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

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The paper will describe the developments in power plants in tankers built in the United States during the 1930, 1945 and 1960 eras, with specific reference to the main propulsion equipment, illustrating advances in steam conditions, component design, metallurgy and other factors in which improvements were made particularly in the post-war years. Further improvements to be expected in this type of power plant in the immediate future are discussed. The trends which might offer further improvement in tanker machinery design in the more distant future are outlined, including comments regarding the concepts of the austerity ship, the automated ship, and the nuclear ship. The design concepts and developments described are those which have taken place, mainly, in the United States during this period.

INTRODUCTION

The author is keenly aware of the honour and is most appreciative of the opportunity afforded him by the Institute to present a paper before this distinguished audience. It is hoped this presentation will induce serious consideration of the subject, stimulate discussion, and provoke interchange of information, all of which could result in further advance of the art.

SCOPE

The paper will cover the developments in steam turbine tanker machinery from 1930 to date and touch upon future trends which are now under consideration both for the immediate future as well as those which might be useful for more long range planning. The descriptions given illustrate developments in the U.S.A. which may differ in some respect from the approach used in Europe.

The paper will concentrate on the main turbine propulsion machinery and closely associated equipment suited for the propulsion of tankers, ore carriers or similar vessels.

In order to illustrate the developments which have taken place, an historical review is included, starting with a description of what might be termed the forerunner of the present day United States tanker power plant.

GROWTH OF TANKER FLEET

Much progress has been made since the construction of the first ocean-going steamer specially designed to transport oil in bulk, which was the s.s. *Glückauf* (*Good Luck*). Incidentally this tanker was constructed in England in 1886 for a German shipowner; she was 300ft. long with a 37ft. beam and was of 3,020 d.w.t.(1) Since then the number of tankers in service has grown tremendously; the size of the average tanker and its speed have also increased significantly. In 1955 the average size of the world tanker was approximately 16,800 d.w.t. with an average speed of 14·1 knots. In 1960 this had risen to 21,900 d.w.t. and an average speed of 15·1 knots. In general, United States ships are somewhat smaller but slightly faster than foreign flag ships.(2)

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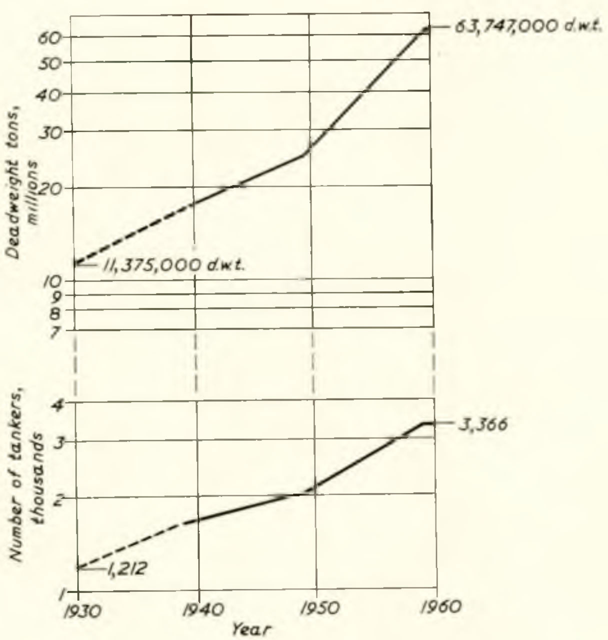
These average figures may be somewhat misleading since in the last few years larger tankers were built from 40,000 d.w.t. to approximately 100,000 d.w.t. having power plants which have increased from approximately 10,000 s.h.p. to 25,000 s.h.p., and, in some special cases, even greater horsepower.

Fig. 1 illustrates the growth of the world tanker fleet and thereby shows the tremendous role that this field of transportation has assumed over the years. In 1930 there were



Note 1—Figures given above include tankers and whaling tankers, 1,000 gross tons and over.
Note 2—Dashed line indicates estimated values

FIG. 1—*Growth trend of world tanker fleet*

1,212 ships world-wide with an estimated total of 11,375,000 d.w.t., and in 1960 there were in operation 3,366 ships totalling 63,747,000 d.w.t., consisting of tankers and whaling tankers.[3]

Of 430 tankers under the United States flag in 1960, 214 were of the T.2 class. Of the remaining 216, 5 were Diesel engine driven vessels and 9 vessels were driven by reciprocating engines. In other words, disregarding the T.2 class, 93·5 per cent of the United States tanker fleet in 1960 were steam turbine driven vessels including 96·5 per cent of the total United States tanker d.w. tonnage other than T.2's.[4]

Since the end of 1945, no turbine-electric machinery has been installed in tankers in the U.S.A. Originally, the lack of manufacturing facilities for mechanical reduction gears, rather than any inherent superiority of turbine-electric machinery, led to the choice of the latter for the wartime T2 design and therefore practically all tankers built in the U.S.A. in the last 15 years were steam turbine driven, although several American owners have built both turbine and Diesel tankers in Europe.

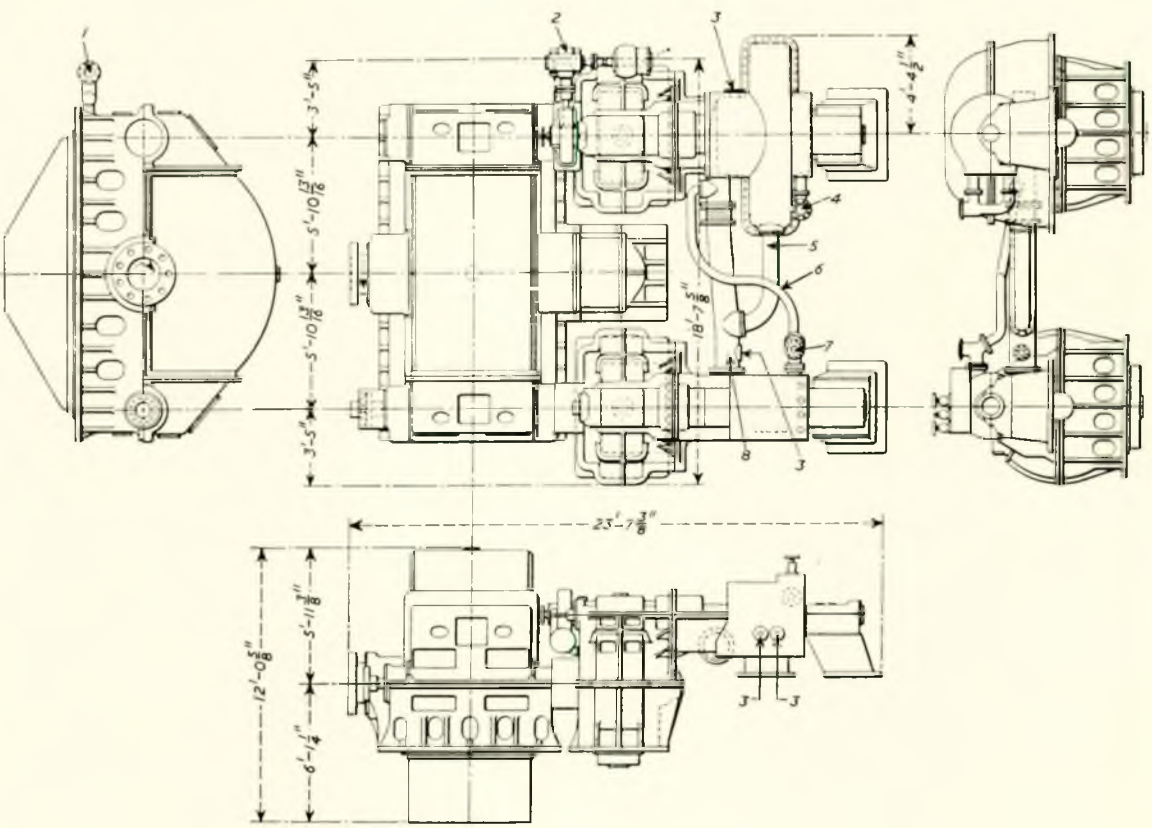
MECHANICAL DEVELOPMENTS OF TURBINE AND GEAR DESIGN

The principles of the steam turbine have not changed since 1897 when Sir Charles Parsons put the ***Turbinia*** to sea and since Dr. De Laval built the turbine power plant for the Stockholm World Exposition. Anyone concerned with steam turbines is very much aware of the contributions made by men like Sir Charles Parsons, Dr. De Laval, Monsieur Rateau and Mr. Curtis.

In the development of turbine tanker machinery, there were no major break-throughs; in most cases progress resulted from the painstaking, day-to-day hard work in improving details.

In order to illustrate the developments which have taken place, three periods were chosen, 15 years apart, beginning with 1930. The 1930 period was chosen because it brought with it the first concepts of the double reduction geared compound turbine installation utilizing superheated steam and a single astern turbine arrangment in the low pressure turbine, which has been the general pattern to date in the U.S.A.

In each era there were certain courageous and forward-



1. Oil inlet
2. Turning gear
3. Bleeder connexion
4. Astern steam inlet
5. Emergency exhaust
6. Emergency steam to low pressure turbine
7. Ahead steam inlet
8. Cross-over pipe

FIG. 2—*Outline of turbines and gears, 1930 era*

looking pioneers who built vessels which were somewhat ahead of their time. Therefore, it cannot be assumed that the examples follow a very rigid pattern, but it is believed they are representative of the period and serve to illustrate the point.

The author's company has traditionally built impulse turbines and articulated gears for all types of merchant vessels, including tankers, and has developed the locked train type of gears for naval combatant vessels. Hence, the author may be forgiven for using in the following discussion illustrations, design parameters, and characteristics of machinery furnished by his company without implying that other design concepts have not been equally successful.

The 1930 Era

In 1930, the s.s. *G. Harrison Smith* and the s.s. *W. S. Farish* were delivered to their owners. Incidentally, the *G. Harrison Smith* was operated as the *Esso Belfast* and the *W. S. Farish* as the *Esso Southampton* until quite recently and from all reports, they have given an excellent account of themselves in spite of their venerable age.

The power plants of these vessels developed 4,000 s.h.p. at a propeller speed of 75 r.p.m. with steam conditions at the turbine throttle of 375lb./sq. in. gauge and a total temperature of 725 deg. F. which corresponds to 283 deg. F. superheat.

These vessels constituted, in their day, two of the most modern, if not the most modern tankers. This was clearly brought out in the discussion of Mr. C. R. Waller's paper presented before the Society of Naval Architects and Marine Engineers in November 1930.[5] The discussions to this paper indicate that the industry regarded double reduction gears and the general arrangement of the turbines as very significant steps forward.

Fig. 2 shows the outline of the main turbines and gears in these ships. Interesting are the massive sliding foot type pedestals supporting the forward turbine bearing and the solid attachment of the aft end of the turbine casings to the reduction gear. The reduction gear had separate cases for each of the first reduction gear trains, and the second reduction pinions were on a common plane with the line shaft. Permanent piping was installed to provide for emergency single turbine operation. Not shown in the drawing were the large rubber bushed couplings connecting the first and second reduction gear trains.

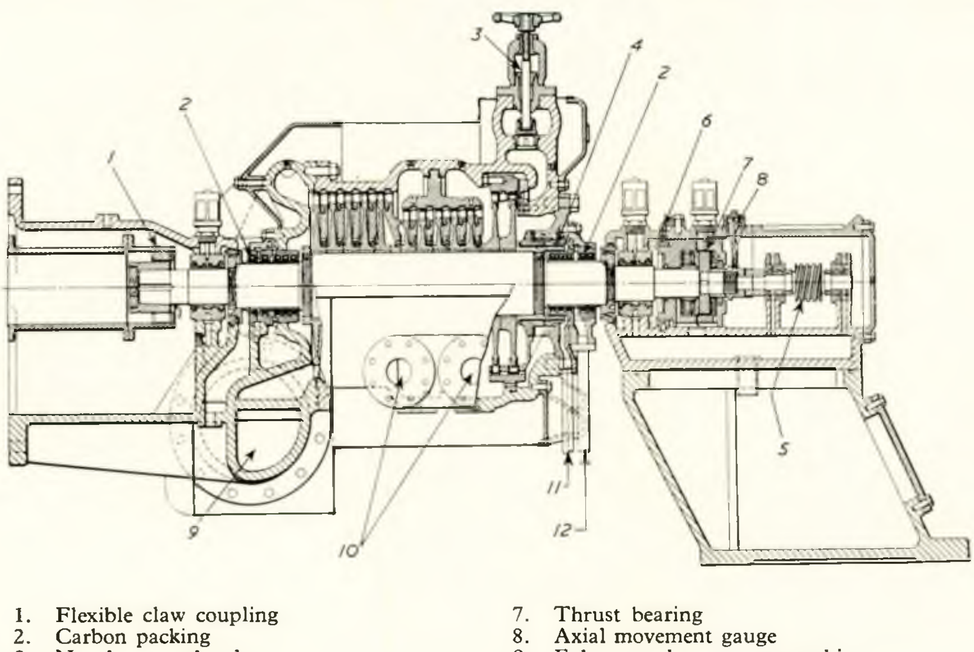
Fig. 3 shows a section through the high pressure turbine. Interesting is the oil sump provided in the pedestal and the right angle drive provided for the governor oil pump submerged in the sump. The thrust bearing housing was adjustable. The turbine wheels were individual discs mounted on the shaft on taper bushings. The buckets were mounted on the wheel with cross pin fastenings for each bucket. The diaphragms were complete rings built up of individual vanes around a steel-ring centre with solidly mounted labyrinth packing at the turbine shaft. To disassemble this rotor it was necessary to remove a wheel, then a diaphragm, then the next wheel, next a diaphragm, etc. Dynamic balancing of the rotor involved building it up without the diaphragms, balancing, stripping down and then rebuilding with the diaphragms, although at that time the individual wheels were carefully balanced separately, so that a complete balance of the rotors was generally unnecessary even if a wheel had to be removed.

The separate inner wheel case in this early design provided for a simplified casting as well as proper belts for extraction steam.

All ahead inlet nozzles were in the turbine cover, divided into groups, one uncontrolled, the others controlled. By proper selection of the groups in service, economical operation was possible over a wide range of power. Speed of this high pressure turbine was 5,480 r.p.m.

The emergency governor as mentioned earlier was controlled by a gear type pump mounted in the pedestal of the forward high pressure turbine bearing. The output of this pump, being proportional to speed, controlled an emergency valve located at the inlet to the manœuvring manifold. Also actuating this emergency valve were provisions for shutting down the turbine in case of low lubricating oil pressure or excessive turbine exhaust pressure.

Fig. 4 shows a cross-section through the low pressure turbine. The turbine rotor was built up of five separate steel forgings, very carefully machined, fitted, and bolted together. The remaining wheels were mounted on the shaft on taper bushings. By avoiding holes in the highly stressed wheels this type of rotor construction provided high strength,



1. Flexible claw coupling
2. Carbon packing
3. Nozzle control valve
4. Labyrinth packing
5. Worm drive for speed limit governor
6. Rotor adjusting device
7. Thrust bearing
8. Axial movement gauge
9. Exhaust to low pressure turbine
10. Steam bleed off connexions
11. Packing box leakoff
12. Packing box drain

FIG. 3—*Typical high pressure turbine, 1930 era*

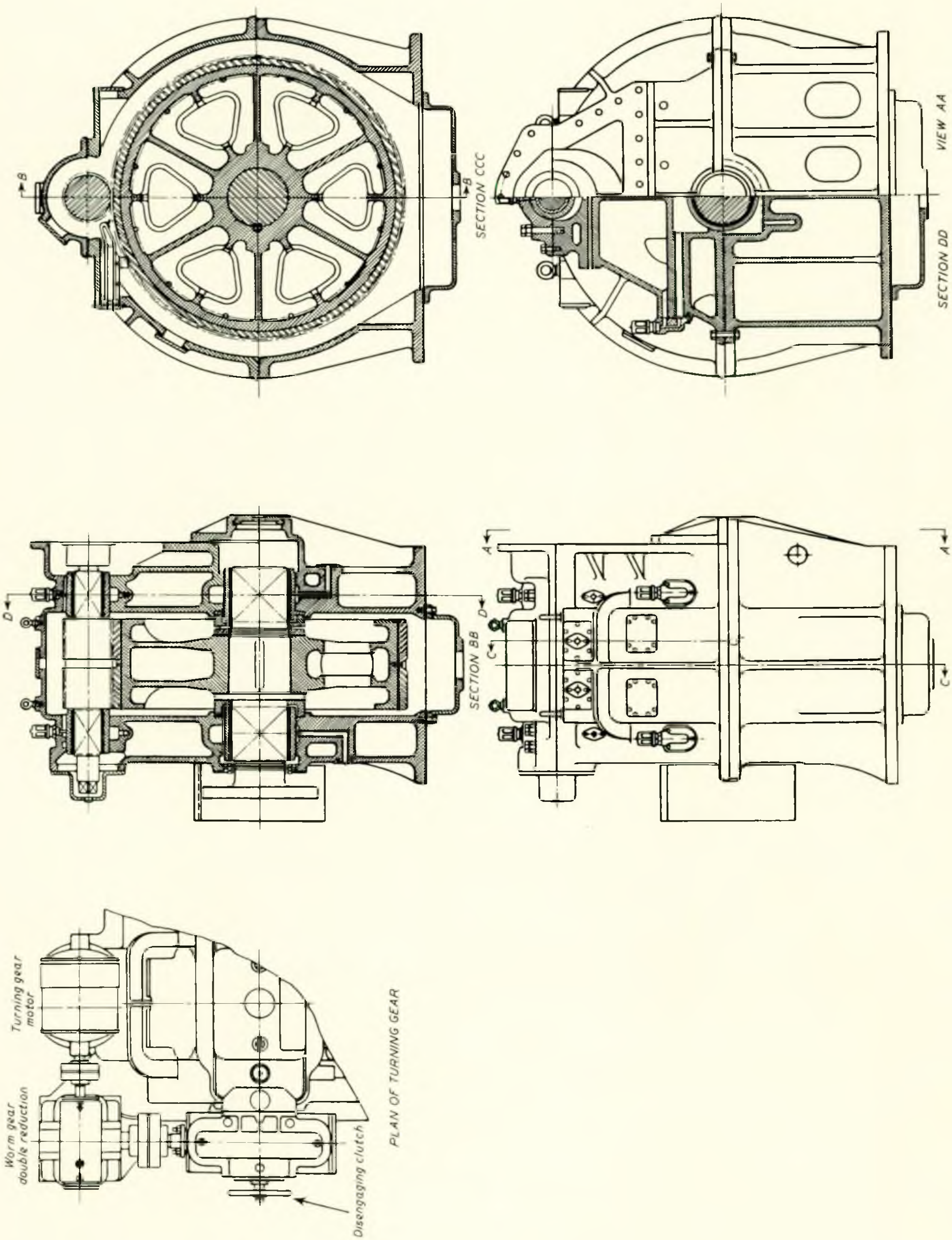
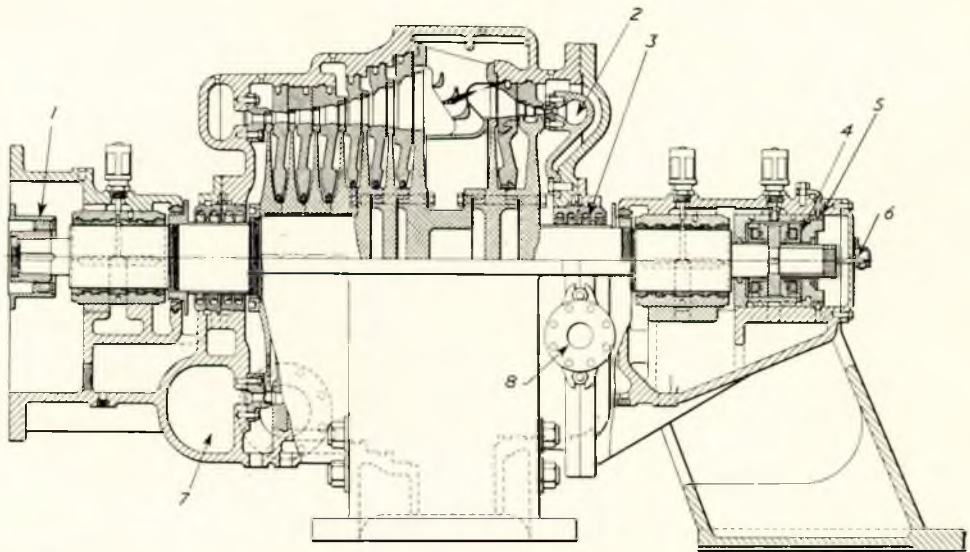


FIG. 5—*Typical high speed reduction gear, 1930 era*



1. Flexible claw coupling
2. Astern steam chest
3. Carbon packing
4. Thrust bearing
5. Rotor adjusting device
6. Axial thrust gauge
7. Ahead steam chest
8. Astern steam inlet

FIG. 4—*Typical low pressure turbine, 1930 era*

homogeneous wheels, requiring less material and space than shaft mounted wheels. The blading, with the exception of the first astern stage, was mounted in the wheel by using the well known De Laval bulb and shank type fastening.

