of these reductions in margin, the normal sea load fell from 600 kW to 520 kW enabling the turbo-alternator frame size to be reduced to 700 kW from 750 kW.

The arguments as to the relative merits of condensing and back pressure turbo-alternators continue to be discussed but may be summarized as follows:

Back pressure alternators offer reduced capital cost and fit neatly into a simple cycle employing a single bleed point provided that the electrical loading is low and the efficiency of the back pressure turbine is high. Due allowance must be made for "hotel" loading, which in modern tankers with full air conditioning is surprisingly high.

In the course of the design study for this class of vessel it was shown that the annual fuel loss from back pressure operation was outweighed by the saving in first cost. It should be noted that the reduction of 80 kW in the normal sea load had a marked effect on the economics of back pressure operation.

The overall effect of these design changes are calculated to show a fuel saving of about £10,000 per annum (assuming a bunker price of £7 10s. per ton) and a reduction in first cost of £10,000 as compared with the 16,000 s.h.p., 850 deg. F. installation in the 42,000-d.w.t. vessels.

A close resemblance will be noted between this design and that described by Main[9]. Both designs were evolved at about the same time although completely independently, and the similarity of conclusions reached is striking.

CONSIDERATIONS LEADING TO THE MACHINERY DESIGN OF 68,000-D.W.T. VESSELS

Originally, the 68,000-d.w.t. class of vessels were programmed to follow approximately one year behind the 50,000-d.w.t. class and it was assumed that the 20,000 s.h.p. machinery required for the larger ships would be a scaled-up version of the lower power installation. During this period, therefore, the higher power design was largely left in abeyance although the basic hull design was progressed. A very short first order look was taken at turbo-electric machinery on the basis that such installations, with electrically-driven cargo pumps, had proved very attractive operationally in the T.2 tankers, but the indications were so definitely against such an installation at this power level that the matter was not pursued.

Later on when the 16,000 s.h.p. design had advanced well into the detail stage with adequate time still available for preliminary work on the 20,000 s.h.p. installation and with service experience building up with the earlier U.K. and Italian-built classes, it was decided to review post-war experience carefully to determine the objectives for any future design. These objectives were defined as follows:

Firstly the machinery must be completely reliable and free from the defects experienced in earlier classes.

Secondly, the machinery should be simple to operate and require minimum maintenance.

Thirdly the fuel rate should be more in line with the best up-to-date world practice and should be not only capable of being demonstrated on trials but should be maintained in service.

The review of past experience included the following features:

Turbine and Boiler Defects in Manufacture and Service

1) *H.P. Turbine Thrust Bearings*

Failures occurred in eight ships in the Company's fleet and in at least two ships of associated fleets. These failures were initially regarded as isolated examples but on occasions repeated failures have occurred in the same ship and these instances have been more fully investigated. It is believed that these breakdowns have not been confined to the Company's experience but it has however been difficult to obtain an accurate estimate of the true frequency of this type of failure. Failures of this nature have been variously attributed to:

a) boiler priming resulting in a heavy overload of the bearing
b) errors in squareness of the thrust collar
c) emulsification of the lubricating oil
d) aeration of the lubricating oil
e) spark discharge through the oil film
f) oil supply failure
g) dirt in lubricating oil.

Each of these causes was investigated in the case of a ship which experienced repeated failures, but it was not possible to lay the blame on any single one of these factors, nor to clear any one from partial blame. All that can be said is that each of the above possible causes was suspected and that steps were taken to eliminate their recurrence. Latest information suggests that white metal fatigue may be added to the list, and this possibility is to be further investigated.

Special care is being taken in new designs to avoid blind pockets and to give straight through feed of oil in order to avoid the possibility of dirt lodging in holes which cannot be sighted.

2) *L.P. Turbine Blade Fractures*

Some fractures occurred of L.P. turbine blades, in all except one case in the last row blades of the double flow turbine rotor. The fractures were through the inner lacing wire hole, and in the early cases were at the forward end of the rotor. This led at first to the belief that they might have been caused by resonant vibration excited by steam flow variations in way of the exhaust steam deflector supports, and the support pitching was modified. Later instances however occurred at the after end of the rotor remote from the exhaust deflectors, and further investigations suggested that steam flow discontinuities in way of the horizontal casing joint might be the source of excitation. It was noted that the casing blade pitches at this point were not uniform, in some instances being up to twice the nominal blade pitch for the row.

Steps have been taken to ensure uniformity of blade pitch in way of the joint and to improve the standard of joint between the lacing wire and blade—which in all cases examined was found to be defective.

Nevertheless, it was felt to be worth while to carry out an extensive investigation (at the Pametrada research station) to provide data to prevent recurrence of this type of failure both in the turbines in which defects had developed and for future design applications.

Tests showed that careful consideration must be given to the batching of L.P. turbine reaction blades, particularly where these are of the tapered and twisted type in the last two rows.

The last two stages fitted with tapered and twisted blades now differ from the preceding stages in that the blades, instead of being batched in large continuous groups, are batched in small groups of 7 or 8 blades. The gaps in the inner wire are in line with those in the outer wire, and a junction wire is fitted at every gap, except at the closing packer where no closing blade is fitted. This method of batching gives the same safeguard against blade vibration as blades batched in large continuous groups, because of the junction wires, but in addition the danger from nodal diameter modes of vibration is reduced by having aligned gaps in the inner and outer wires.

The remaining stages are batched with the gaps in the inner wire midway between the gaps in the outer wire, and the number of blades between gaps is chosen to give maximum safeguard against any vibration of the upper half of the blades about the inner wire.

Resulting from these investigations criteria to inhibit vibration in reaction blade systems have been established which will be applied to future designs.

3) *H.P. Turbine Blade Failures*

Isolated failures have occurred in H.P. turbines, notably in the last (14th) row. These failures have not been fully investigated and have not recurred. They have therefore been attributed to defects in the blade as manufactured.

4) *H.P. Diaphragm Blade Roots*

An isolated case of diaphragm blade root fracture in an H.P. turbine at the horizontal joint has led to an improvement in design to overcome this trouble. In the failed installation the blade roots projected above the casing joint in a series of triangular projections, these being accommodated in corresponding recesses in the mating half-cylinder. It is believed that these fractures were due to differential expansion between the triangular projections and their mating recesses. A new design, giving a substantially flush finish, has been developed for more recent turbines of this design.

5) *L.P. Casing Distortion*

The double flow L.P. turbines installed in all classes of machinery so far in service have been prone to cylinder distortion. The trouble is indicated by heavy blade rubs along the bottom half-cylinder, sometimes occurring in each row of both forward and aft flows. Each class has suffered to some extent, the later classes perhaps more than the earlier due to smaller blade tip clearances being adopted in an attempt to improve the steam consumption of the machinery. Early designs may now have over 0·1in. blade tip clearance, achieved as the result of repeated dressing of blade tips. At this clearance they would seem to be free from further trouble, but as it amounts to over 1 per cent of the blade height it is reflected in the poor service results of the early classes.

This rubbing has been attributed to casing distortion due to exhaust steam impingement when "dumping" to the condenser, and to seized guide keys preventing free expansion of the cylinder. Both these features have received attention by modification but without apparently overcoming the trouble and blade rubs are still encountered. It is now considered that such distortions may be fundamental to the basic design, possibly allied to the tanker operating schedule which unavoidably entails extended periods at standby during loading and discharge. It is expected that these troubles will be eliminated in the single flow designs now entering service, and shore prototype tests on these will simulate the worst possible stand-by conditions.

6) *Blade Erosion*

L.P. blade erosion in the last four rows of each flow in the earlier classes has been overcome by fitting erosion shields to these blades. This has been done progressively as the rotors are removed for survey, and erosion shields have also been fitted in later machinery with this type of rotor. Following this experience it has been deemed wise to fit erosion shields to the later stages of the single flow L.P. turbines now entering service which despite slightly reduced exhaust wetness, are running with high blade speeds.

7) *Primary Wheel Thrust Bearings*

Trouble has been continually encountered with failures of the primary wheel thrust bearings of the articulated double reduction gearing. These have been overcome to some extent by improved oil supply to the bearing, but trouble still arises from time to time, due apparently to locking of the quill shaft couplings, particularly where the thrust was of indifferent design. In the Italian-built ships, where a tilting pad bearing was employed, trouble has not been experienced. The whole subject has been fully examined recently by Pametrada and Lloyd's Register of Shipping, with the result that new recommendations for design loading, and design of the quill shaft couplings have been made. These recommendations are being considered for installation in the latest class of machinery, which will also be fitted with tilting pad primary wheel thrust bearings.

8) *Turbine Primary Couplings*

Ever since the early 28,000-ton class, trouble has been experienced with wear of turbine primary couplings and progressive attempts have been made to improve conditions and extend coupling life. These have not been wholly successful, however, and the 35,000-d.w.t. class machinery in particular has had a poor record in this respect, all couplings being due for renewal after four years service.

This rapid wear has been variously attributed to coupling design, tooth loading, and the method of oil feed to the coupling teeth.

All these turbines employ the fine-tooth coupling which replaced the older claw type coupling because it removed a mode of resonance which could be established between the coupling tooth frequency and certain expansions of reaction blading, as well as promising to provide a more flexible connexion between turbines and gearing. The 35,000-d.w.t. class couplings were supplied by a radial jet of oil impinging between the teeth at the centre of the face width. Loading marks at the pinion end of the couplings indicated, however, that lubrication transverse to the radial direction was also required.

Analysis of a wide range of H.P. flexible couplings revealed that failure could not be attributed to any of the following characteristics alone:

Number of teeth
Loading per inch width
Lubricating oil feed
Length/diameter ratio
Tooth surface treatment
Face width of teeth
Tooth form
Mating material combination.

It is possible however that this was because of the lack of clear cut design criteria. Couplings loaded to 320lb./in. face width were satisfactory while those loaded to 277lb./in. failed. These latter, however, were lubricated at the centre of the face only, while those with satisfactory performance and higher loading were lubricated either at the end only, or at the centre and the end. In all cases the mating materials were EN.8/EN.25 and the surface treatment was the same.

In the authors' opinion the importance of ensuring an adequate flow of oil from one end of the teeth to the other, combined with conservative tooth loadings, cannot be over-emphasized if longevity is to be achieved.