The astern turbine consisted of one Curtis stage, followed by one Rateau stage, and the astern nozzle chest had two semi-circular rings mounted in such a manner that they were able to expand radially. Steam was admitted into both chests through separate inlet pipes penetrating the turbine cases with slip joints.

The diaphragms of the low pressure ahead turbine were horizontally split and made of cast semi-steel using preformed vane sections of steel cast integrally.

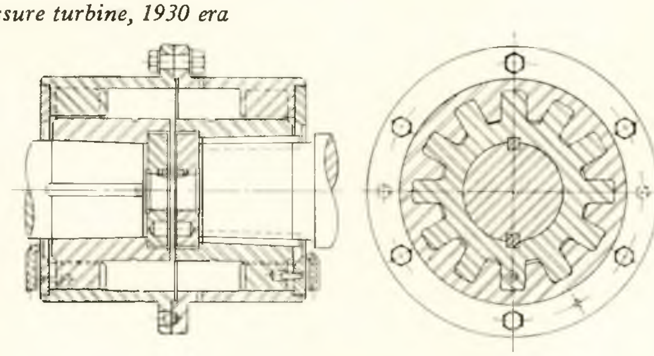


FIG. 6—*Typical claw type coupling, 1930 era*

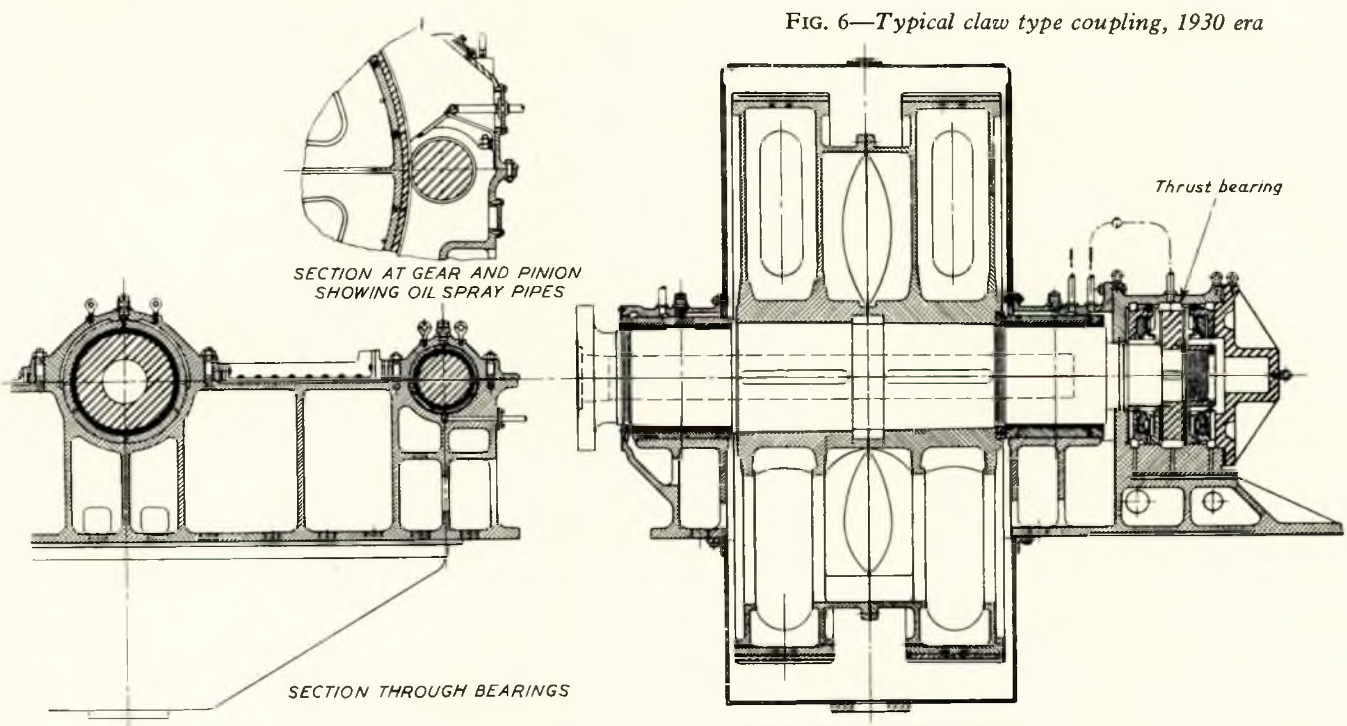


FIG. 7—*Typical low speed reduction gear, 1930 era*

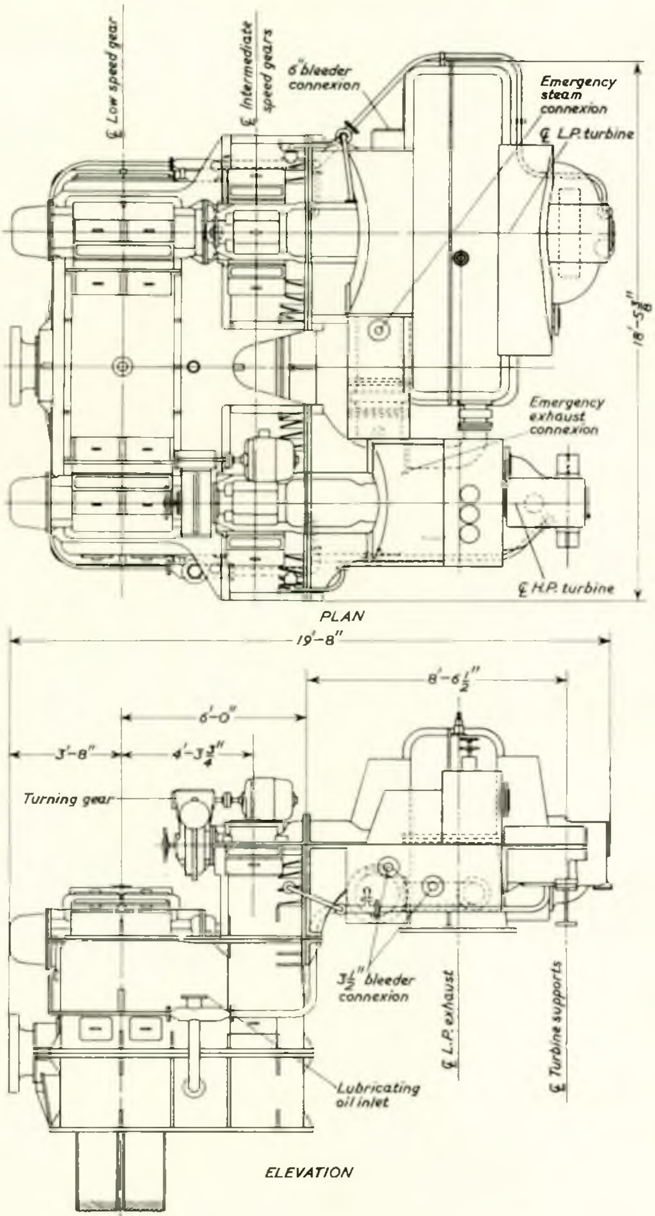


FIG. 8—*Outline of turbines and gears, 1945 era*

The ahead and astern exhaust section were each provided with an elaborate vaned guide to divert the respective exhausts away from the opposite blading.

The speed of this low pressure turbine was 4,270 r.p.m.

The steam seal and leakoff system on these turbines was controlled manually with the leakoff going directly to the condenser and with a number of valves to adjust the glands at each condition of loading.

Fig. 5 shows a section through the first or the high speed reduction gears. These were in separate housings mounted directly on the ship's foundation using suitable liners. Of particular interest in this design was the heavy cast iron construction of the gear cases. The gear wheels were also of cast iron mounted on steel shafts with a taper fit. The gear rims were shrunk onto the wheels and further secured by means of axial screws between the gear rim and the body.

The oil sprays are on the incoming side of the mesh.

The couplings between the turbines and the high speed pinions were of the so-called claw type shown in Fig. 6 with provisions for continuous lubrication.

Fig. 7 shows several sections of the second reduction with the main thrust mounted forward of the gear wheel. The gear wheels were of similar construction to the first reduction, but used two wheels bolted back to back. It should be noted that the bolting flange is of sufficient diameter to provide needed rigidity between the two wheels. The pinions were supported in three bearings to avoid excessive deflexion. The riveted construction using lightweight plate in the cover was actually a forerunner of complete welded gear case construction.

The 1945 Era

Remembering that the description just presented illustrated a unit which was somewhat ahead of its time in 1930, now a unit, which represents a design used just before, during and immediately after World War II, will be described.

The average horsepower had by this time increased to about 6,000 s.h.p. with propeller speeds increasing to about 92 r.p.m. The illustrations used describe the turbines and gears installed in the s.s. *Flagship Sinco* built in 1941 with steam conditions which were practically standardized at 450lb./sq. in. gauge, 750 deg. F. in the U.S.A. in the 1945 era.

In reviewing the mechanical construction of this unit, Fig. 8 shows an outline of the turbine and gears. The overall length, which was 23½ft. for the 1930 type of 4,000 s.h.p. unit, was decreased to slightly less than 20ft. for the 6,000 s.h.p. unit. The combining of the reduction gears into a single case is clearly shown, as well as the use of welded construction with attendant decrease in weight and increased resistance to shock, which became particularly important in those years. The elimination of the permanent emergency piping gave testimony to the increased confidence in this type of machinery.

In Fig. 9 a cross-section of the high pressure turbine is shown. The forward bearing pedestal was simplified by using a direct drive arrangement of an IMO pump for governing. The sliding type pedestal was eliminated and the simpler, yet more reliable, flex plate concept was introduced.

The journal bearing design was much improved and modernized, mainly by shortening the length of the bearing.

The rotor is a solid forging with the wheels integral with the shaft, permitting higher turbine speeds and a more rugged design. The buckets are held in the wheel by a dovetail or inverted T-type fastening. The diaphragms are of a built-up all-steel construction, split on the horizontal centre line permitting much better accessibility and easier maintenance. While the carbon rings have been retained as shaft seals, the pressure breakdown seal at the high pressure end and the interstage labyrinth packing are segmental, easily installed and maintained.

Turbine speed was 5,265 r.p.m.

The emergency overspeed governor was still retained on the high pressure turbine. However, the separate emergency valve had been eliminated and had been made part of the manœuvring manifold, which now included the steam strainer, emergency valve, ahead manœuvring valve, astern manœuvring valve and astern guarding valve.

The low pressure turbine is shown in Fig. 10. In the same manner as in the high pressure turbine, the forward bearing pedestal is flex mounted simplifying the construction and providing more effective provisions for expansion than in the 1930 concept. Similarly the L/D ratio of the bearings has been reduced in keeping with the advancements in bearing design.

While the other general design features are similar to the 1930 unit, the advanced design of the astern steam chest is of interest. The astern turbine nozzle chest formed a complete ring, retained in the casing by keeper plates and radial guides, so as to allow free radial expansion. The inlet was through a nozzle from the lower case, sealed by piston rings in a slip joint. This astern steam chest is split at the horizontal joint and held together with internal bolts accessible through manholes on either side.

The low pressure turbine speed was 4,518 r.p.m.

The steam seal and leakoff system on these turbines was

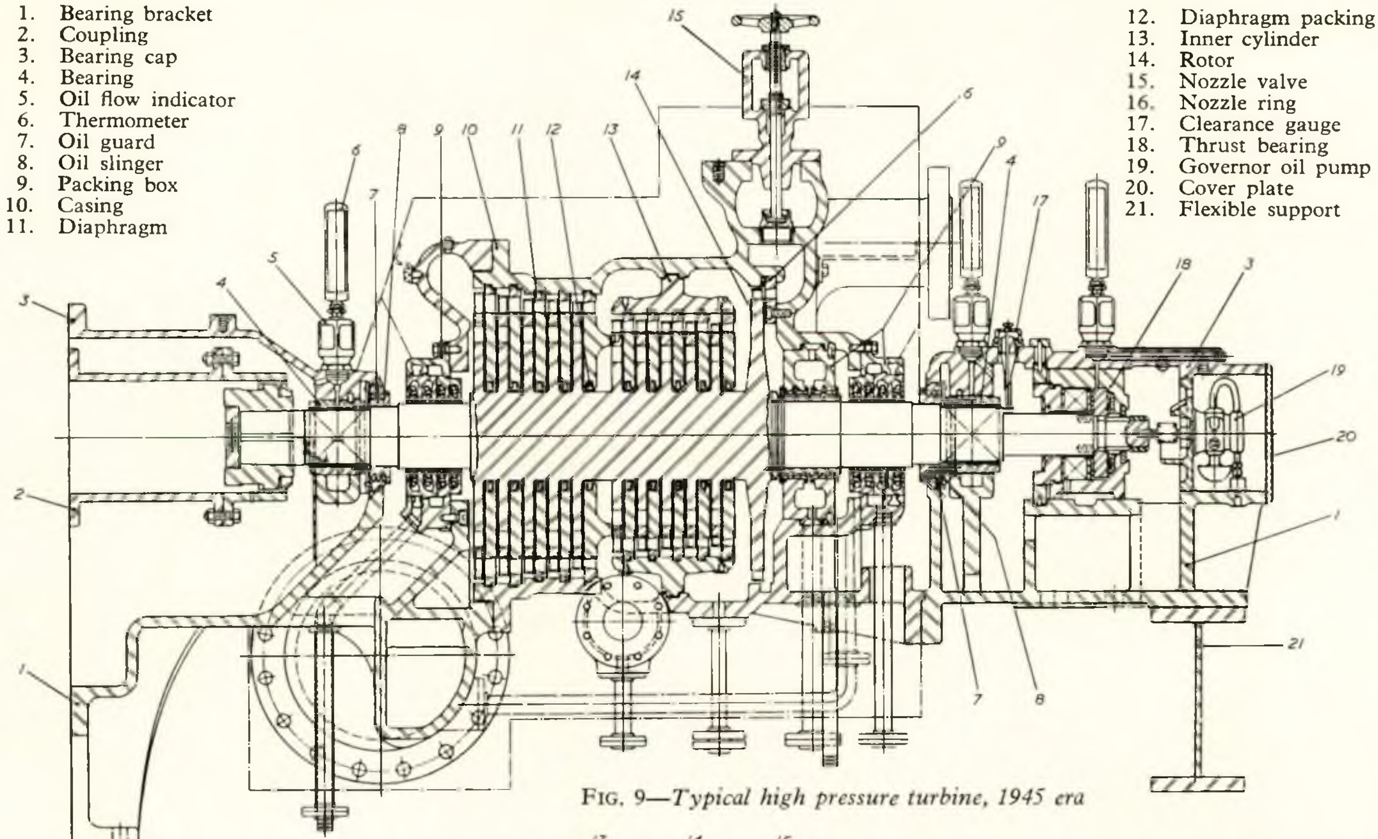


FIG. 9—*Typical high pressure turbine, 1945 era*

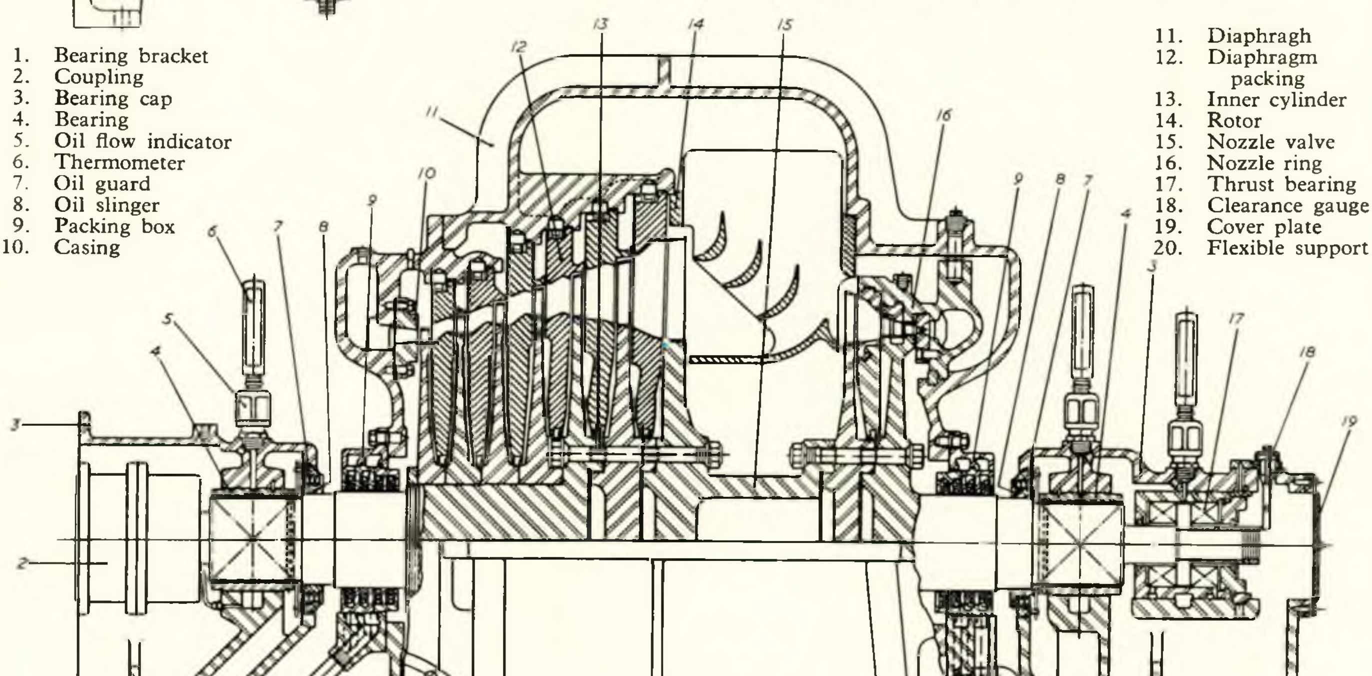
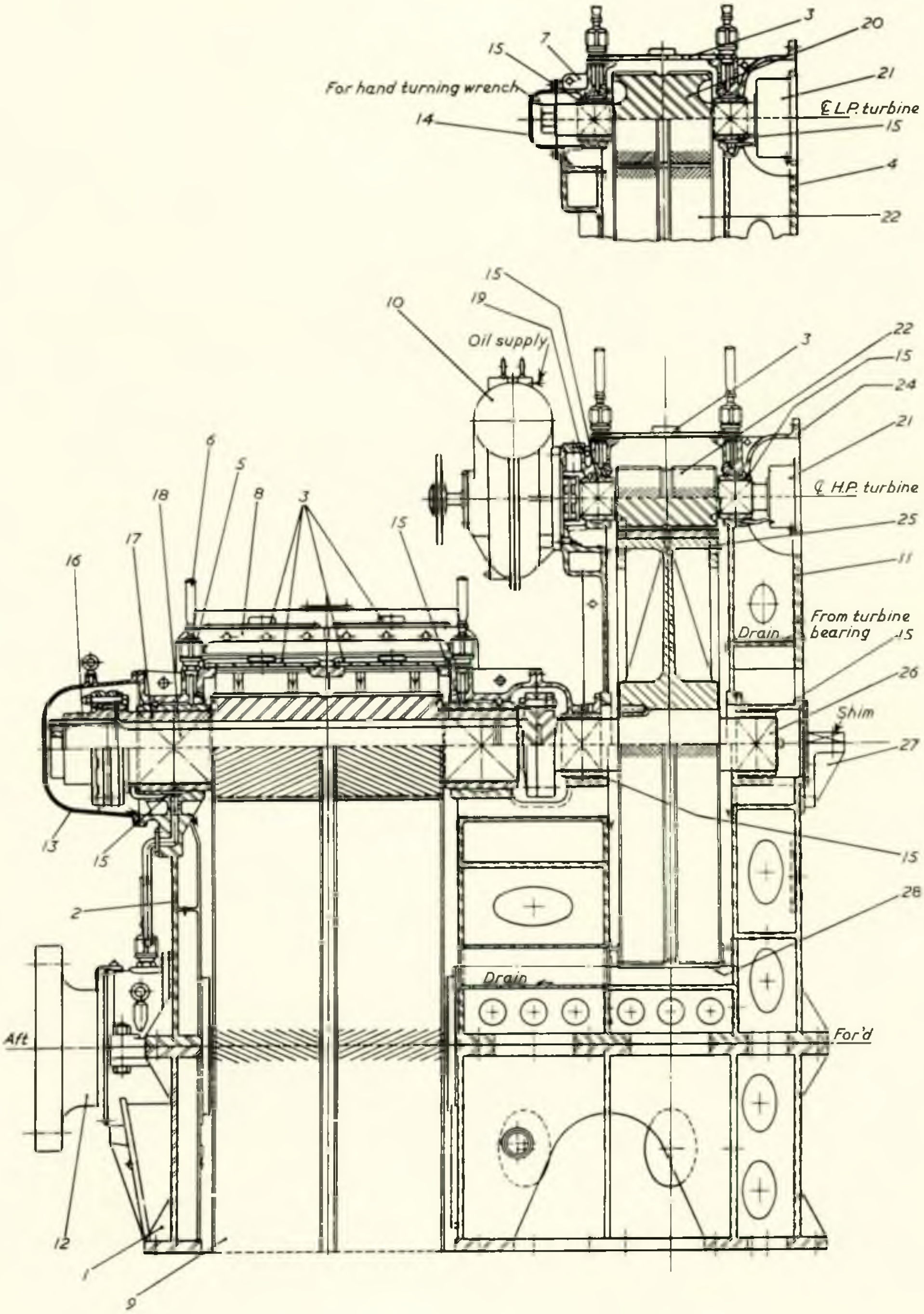
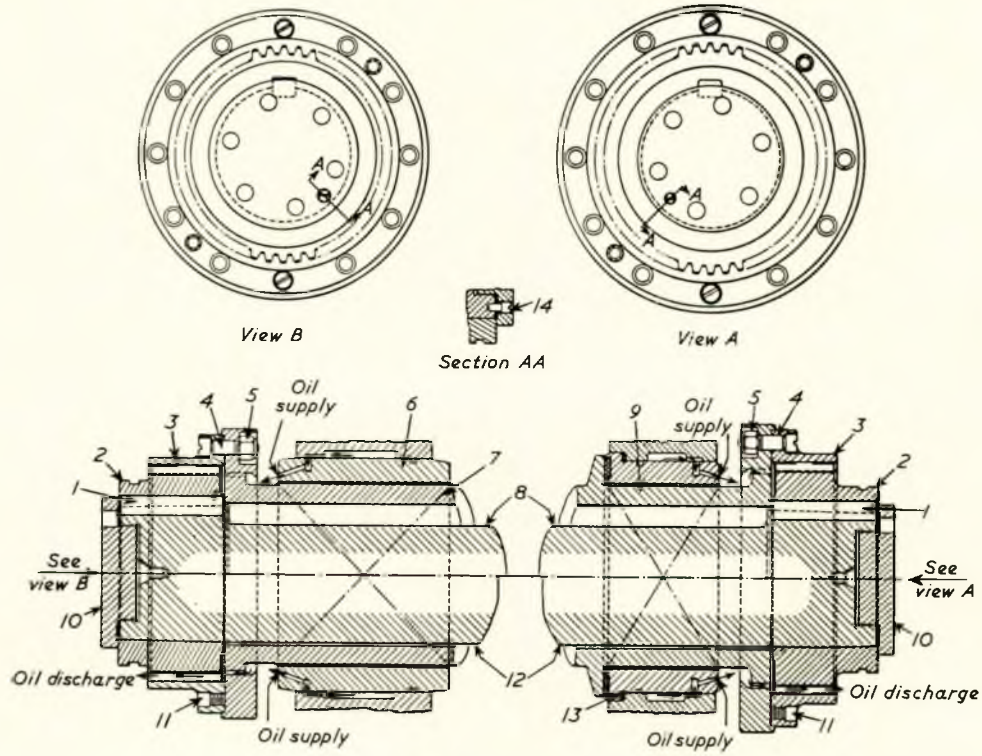


FIG. 10—*Typical low pressure turbine, 1945 era*



1. Lower case
2. Intermediate case
3. Inspection plate
4. Upper case L.P.
5. Oil flow indicator
6. Thermometer
7. Upper case cover, L.P.
8. Intermediate case cover
9. Low speed gear
10. Turning gear
11. Upper case, H.P.
12. Low speed gear shaft
13. Coupling guard
14. Cover
15. Bearing
16. Intermediate speed coupling
17. Intermediate speed pinion, H.P.
18. Quill shaft, H.P.
19. Inspection plug
20. High speed pinion, L.P.
21. High speed coupling
22. Intermediate speed gears, L.P.
23. High speed pinion, H.P.
24. Upper case cover, H.P.
25. Intermediate speed gear, H.P.
26. Intermediate speed gear shaft, H.P.
27. Turbine bracket
28. Baffle

FIG. 11—*Typical articulated reduction gears, 1945 era*



1. Key
2. Hub gear
3. Ring gear
4. Bolt
5. Nut
6. Pinion bearing
7. Intermediate speed pinion
8. Quill shaft H.P. side
9. Intermediate speed gear shaft
10. Lock nut
11. Hob for jacking (use coupling bolt)
12. Quill shaft L.P. side
13. Gear bearing

FIG. 12—*Typical gear tooth type coupling, 1945 era*

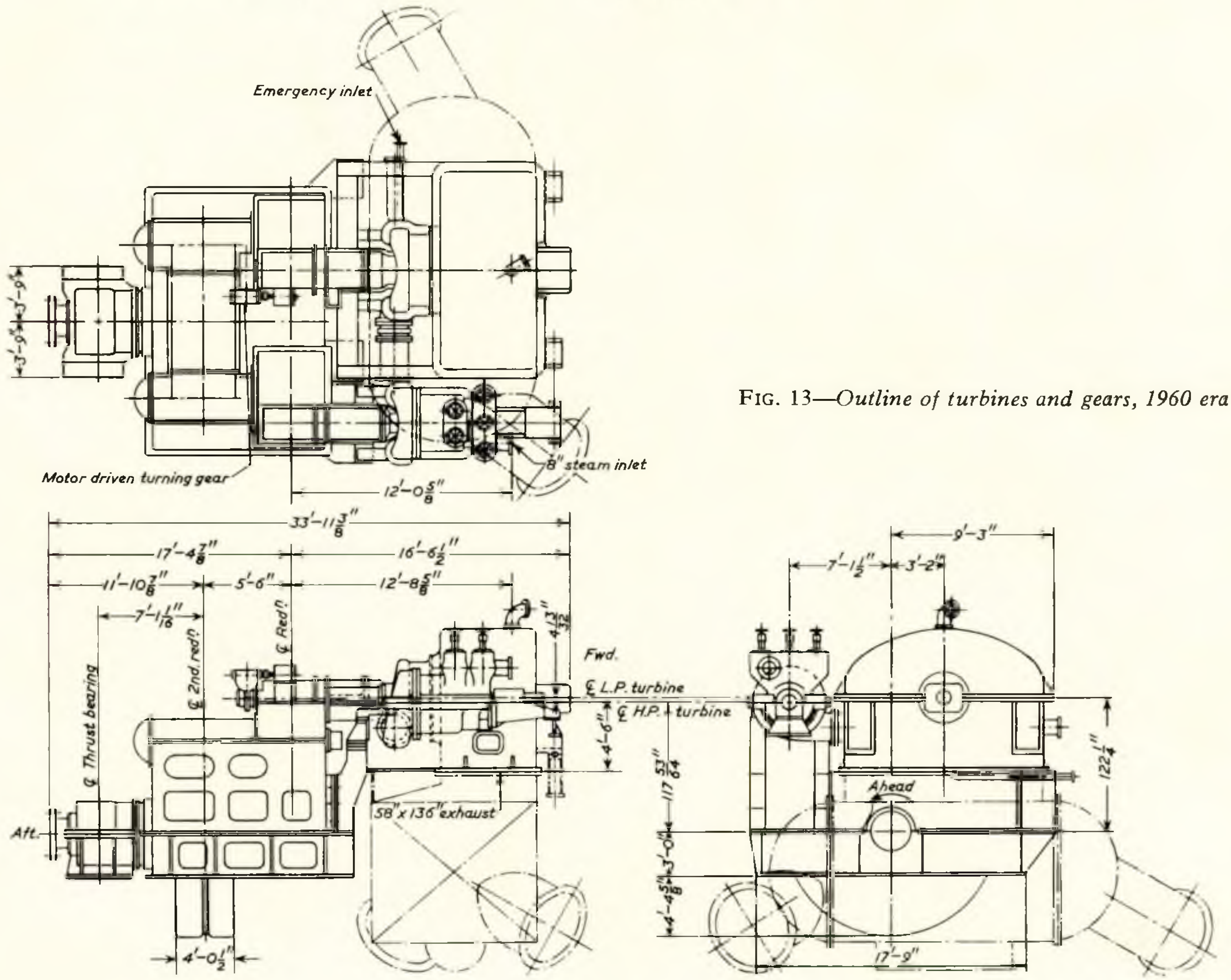
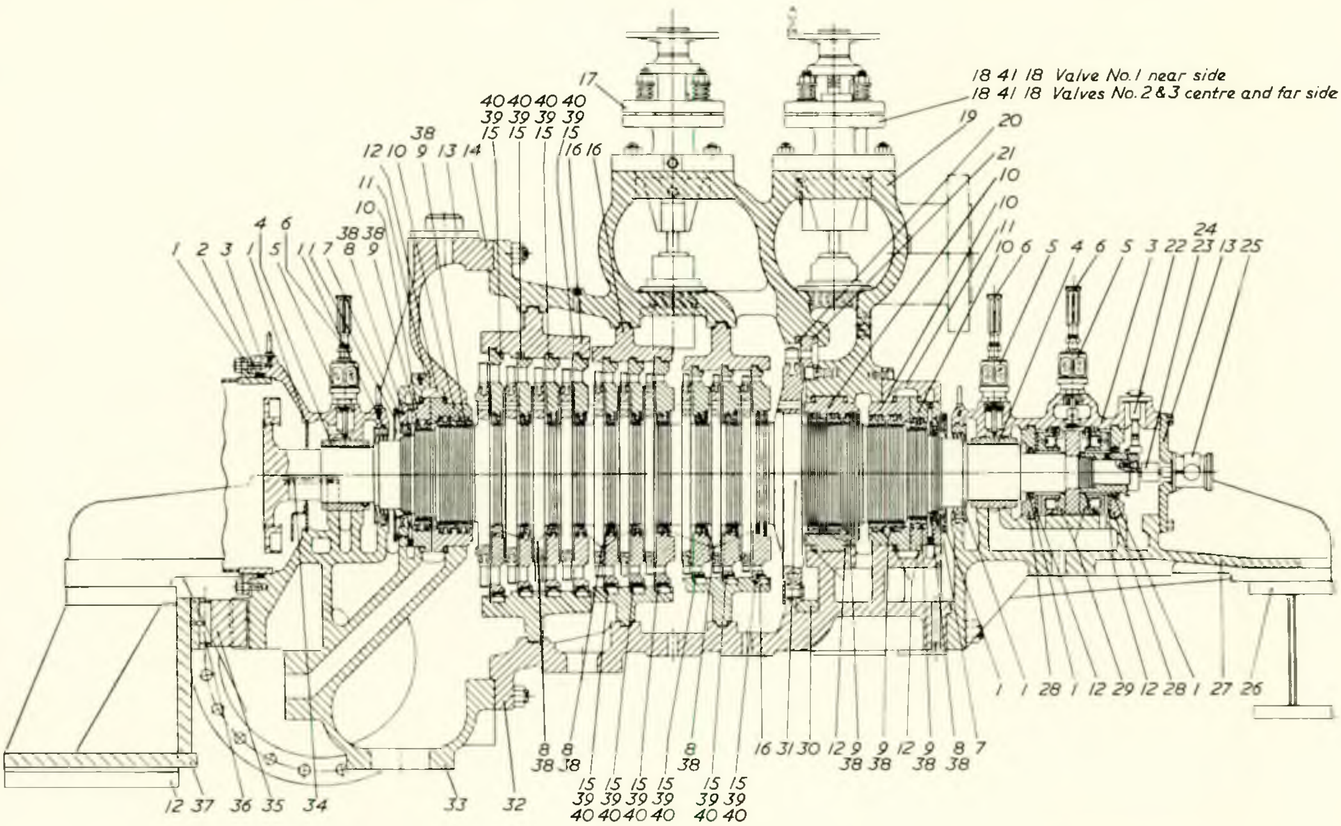


FIG. 13—*Outline of turbines and gears, 1960 era*



1. Oil guard
2. Metallic packing
3. Bearing bracket cap
4. Turbine bearing
5. Oil sight feed
6. Thermometer
7. Collar
8. Tightening ring
9. Labyrinth ring
10. Labyrinth housing
11. Packing box cap
12. Shim
13. Cover
14. Wheel case cover
15. Diaphragm
16. Inner wheel case
17. Bypass valve
18. Nozzle valve
19. Steam chest (upper)
20. Wedge segment
21. Nozzle ring
22. Thrust gauge
23. Coupling hub
24. Coupling sleeve
25. IMO pump
26. Support
27. Bearing bracket
28. Thrust bearing ring
29. Kingsbury thrust bearing
30. Baffle
31. Turbine rotor
32. Steam chest (lower)
33. Wheel case
34. Oil spray
35. Guide
36. Guide pin
37. Turbine support
38. Spring
39. Diaphragm guide pin
40. Diaphragm guide key
41. Hand valve

FIG. 14—*Typical high pressure turbine, 1960 era*

somewhat simplified to minimize the adjustments during manœuvring. Also a gland leakoff condenser was incorporated in the feedwater cycle to recover the heat from the excess steam.

The general features of the reduction gear are shown in Fig. 11.

Still retained at that time was the built-up gear wheel construction. Although the author's firm had already a very considerable number of all welded gears in service at that time, limitations in available manufacturing facilities during the war led to this choice. The second reduction pinions were now of the two bearing design, hollow bored and connected to the first reduction gears through quill shafts.

The couplings shown in Fig. 12 were of the fine tooth or dental type. These were successfully developed during this era and exhibited better load carrying and operating characteristics at high speed. The manufacture of these, with the teeth hobbed on a conventional gear hobber ensured complete interchangeability, greater accuracy and, at the same time, the elimination of tedious hand fitting required by the claw type.