The more recent designs of flexible coupling are based on the concepts of ease of inspection, increased number of teeth (suitably barrelled and crowned), the length/diameter ratio increased to greater than unity, and the male teeth of the coupling arranged on the torque tube, with mating female teeth on the short sleeves attached to the turbines and pinions respectively.

It would appear that when loading marks appear to be greater towards the pinion end of the coupling, end lubrication plays an important part in extending coupling life.

9) *Condenser Defects*

Condenser tube perforations have occurred from time to time in all classes and were thought to have been overcome by the adoption of aluminium brass condenser tubes. Recently however, trouble has occurred on 42,000-d.w.t. class ships with tube perforations adjacent to the tube inlet end. These failures have been investigated and it is now the practice to fit short tube inserts into the tube inlet ends to shield the condenser tube in the perforation zone. These inserts appear to have been effective and are being continued in the later classes.

It is of interest that the Italian-built 35,000-d.w.t. class ships have condenser tubes expanded into the tube plate at both ends of the tube, and have a bellows fitted in the condenser shell to accommodate thermal expansion. This has been most successful in service and there has not been a single case of tube end leakage in over three years service in these ships. During the same period of time the 42,000-d.w.t. class ships were being taken over in the U.K. and several cases of tube end leakage have occurred, usually during the early running of each ship.

The service experience obtained with expanded tubes has been so satisfactory that condensers with this detail have been specified in 68,000-d.w.t. class machinery.

10) *Boilers*

While much has been said in the foregoing of factors affecting reliability in the turbines themselves, it will be noted that most of the criticisms raised refer to minor defects of an otherwise reliable machine. Indeed, the very fact that so much attention has been drawn to these points of detail is, in a way, a commendation as to the turbine's reliability. However, the main turbines themselves are only one factor in the reliability of a turbine driven ship.

Boiler defects are undoubtedly the greatest single item prejudicing the reliability of turbine driven vessels. Reference has already been made elsewhere in this paper to air heater chokage and boiler tube failures. These defects have now been largely overcome by the methods described. One factor affecting tube failures has not however been mentioned. It became apparent some three years ago that tube failures of the type described were occurring with predictable regularity after about one year's service. Further, the tubes most likely to fail could be indicated in advance. Invariably these tube failures occurred in fire row and water wall tubes located about three feet from the front wall. The failures always occurred about three feet above the floor and were in the form of blisters on the tubes which eventually perforated or split. Similar defects associated with the presence of metallic oxides have been reported in high pressure land boilers. The mechanism of failure appears to be that the oxides build up on the tube wall, causing a local increase in tube thickness with resultant rise in metal temperature. Thereafter the film of feroso/ferritic oxide which normally protects the tube from corrosion breaks down locally exposing the underlying metal which corrodes, the resultant corrosion products forming what has been aptly described by Hankinson and Baker[6] as a "barnacle". Due to the porous nature of the "barnacle" and the locally high metal temperature, water seeping under the "barnacle" becomes a progressively concentrated solution of impurities. Accelerated corrosion, probably of an electrolytic nature, thereafter occurs, which leads to a progressive reduction in the thickness of the tube wall which first bulges and then finally perforates, the perforation being sometimes in the form of a pin hole and at other times in the form of a split.

A secondary effect, which is suspected in high pressure land boilers, is that the nascent hydrogen produced as a corrosion product, being in a highly active form, attacks the carbon within the structure of the steel to form methane. This "hydrogen embrittlement" effect is of particular significance in the grain boundaries since these are weakened until, finally, almost completely separated grains are left. Thereafter the tube bulges as previously described. This secondary effect is probably of little significance in a comparatively low pressure marine boiler where, despite the high furnace rating, the corrosive action is probably entirely electrolytic.

The only cure for these troubles is to prevent the ingress of metallic oxides to the boilers and, as already described, the separation of the main and auxiliary steam and condensate systems had not, apparently, cured the trouble. It is now believed that the root cause of the trouble can be traced to oxidation of the steam and feed systems during construction. It is suspected that the oxides are flushed into the boilers during the initial steaming period and tend to accumulate in pockets so that normal blow-down procedures are ineffective in removing these deposits.

It is therefore now the Company's practice to clean the boilers chemically after trials and again after one year's service. This approach has proved effective and boiler tube failures early in service have now been virtually eliminated. This experience leads to the consideration whether the geometry of the modern marine boiler, based on naval prototypes, is correct for a merchant ship. It is clear that, despite the nominally conservative ratings of merchant marine boilers in terms of heat release per cubic foot of furnace volume and per square foot of radiant heating surface, the actual local heat transfer rate in the area prone to tube failures of this type must be higher than that in the highest pressure boilers in land use, particularly if flame clearances are inadequate. Conditions favourable to hydrogen embrittlement could therefore conceivably arise.

Frequent brick work renewals are a further indication of insufficient furnace volume since, although vibration may be a contributory factor, flame impingement is probably the major cause.

All these considerations lead one to consider that a "new look" in boiler design is overdue and that the ideal boiler for tanker service should include the maximum furnace volume which can be accommodated within the contours of the ship. Bearing in mind that height is seldom a problem in a tanker, one inclines to the view that a boiler, based on larger water cooled furnaces of conservative rating and generous flame clearances, might well prove to be the ideal.

Mention should be made at this stage of the advances in oil-burning techniques within recent years. New designs of air register have appreciably reduced the furnace volume required for complete combustion, while the re-introduction of steam atomizing, in the form of steam assistance, is expected to assist in maintaining boiler fireside cleanliness.

The fact remains however that considerable scope exists for new measures to improve boiler reliability and to reduce boiler maintenance costs.

DEVELOPMENT OF THE 20,000 S.H.P. DESIGN

Basic Design Requirements

Following the analysis of experience in previous classes of vessel, already described, the requirements were laid down for the design study which was carried out by John Brown and Co. (Clydebank) Ltd.

a) While following the concept of maximum economy at normal service power, combined with simplicity, it was accepted that the machinery should be an advance on the previous 16,000 s.h.p. design, in that greater use would be made of regenerative feed heating.
b) The steam conditions were to be based on 600lb./sq. in. gauge, 900 deg. F., but an investigation was to be made covering the range from 600lb./sq. in. gauge, 900 deg. F. to 850lb./sq. in. gauge, 950 deg. F.
c) The main machinery was to consist of a single-screw geared turbine installation operating at a normal service power of 20,000 s.h.p. at 105 r.p.m. with no overload and no nozzle control requirements.
d) The H.P. turbine of all impulse type incorporating a double casing design similar to the 50,000-d.w.t. class design was to be taken as the basis for comparison, but alternative designs were to be considered as required to suit alternative feed systems. For the L.P. turbine, both single flow and double flow arrangements were to be considered.
e) To keep down gearing sizes, K values up to Lloyd's revised limits were acceptable. Consideration was to be given to both articulated and locked train designs. B.S. 1807 Class A2 was considered suitable for such gearing.
f) The most suitable condenser vacuum was to be assessed taking into account the specified operating schedule, the condenser tube water velocity and tube diameter, the condenser first cost, the main circulating pump first cost and power requirements, and the overall machinery fuel consumption.
g) The boilers were to be designed for maximum efficiency at an evaporation corresponding to normal service power with the maximum output arranged to provide additional steam sufficient for hot washing with two tank cleaning machines and one stripping pump.
h) Special consideration was to be given to the assess-

TABLE 1—COMPARISON OF SIMPLE AND SPLIT ECONOMIZER FEED CYCLES FOR 68,000 D.W.T. CLASS

System	Simple Cycle		Split economizer cycle	
Proposal	A.	B.	C.	D.
H.P. turbine	Double casing	Double casing	Single casing	Single casing
L.P. turbine	Single flow	Double flow	Single flow	Double flow
H.P. turbine power	11,000 s.h.p. at 4790 r.p.m.	11,000 s.h.p. at 4790 r.p.m.	11,000 s.h.p. at 4790 r.p.m.	11,000 s.h.p. at 4790 r.p.m.
L.P. turbine power	9,000 s.h.p. at 2370 r.p.m.	9,000 s.h.p. at 3310 r.p.m.	9,000 s.h.p. at 2370 r.p.m.	9,000 s.h.p. at 3310 r.p.m.
Total machine weight (tons)	Basis	−7·0	+45	+38
Initial cost of machinery installations	Basis	−£3,300	+£18,600	+£16,100
Turbine steam rate (non-bleed) lb./s.h.p.hr.	5·76	5·79	5·76	5·79
Fuel rate (all purposes) lb./s.h.p.hr.	0·525	0·527	0·512	0·514
Fuel consumption per annum (tons)	30,375	30,490	29,625	29,740
Fuel cost per annum	£227,800	£228,700	£222,200	£223,000
Differential fuel cost per annum	Basis	+£900	−£5,600	−£4,800
15 per cent amortization	Basis	−£500	+£2,800	+£2,420
Overall saving or loss per annum	Basis	£400 loss	£2,800 saving	£2,400 saving

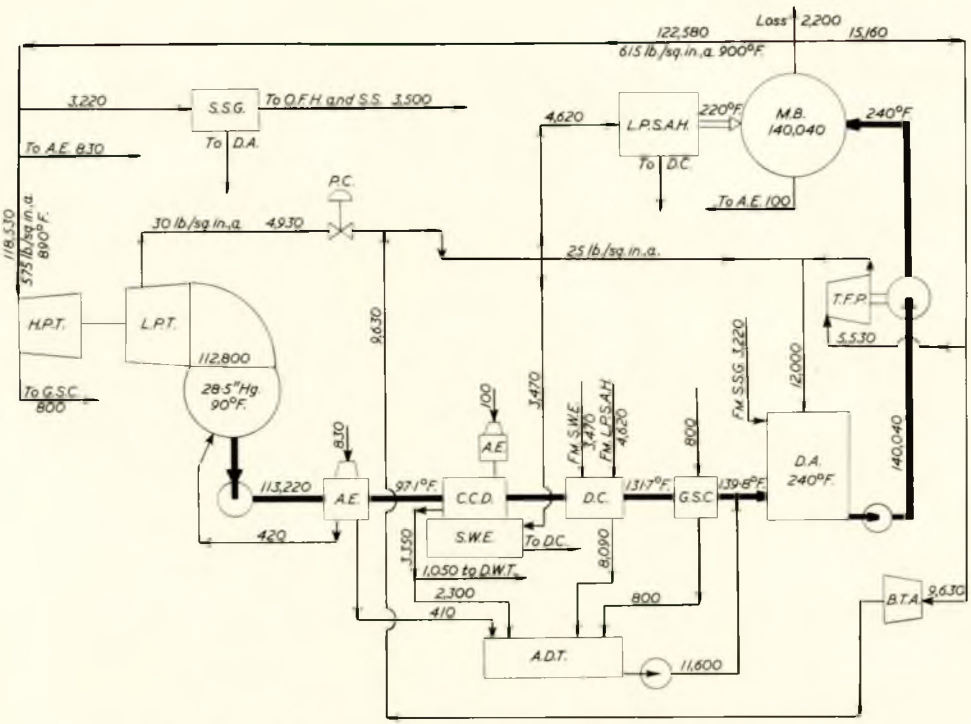


FIG. 16—*Heat balance—Simple system for 68,000-d.w.t. vessels*

ment of margins on heating surfaces and associated boiler auxiliaries in order to ensure that these were only as large as were necessary.

j) The maximum steam temperature leaving the superheater was to be arranged to correspond to the normal service power evaporation.