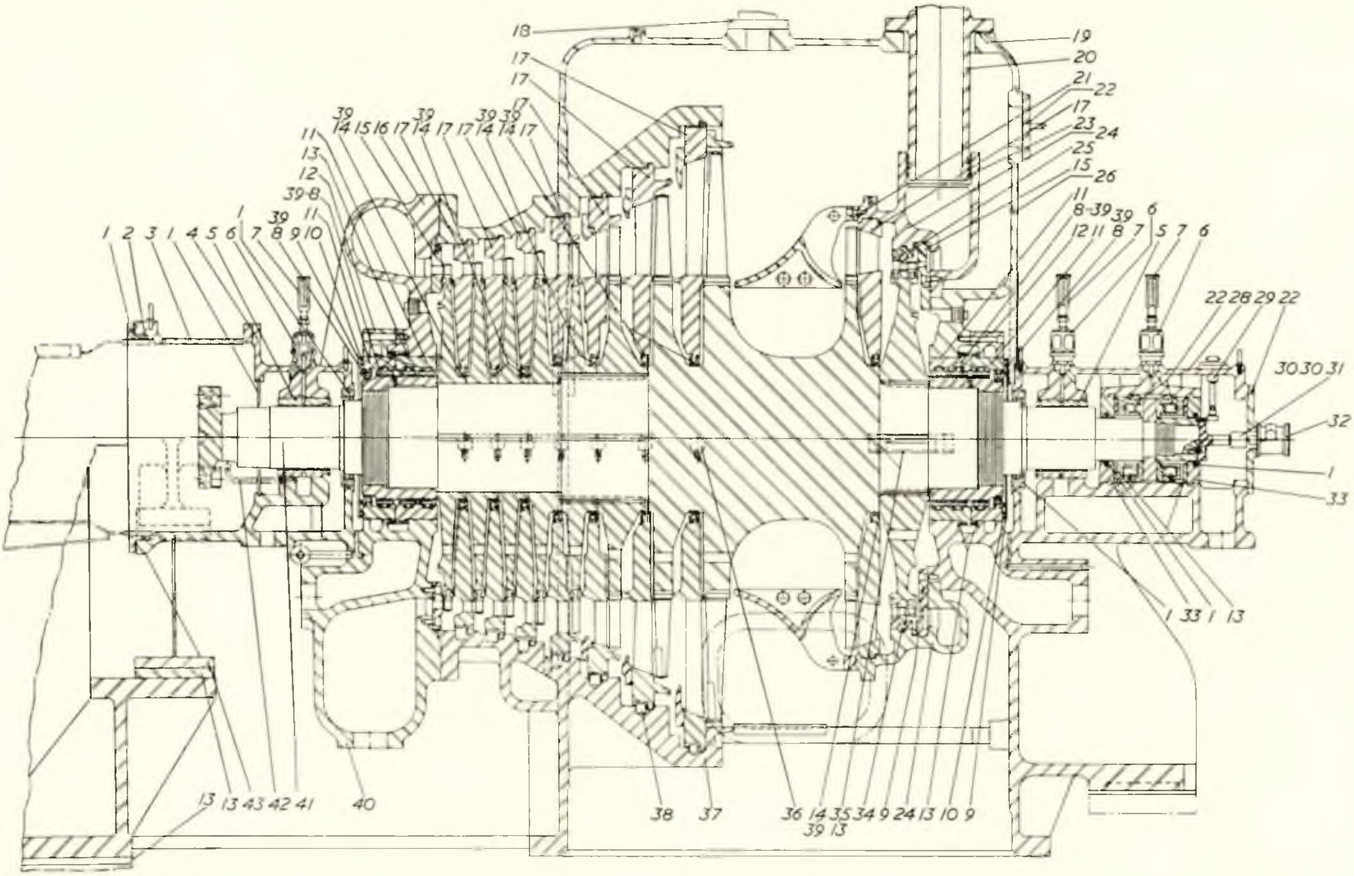
The 1960 Era

Fifteen years later, or in what the author regards as the 1960 era, steam conditions had increased to a nominal 600lb./sq. in. gauge, 850 deg. F. which might be regarded as the most popular conditions in the U.S.A. at the present time, although other conditions, of course, have also been used.

These years also brought with them a very significant increase in horsepower. While in the late 1940's the average United States tanker was equipped with 8,000 to 12,000 s.h.p. machinery, this has now been increased to a range of 17,500 to 25,000 s.h.p. The increased size of the ship made possible the economies which justified these larger powers and made it easier to use the higher steam conditions and to obtain better specific fuel rates.

Fig. 13 shows the outline of a typical 20,000 s.h.p. unit used in the 1960 era, designed to operate at 650lb./sq. in. gauge and 850 deg. F. with a propeller speed of 100 r.p.m., with a high pressure turbine speed of 5,448 r.p.m. and a low pressure turbine speed of 3,279 r.p.m.

While the power has been increased from 6,000 s.h.p. in the 1945 era to 20,000 s.h.p., the comparable overall length of the unit has increased less than 7ft., excluding the thrust bearing. The thrust bearing has been located aft of the main gear increasing the actual overall length to about 34ft. This, as discussed later in the paper, has provided the shipbuilder



1. Oil guard
2. Metallic packing
3. Coupling guard
4. Bearing bracket cap
5. Turbine bearing
6. Oil sight feed
7. Thermometer
8. Labyrinth ring
9. Retainer ring
10. Collar
11. Labyrinth housing
12. Packing box cap
13. Shim
14. Tightening ring
15. Nozzle ring
16. Steam chest (upper)
17. Diaphragm
18. Relief valve
19. Flexible gasket
20. Astern inlet pipe
21. Exhaust guide
22. Cover
23. "Step Seal" piston ring
24. Astern steam chest
25. Wedge segment
26. Wheelcase cover
27. Kingsbury thrust bearing
28. Bearing cap
29. Thrust gauge
30. Coupling hub
31. Coupling sleeve
32. IMO pump
33. Thrust bearing ring
34. Guide bucket ring
35. Retainer block
36. Diaphragm guide key
37. Wheelcase
38. Diaphragm guide pin
39. Spring
40. Steam chest (lower)
41. Turbine rotor
42. Oil spray
43. Turbine support

FIG. 15—*Typical low pressure turbine, 1960 era*

with more suitable provisions for absorbing the thrust of the unit and has removed thrust loading from the gear case proper. The other general arrangements are quite similar to previous well proven designs but with many interesting advancements in detail design.

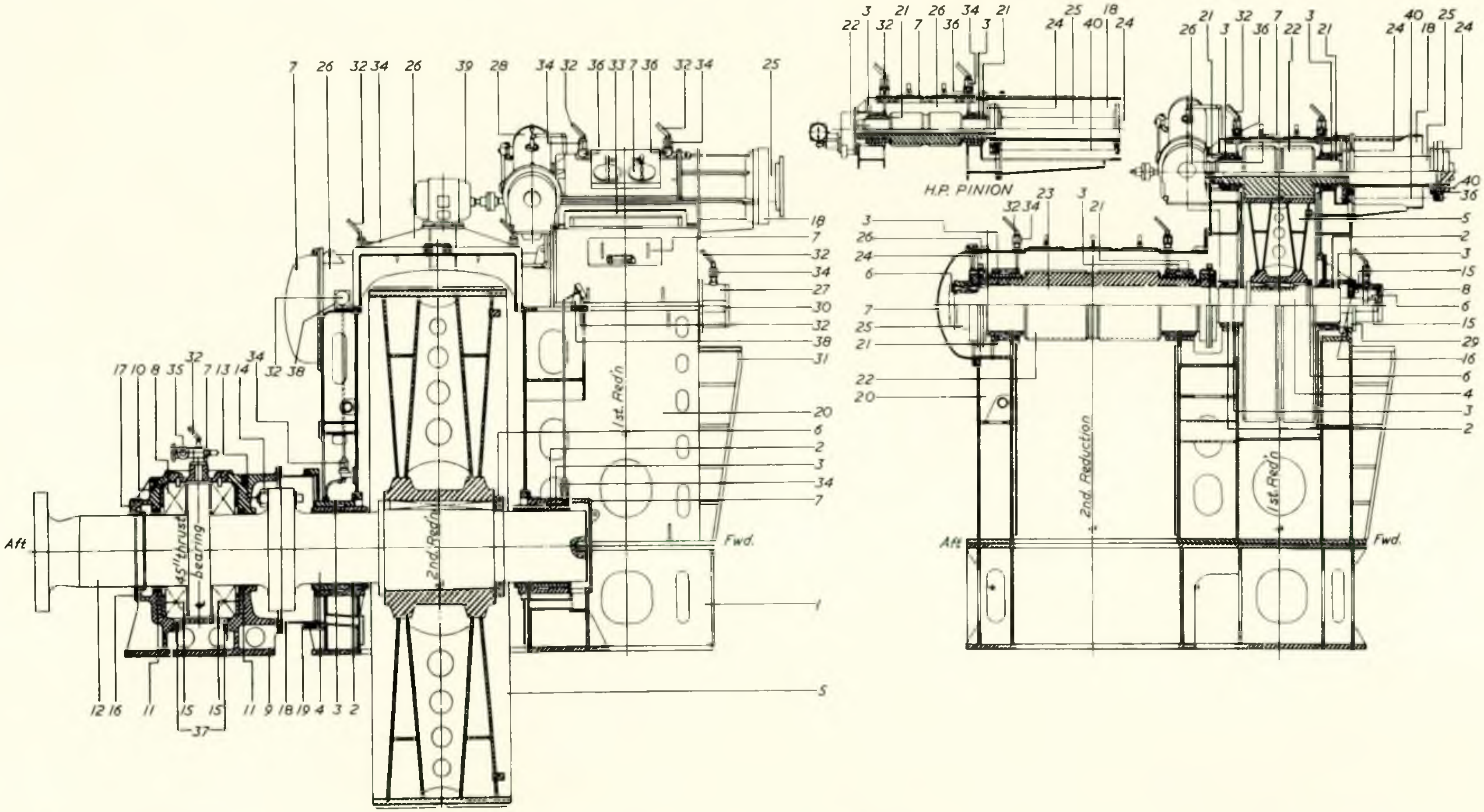
Fig. 14 shows a cross-section of the high pressure turbine. The forward bearing pedestal has been further simplified, with the governor oil pump taking suction directly from the lubricating oil header. The flex plate support has been refined by the use of parallel plates to further increase the strength and shock resistance. There are improved provisions for securing the thrust collar, both sides of the turbine thrust using tilting-pad type shoes. The adjustable feature has been removed from the thrust bearing. The L/D ratio of the journal bearings has been further improved with advancements in bearing design. The use of steel bearing shells was introduced. A refined type of rotor position indicator gives the operator a reliable running check of the rotor position. The oil seals are simpler, more reliable, and can be changed without lifting the rotor.

The rotor is a solid forging which has been checked by heat indication following all machining operations. The coupling is bolted to a flange on the turbine shaft, precluding the possibility of a poor taper fit.

The aft end of the turbine is supported by a combination of fore and aft flex plates off the centre line and a centre line key that allows the turbine to expand freely about its centre line.

All buckets are fastened to the wheel with inverted T-type roots. The carbon rings have been completely eliminated and replaced with labyrinth packing. All of this packing, both end seals and interstage is segmental and spring loaded so that neither the rotor nor the packing will be damaged by a temporary bow or unbalance.

All of the stages except the first are contained in the three inner wheel cases, providing a maximum of flexibility and generous extraction belts. The diaphragms are supported within these inner wheel cases on centre line keys, so that the diaphragm and case are both free to expand and contract



1. Lower gear case
2. Gear bearing
3. Bearing cap
4. Gear shaft
5. Gear wheel
6. Check nut
7. Cover
8. Thrust bearing
9. Thrust bearing pedestal
10. Thrust bearing cap
11. Sealing ring
12. Thrust bearing shaft
13. Bolt—tapered
14. Nut
15. Thrust bearing shim
16. Oil guard
17. Rotor position indicator
18. Coupling guard
19. Gland
20. Intermediate gear case
21. Pinion bearing
22. Pinion
23. Quill shaft
24. Coupling sleeve
25. Coupling hub
26. Gear case cover
27. Thrust bearing housing
28. Turning gear
29. Retainer ring
30. Thrust bearing cover
31. Turbine support
32. Thermometer
33. Oil spray cover
34. Oil sight flow indicator
35. Angle regulating valve
36. Adapter
37. Ball bearing and housing
38. Bracket
39. Electric motor
40. Coupling bolt

FIG. 16—*Typical articulated reduction gears, 1960 era*

while providing means to maintain their cold condition concentricities. All diaphragms are of completely welded construction.

The particular high pressure turbine design illustrated has bypass valves by which three stages can be bypassed to provide additional maximum power. These have been incorporated in many United States ships as a national defence feature and as such are not used in normal commercial operation.

The ahead nozzles are all located in the upper half of the turbine and are divided into groups which provide economical operation over the operating range of the ship.

The hand valve assemblies for both the normal ahead nozzle blocks and the overload nozzles utilize stellite facing on the valve seats and discs, nitriding of the valve stems, leakoff bushings on the stems eliminating all soft packing, and a spring loading feature to preclude jamming and bending the valve stems.

The low pressure turbine for this class is shown in Fig. 15. The turbine case design incorporates on each side a steel girder which supports the condenser. These girders are supported on the forward end by the shipbuilder's foundation and on the aft end by the gear case.

The turbine case itself is fabricated entirely of steel castings and plates largely eliminating temperature restrictions during astern or abnormal operation.

An emergency governor oil pump has been added to the forward end of the low pressure turbine also, thus affording protection in case of any derangement in either the high or the low pressure train.

Similar improvements to those noted in the high pressure turbine have been made.

The rotor is a combination of built-up and solid rotor, endeavouring to preserve the advantages of both. Those wheels which are not integral are shrunk directly on the shaft without the use of taper bushings.

The author's company has also built a number of units with solid rotors. However, some owners prefer to retain the possibility of removing a wheel in the event of damage. Nevertheless, it is the author's opinion that solid rotors will become the rule rather than the exception because such derangements are becoming more and more scarce.

Blade mounting in the wheel is with inverted "T" roots except the last three ahead stages and last astern stage which utilize the bulb and shank type fastening.

As in the high pressure turbine the diaphragms are of all welded design with centre line support, thereby permitting freedom of expansion. Carbon packing has been eliminated. All clearances are extremely ample with the exception of the labyrinth packing and this, by virtue of its spring loading, can readily adjust itself. Continued low maintenance is assured by the absence of any rotating fins on the shaft and the characteristic clearances of the impulse turbine.

Moisture separators are provided in the diaphragm design of the last few stages of the ahead turbine, which remove a significant portion of the entrained moisture from the steam. Supplementing this, there is stellite shielding on the leading edges of the last two stages. Experience has shown that the combination of the two effectively control erosion in the low pressure turbine.

The astern turbine case, nozzles and diaphragms are again mounted in the forward end of the low pressure turbine, retained axially by a keeper plate arrangement, supported by guides but otherwise completely free to expand or contract. Fig. 15 clearly illustrates the method of steam admission. Also of interest are the stainless steel protective sleeves on the internal braces, protecting the braces from erosion due to the moisture in the exhaust.

Not shown in the illustrations, but of considerable interest, is the gland seal steam and leakoff system. This is controlled by a completely automatic gland seal regulator operated by lubricating oil, measuring and controlling the gland seal pressure with less than 1lb./sq. in. gauge variation between no load and full load.

Complementing this is the present practice of inducting the gland leakoff steam into the air ejector condenser where its heat is regained by the feed water.

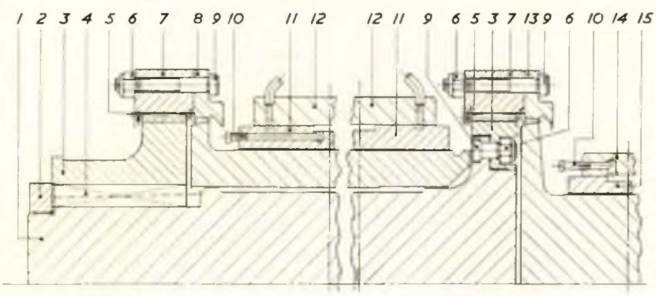
Also of interest is the present design of manœuvring valves. The emergency valve and ahead manœuvring valve are combined into a single valve, thus making a simple and compact arrangement. By operating the emergency valve entirely with oil, the difficulties formerly experienced with steam actuated valves have been eliminated. The emergency valve functions in the event of: overspeeding of either turbine, low lubricating oil pressure, low gravity tank level. By having the emergency valve control only the ahead steam, the astern steam can be applied quickly to bring the unit to a stop in case of an oil failure. Conversely, protection in the astern direction is not considered necessary as the throttle is constantly attended while going astern. With the advent of all steel low pressure turbine cases, the low vacuum trip is usually eliminated in the interest of simplification.

Fig. 16 is a section of the main reduction gear. The thrust bearing has been mounted aft and independent of the main gear. No journal surfaces are provided in the thrust, although there are provisions for dummy bearings to be used for shipyard alignment.

The most noticeable feature is the use of the all welded gear wheel for both the first and second reduction and the elimination of the shrunk on band.

The gear case is of all welded construction and so designed that all bearings can be rolled out without lifting any of the gear elements or gear case. There are improved access openings to allow adequate inspection of gear tooth contact, oil sprays, etc. Visible oil sight flow indicators as well as thermometers are provided for all bearings. Notable in the first reduction is the use of oil sprays of both the ingoing and outgoing mesh to properly control the lubrication and cooling.

Means are also provided to check the alignment of the gear case after assembly, and even after the ship is in service, with no major disassembly. Of interest is the use of tilting-pad type thrust bearings in the first reduction gear wheel and the use of full quill shafts through the pinions with the flexible couplings described below.



1. Quill shaft
2. Check nut
3. Coupling hub
4. Key
5. Retaining ring
6. Coupling bolt nut
7. Coupling sleeve
8. Second reduction pinion
9. Coupling bolt
10. Oil spray nozzle
11. Pinion bearing
12. Pinion bearing cap
13. First reduction gear rotor
14. Gear bearing cap
15. Gear bearing

FIG. 17—*Typical intermediate speed coupling, articulated gear 1960 era*

Typical quill shaft couplings for this design are shown in Fig. 17. These are of the fine tooth design using an involute tooth of high accuracy. Recent developments which have resulted in increased service and dependability have been the increased accuracy both in tooth spacing and surface finish, all teeth being finished by crown shaving and radial clearances being held to a minimum. Paralleling this has been the use of harder materials in the 300 to 350 Brinell hardness range. Crown shaving of the hub teeth has been very effective in preventing load concentrations on the ends of the teeth. Oil dams have been added in the coupling sleeve to ensure adequate lubrication by providing an annular ring of oil surrounding the mating teeth.

BOILER DEVELOPMENT

The 1930's found marine engineers struggling to advance steam conditions beyond those of the common cycle employing Scotch boilers generating steam at 225lb./sq. in. and with superheat of only 50 to 100 deg. F. It was recognized that boiler steam conditions were a function of the desired turbine efficiencies and were governed by economic considerations such as investment and return, evaluated plant weight and space, oil rate per shaft horsepower, labour costs and fuel costs—the very factors still so important today. Clearly, a more widespread use of a better boiler was indicated.

When the tanker *G. Harrison Smith* was designed it was decided to fit the most modern watertube boiler available. Culminating many years of development, the three pass sectional header boiler with 2-in. outside diameter tubes, riveted steam drum, all refractory furnace and interdeck superheater, was well suited to supply the 375lb./sq. in. gauge, 725 deg. F. steam at the turbine inlet required for the vessel's

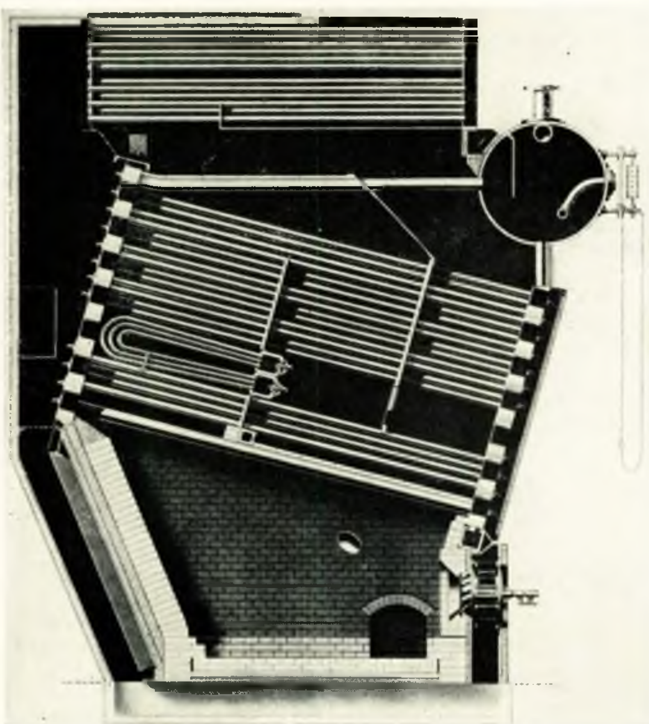


FIG. 18—*3-pass, sectional header, marine boiler, 1930 era*

4,000 s.h.p. In addition, air-heaters supplied added economy and heated air to assist in burning the cracked fuels then just being introduced. The refractory furnace was deemed a necessity since it was believed that the new bunker fuel could not be burned successfully in a relatively cold, water cooled furnace. Much thought, too, was given to arranging the soot blowers to clean the soot accumulating shelves formed by the three pass gas baffle installation.

Steam pressures of 400 to 450lb./sq. in. and temperatures of 700 to 750 deg. F. were then considered the ultimate. Designers were sure that above these figures the cycle gains would diminish rapidly as the engineering risks increased greatly. It was not without some misgivings, therefore, that they turned their backs for the last time on the Scotch boiler, for which time had finally run out.

By the late 1930's, and early 1940's ships of 4,000 to 8,000 s.h.p. were in demand and steam conditions of 450lb./sq. in., 750 deg. F. had become standard and considerations were given to the use of still higher pressures and temperatures. The same arguments used against going to the 450lb./sq. in. cycle were revived—"materials were unsuitable, piping joints would not stay tight, and complicated superheat control would be required".

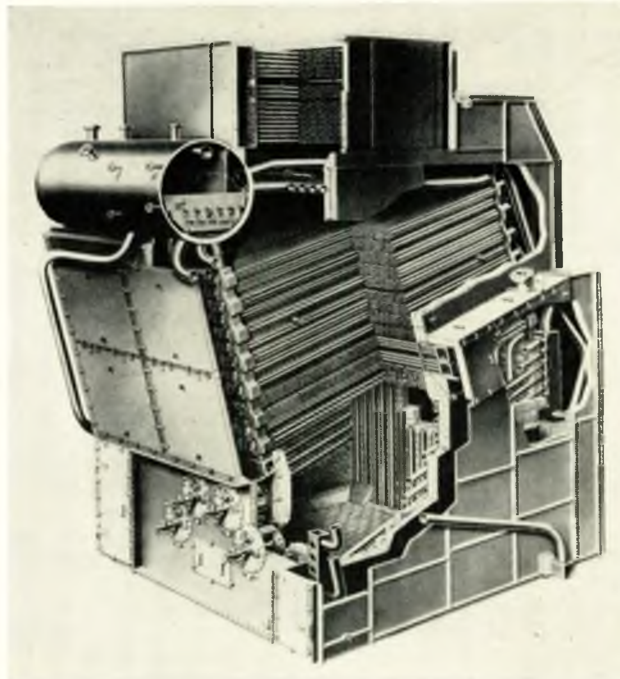


FIG. 19—*Single-pass header type marine boiler, 1945 era*

However, progress had continued in boiler design and two types of boilers were available to meet the increased demands. The most prominent type was a descendant of the three pass header boiler, while the other was an integral furnace two-drum type developed from contemporary land practice. The former, a single gas pass sectional header boiler, was designed to provide more heating surface for improved efficiency with less draft loss. The elimination of baffles also improved the problem of cleaning boiler and superheater tubes. To reduce refractory maintenance costs, water walls were applied to the furnace. This surface was of the stud tube-refractory protected type to keep furnace heat absorption within the limits of the then existing techniques of water side chemical treatment. In addition, the use of air-heaters, economizers, or both, to improve efficiency was no longer considered unusual and automatic controls were applied with greater frequency to reduce labour costs, and improve operation.

It was this type of boiler that was selected for installation in the 6,600 s.h.p. tanker *Flagship Sinco,* to provide 450lb./sq. in., 740 deg. F. steam with the higher reliability inherent in a time tested design. Similar boilers were selected for widespread application in ships designed by the U.S. Maritime Commission.

It became increasingly clear to the boiler designer that the two-drum integral furnace boiler offered the most promise of successfully meeting the new cycle requirements economically. It was better equipped to take the large superheaters required, more flexible in furnace and saturated boiler bank arrangement, and was capable of being operated at much higher ratings than the corresponding header type boiler. Equipped with air-heaters, or economizers, or both, high efficiencies were obtainable with high reliability. As a consequence, following the respite granted by World War II, the header type boiler found fewer advocates.

During the years following the war the continuing deterioration of the available fuels had a pronounced effect on

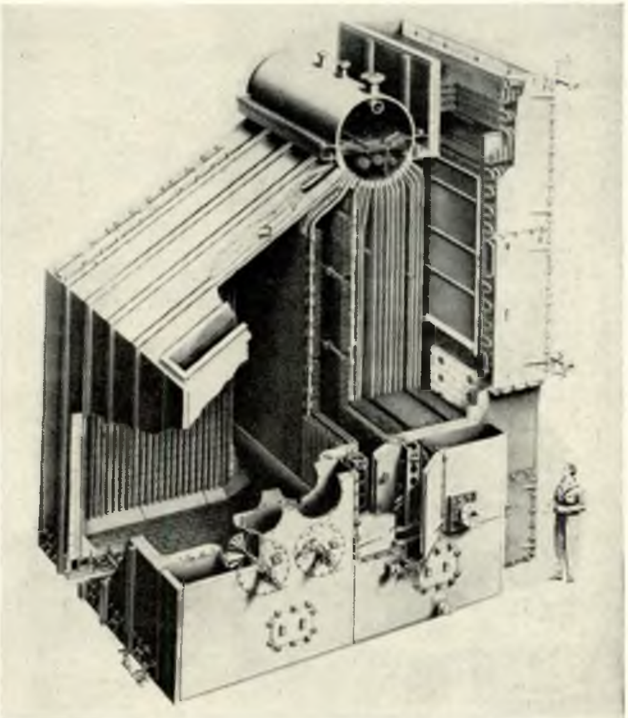


FIG. 20—*2-drum type marine boiler, 1960 era*

two-drum boiler design. Use of fuels with an ash content of up to 0·3 per cent and rich in sodium and vanadium compounds, produced severe slag deposition on superheater tubes, rapid deterioration of superheater tube supports and premature failure of furnace refractory. With up to 5-6 per cent sulphur, in some cases, extremely rapid failure of cold-end heat exchanger surface also was experienced.

To alleviate these difficulties, when the boilers for the 1960 era tanker were designed, important changes in arrangement were included for operation at 600lb./sq. in., 875 deg. F. at the superheater outlet. Bare furnace tubes in the side and rear water wall were arranged on close spacing to prevent slag erosion and wastage of brickwork. Floor supply tubes were utilized to provide increased cooling and longer life for furnace floor refractory.

The most significant departure from previous standard practice was the use of a cavity type superheater. With "walk-in" space between the legs of the superheater tubes, access for cleaning, inspection and repair was greatly improved. The cavity, too, permitted the installation of an expendable and readily replaceable type of superheater support. Prior to the use of the cavity superheater, replacement of superheater supports was a major task requiring the complete renewal of superheater tubes as well as supports.

The cavity superheater also provided the space necessary to install long, retracting, mass action sootblowers. Because of the concentrated force obtained from this type of blower, slag deposits were reduced to an absolute minimum, with the result that the period between necessary water washing or manual cleaning of the superheaters was greatly increased.

The problem of cold-end heat exchanger corrosion was approached by two means. Firstly, the economizers were built of a copper bearing low alloy steel having 2 to 2½ times the corrosion resistance of ordinary steel. Secondly, the use of feed water at a temperature of 277 deg. F. greatly reduced the chance of the tube temperature falling below the dew-point, therefore acid deposits were minimized and long life of the economizers assured.

These and numerous other developments have contributed greatly to increased reliability, decreased maintenance requirements and the capability of successful operation at even higher pressures and temperatures than are at present considered standard.

The 1930 tanker had little or no automatic feed water control or automatic combustion control. It relied heavily upon hand lances for boiler cleaning and for a gravity salinometer to test boiler water.

By the early 1940's, many tankers had fully automatic feed water control. Manually operated sootblowers had taken the place of the hand lance. The proper control of boiler water was recognized and salinity, alkalinity and hardness were listed and controlled chemically.

In 1960 even more sophisticated controls of feed water, combustion and superheat were developed. Soot blowing has become almost completely automated with electrically or air powered blower heads, automatic draining of steam lines and retracting superheater blowers.

CONDENSER DEVELOPMENT

Maritime steam condensers as furnished for tankers are generally designed for two passes of sea water through the tubes at velocities of 5-7½ft./sec. Due to the larger quantity of cooling water required, single pass condensers are rare except on naval vessels which rely upon scoop injection rather than pumping capacity for maximum flow. Occasionally, in small auxiliary condensers, three or four passes are used. Usually, condenser design is based upon operation at 1·5in. Hg. absolute with 75 deg. F. inlet water. To allow for tube fouling, 85 per cent of the clean tube heat transfer rate is a normal design allowance. Tube sizes have been standardized for many years at ¾-in. outside diameter by 18 B.W.G. (0·049-in. wall).

There has been little change in the material of condenser shells, support plates, air hoods, hot wells and baffles which are constructed of steel. Material changes have occurred in waterboxes, and predominantly in tubes and tube sheets.

Around 1930 the majority of main condenser shells were round and usually contained a single bundle of Admiralty tubes rolled at one end and packed at the other end into Muntz metal tube sheets. Cast iron waterboxes were the general rule.

By 1945 aluminium brass tubes were preferred in the U.S.A., either rolled and packed, or alternatively rolled at both ends into Muntz metal or naval brass tube sheets. In the latter case of expanded tubes, an expansion joint in the shell permitted differential expansion between tubes and shell. Waterboxes were either cast iron or steel rubber lined.

By 1960, with larger propulsion plants, main condensers generally became rectangular in cross-section with tube bundles split to provide more favourable steam distribution throughout the tubes by means of a centre steam lane. Tubes became 90-10 copper/nickel alloy with Muntz metal, naval brass, or 90-10 tube sheets. Waterboxes remained alternately cast iron or steel rubber lined.

At present, with more concentration on salt water corrosion and erosion at inlet tube end joints, waterboxes are generally deeper to lessen turbulence. Some cast iron or steel rubber lined waterboxes are being replaced by waterboxes fabricated of 90-10 copper/nickel, and possibly there is a trend to use 90-10 copper/nickel tubes and tube sheets. Packed tubes continue to remain popular but tubes expanded at both ends are also used.

It is possible in the future that "U" tube bundles will replace the conventional straight tubes with their associated return waterboxes in two pass condensers. Tubes may be specified as stainless steel with higher water velocities to maintain the present heat transfer rates and to maintain tube cleanliness by high velocity scouring action. Also, with increasing stress being placed on low oxygen content in the condensate, it is probable that separate steam heated de-aerating tubes will be installed above the hot well in future maritime condensers.

TURBINE GENERATOR DEVELOPMENT

Electric load capacity aboard tankers naturally increased, not only with the size of the ship but also with additional requirements for electric auxiliaries and various other uses. The actual capacity required for a given size of vessel depends largely on whether motor-driven or turbine-driven cargo pumps are used. In their development, turbine generators have followed a pattern somewhat paralleling that of the main turbines.

The 1930 tanker was supplied with two turbine generators of 300 kW., 240/120 volt d.c., operated at main unit steam conditions. The specific ships mentioned for this era, also had an attached generator in line with the endeavour to drive as many auxiliaries off the main engine as possible.

The first vessels built in the U.S.A., completely equipped with a.c., were the Atlantic Refining Co. tankers of the *Van Dyke* class, followed shortly thereafter by the well known T.2 class which introduced the advantages of a.c. for ship machinery to the U.S.A. Merchant Marine, whereupon a.c. soon became standard.

While the basic design of high speed turbine with bottom exhaust, driving a 1,200 r.p.m. generator through a reduction gear, has remained approximately the same, significant advances have been made in mechanical details contributing to reliability and reduced maintenance consistent with good operating economy.

Turbines are of the multi-valve type using a bar lift or other devices to open nozzle groups in sequence. The use of all labyrinth packing has contributed to reduced maintenance. Governing continues to be with flyball type governors through an oil operated servo-motor giving a high degree of reliability. The four-bearing unit with a gear tooth flexible coupling between the turbine shaft and pinion has proven itself in marine service.

The open wound d.c. generator was followed by the open a.c. generator and that, in turn, by the completely enclosed a.c. generator, water cooled and now generally statically excited.

PUMP DEVELOPMENT

The scope of this paper does not permit a detailed treatment of pump design. However, as far as the operator and the naval architect are concerned, it is significant that, in general, in the 1930 era the industry had just gone from the reciprocating pump to the centrifugal pump, and used the latter in the horizontal arrangement in most cases. Since then, the use of vertical pumps, usually motor driven, wherever possible, has become the general practice. The main feed pump is still usually turbine driven because of the advantages inherent in better speed control, reliability and the fact that the feed pump turbine exhaust can be used advantageously.

During the early 1930's, rotary pumps for lubricating oil and fuel oil service became available both in Europe and in the U.S.A. The author's company introduced the De Laval IMO pump to the marine field in the U.S.A., where it rapidly gained universal acceptance during the next decade in marine power plants, due to its simplicity of construction achieved by the absence of external bearings and timing gears. Its ability to use direct motor drive as well as its very high reliability, its ability to handle difficult fluids, including lubricating oils containing large amounts of entrained and dissolved air, are outstanding advantages. Today, fuel oil service pumps have been developed and are currently in use over a wide range of capacities with discharge pressures of 1,150lb./sq. in. or more. Fuel oil transfer and cargo pumps are available for practically any desired discharge pressure, with capacities upwards to 1,000 gal./min.

EVAPORATORS

Perhaps one of the most significant advancements has been in the evaporator plant. The 1930 evaporator was a high pressure, single stage unit, grossly inefficient, requiring frequent and time consuming cleaning. The ship generally relied on its fresh water tanks for the normal voyage.

The great steps forward in evaporators came during the war years when ships no longer pursued their normal trade routes and the additional personnel greatly increased the potable water requirements.

Today the evaporating plant is a low pressure unit utilizing extracted steam from the main turbine, operating at low pressures and temperatures thus minimizing scaling with its operation automatically controlled and monitored.

The 1960 ship is able to rely entirely upon its own evaporators and carry only sufficient fresh water for emergency requirements.

STEAM CONDITIONS

One of the most significant developments in marine power plants during the past 30 years has been the advancement of steam conditions. The elevation of steam conditions for marine power plants was accomplished with the minimum of difficulties since in each case reliability had previously been established by land installations, primarily in electric utility plants.

By the use of higher steam pressures and temperatures, and other associated advances, appreciable reductions were made in fuel consumption, space and weight of the propelling machinery.