With these basic requirements defined, specific design studies were carried out on the many alternatives, the more important of which are described below.

Steam Conditions and Comparison of Feed Cycle

Taking into account both fuel consumption and first cost,

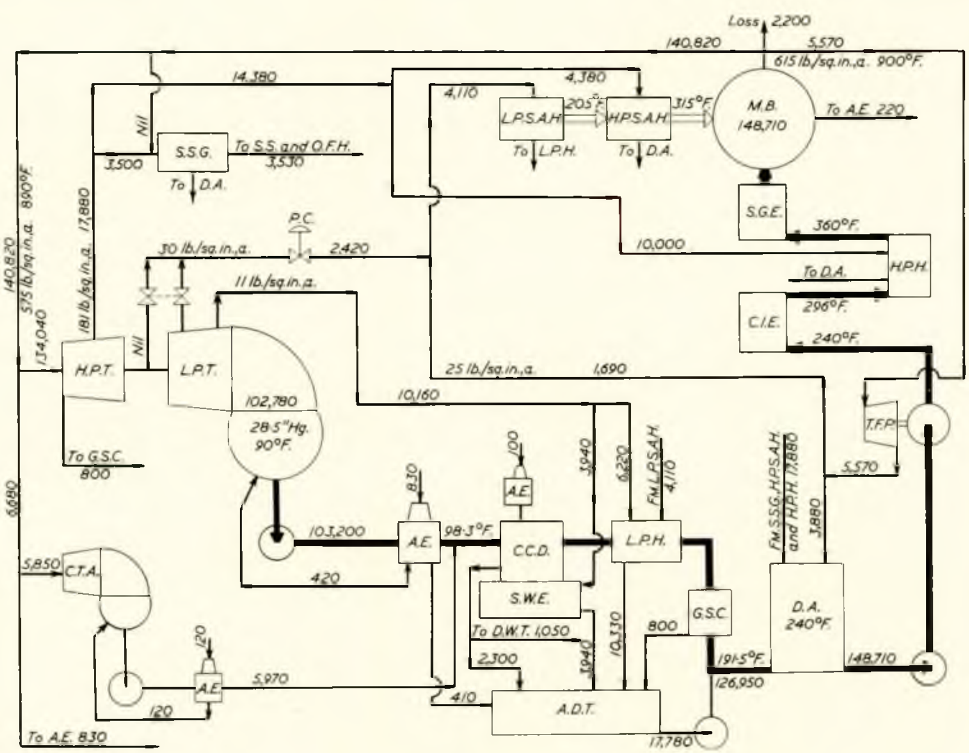


FIG. 17—*Heat balance—Split economizer system for 68,000-d.w.t. vessels*

it is generally true to state that a more complicated feed cycle shows greater return at more advanced steam conditions than at lower steam conditions. There is, of course, an economic limit to both the degree of complication and the steam temperature and pressure.

It was therefore necessary to examine each of several feed cycles in turn with selected steam conditions to obtain a true picture of the optimum arrangement.

An analysis of alternative heat balances was made including single-bleed and multi-bleed arrangements, back pressure and condensing turbo-alternators, electric and turbo-driven feed pumps, etc.

It is not proposed to deal with all the alternatives considered in this paper. Suffice it to say that the choice was reduced to a short list of two cycles, shown in Figs. 16 and 17 respectively, both based on steam conditions of 600lb./sq. in. gauge, 900 deg. F. at the superheater outlet. Table I summarizes the results for these two cycles including annual fuel consumption and cost (assuming fuel at £7 10s. per ton) and annual costs, allowing 15 per cent of the first cost for depreciation, interest, insurance and maintenance charges.

alternative turbine designs were studied, as follows:

a) H.P. turbine double casing design
b) H.P. turbine single casing design with diaphragm carriers
c) L.P. turbine double flow design
d) L.P. turbine single flow design

a) *H.P. Turbine Double Casing Design*

The double casing H.P. turbine followed the improved 16,000 s.h.p. design already discussed.

This design of turbine was proposed for the simple feed cycle shown in Fig. 16.

b) *H.P. Turbine Single Casing Design with Diaphragm Carriers*

This design of turbine which was proposed for the feed cycle shown in Fig. 17 was suitable for multi-bleed points and had the inner barrel made up of separate short barrel portions. Unlike the double casing turbine, the outer casing and its bolting are subjected to the pressure at the various bleed points in the turbine.

TABLE II—COMPARISON OF TURBINE PROPOSALS FOR 68,000 D.W.T. CLASS

Steam conditions at turbine 560 lb./sq.in. gauge 890 deg. F.

System	Simple feed system	Split economizer feed system	
Proposal	1	2	3
Casing construction			
H.P. turbine casing	Double	Single + diaphragm carriers	Single + diaphragm carriers
No. of bleed points	Nil	One	One
L.P. turbine casing	Double casing, single flow	Double casing, single flow	Double casing, double flow
No. of bleed points	Nil	Two	Two
Blading and rotor details			
H.P. turbine ahead	11 rows—all impulse blading 24·0 in.—25·95 in. mean diameter	12 rows—all impulse blading 24·0 in.—25·95 in. mean diameter	
astern	Overhung 2-row Curtis wheel	Overhung 2-row Curtis wheel	
L.P. turbine ahead	6 impulse + 6 reaction rows 46·9 in.—58·4 in. mean diameter	6 impulse + 6 reaction rows 47·2 in.—58·4 in. mean diameter	2x6 impulse + 2x5 reaction rows 34·4 in.—40·1 in. mean diameter
astern	2-row Curtis + 1-row impulse wheel	2-row Curtis + 1-row impulse wheel	
Type of gearing	Double reduction articulated	Double reduction articulated	
Power and speed distribution			
H.P. turbine	11,000 s.h.p. at 4,760 r.p.m.	11,000 s.h.p. at 4,790 r.p.m.	11,000 s.h.p. at 4,790 r.p.m.
L.P. turbine	9,000 s.h.p. at 2,370 r.p.m.	9,000 s.h.p. at 2,370 r.p.m.	9,000 s.h.p. at 3,310 r.p.m.
Weights			
H.P. turbine and supporting beam; L.P. turbine and gearing complete	136·0 tons	136·5 tons	132·0 tons
Initial cost differentials			
H.P. turbine, L.P. turbine and gearing complete	Basis	−£4,000	−£6,500
Non-bleed turbine water rate lb./s.h.p. hr.	5·76	5·76	5·79

For the purpose of estimating the annual fuel consumption the figure of 270 days per annum full power steaming at sea has been taken. This figure is regarded as more realistic than those frequently quoted elsewhere. It does not indicate the total number of days per annum in service and refers only to the length of time spent at the full power level. The importance of interpreting comparative heat balances in association with an accurately assessed operating schedule cannot be emphasized too strongly. It is not enough merely to arrive at a correct figure for the service power days steaming, and it is indeed of at least equal importance to assess correctly the periods under reduced power for manœuvring, canal passage, and under pilotage as well as the port times loading, discharging and under repair. The authors' companies have learned a great deal about the fundamental problems of machinery installation design in the course of their collaboration in this work.

Comparison of Turbine Designs

Concurrently with the examination of these systems

The astern turbine was similar to that for the double casing design.

c) *L.P. Turbine Double Flow Design*

This was an improved version of the conventional pattern of double flow turbine previously employed in the early 16,000 s.h.p. design, the first six stages being impulse and the remainder being reaction. Each half of the inner barrel was cast separately and was tied together at the steam inlet belt, the whole being supported in palms by the outer casing.

The astern turbine was arranged at the forward end of the casing.

d) *L.P. Turbine Single Flow Design*

This again was a mixed impulse/reaction turbine with the astern at the forward end, and followed the improved 16,000 s.h.p. single flow L.P. turbine design. The inner barrel carrying the diaphragms and fixed blading was of cast and fabricated design.

A recent paper by Veitch[11] discusses the choice of L.P.

turbines and it is interesting to note the improvement in performance of the single flow turbine of modern design over the conventional double flow turbine.

Table II shows a comparison of these turbines and includes brief design particulars, costs, weights and turbine water rates.

The conclusions from the above studies may be summarized as follows:

1) For any of the feed cycles considered an overall annual saving can be achieved by raising the steam temperature to 950 deg. F. with the pressure retained at 600lb./sq. in.
2) The overall annual savings that can be achieved at steam conditions of 850lb./sq. in. gauge, 950 deg. F., although being greater than at 600lb./sq. in. gauge 950 deg. F., do not show the same proportionate increases, there being a diminished return due to the high cost of the boiler and feed pumps and piping required for 850lb./sq. in. gauge.
3) The split economizer feed system shows definite overall gains in spite of increased costs over the simple feed system with either steam air heating or with a combination of both steam and gas air heating.
4) The choice of the H.P. turbine design is tied with the choice of the feed cycle although the reduced cost of design b) for the multi-bleed split economizer feed system naturally contributes to making the latter more attractive.
5) The single flow L.P. turbine showed no appreciable difference in efficiency or cost over the corresponding double flow turbine, but its short distance between bearing centres is considered to be a significant contribution towards avoiding the casing distortion experienced with the double flow designs under standby conditions.

Recommendations

On the basis of the above conclusions, recommendations were made that the installation should incorporate a split economizer feed system with an H.P. turbine of the single casing design having diaphragm carriers and a single flow L.P. turbine.