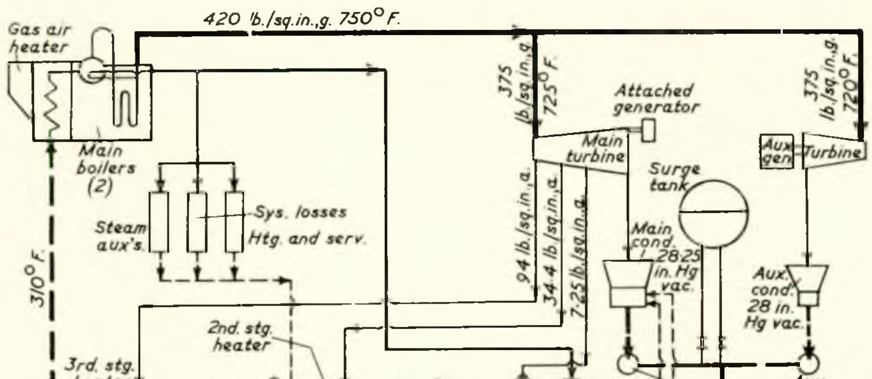


FIG. 21—*Simplified flow diagram for 1930 era tanker*

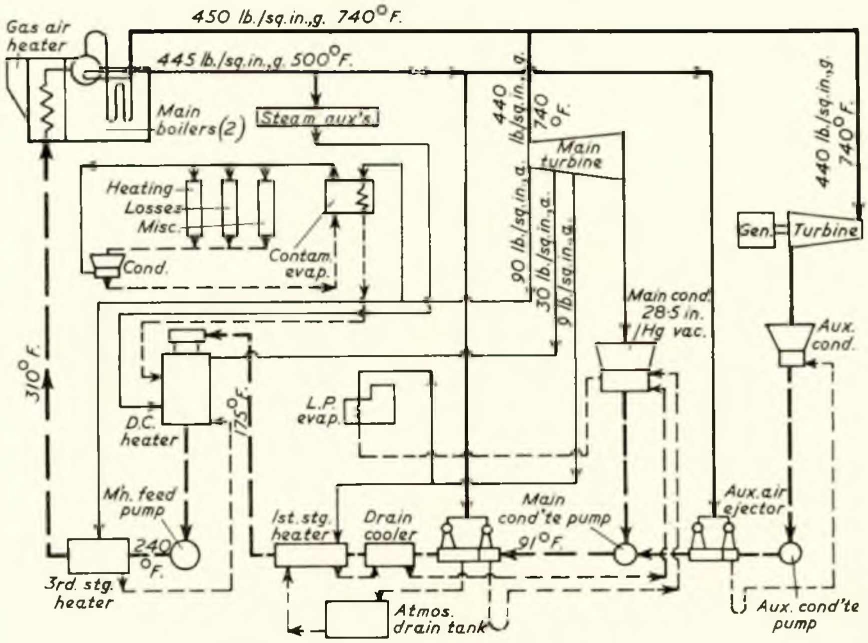


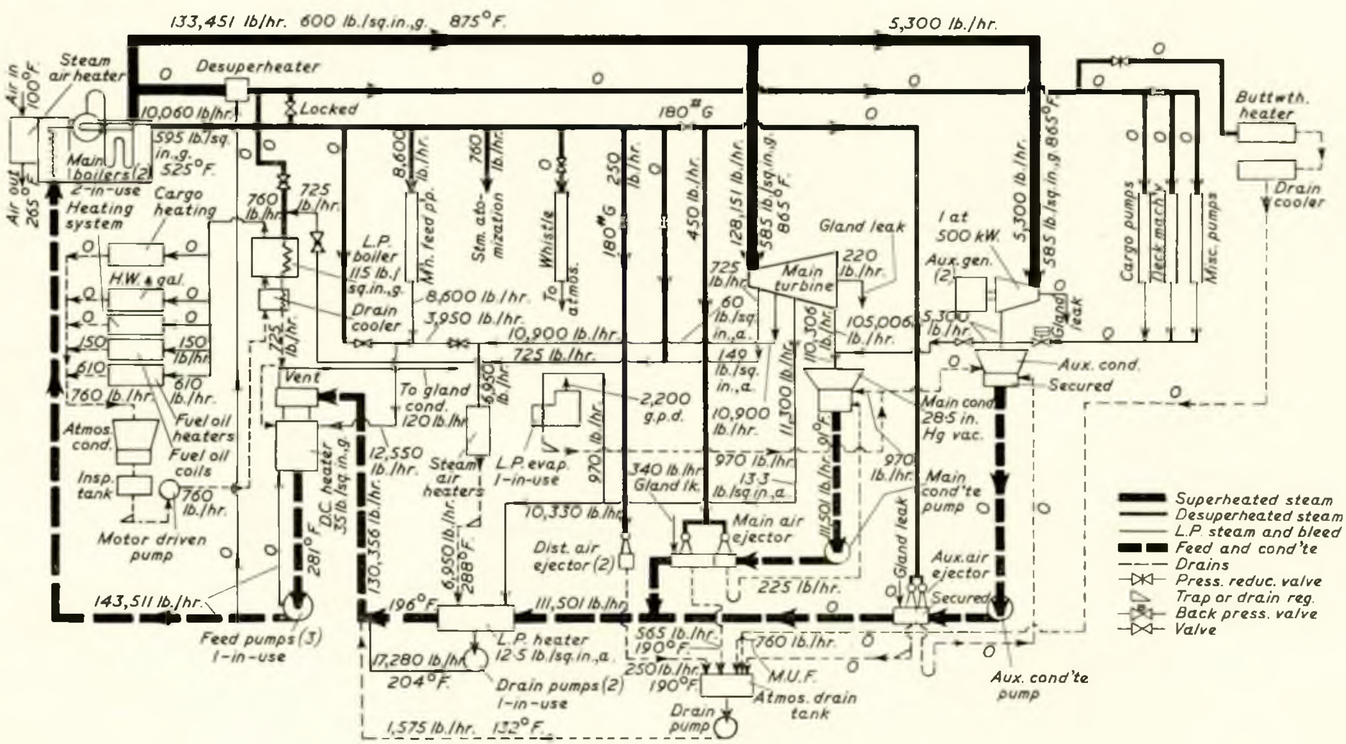
FIG. 22—*Simplified flow diagram for 1945 era tanker*

It is interesting to note that at present merchant ships have, in general, standardized on steam conditions at the superheater outlet of 600-650lb./sq. in. gauge and 850-900 deg. F., while the U.S. Navy has standardized on 1,200lb./sq. in. gauge, 950 deg. F. for combatant surface vessels using fossil fuel.

CYCLE CHANGES

Associated with the advancing of the steam conditions, changes in the power plant cycle have also been accomplished in order to effectively utilize the different steam conditions.

Fig. 21 shows in simplified form the flow diagram for the 1930 era tanker. This cycle was of the "open type", utilizing a surge tank and three stages of regenerative feed water heating, as well as an attached generator in addition to the conventional turbine generators, a booster pump taking suction from the condensate pump discharge, and a split inter and after condenser.



Fuel consumption, lb./hr. = 10,000
Fuel rate, lb./s.h.p./hr. = 0·500
20,000 s.h.p. Trial condition

FIG. 23—*Typical heat balance flow chart for 1960 era tanker*

Fig. 22 illustrates the refinements that were made in the 1945 era cycle. This cycle was of the "closed type" with a direct contact feed water heater in order to further reduce the amount of dissolved oxygen in the boiler feed water. The attached generator and the condensate booster pump were eliminated, but the three stages of feed water heating were maintained. A contaminated evaporator, or low pressure steam generator, was also used to further maintain the purity of the boiler feed water. Purity of feed water is essential with higher steam conditions.

Fig. 23 represents a typical heat balance flow chart for the 1960 era tanker. This cycle has become essentially the standard cycle today for turbine tanker machinery with steam conditions in the neighbourhood of 600lb./sq. in. gauge, 850 deg. F. It consists of two stages of regenerative feed water heating, one of which is of the direct contact type, boiler economizers and steam air-heaters replacing the gas air-heaters of the 1930 and 1945 era. The direct contact heater operates at a constant pressure around 35lb./sq. in. gauge in order to maintain a feed water temperature sufficiently high to minimize economizer corrosion. The cycle has one or two auxiliary condensers depending upon owners' or ship designers' preference.

The use of one auxiliary condenser reduces initial cost and also eliminates the need for a second set of auxiliary circulating and condensate pumps, but somewhat complicates the piping system. If one auxiliary condenser is used, it is not normally in use at sea since additional surface is provided in the main condenser for the turbine generator exhaust.

The use of back pressure turbine generators may also be considered as an alternative to the basic cycle. This type of cycle has the advantage of lower initial cost; but whether or not it proves economically advantageous depends upon the electric load, the amount of time the vessel is operated at lower than normal shaft horsepower, and the frequency and length of time in ports.

Other alternatives include the use of an atmospheric condenser for the cargo pump exhaust, a condensate cooled distillate condenser, a filtering system in lieu of the low pressure steam generator, and various other modifications.

The power plant depicted on Fig. 23 is basically simple, with relatively low initial cost. The components have proved reliable in tanker service which, in turn, has minimized personnel and maintenance requirements.

Reheat cycles have been extensively studied and applied successfully, but have not demonstrated an economic advantage over the non-reheat cycle and have probably not been applied more frequently in the U.S.A. because of the added complication involved.

FUEL RATES

Fig. 24 presents typical expected all purpose fuel rates for present day geared turbine tankers with various steam conditions and for a range of horsepowers. For convenience, the fuel rates for the 1930 and 1945 era tankers are also plotted on Fig. 24.

It should be noted that the fuel rate shown on Fig. 23 represents a trial fuel rate; and, therefore, is lower than the all purpose fuel rate read from Fig. 24 for comparable steam conditions and horsepower.

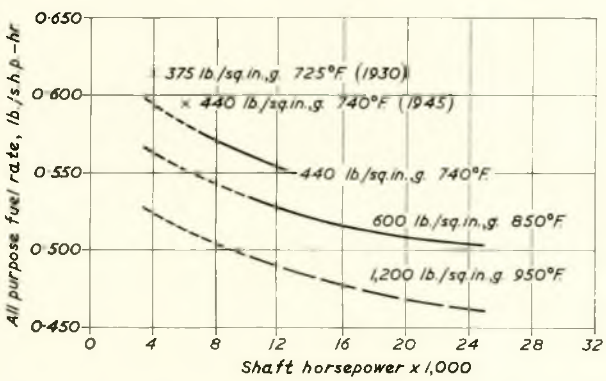


FIG. 24—*Present day typical fuel rates for geared turbine tankers with representative fuel rates for 1930 and 1945 eras*

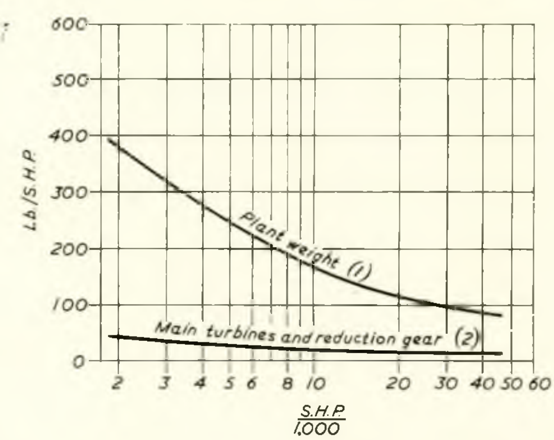
In the preceding discussions, the mechanical advances in design, as well as cycle and steam condition changes have been described, and now Fig. 24 illustrates the gains in fuel rate. It is difficult to make a direct comparison of the 1930, 1945 and 1960 era tankers due to the wide difference in horsepower; however, it can be seen that if the 1945 era tanker was built today with the same horsepower and steam conditions, the fuel rate would decrease from 0·595lb./s.h.p. hr. to 0·580lb./s.h.p. hr. which amounts to approximately 2·5 per cent reduction.

This comparatively small gain in fuel consumption is in reality a tribute to the early designers.

If this same horsepower tanker had steam conditions of the 1960 era, namely 600lb./sq. in. gauge, 850 deg. F. instead of 440lb./sq. in. gauge, 740 deg. F., at the turbine throttle, the expected fuel rate would be 0·552lb./s.h.p. hr. or approximately 7·2 per cent better than the 1945 tanker, or an additional gain of 4·7 per cent. This hypothetical example illustrates the gains in fuel consumption that are possible with advancing steam conditions. Further elevations of steam conditions will be discussed later.

MACHINERY WEIGHTS

This paper would not be complete without a discussion of machinery weights, and in view of the difficulties in comparing weights due to the many variables, the author will not attempt to make a comparison for the three eras chosen, but instead will briefly discuss present day machinery weights. Fig. 25 presents a curve of a complete machinery plant weight[(6)]



Note 1—Plant weight includes auxiliaries, piping, wiring, ventilation, ladders, gratings, etc. In engine room. For single screw geared turbine installations with watertube boilers built to A.B.S. requirements.

Note 2—Main turbine and reduction gear weight includes main thrust bearing and manœuvring valve.

FIG. 25—*Present day average machinery weights*

to which has been added a curve showing separately the weight of the main turbines and reduction gears.

In the opinion of the author, the most significant point to be seen from Fig. 25 is the fact that the weight of the main turbines and reduction gears is only a small fraction of the complete power plant weight. It emphasizes that, if further weight reduction is required in a marine power plant, it should not be expected that significant reductions in the overall plant weight could be made by concentrating on the main turbines and reduction gears only.

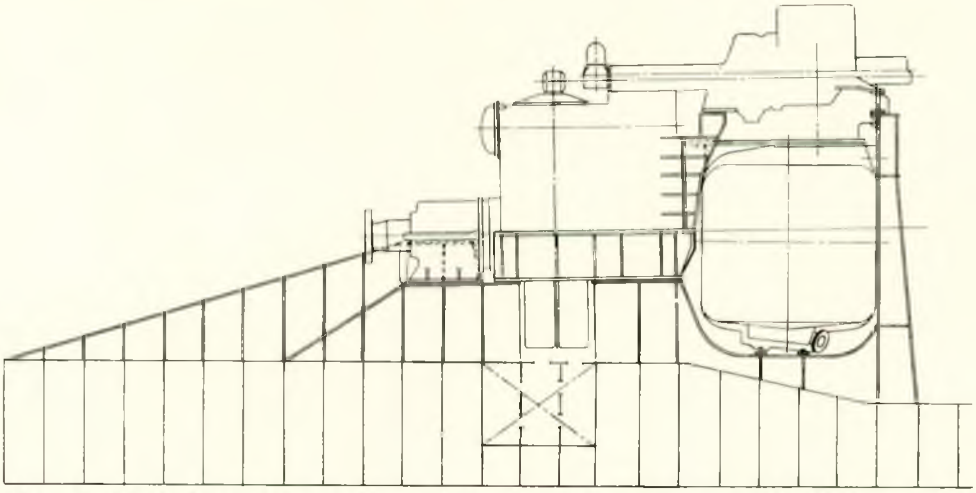


FIG. 26—*Typical propulsion turbine and gear foundation arrangement for 1960 era tanker*

FOUNDATION AND ALIGNMENT

The design of the foundation illustrated in Fig. 26 is of particular importance. The fact that all important weights are carried by a continuation of the individual frames, the liberal bracing of the foundations, both fore and aft and in an athwartship direction, the centre line support of the thrust bearing in a substantial cradle and flexible support of the forward ends of the turbines forming, at the same time, a support from which the condenser hangs free to expand downward, can all be noted from Fig. 26. Other features worthy of note are the location of the oil sump, which should preferably provide a suction for the oil pump as near the centre line of the ship as possible, and the sway chocks to restrain the condenser from athwartship movement without limiting its free vertical movement. Sway braces instead of sway chocks, as indicated, can also be used with equally satisfactory results.

The proper alignment of the reduction gear to the line shafting is another important matter. No line shaft bearing should be provided in a position too close to the main reduction gear wheel bearings, in order to ensure that the main reduction gear wheel remains fully supported by both of its bearings. Line shafting must be aligned in a cold condition with proper corrections for thermal movement under normal operating conditions. Such corrections, generally known as "gap and sag", are calculated and indicated on a shaft alignment drawing, such as Fig. 27.

After the alignment has been accomplished on the ways, it should be again checked when the ship is water-borne by breaking the couplings and by checking the tooth contact. In any event, acceptable tooth contact will be the final proof of proper alignment of the shafting to the reduction gear.

Uniform tooth contact is obtained under shop conditions and recorded on a form similar to that shown in Fig. 28. Upon re-assembly in the ship, various procedures are used to duplicate the alignment of pinions and gears which existed in the manufacturer's shop. Regardless of the measurements and checks made during these procedures, the final criterion is the uniform distribution of load along the teeth of pinions and gears under load. The form shown in Fig. 28 is used in recording tooth contact during shipboard installation and is also useful for recording tooth contact information throughout the life of the ship.

The problem of installation and alignment of reduction gears was considered sufficiently important in the U.S.A. that The Society of Naval Architects and Marine Engineers formed a panel consisting of representatives from the shipbuilding and

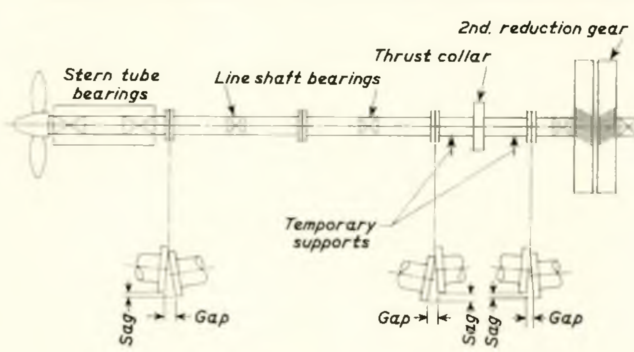


FIG. 27—*Gap and sag diagram for line shafting*

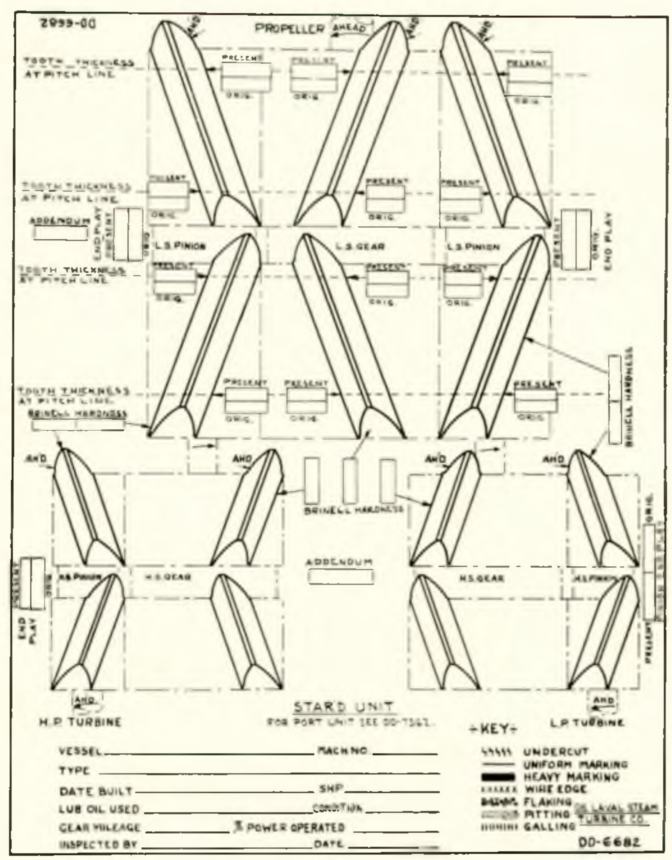


FIG. 28—*Geared tooth contact chart*

gear industries, including representatives from the author's firm, to develop and publish a set of recommended procedures.(7)

LUBRICATION

The lubrication of turbines and gears, as well as the auxiliaries, is another extremely important factor in the satisfactory operation of the ship. Recommended practices for marine propulsion turbine lubricating systems were prepared

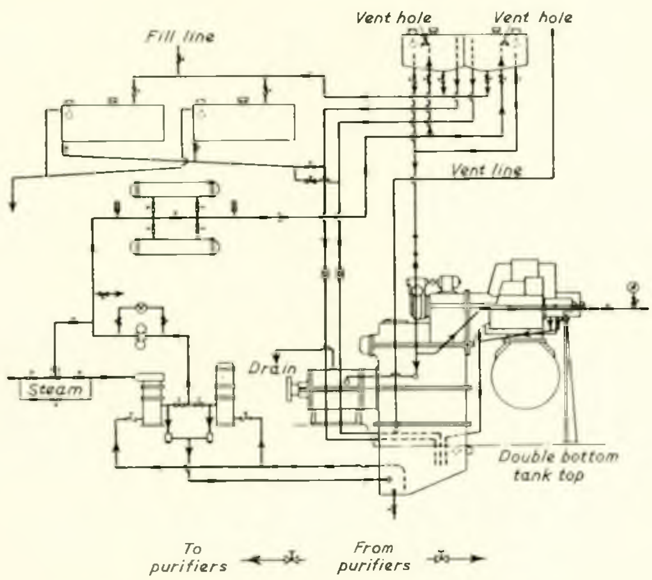


FIG. 29—*Gravity type lubricating oil system*

by a joint committee of the American Society for Testing Materials, the National Electrical Manufacturers Association and the American Society of Mechanical Engineers, on which committee the author's company was actively represented,(8) together with most of the major turbine and gear manufacturers and shipyards in the U.S.A.

These recommendations described three groups of systems; namely; the gravity system, the pressure system and the pressure gravity system. Fig. 29 showing the gravity system and Fig. 30 showing the pressure gravity system are self-explanatory and are the most common for tanker application.

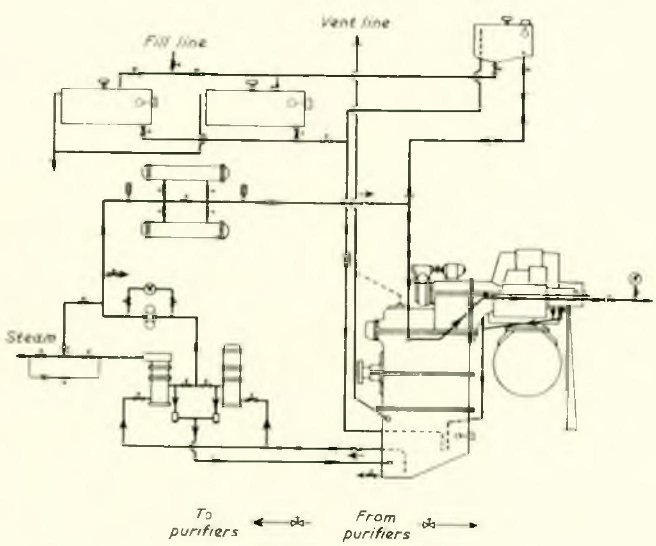


FIG. 30—*High elevation pressure-gravity type lubricating oil system*

FUTURE TRENDS

The last decade has brought with it a considerable increase in the size of the newer tankers. Various types of standard tankers have been developed during this time, ranging from 30,000 d.w.t. to 46,000 d.w.t. to 65,000 d.w.t., and a number of tankers have been built reaching sizes of 100,000 d.w.t. or more.

In view of the available port facilities and other reasons, it is doubtful that tankers much beyond 100,000 d.w.t. will be built in the very near future, in spite of the advantages which a large ship offers to the owner.

The largest power plants for any of these ships were in the approximate range of 20,000-25,000 s.h.p. per shaft, and very few of these vessels have been twin-screw.

It seems that at present, as if the large gear wheel and propeller required for a 25,000 s.h.p. power plant will probably constitute limitations which will not be exceeded, as is proven by the choice of power plants for the most recent tankers built.

If larger powers are required or if a higher propeller speed should be chosen, it is quite likely that consideration will have to be given to a double flow low pressure turbine and a locked train gear; although a single flow low pressure turbine is probably just as advantageous, or more so, in the range of 25,000-30,000 horsepower per shaft.

Assuming then for the immediate future, that the currently popular power range of 19,000 to 25,000 s.h.p. for a large seagoing tanker will remain and that lower horsepower units will be required for the coastwise or shorter haul tankers, the next major development would probably be the introduction of even more elevated steam conditions than are in general practice today.

Steam conditions of 1,200lb./sq. in. gauge and 950-1,000 deg. F. are in operation today. In the comparatively near future, it is anticipated that sufficient experience will have been accumulated to make these conditions suitable for reliable operation without undue maintenance, such as required for tanker operation. The necessary metallurgy will undoubtedly advance to a state that is adequate to deal with these conditions without having to resort to materials which may not be readily commercially available.

In order to illustrate the advancement in metallurgy, reference is made to Appendix A which gives a brief outline of this important subject.

It is realized that economic studies prepared in the last few years do not yet justify the use of steam conditions of 1,200lb./sq. in. gauge, 950 deg. F. This should not be a deterrent to progress, because the same was true when previous advances in steam conditions were proposed. Commercially obtainable design components and hardware, made of the proper materials, will contribute a great deal to the solution of this problem.

The result of the choice of more advanced steam conditions used with a suitable cycle may be seen when referring back to Fig. 24, where a comparison of the expected fuel rates per s.h.p. for all purposes is given.

Advanced Steam Conditions and Cycle

Associated with the advancing of the steam conditions to 1,200lb./sq. in. gauge, 950-1,000 deg. F. one can anticipate some changes from the standard cycle of today. Fig. 31 has been prepared to show a cycle that could be used for the advanced steam conditions with good economy.

This cycle is in reality quite conventional and in use today for certain applications. It consists of four stages of regenerative feed heating, gas economizers and gas air-heaters. Other cycles will probably receive serious consideration and possible application in the ever increasing desire to lower operating costs.

Consideration may be given to driving auxiliaries, such as generators and feed pumps, with steam bled from the main turbine if it will not complicate the operation and reliability of the plant. Consideration may also be given to inducting excess auxiliary exhaust steam into the main turbine.

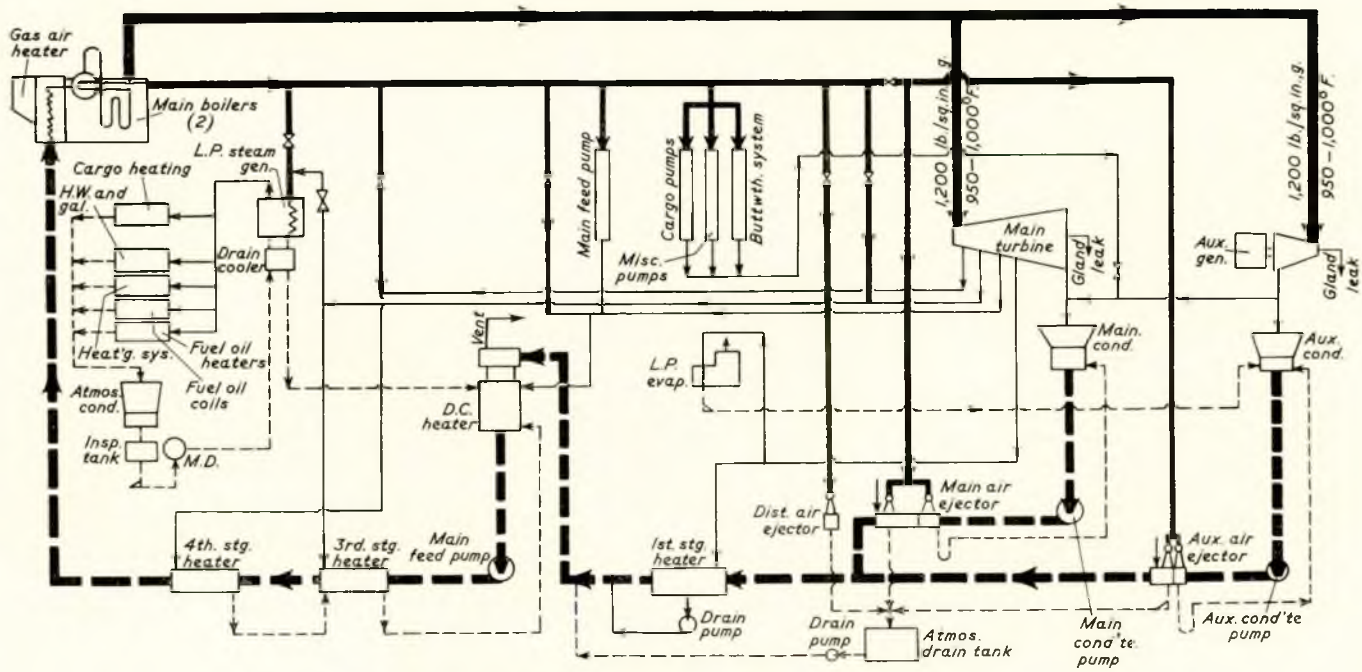


FIG. 31—*Representative 1,200lb./sq. in. gauge 950-1,000 deg. F. cycle flow diagram*

Fig. 24 shows that, even with the conventional cycle shown on Fig. 31, savings in fuel consumption can be obtained by the use of more advanced steam conditions. The following table presents a comparison between the fuel rate of the 600lb./sq. in. gauge, 850 deg. F. standard cycle and a conventional four stage feed heating cycle using 1,200lb./sq. in. gauge, 950 deg. F.

TABLE I—ALL PURPOSE FUEL RATES
lb./s.h.p. hr.
(Data taken from Fig. 24)

s.h.p.	600lb./sq. in. gauge at 850 deg. F	1,200lb./sq. in. gauge at 950 deg. F.	Per cent savings with 1,200lb./sq. in. gauge at 950 deg. F.
10,000	0·533	0·496	6·95
15,000	0·518	0·480	7·34
17,500	0·513	0·473	7·80
20,000	0·508	0·468	7·90
22,000	0·506	0·465	8·10

From the above, it can be seen that there is a considerable fuel rate gain that can be realized by advancing steam conditions.

It should be understood that with advancing metallurgy, 1,200lb./sq. in. gauge, 950-1,000 deg. F. will not represent the ultimate in steam conditions. It is believed that the pressures and temperatures will continue to increase. This is almost inevitable, since the greatest gains in economy can be realized in this area.

The advanced steam conditions will, of course, be accompanied by other design changes along the lines pursued during the recent past in order to take advantage of the available metallurgy to take care of more severe problems of heating and cooling and also to reduce the cost steadily and increase the reliability. As an example, piping arrangements will avoid flanges more than ever and control valves will be simplified.

Fig. 32 shows a bar lift type of nozzle valve operating gear which automatically provides the optimum nozzle setting for any power. All that is provided for the operator is one ahead and one astern hand wheel.

The marine turbine has been refined to a point where gains in internal efficiency are very small, although every turbine manufacturer is actively engaged in attempting to obtain the last fraction of a per cent which may be possible without jeopardizing reliability.

Whether or not more ambitious steam conditions are accepted, there will still be many design features which will be improved in the interest of reduction in cost and improvement in reliability. Among these, is the method of reversing the turbine, which is the subject of frequent discussions today.

Reversing and Manœuvring

This important subject has been widely discussed in recent years and some comments seem in order with respect thereto. At the present time, it is common practice in the U.S.A. to incorporate an astern turbine for reversing purposes in the low pressure turbine, and this has been found entirely adequate to comply with present standard specifications for ocean-going vessels that the astern turbine should develop 50 per cent of the ahead torque at 80 per cent of the ahead r.p.m. with not more than 100 per cent of the ahead normal steam flow. The design of such an astern turbine, which was described elsewhere in this paper, has not presented too many difficulties, and the rate of maintenance or repair, or derangements stemming from the reversing turbine itself, or even from the action of reversing, from the author's experience over many years, is almost negligible. The loss of power in a 20,000 h.p. compound double reduction geared main propulsion unit with modern steam conditions is in the range of approximately 120 h.p.

Proposals have been made to incorporate the astern turbine in the high pressure turbine. In the author's opinion this would offer no real advantage and would result in a seal between the ahead turbine and the astern turbine that would be inaccessible. Considering the fact that no difficulties have been experienced with the astern turbine in its traditional location in the low pressure turbine, it does not seem prudent to make a change without an accompanying improvement.

The arrangement of the astern turbine in a separate casing has been practiced for many years, particularly in Europe, but seems to require greater length of the turbine, and certainly higher cost, because a separate casing has to be provided.

Parenthetically, in case a double flow low pressure turbine is used for higher horsepowers, or other reasons, American practice has always provided an astern turbine on both ends of the low pressure turbine in order to avoid unsymmetrical thermal conditions.

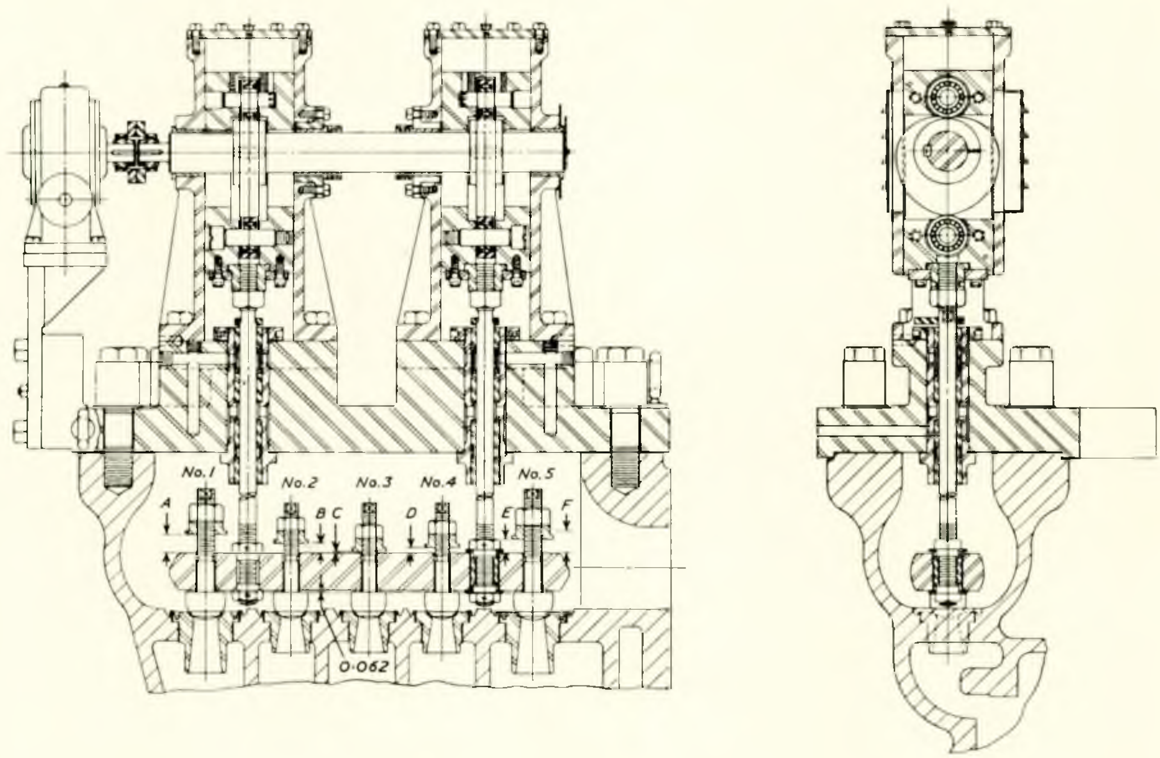


FIG. 32—*Nozzle valve operating gear*

Frequently in recent years, means have been discussed to eliminate the astern turbine and accomplish the necessary manœuvring by other means.

The reason for these proposals is not entirely obvious as far as a steam turbine is concerned, although the introduction of other reversing means seems imperative in a gas turbine power plant.

At the present state of the art and upon factual evaluation, the astern steam turbine cannot be considered either hazardous or unreliable, and its losses, when operating the unit ahead are insignificant and its capability to reverse the ship adequately has been proven. The availability of greater astern power than is usual today may be advantageous in special vessels, but does not seem to be mandatory in the average tanker today.

The only advantages in eliminating the astern turbine lie in the possibility of designing an ahead turbine by utilizing the decreased centre distances between bearings for a slight improvement of the critical speed condition or, if this is satisfactory, to reduce the shaft diameter to obtain a very minor improvement in efficiency.

Nevertheless, there are several promising approaches for other means of reversing on which distinguished and experienced engineers are working today. One of these schemes employs a hydraulic coupling and the necessary controls; the other employs an ingenious synchronizing clutch; while other approaches include the use of epicyclic gears which effect the reversing by use of a brake.

In addition to the various reverse gears, there have been an increasing number of applications of a reversible pitch propeller, which is gaining in reliability and acceptance.

Reduction Gears

It has long since been found to be obvious that the inherent characteristics of the turbine favour a high rotative speed, while the propeller to drive the ship has an inherent characteristic which demands that it rotate at a comparatively low speed for peak performance. In the beginning, the decision and construction of reduction gears for large powers seemed a difficult undertaking and fluid transmissions were used in several instances. Very soon, however, the state of the art had advanced far enough so that large reduction gears, reducing the speed of the turbine to that required by the propeller, could be built very satisfactorily and these techniques are still progressing.

Techniques to produce accurate tooth form and spacing, and advances in metallurgy have made it possible to increase considerably the specific loading on gear teeth over the years. While in the 1930's, K factors of between 45 and 55 for the second reduction gear were common and, during World War II, K factors of between 65-75 were prevalent, in American practice today K factors of between 80-90 are regarded as perfectly satisfactory. Gear loadings in first reduction elements have always been correspondingly higher with good results.

As can be seen from Fig. 33, a further increase in gear tooth loadings will have less and less effect in the reduction of overall gear weight, simply because the transverse girders of the gear case are not decreased in relation to the reduction in width of face of the rotating elements, and the gear case dimensions are generally dictated by the turbine arrangement. Nevertheless, further progress in gear finishing techniques and metallurgy is still being made and will probably permit still higher loadings on hobbed and shaved gears.

Better manufacturing techniques and the ability to handle harder materials will undoubtedly contribute further to decrease

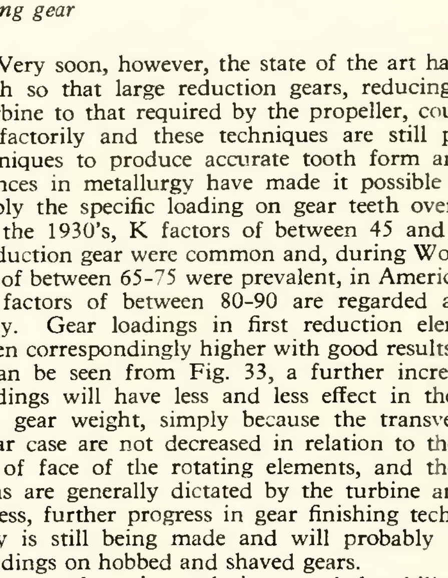


FIG. 33—*Influence of K factor on gear weight*

the weight and cost of gears. Another very interesting development is that of reliable epicyclic gears, a large number of which have already been built in smaller sizes than required in some of the large tankers. The introduction of this type of gear may be able to bring with it other modifications, mainly because of its small size and the co-axial arrangement which it makes possible in conjunction with a single cylinder turbine.

Auxiliaries

The field of power plant auxiliaries presents a large area in which further improvements will undoubtedly be made in space requirements, reliability, efficiency and cost.

The possibilities of supplying steam driven auxiliaries with bled steam from the main unit and the possibilities of exhausting back into the main turbines have already been previously mentioned.

In the early 1930's some distinguished designers made various efforts to improve the efficiency of the plant by attaching as many auxiliaries as possible to the main unit in order to take advantage of its better steam rate. Most of these efforts have been discarded owing to the complexities introduced when manœuvring. The present trend seems to be a simplification of all engine room auxiliaries, to increase their reliability and reduce the necessary maintenance as much as possible.

THE AUSTERITY SHIP

A recent study has shown that the most important items of expense in the operation of a tanker are first cost or capital charges, crew, maintenance, and fuel. It is obvious, then, that future trends in power plant design will have to concentrate in these areas in order to obtain the most economical method of transportation.

Many of the developments described as having taken place over the last 30 years have, of course, gone in this direction and many developments contributing to decreased first cost and decreased fuel consumption have been discussed.

As the economic climate in the transportation industry becomes more competitive, greater austerity in the design of the power plant and in each piece of equipment connected therewith will certainly be carried out, possibly even at a sacrifice in fuel consumption. It is visualized that such a power plant will avoid the use of any unnecessary standby equipment, bringing with it maximum requirements for reliable machinery. Any refinements which will complicate the operation or the manœuvring of the plant will be avoided at all costs.

If reliability, ease of installation, maintenance, and repair are made the primary design concepts, machinery can be developed to accomplish these aims. Unit replacement, for instance, may be possible in much less time than that required for present day shipboard repairs.