Practical and economic considerations suggested a preference for steam pressure lower than 850lb./sq. in. gauge, and it was felt that this should be retained at 600lb./sq. in. gauge, a pressure at which operational experience had already been gained. As regards steam temperature, it was recommended that advantage should be taken of benefits to be obtained with 950 deg. F., and the machinery will be designed for this temperature, but operated initially at 900 deg. F. until a satisfactory background of service experience has been obtained at this temperature in the 50,000-d.w.t. class, which will only enter service 18 months ahead of these larger vessels.

Selection of Feed Cycle

The finally selected heat cycle therefore was that corresponding to Fig. 17, incorporating the maximum possible use of bled steam from the L.P. turbine, and including a condensate circulated distiller, combined L.P. feed heater and drain cooler, de-aerator, split economizer with H.P. feed heater, and two stages of bled steam air heating, and was calculated to give an average service fuel rate of 0·512lb./s.h.p. hr. corresponding to a daily fuel consumption of 110 tons, based on a turbine water rate of 5·82lb./s.h.p. hr. at 900 deg. F.

TABLE III—COMPARISON OF BACK PRESSURE AND SELF-CONDENSING ALTERNATORS FOR THE SPLIT ECONOMIZER FEED SYSTEM FOR THE 68,000 D.W.T. 20,000 S.H.P. CLASS

	Feed system	Specific fuel consumption lb./s.h.p. hr.	Differential annual fuel bill	Differential cost of system	Capitalization and amortization differential at 15 per cent	Overall annual saving or loss
1	2 low efficiency back pressure turbo-alternators (29·0 lb./kw.-hr.) 1 H.P. bleed point at 181 lb./sq.in.abs. 1 H.P. feed heater—final feed temperature 360 degrees F. 1 condensate circulated distiller 1 drain cooler Bleed available from H.P. crossover at 57 lb./sq.in.abs.	0·527	Basis	Basis	Basis	Basis
2	As for arrangement 1 but with 2 high efficiency back pressure turbo-alternators (18·7 lb./kw.-hr.) Bleed from H.P. crossover at 57 lb./sq.in.abs. supplements exhaust line	0·513	−£7,300	+£18,000	+£2,700	£4,600 saving
3	2 high efficiency self-condensing turbo-alternators (10·9 lb./kw.-hr.) 1 H.P. bleed point at 181 lb./sq.in.abs. 1 H.P. feed heater final feed temperature 360 degrees F. 2 L.P. bleed points at 30 lb./sq.in.abs. and 11 lb./sq.in.abs. 1 L.P. heater/drain cooler 1 condensate circulated distiller	0·512	−£8,200	+£36,700	+£5,505	£2,695 saving
4	As for arrangement 3 but with 1 high efficiency and 1 low efficiency self-condensing turbo-alternator (10·9 and 26·0 lb./kw.-hr. respectively)	0·512	−£8,200	+£21,000	+£3,150	£5,050 saving

To determine the choice of self-condensing or back pressur turbo-alternators, several studies were made and these an summarized in Table III. It will be seen that whilst there is negligible difference at normal service power conditions between a cycle including a high efficiency back pressure set and a high efficiency self-condensing set, there would however be an overall loss in fuel consumption per annum with the back pressure arrangement due to the fact that dumping would take place at and below 95 per cent of normal service power, and under harbour conditions.

It was decided therefore to install one high efficiency condensing alternator which is intended to be in use under all normal conditions and one low efficiency alternator exhausting to the auxiliary condenser as a standby unit.

It will be noted that this arrangement of one high efficiency and one standby low efficiency set, whether it be for a back pressure or for a condensing unit, gives a considerable saving in first cost over the more usual arrangement of two identical high efficiency units.

The cost saving of £15,700 accruing from the artifice of a low efficiency standby alternator will be beneficial provided that the maintenance costs of the high-efficiency turbo-alternator are not increased substantially by its almost continuous use. It is confidently expected that this machine, which is built to the same standards as a main propulsion turbine, will benefit rather than lose from this continuous usage. Consideration is being given to the extension of this philosophy to other auxiliaries.

Further consideration indicates that a turbine water rate of 5·75lb./s.h.p. hr. at 900 deg. F. is likely to be achieved which reduces the fuel rate to 0·508lb./s.h.p. hr. Using cold weather electrical loadings, and reducing the losses to the more optimistic values frequently quoted, the fuel rate for this particular installation would calculate at 0·495lb./s.h.p. hr. An increase in superheat temperature to 950 deg. F. would reduce the fuel rate to 0·489 and an increase in pressure to 850lb./sq. in. would further reduce the consumption to 0·483lb./s.h.p. hr. The possibilities of this type of cycle therefore are by no means exhausted. Impressive though such figures may look when quoted, however, the figure of 0·512lb./s.h.p. hr. is more important since this represents the average performance which can reasonably be expected in service.

Selection of Turbine Design

Having decided upon a heat cycle, the next step was to select a turbine design which was

a) suitable for multi-bleed operation
b) capable of further development to permit of operation at higher temperatures and pressures
c) of an inherently rugged design, capable of absorbing transient thermal conditions without distortion
d) capable of maintaining a reasonable efficiency at part load operation.

The double casing H.P. turbine selected for the 50,000-d.w.t. vessels was eliminated on count a) since this turbine is only suitable for the simplified type of installation. Several alternative turbines of both U.K. and foreign design were examined, criticized and discussed. The close co-operation of the builders and of Pametrada were willingly forthcoming and as a result of these discussions the design offered by Pametrada (Fig. 18) was selected. This design, in effect, synthesized the more attractive features of several recent Pametrada basis designs, and is now offered as a standard frame size turbine covering the power range of 15,000 to 20,000 s.h.p. over a range of steam conditions varying from 600lb./sq. in. gauge, 900 deg. F. to 800lb./sq. in. gauge, 950 deg. F.

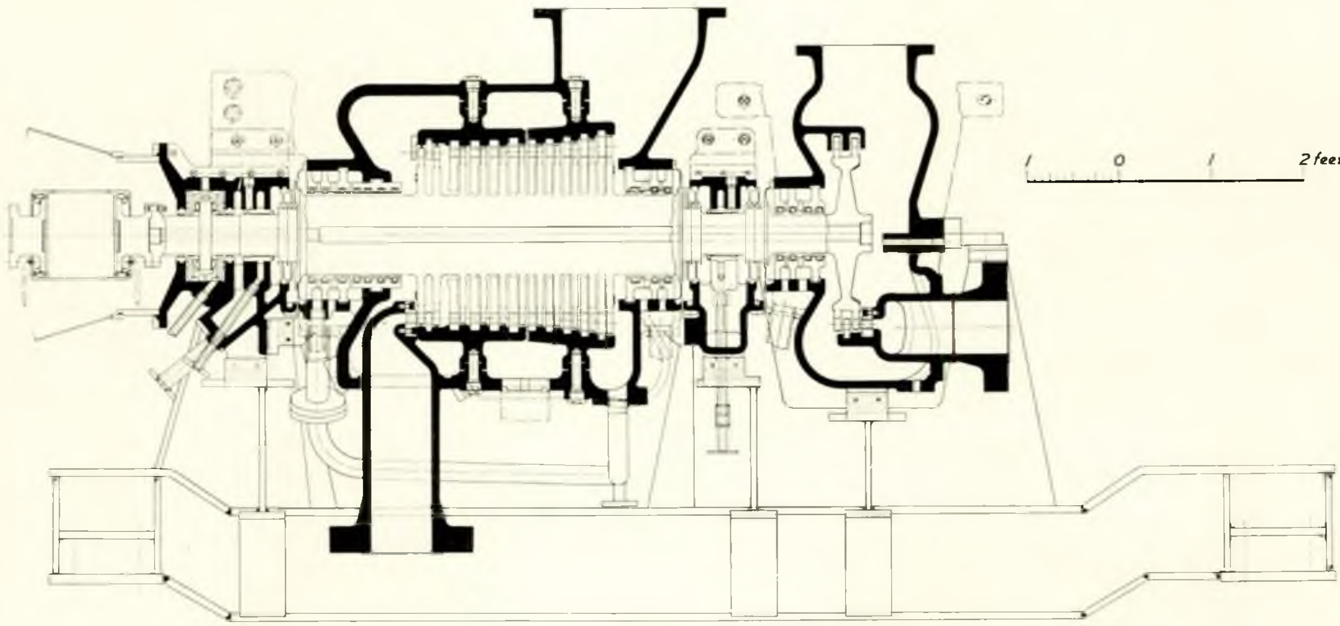


Fig. 18—*H.P. Turbine basis plan—68,000-d.w.t. vessels*

The design incorporates a divided nozzle belt for part load economy and a two row Curtis wheel in the first stage. These features were contrary to the original Statement of Requirements, but were considered to be acceptable for the projected vessel since experience in recent years had indicated that a certain amount of flexibility can be valuable in tanker operation. Such ability to operate economically at two power levels is a useful bonus if obtainable without undue penalty at the normal service power condition. Furthermore, since this H.P. turbine was intended as a "standard frame" to cover both a range of power and a variety of owners' requirements, the inclusion of a Curtis wheel stage was clearly desirable.

Following the two-row impulse wheel there are nine single-row impulse stages. The nozzles and diaphragms are of welded construction and the diaphragms are supported in plain grooves in the short inner cylinder sections—and are thus easily accessible for inspection, repair or renewal. An arrangement of supporting palms, key and dowel pins serves

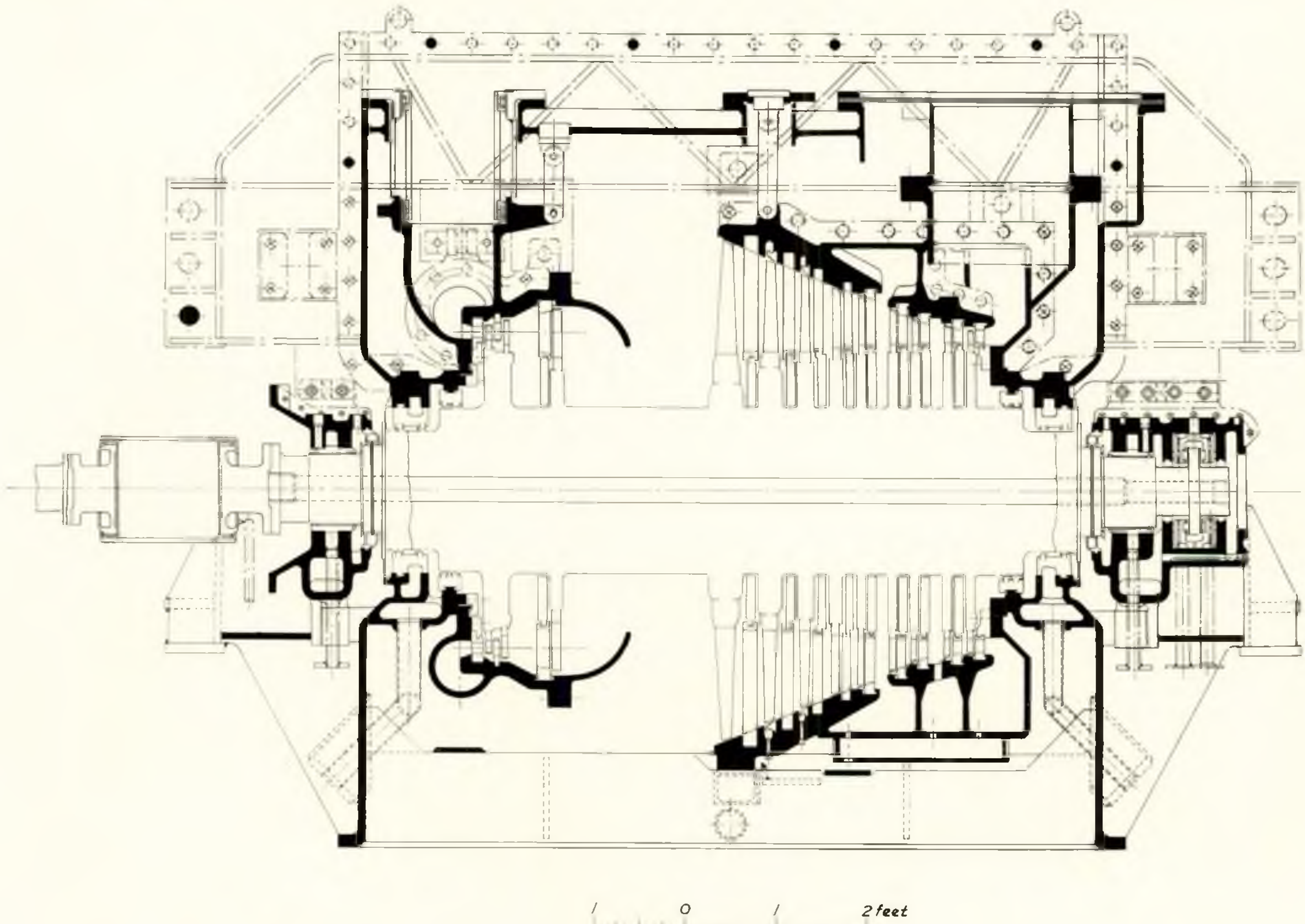


FIG. 19—*L.P. Turbine basis plan—68,000-d.w.t. vessels*

to maintain the co-axiality of the bearing journals, glands and diaphragms and it is expected that this design will be more able to stand up to transient conditions than any of the earlier designs. The two-row impulse wheel of the H.P. astern turbine is overhung from the H.P. ahead turbine and the nozzle plate in this turbine is also of welded construction. The thrust block is at the after end of the ahead turbine and the lubricating oil to the thrust face is led through holes to the inner radius of the pads. The lubrication arrangement is such that the oil can only pass once over the thrust face before passing to the drain pipe.

The design of the L.P. turbine selected was similar to the Pametrada "Prototype I" L.P. turbine (Fig. 19).

This is a single flow turbine and consists of five impulse stages followed by four reaction stages. The L.P. astern turbine, which is arranged in the same outer casing as the L.P. ahead turbine, consists of a two-row Curtis wheel followed by a single-row impulse wheel. The cylinder is of double casing design, the ahead and astern inner cylinders, which are steel castings, being supported by palms and keyed to the outer fabricated cylinder with a view to maintaining co-axiality of the inner casings and the rotor. Water extraction from the ahead inner barrel has been arranged at two positions near to the exhaust end of the turbine.

Balanced spring-backed glands are fitted throughout the H.P. and L.P. turbines.

This turbine is the L.P. unit of an advanced turbine design of 22,000 s.h.p. employing steam conditions of 850lb./sq. in. gauge, 1,050 deg. F. which has been described by Brown and Veitch[4]. In the present application, it is suitable with minor modifications to the blade path as the L.P. unit of a 20,000 s.h.p. installation with steam conditions of 600lb./sq. in. gauge, 950 deg. F. It was clear that this combination of turbines, while fully meeting the requirements for the present design, possessed appreciable development potential for the future.

Gearing Design

Table IV shows a comparison of articulated and locked train gears suitable for the 20,000 s.h.p. design.

The primary and secondary gears are of built-up construction except for the primary wheels of the locked train gearing which would be solid construction.

It will be seen from the table that there is a considerable saving in weight, size and also of materials in the locked train design particularly with the reduced face width of the secondary elements. However, the cost of workmanship is considerably more, thus offsetting the saving in material costs.

It was, however, considered that the greater complication, particularly with regard to maintenance at inexperienced hands of locked train gearing, would only be justified if necessitated by gearing load factors, and as this did not arise the simple articulated design was selected.

In this design the primary gear wheels are of the fabricated type, the toothed rims being attached to collars on the spindles by means of bolted-on side plates, stiffened by welded cone plates. The second reduction wheel has a cast iron centre with shrunk-on rim, mounted on and keyed to the main shaft. The primary pinions are solid and the second reduction pinions are hollow to accommodate the quill shafts. The quill shafts have solid couplings at the forward end and flexible couplings at the after end. All pinions are of material to B.S. specification EN.26 and the wheel rims to B.S. specification

TABLE IV—COMPARISON OF ARTICULATED AND LOCKED TRAIN GEARING FOR THE 68,000 D.W.T. CLASS

Type		Double reduction articulated		Double reduction locked train	
Turbine		H.P.	L.P.	H.P.	L.P.
Normal service power		11,000 s.h.p.	9,000 s.h.p.	11,000 s.h.p.	9,000 s.h.p.
Normal service speed		4,790 r.p.m.	2,370 r.p.m.	4,760 r.p.m.	2,360 r.p.m.
Primary pinions;	Pitch circle diameter	12·1195 in.	24·4594 in.	9·9184 in.	20·0204 in.
	No. of teeth	55	111	54	109
Primary wheels;	Pitch circle diameter	82·1922 in.	82·1922 in.	2 at 47·2042 in.	2 at 47·2042 in.
	No. of teeth	373	373	257	257
Primaries;	Face width	25 in. + 3 in. gap.		20 in. + 3 in. gap.	
	Tooth form	6/10 in.		5/10 in.	
	Flank angle	16 deg.		16 deg.	
	Helix angle	29 deg. 58·805 min.		29 deg. 59·174 min.	
Secondary pinions;	Pitch circle diameter	24·3736 in.	24·3736 in.	2 at 17·2194 in.	2 at 17·2194 in.
	No. of teeth	83	83	67	67
Secondary wheel;	Pitch circle diameter	164·1548 in.		163·9696 in.	
	No. of teeth	559		638	
Secondaries;	Face width	48 in. + 3 in. gap.		32·5 in. + 3 in. gap.	
	Tooth form	8/10 in.		7/10 in.	
	Flank angle	16 deg.		16 deg.	
	Helix angle	29 deg. 58·057 min.		29 deg. 58·373 min.	
Construction;	Primary wheels	Built		Solid	
	Secondary wheel	cast iron centre—shrunk on rim		cast iron centre—shrunk on rim	
"K" Values;	Primaries	90		90	
	Secondaries	79		65	
Gearing weights		96·0 tons		87·0 tons	
Gearing materials;	Wheels	EN.8		EN.8	
	Pinions	EN.25		EN.25	

EN.9 All bearing housings, except those for the main wheel, are adjustable for alignment purposes, the main wheel bearing housings being of a rigid type welded into the lower half of the gear case. A Michell thrust block has been arranged at the forward end of the primary gear wheel spindle to locate the primary gears axially.

Main Condensers

An analysis was made of sea water temperature conditions throughout the year during the passage from U.K. to the Persian Gulf. The average sea water temperature recorded was 75 deg. F., consequently this temperature was taken for the basic condenser design conditions.

The effect on condenser cost and overall fuel consumption was explored with the following variations in other factors.

a) Vacuums between 28·0in. and 28·8in. of mercury
b) Circulating water tube velocities between 6·0 and 7·0 ft./s.
c) Condenser tube sizes between ⅝in. and 1in. outside diameter.

The effect of these variations on the main circulating pump costs and horsepowers were also taken into account, it being assumed that there were to be two single-speed electrically-driven centrifugal units, each being capable of supplying sufficient water to maintain 70 per cent of the service power at a sea water temperature of 75 deg. F.

Fig. 20 shows the effect of variation in the design vacuum from 28·0in. to 28·8in. of mercury, at a constant tube velocity,

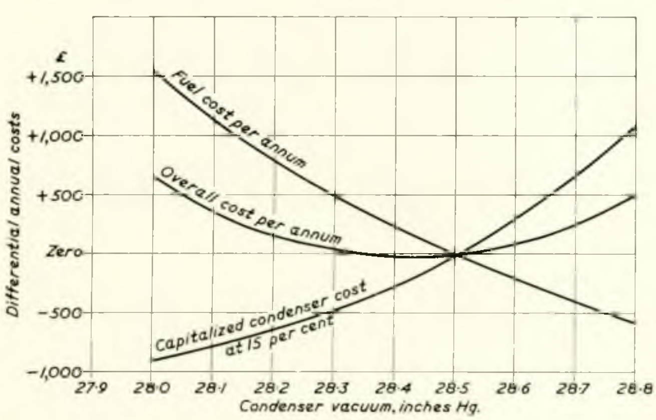


FIG. 20—*Comparison of condenser vacuum, costs and fuel savings*

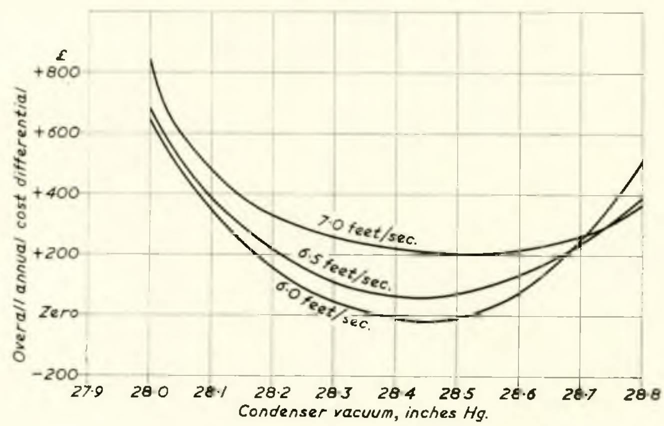


FIG. 21—*Comparison of condenser vacuum tube velocity and overall annual costs*

TABLE V—COMPARISON OF PARTICULARS FOR MAIN CONDENSER AND CIRCULATING PUMPS FOR ¾ IN. AND 1 IN. OUTSIDE DIAMETER TUBE SIZES FOR THE 68,000 D.W.T. CLASS

Tube outside diameter		¾ in.	1 in.
Condenser surface	sq. ft.	15,750	14,000
No. of flows		2	2
No. of tubes		7,460	4,990
Tube thickness	S.W.G.	18	18
Tube length overall		10 ft. 11½ in.	10 ft. 11½ in.
Tube length between plates		10 ft. 9 in.	10 ft. 9 in.
Design vacuum	in. Hg.	28·5	28·5
Corresponding sea water temperature	deg. F.	75	75
Circulating water quantity	gal./min.	19,600	24,900
Total frictional head	feet	26	25
No. of circulating pumps		2	2
Motor size	h.p.	105	130
Total differential b.h.p. absorbed			+47

on the capitalized cost of the condenser and main circulating pumps, the annual fuel costs and also on the overall annual costs.

Fig. 21 shows, for three tube velocities, the effect of variation in vacuum on the overall annual cost.

Table V shows the effect of condenser tubes of ¾in. and 1in. outside diameter on the condenser and circulating pump design, cost and fuel consumptions.

From the table and figures the following facts emerged:

1) It was apparent that any increase in water speed through the tubes above 6 ft./s. was of no advantage, the extra cost of circulating pumps and the power required for driving them outweighing any savings from reduced condenser size.

   Apart from this consideration, the effect of increased velocity on tube life had to be kept in mind.

2) The more detailed analysis on the 6ft./s. tube velocity was made and it will be seen than an optimum vacuum condition of 28·45in. of mercury was established.

3) The saving in weight per condenser with ⅝in. diameter tubes was about 1¾ tons compared to the ¾in. tubes and about 4½ tons compared to the 1in. tubes.

   There was a considerable saving in the number of tubes required for the 1in. size over the other sizes thus reducing, for a similar total surface, the labour costs for fitting and maintenance. However, due to the increased quantity of circulating water in the case of the 1in. diameter tubes, an increase in the size and power of the main circulating pumps was required.

On the basis of the above, it was decided to adopt a condenser designed for a vacuum of 28·5in. Hg. with a tube velocity of 6·0ft./s. Whilst the inclusion of 1in. diameter tubes appeared most attractive, it was decided to delay a decision on this until a later vessel due to the provision in the Pametrada standard frame for a condenser with a tube plate length suitable for ¾in. diameter tubes.

For the first of the class the condenser being adopted is of Pametrada regenerative type having ¾in. outside diameter tubes, both ends being expanded and flared into the tube plates and the condenser shell having an expansion piece at one end to take up differential expansion. The tubes are of aluminium brass and have inserts of the same material at the inlet ends.

Boiler Design

The previous classes of vessel already described were mainly fitted with Foster Wheeler boilers of the E.S.D. type, with steam temperature control by means of air attemperators or with Babcock and Wilcox selectable superheat boilers, in which the steam temperature is controlled by means of gas bypass dampers.

At the same time that the preliminary designs were being considered, a prototype boiler of Foster Wheeler design was under construction in which the steam temperature was controlled by means of a superheater bypass unit in which feed from the economizer, before entering the steam drum, circulates a small control unit. Regulation of the proportion of gases through the superheater and through this unit by means of dampers provides the method of controlling the superheated steam temperature.

This type of boiler was recommended on the grounds of improved method of control, reduction in weight and reduction of cost and was adopted for the 68,000-d.w.t. class.

As far as the heat recovery side of the boiler design was concerned, this was entirely dependent on the choice of the feed cycle and has already been discussed.

THE PRESENT POSITION

Summarizing the preceding paragraphs, it will be seen that the design chosen for the 68,000-d.w.t. class of vessel includes as its main feature a new design of turbine which has evolved from several previous Pametrada designs. This evolution has resulted from analysis of previous experience with Pametrada designs and from ventilating and discussing

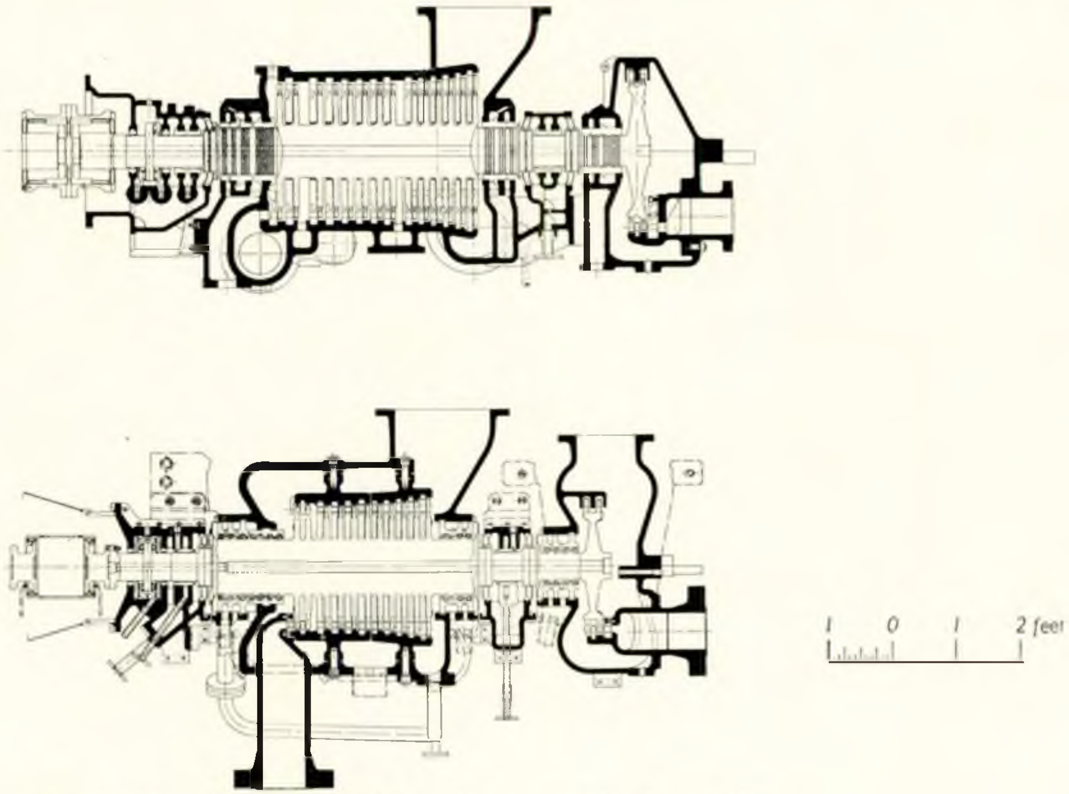


FIG. 22—*Comparison between H.P. turbines for 12,500 s.h.p. (1950) and 20,000 s.h.p. (1961)*

TABLE VI—PRINCIPAL MACHINERY PARTICULARS FOR THE POST WAR CLASSES OF TANKERS

Tanker class tons d.w.	28,000/32,000	32,000/35,000	42,000/49,000	50,000	68,000
Year completed	1953	1957	1959	1962	1963
Normal service power	12,500 s.h.p. at 112 r.p.m.	14,000 s.h.p. at 105 r.p.m.	16,000 s.h.p. at 105 r.p.m.	16,000 s.h.p. at 105 r.p.m.	20,000 s.h.p. at 105 r.p.m.
Overload power	13,750 s.h.p. at 115·6 r.p.m.	15,500 s.h.p. at 109 r.p.m.	17,600 s.h.p. at 108 r.p.m.	—	—
Steam conditions	450 lb./sq.in. gauge 750 deg. F.	600 lb./sq.in. gauge 850 deg. F.	600 lb./sq.in. gauge 850 deg. F.	600 lb./sq.in. gauge 900 deg. F.	600 lb./sq.in. gauge 900/950 deg. F.
Final feed temperature	310 deg. F.	240 deg. F.	240 deg. F.	250 deg. F.	360 deg. F.
Boiler efficiency	87·5 per cent.	87·5 per cent.	87·7 per cent.	87·7 per cent.	88·1 per cent.
Main turbines:					
Non-bled steam rate lb./s.h.p. hr.	6·91	6·20	6·15	5·77	*5·75/5·62
H.P. turbine: Type of casing	Single casing	Single casing	Single casing	Double casing	Single casing
No. of stages	14 rows all impulse	15 rows all impulse	15 rows all impulse	12 rows all impulse	10 rows all impulse
Normal speed	3,989 r.p.m.	3,940 r.p.m.	4,450 r.p.m.	4,996 r.p.m.	5,402 r.p.m.
L.P. turbine: Type of casing	Double casing, two flow	Double casing, two flow	Double casing, two flow	Double casing, one flow	Double casing, one flow
No. of stages	2 x 13 rows all reaction	2 x 15 rows all reaction	2 x 15 rows all reaction	5 impulse+7 reaction	5 impulse+4 reaction
Normal speed	3,662 r.p.m.	3,700 r.p.m.	3,685 r.p.m.	2,733 r.p.m.	3,556 r.p.m.
Gearing:					
Material: Wheels	F.S. 34-38 tons/sq.in.	F.S. 34-38 tons/sq.in.	EN.8	EN.8	EN.9
Pinions	3½ per cent Nickel Steel	3 per cent Cr Mo	EN.25	EN.25	EN.26
"K" factor: Primaries	67	80	80	97	92
Secondaries	60	71	76	80	77
Main condensers:					
Design vacuum in. Hg.	28·5	28·5	28·5	28·5	28·5
Surface sq. ft.	13,000	13,000	15,000	13,500	14,300
Main boilers:					
Maximum evaporation lb./hr.	2 x 75,000	2 x 82,000	2 x 90,000	2 x 75,000	2 x 90,000
Normal service evaporation ex tank heating lb./hr.	2 x 55,600	2 x 55,600	2 x 63,000	2 x 57,000	2 x 74,300
Turbo-alternators:					
Number, size and type	2–600 kW. self-condensing	2–600 kW. self-condensing	2–750 kW. self-condensing	2–700 kW. back pressure	2–750 kW. condensing
Fuel consumption:					
Design fuel consumption lb./s.h.p. hr.	0·615	0·577	0·573	0·525	*0·508
tons/day	82	86·5	98	90	*109
Percentage improvement in fuel rate	Basis	6·2	6·7	14·6	17·4

* at 900 deg. F.

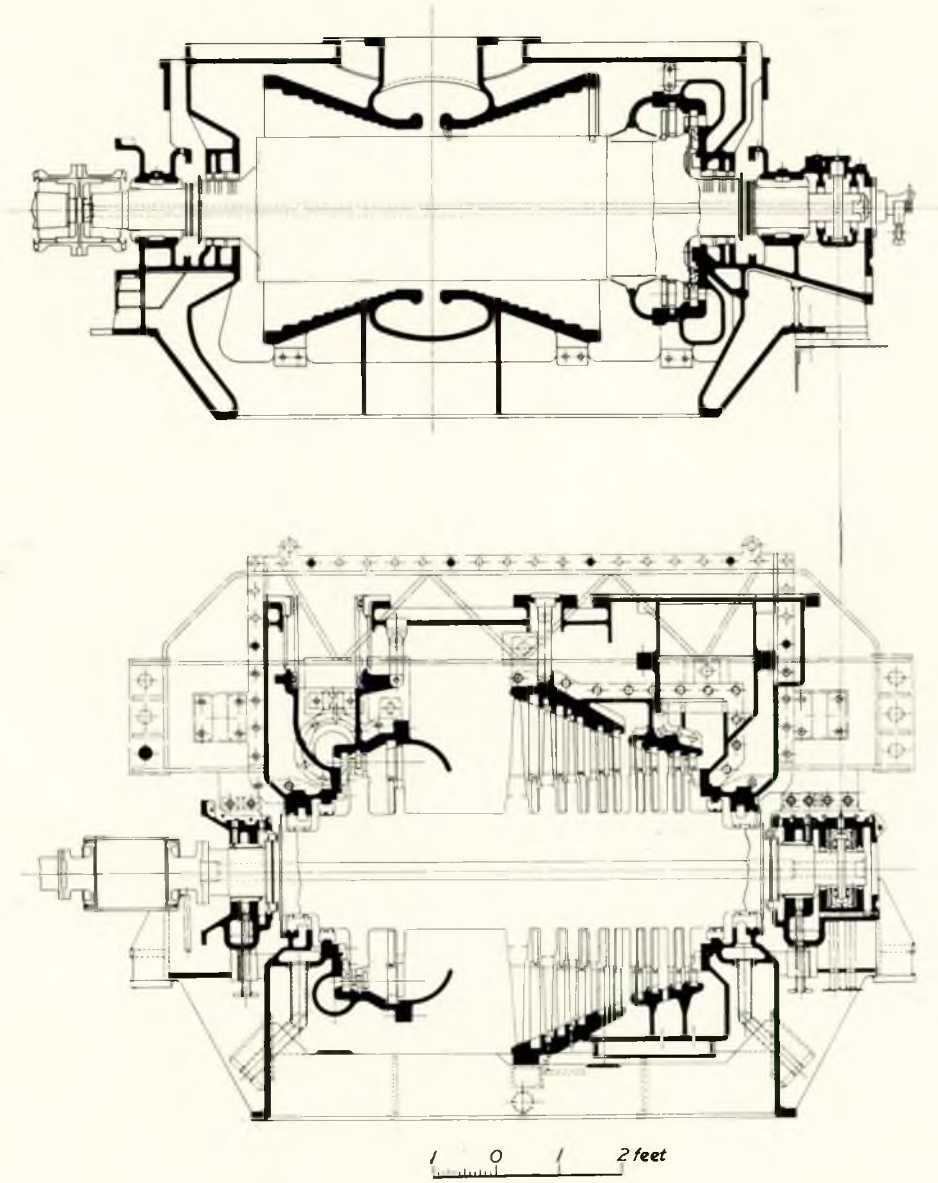


FIG. 23—*Comparison between L.P. turbines for 12,500 s.h.p. (1950) and 20,000 s.h.p. (1961)*

previous short-comings. The criticisms by the Company have been forthright and uninhibited but, it is believed, constructive. It is to the credit of the Pametrada organization that criticism, where constructively intentioned, has been accepted and analysed. The whole-hearted co-operation between owners and designers which has been a feature of this design study is confidently expected to result in turbine designs at least equal on all counts to the best available from competitors.

Before concluding this survey it is interesting to observe the progressive advance in tanker machinery over the last twelve years from the designs shown in Table VI and as shown in Figs. 22 and 23.

FUTURE TRENDS

Introduction

Considerable as the advances indicated by the figures given in Table VI may appear, it is not to be assumed that designs of steam turbine machinery installations are yet approaching the end of the road. Improvements in the internal efficiency of the turbines themselves are now becoming difficult to achieve, other than by advancing thermal conditions, and it is from the overall design of the plant that most advantage can be taken for further improvements in efficiency.

The Trend towards Higher Powers

Although the urgent demand for new tanker tonnage which was characteristic of the post-war period has resulted in a temporary surplus, the position is gradually becoming more in balance and shortages can at least now be foreseen. As bulk crude oil can be transported more economically in large vessels it would seem logical that the trend towards still larger ships will continue, subject to flexibility of terminal facilities and suitable dry docks. Continuing requirements will, however, exist for smaller ships in the refined products trade.

This trend towards larger vessels dictates a trend towards higher powers and it is in this higher powered field that the turbine installation will show the greatest advantage, even when due allowance has been made for the remarkable progress of the Diesel engine during the period covered by this paper.

The provision of such higher powers presents new problems which, while they are at present receiving attention, have yet to be solved. During the past decade the tendency has been to retain single screw propulsion because of its higher propulsive efficiency. There is no difficulty in providing 20,000/25,000 s.h.p. on a single screw and it is understood that with two vessels currently building in Japan 28,000 s.h.p. at 105 r.p.m. is to be applied to a single shaft. The economic upper limit of power which can be applied to a single shaft is principally allied to the weight and size of the propeller required to absorb the power at r.p.m. low enough to retain the propulsive efficiency substantially above twin screw figures. Present indications are that there is little likelihood of single shaft powers proving feasible above 30,000 s.h.p.

Where greater power than this is required, twin screws will have to be employed, although for ships of the same speed and deadweight tonnage a substantial reduction in propulsive efficiency with twin screws is unavoidable.

The designer of turbine machinery for tankers will therefore be faced with two alternatives:

a) to provide up to 30,000 s.h.p. on a single screw at about 105 to 110 r.p.m.
b) to provide upwards of 40,000 s.h.p. on twin screws at about 110 to 120 r.p.m.

Such powers are not, of course, remarkable in the context of passenger ship or warship turbine machinery, but will present new problems for tankers where it is essential to keep the length of engine rooms down to a minimum in order to obtain the maximum cargo space. The problem is rendered more difficult in larger, faster vessels with finer hull form aft, which reflects particularly in twin screw installations.

The turbines themselves do not present any particular problems, and the concern will be mainly with improving detail design. Similarly, for gearing the use of locked train drive will still permit the use of conventional materials such as EN.8 and EN.25 for powers between 20,000 and 27,000 h.p. per shaft without recourse to hardened or ground gears.

There seems little possibility that carburized and hardened gears of tanker main wheel size are economically feasible, but substantial improvements are possible by employing hardened and ground pinions in association with "soft" main wheels in materials such as EN.26. The most promising long term development would seem to be in tooth-by-tooth induction hardening which, when problems of quality control have been solved, should provide gears equal in hardness, accuracy and reliability to the best gas-carburized practice. Some indication of the benefits available from a conservative approach to such advances in gear technology have been given by Braddyll[3] and Nicolson[10].

It has still to be proved whether the reduction in size and weight offered by hardened gearing over conventional and well tried materials can ultimately be made to offset their present high production costs.

The Trend towards Higher Steam Conditions

The 20,000 s.h.p. design study showed that increase in steam pressure to 850lb./sq. in. gauge provided marginal economic advantage at the 20,000 s.h.p. level, which was outweighed by practical considerations.

At higher power levels, however, further increase in pressure cannot be overruled. It is necessary in the interests of reliability, that increase in pressure beyond 600lb./sq. in. gauge should be accompanied by the introduction of welded pipe joints which, while new to most marine engine builders in this country, are common in land practice. There would appear to be considerable latitude for the development of prefabrication techniques as far as pipe work is concerned, thereby reducing the number of site welds to a minimum. Further increase in steam pressure tends to lead to increased losses in the turbines themselves unless the steam flow through the turbines is further increased by the greater use of regenerative feed heating, a subject to which further reference will be made.

Increase of steam temperature is dependent firstly upon further knowledge of combustion conditions within the furnaces of marine boilers. At the moment it would appear reasonable to suppose that satisfactory operation at 950 deg. F.—1,000 deg. F. can be expected from marine boilers of current design, provided that superheater metal temperature can be limited to 1,100 deg. F. Such limitation of temperature involves a high pressure drop through the superheater and this is in itself an economic loss tending to offset the gain from increase in temperature.

Research into combustion techniques continues however and it is hoped that fundamental research now being carried out[8] will enable steam temperatures exceeding 1,000 deg. F. to be attained before the end of the next decade.

If success attends the research into combustion problems leading to a solution of the superheater slagging menace, the metallurgical problems associated with creep will come to the fore. Steam turbines operating up to 1,200 deg. F. present no insoluble difficulties, the high temperature parts being small, but the problems associated with long steam lines of austenitic steel in the marine environment, where load rate changes are often not controllable, would appear to be formidable. The inclination, therefore, is to suggest that the solution of superheater slagging problems with vanadium-bearing fuels may pave the way to high temperature gas turbine machinery rather than to advanced steam installations.

The subject of steam conditions cannot be left without reference to reheat. There can be no doubt that theoretically this technique offers a most useful further range of development when superheater outlet temperature limits have been reached, and it has been very successfully exploited for very large installations in the electrical generating industry. Turbo-electric installations would seem to offer a more attractive field for exploitation of reheat than reversible geared turbines, but the cost, weight and ancillary penalties, involved in electric transmission, make it unattractive. Application of reheat to geared machinery is feasible, as exemplified by the successful passenger ship installation described by Davis[5] and with developments in automatic controls tending to simplify operation, it may well be that such systems will in due course find application in tankers, but probably only after the top levels of pressure and temperature have been exploited.

The Trend towards Sophisticated Feed Systems

Increase in steam pressure leads to an increase in regenerative feed heating for the following reasons:

a) Increase in extraction steam flow increases the internal efficiency of the turbine, particularly in the H.P. stages, and offsets the increased losses due to increase in pressure.
b) Higher steam pressures increase the saturation temperature and raise the economic optimum feed heating temperature.

These considerations lead to the possibility that greater use may be made of regenerative feed heating in the future. From the point of view of maximum fuel economy, combined with low first cost, the maximum use should be made of L.P. bled steam, and, as explained earlier, this was one of the major factors affecting the decision to adopt condensing turbo-alternators in the 68,000-d.w.t. class.

The requirement for H.P. bled steam extraction is largely determined by the type of cycle selected. In an economizer cycle, the optimum final feed temperature is 280 deg. F., assuming an all steel economizer, irrespective of pressure conditions. Given a temperature of 280 deg. F. at the de-aerator outlet and an all steel economizer, there is no advantage in adopting the split economizer system at 600lb./sq. in. gauge since the amount of H.P. bled steam and the thermodynamic gain thereby involved is too small to justify the capital expenditure.

Where a cast iron economizer is employed, a final feed temperature of 250 deg. F. enables adequate waste heat recovery to be obtained and still leaves room for H.P. feed heating, particularly if the pressure is increased to 850lb./sq. in. gauge. In this case, of course, the high temperature section of the economizer would be steel.

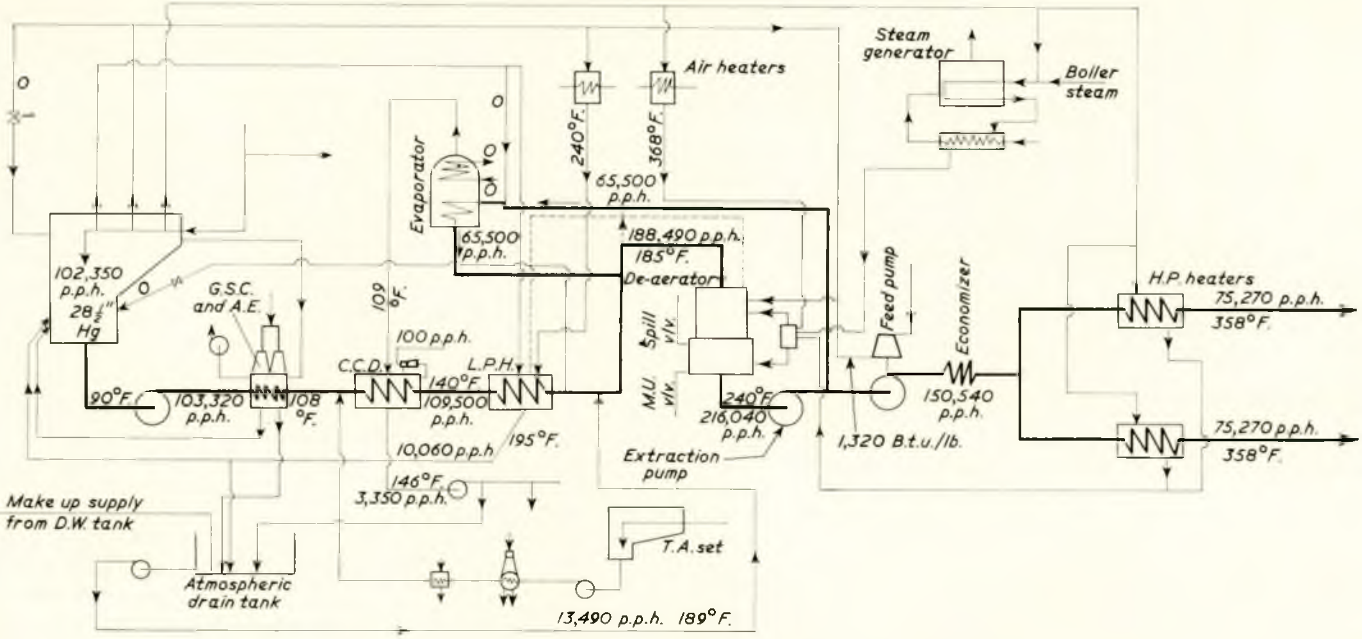


FIG. 24—*Heat flow diagram with water circulated evaporator*

Great ingenuity has been demonstrated in the preparation of heat balances to obtain the maximum benefit from regenerative feed heating. The use of water circulated units has recently attracted attention. While originally intended for jacket water heat recovery in motor ships, the possibility of employing this technique in steam ships has not been neglected as is shown in Fig. 24, which shows a proposed heat cycle employing a hot water circulated (motorship) evaporator in the heat cycle of a turbine driven vessel.

The Trend towards Complexity and its Influence on Reliability

In the immediately preceding paragraphs, some of the opportunities for refinement, which are open to the designer with increase in power, have been considered and it is only necessary to examine the trends of central power station practice, to see the savings in direct fuel cost in large installations which can accrue from the employment of reheat and the more extensive use of regenerative feed heating in order to take maximum advantage of higher steam conditions. It is necessary, however, to view these attractions with caution when considering the machinery installations of the very large tanker. The economic advantage of investment in a large ship is justified only provided that the vessel continues in service, without breakdown, to fulfil regularly the operating schedule upon which her machinery design was based.

In contrast with a tanker, a large passenger vessel operates with a comparatively low usage factor, and can take advantage of this to provide greater opportunity for routine maintenance in order to avoid breakdown. It seems certain, therefore, that even should tanker size rise to the level at which the power required to propel her approaches that installed in medium sized passenger vessels, it will be prudent to keep a step behind such vessels both in steam conditions and system complexity. Thus it will be seen that the lag behind passenger ship machinery which is generally discernible in tanker practice is based on sound economic reasoning rather than any innate wish to remain conservative in outlook. The delaying period for increasing operating steam temperature to 950 deg. F., which has already been described, is a typical example of this policy operating in practice. At the same time, the economic advantage of more complex systems must not be dismissed altogether through fear of breakdown, for really good design should always be able to eliminate such risks. The authors believe that the philosophy, expressed earlier in the paper, of accepting that inanimate objects, such as heat exchangers, can be made fundamentally reliable, indicates that improved heat balance can be achieved without loss of reliability.

Packaged Machinery

The use of this term has in many cases been applied by American engineers to the supply of and responsibility for a complete machinery installation of boilers, turbines and auxiliaries and in this context is the normal practice of British marine engine builders.

What can however be envisaged, is the greater use of packaged auxiliaries, in which a complete system will be designed, built, piped and fitted out on board as a single unit. It is not often appreciated that the cost of valves and piping is a major factor in the cost of a machinery installation and could be equal to the combined cost of the turbines, gearing and condenser.

It will be seen that there is more scope for cost reduction in this section than in any other and by careful design of packaged auxiliaries much can still be achieved.

The Trend towards Automatic Controls

Automatic control methods have been applied increasingly in all branches of engineering to enable complex processes to be operated efficiently and reliably, and they provide a means of ensuring that the more sophisticated machinery foreshadowed in the preceding paragraphs shall not fail in these respects.

A great deal is being said and written today about "automation" in ships as a possible "solution" to difficulties currently being experienced in obtaining sufficient seagoing engineers of the quality necessary to man modern vessels. The term "automation" properly used, refers to the elaborate programming techniques now common in factory process engineering. Whilst the application of such techniques to ships' propulsion machinery is already scientifically feasible, it seems unlikely that they will be introduced in the foreseeable future and the term "control engineering" is more apposite to the subject which will now be discussed.

The objects of control engineering may be defined as follows:

a) To ensure correct operation of the machinery at its design point during normal steaming without the need for judgement or manual intervention on the part of the watchkeepers.

b) To provide complete sequential control over the machinery throughout the power range without manual intervention other than the setting of the main throttles.

c) To reduce the burden of responsibility on the ship's senior engineers by enabling them to select the correct settings for the automatic controls in the knowledge that these settings will be maintained.

The achievement of these objectives should not prove difficult. Experience has been gained with automatic controls of the pneumatic type and the necessity for co-ordination between the control designers and the system designers has been demonstrated by such examples as that given earlier in the paper, when reference was made to de-aerator controls.

Sufficient experience has now been gained to enable a step forward to be taken in the 68,000-d.w.t. class, to which automatic controls will be applied extensively. For these vessels consideration is being given to providing a control room in which all essential instruments will be grouped and from which all essential functions can be performed. It has been taken as a basic concept in this design that all main and auxiliary units will be brought on load manually under the supervision of a senior engineer and then transferred to automatic operation. No attempt has been made to bring units onto load automatically, nor is this thought to be desirable at this stage.

Further developments will be possible as experience is gained in earlier ships of this class. One attractive line is indicated by the fact that all the basic pneumatic control elements have precise electronic counterparts which, in transistorized form, should prove quite suitable for use at sea.

CONCLUSIONS

Progress in the last twelve years has indeed been rapid. Turbine machinery has increased in reliability and efficiency to a degree which would not have been thought possible in 1950. The lessons learnt in the period reviewed in this paper, during which an unprecedented peacetime demand for ships led inevitably to some errors and omissions, both in design and manufacture, are now being applied to new construction. It is expected that the next twelve years will witness further advances in turbine plant engineering, leading to higher efficiency, by maximum exploitation of the potential of available steam conditions. Automatic controls will play an important part in ensuring that reliability and ease of operations are maintained.

It is suggested that, rather than competing, the one to displace the other, Diesel and steam turbine installations should advance side by side, being selected according to the dictates of local preference and trading conditions. The period beyond the next twelve years is outside the scope of this paper, but it seems reasonable to suggest that the gas turbine could by then enter the lists as a serious contender for medium and high powered applications.

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