It can readily be visualized that by the elimination of duplicate equipment the entire machinery plant will be much simpler with gains, not only in first cost, but also in maintenance and attendance, particularly if consideration is given to the fact that, with the elimination of standby equipment, a significant amount of valves, piping, cables, controls, etc., can be eliminated.

After this point is reached, and some operational experience gained, it should not be too difficult to provide a simpler machinery plant which will also readily lend itself to the additional gains of automation.

THE AUTOMATED SHIP

The subject of automation is receiving a considerable amount of attention throughout the world at the present time in order to further reduce operating costs of ships. The author will not discuss this in detail because there are many papers available on the subject. However, he would be remiss not to express some brief comments.

Certain fundamentals of automation, of course, apply to the ship in general. In order to evaluate the economic effects of automation, it should be borne in mind that the ship can actually be operated effectively by two men on the bridge and two or three in the engine room or for three watches a total of 15 men out of a total crew of probably fifty or sixty. Consequently, the saving in crew through automation of the power plant alone will not produce as much economy as is generally assumed unless other functions in operating and maintaining the ship are also reduced.

Today we have several examples of successful automation in certain areas such as turbine generator sets, voltage regulators, and boiler combustion controls. The next step probably will be to have the main propulsion turbines uprated, first by much simpler controls in the engine room, from where it will be only a short step to operate the propulsion unit from the bridge.

The degree of automation will determine its complexity, hence, the first question to be decided is what to automate. If automation produces tangible savings in cost, then it should be considered but it should not be carried out for the sake of automation alone.

It is believed that the problem will be aproached gradually and that various systems will be automated independently rather than attempting to completely automate an entire ship. Start-up and shut-down of some systems will probably remain semi-automatic. Ultimately, we might see computers controlling the complete propulsion plant and the operating personnel would really be acting in a surveillance capacity only.

The basic functions to be accomplished by automation or the operators are the following:

1) Sensing (r.p.m., power, flow, pressure, temperature).
2) Measuring the above.
3) Recording the data measured.
4) Controlling the operation of the equipment.
5) Maintenance and repair of the equipment.

All of these are, at present, accomplished by the operating personnel with the aid of instruments. In the future, the operators will perform less and less of the above functions but will act in a supervisory capacity and, of course, will have to be responsible for seeing that the equipment is properly maintained and repaired.

The ability to automate the machinery is available today and the need to do so is pressing but it will take some pioneering company or agency to provide the necessary funds in order to demonstrate conclusively the advantages of automation.

The author's company is at present engaged in a preliminary design study of an integrated steam turbine propulsion plant in the range of 20,000 s.h.p. which forms part of the research programme of the U.S. Maritime Administration.

THE NUCLEAR SHIP

All previous discussions have referred to conventional fossil fuel types of power plants. Many investigations are now going on all over the world to determine the economics of a nuclear plant. It is not the intent of this paper to discuss this subject except insofar as it might affect the further development of turbine machinery. The technical feasibility of the use of nuclear reactors in the propulsion of ships has been amply proven to date by the excellent results obtained by the U.S. Navy and in many land-based power plants.

The economic justification for the nuclear tanker is probably not fully determined mainly because the cost of fuel is not definitely determined and because the first cost of such an installation is still much higher than that of a conventional power plant. The nuclear tanker will not become popular until the first cost of a nuclear plant is comparable to the first cost of a present day conventional power plant, since there is not much difference in the cost of fuel between the fossil fuel power plant and the nuclear power plant. Neither the effect of saving port time nor any other advantages of the nuclear plant will significantly change this comparison, particularly on short hauls where the additional carrying capacity of the nuclear vessel, due to absence of bunkers, may not be great enough to tip the balance in its favour.

In this connexion, it may be of interest to describe briefly the turbine, gears, and auxiliary machinery of the n.s. *Savannah,*

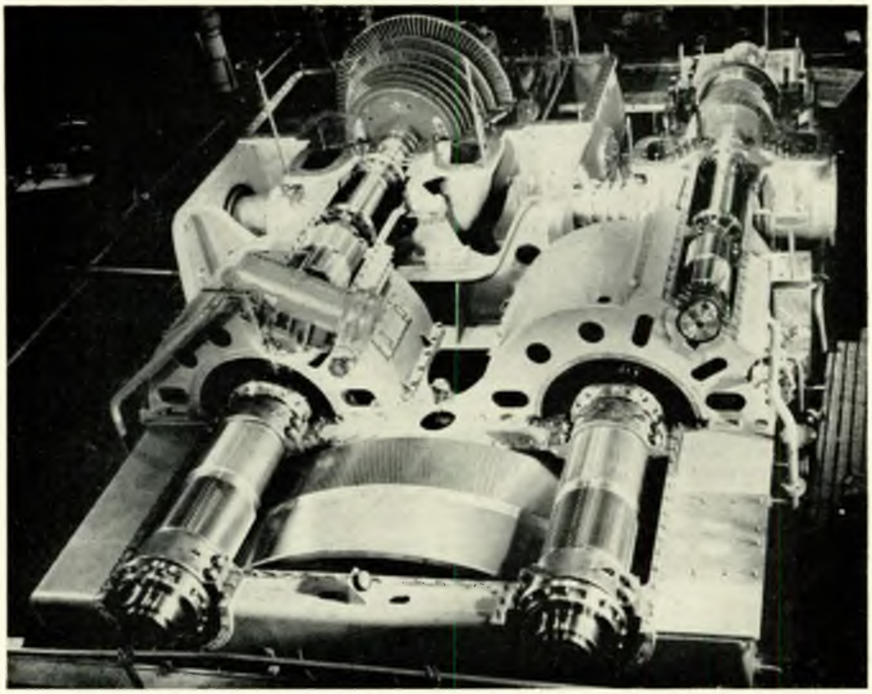


FIG. 34—*Propulsion turbines and gears, n.s.* Savannah

the first nuclear merchant ship, which were designed and manufactured by the company represented by the author.

The main propulsion unit is essentially the same type as used in a conventional steam turbine driven ship today and is shown in Fig. 34. However, the design of the n.s. *Savannah* equipment presented some unique problems. First of all, the main turbines which develop 22,000 s.h.p. had to accomplish this with saturated steam ranging in pressures from approximately 430lb./sq. in. abs. to over 700lb./sq. in. abs.

In a conventional power plant, the low pressure turbine is generally designed so that the steam leaving the last stage contains about 10 to 12 per cent moisture. In the case of the n.s. *Savannah,* the steam enters the high pressure turbine in a saturated condition and as it expands in the turbine the amount of moisture increases rapidly. It was, therefore, necessary to remove this moisture from the turbine in order to prevent excessive erosion of the turbine blades. Greater attention, therefore, had to be devoted to methods of moisture extraction, a problem which was dealt with in the days of saturated steam, 40 years ago, except that today it has to be carried out in turbines with much higher tip speed.

In the design of the main reduction gear, provision had to be made for a 750 h.p. electric motor to be coupled to the gear by means of a special clutch and a simple lever. The purpose of this motor which would be driven either from the ship's service turbine generator or a Diesel generator, would be to provide the ship with emergency power in case of reactor failure.

The electric power for n.s. *Savannah* is supplied by two 1,500 kW. turbine generators, shown in Fig. 35. These sets incorporate De Laval-Stoeckicht planetary gears to reduce the speed of the turbine to that required by the generator which assists greatly to meet the stringent space requirements.

In the vessel virtually all components of the power plant

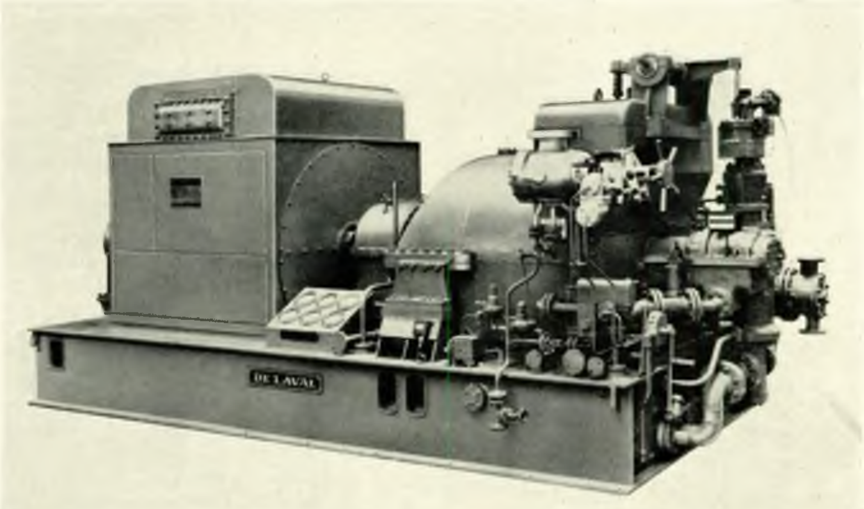


FIG. 35—*1,500 kW ships service turbine generator set, n.s.* Savannah

Period
Nominal steam conditions
H.P. turbine rotor
H.P. turbine blades (buckets)
H.P. turbine case
H.P. turbine diaphragms
L.P. turbine rotor
L.P. turbine blades (buckets)
L.P. turbine case
Astern turbine steam chest
L.P. turbine diaphragms
H.P. and L.P. turbine outside packing
H.P. turbine interstage labyrinth
L.P. turbine interstage labyrinth
H.P. and L.P. turbine bearings

Period
K factor (second reduction)
Gear case
First and second reduction pinions
First and second reduction gear bands
First and second reduction gear body

can be operated from a single remote control centre. For example, the main turbines can be operated throughout the power range simply by moving a lever and through various mechanical and electrical devices, more or less steam will be admitted to the turbines.

CONCLUSION

It is hoped this paper may give an overall view of the progress made by industry as a whole in the development of turbine tanker machinery over the last 30 years and it describes the main features of an up-to-date United States tanker power plant.

The main purpose of this paper, however, is to highlight the reliability and flexibility of the steam turbine as a prime mover providing, as it does, the most essential elements for tanker operation.

The many various combinations in which a steam turbine can be used and the inherent characteristics of the geared steam turbine not only offer many advantages to the owner at the present time, but also give promise of providing even better and more economical power plants which undoubtedly will keep the transporatation of bulk cargoes by tanker competitive with other means of transportation.

The author is convinced that further improvements in turbine tanker machinery will make it possible for this type of power plant to retain its leading position which it certainly has occupied, at least in the U.S.A. for the last three decades, even in the face of the most intensive scrutiny by the experienced owner and financier

ACKNOWLEDGEMENT

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The statements and opinions made in this paper represent the individual expressions of the author, and while the information is believed to be accurate, any application of the contents must be at the sole discretion and responsibility of the user.

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APPENDIX A

Table II illustrates the metallurgical advancements over the past three decades that have made possible the advances in steam conditions, gear tooth loading and other factors such as size, reliability and cost of manufacture. Using the experience now being obtained on naval propulsion units it is possible to forecast the metallurgical requirements shown for the future tanker using steam at 1,200lb./sq. in. gauge, 950-1,000 deg. F.

It may be of interest to discuss a few of these developments very briefly: Turbine rotors were formerly assemblies consisting of a carbon steel shaft on which were fitted alloy steel turbine wheels, usually of 3·50 per cent nickel steel with a tensile strength of approximately 120,000lb./sq. in. Modern turbine rotors are frequently the so called solid-forged type where the wheels and shaft are forged integrally and then the individual wheels machined out. The majority of the rotors today are made of nickel/molybdenum steel, except in the higher temperature ranges where chromium/molybdenum/vanadium alloy steel is used. In the future ship it is anticipated that all rotors will be solid forgings and the high pressure rotor will be a chromium/molybdenum/vanadium alloy steel forging.

Practically all steel turbine rotor forgings today are made of vacuum treated steel. This treatment lowers the dissolved hydrogen level to a very low order and is considered almost a guarantee against the formation of thermal flakes as the forging cools from the press.

The heat stabilization test that is applied to forged rotors today has contributed to making these rotors dependable and stable even under abusive service. Only with a turbine rotor that is thermally stable is operation practical throughout the range of speed and temperature conditions experienced in propulsion units, without undue vibration.

Originally, turbine blades were made from 3·50 per cent nickel steel, and later on from 5 per cent nickel steel, and finally from monel, a natural nickel/copper alloy. Eventually, the 12 per cent chromium stainless steel was developed in Great Britain and was rather quickly accepted as a fine turbine blade material. It has satisfactory corrosion resistance, good dampening characteristics, a high degree of ductility and renders commendable service in most steam conditions.

In some of the higher duty turbines, with more advanced steam conditions, a substantial amount of Type 422 steel will be used. This is a modification of the 12 per cent chromium containing slightly more carbon plus 1 per cent molybdenum, 1 per cent tungsten and 0·4 per cent vanadium. This material has a high yield strength cold and has better hot strength than the standard 12 per cent chromium. There are also modifications of the 12 per cent chromium with a little higher carbon, with additions of either tungsten or molybdenum to increase the hot strength.

In the low pressure turbine the leading edges of the last row or last few rows of blades are protected against water erosion with precision cast stellite (cobalt/chromium alloy) silver brazed to the outer third of the leading edge whenever the entrained moisture build-up exceeds 10 per cent and blade velocities are in excess of 90ft./sec.

The turbine diaphragms have progressed from the composite construction shown for 1930 with blades of nickel/bronze, 80 per cent copper, 10 per cent nickel, 3 per cent tin, 7 per cent zinc, to the cast construction and then to the all welded construction introduced in the late 1940's.

Cast iron is no longer used for turbine case material due to its poor resistance to shock and the fact that above 650 deg. F., it undergoes a slow graphitization and oxidation resulting in growth and loss of dimensions, and cracking. The availability of high quality alloy steel castings as well as the development of welding techniques with a complete stress relief after fabrication has resulted in more dependable, as well as lighter, turbine cases particularly in the L.P. case.

The development from cast iron gear cases to the low carbon steel weldment has naturally followed the advancements in welding techniques and the availability of fabrications. This, like the turbine case development, was accelerated by wartime experience with severe shock but due to limited facilities for manufacture of large weldments some cast iron cases were made through the war years.

Pinion materials were at first annealed steel—3·5 per cent nickel and 0·4 per cent carbon at a hardness of about 170 Brinell. In these earlier days it was considered quite dangerous to quench heavier sections of steel. The most drastic treatments were normalizing and tempering. This limited, to some extent, the hardness and tensile properties. Later, pinions were still normalized and tempered, but in addition to nickel, molybdenum was added and hardnesses were raised to slightly over 200 Brinell. In the post-war years, it was felt that by hollow boring the pinions they could be safely quenched and it has been possible to obtain higher hardness levels and tensile strengths.

The same in general was true as far as gear bands were concerned. These were originally annealed 0·25 per cent carbon steel and, therefore, hardened at approximately 135 Brinell hardness number. With the advent of welding and weld fabricated gears, there was a tendency to hold down the carbon steel to 0·35 per cent maximum. Water quenching and tempering were later used to raise the hardness levels to approximately 170 Brinell hardness number. Improvements in the art of welding now make it possible to go to a higher carbon and introduce alloys into the bands, thereby increasing gear band hardnesses even further.

Many new non-destructive tests and methods of inspection have come into usage in the past 25 years.

The magnetic particle inspection of today is done under white light using coloured magnetic powders. It is also done under ultra-violet light with fluorescent particles suspended in a liquid. This latter is a very sensitive test and is applied to all turbine blades. The powder method is more often used on heavy castings and also in the inspection of weldments.

The penetrant methods are really successors to the old oil and whiting method and there are a number of adaptations to these. The so called dye penetrants use a red dye in a vehicle, having very highly penetrating characteristics. The fluorescent penetrant is a similar test but a fluorescent dye is used and observed under ultra-violet light when being developed. The penetrant methods may be used on forgings, castings and weldments but must be used on the non-magnetic materials where the magnetic particle inspection cannot be used.

Radiography has come into rather wide use as an inspection method for castings and the development of casting techniques and also as a tool for inspection of weldments. There are a number of modifications of radiography, using different source energy such as X-ray machines, Betatrons, radiographic isotopes, such as Cobalt 60 and Iridium 192. Their usefulness varies with the thickness of penetration required, the Iridium 192 for very thin plates and castings, the Betatrons and Cobalt 60 for very heavy, thick castings. Radiographic quality castings are required for the higher temperatures and pressures.

Considerable interest has developed in recent years on determination of notch brittleness transition temperatures using V-notch Charpy tests for determination of these transition temperatures. The transition temperature is the temperature below which the material exhibits notch brittleness characteristics. This is becoming standard practice on turbine rotor materials. It has resulted in raising the nickel content and even quenching some low pressure turbine rotors.

In the future many of these same tests and inspections will be more widely used and further developed as the higher temperatures and pressures require more effective material inspection.

Generally, there has been little inclination at present to elevate temperatures above 1,050 deg. F. This appears to be a limiting temperature for the ferritic type steels, which are those materials that transform from face centred cubic lattice to body centred cubic lattice as they cool to room temperature. Alloy steels of this type, properly treated, have good properties but at higher temperatures more tungsten, more molybdenum and more vanadium are required to give them better high temperature properties. Above 1,050 deg. F., it is generally felt necessary to go to the austenitic type of materials, the most common and familiar being the 18 per cent chromium, 8 per cent nickel stainless steel. These materials have some undesirable characteristics, namely expansion coefficients considerably higher than the ferritic types which makes them somewhat crack sensitive to sharp temperature gradients; they also have poor coefficients of thermal conductivity which have almost the same effect as sharp thermal gradients. Most of these materials have poor journal characteristics and tend to seize and pick up readily; however, this can be avoided by sleeving, overlaying, plating and other methods. Austenitic materials, as a class, tend to have relatively low yield strengths (cold) although their hot strength and creep resistance are better than the ferritic types.

Some of these austenitic materials have a tendency towards precipitating chromium carbides from the grain boundary of the austenitic grains when held for long periods in temperature ranges between 800 and 1,400 deg. F. This produces brittleness and loss of corrosion resistance and must be compensated for by addition of strong carbide formers, such as columbium, tantalum and titanium. With the development of some of the strong, satisfactory high temperature alloys for gas turbine operation it does seem probable that suitable materials for higher temperatures in steam turbines will be available.

Discussion

MR. H. ARMSTRONG (Member) said it was his pleasure to open the discussion on the two papers which had just been presented, but first of all he would like to thank and to congratulate the authors on these interesting presentations.

Referring to the paper by Commander Platt and Mr. Strachan, the first portion of the paper was an historical review of the evolution of a large tanker fleet, which was extremely interesting and instructive to say the least. It was particularly interesting to him since it gave him the opportunity of comparing his company's advance through the various stages. Considering that the authors and Mr. Armstrong's company's experience ran in parallel and without direct liaison, it was remarkable how similar their problems had been. As might be expected, those who were associated with the various oil and tanker-owning companies, did at times exchange views and ideas for developing the tanker as an efficient and economic carrier, and while individual company requirements very often dictated differences in design, many of their problems were common. Reference was made on page 389 of the paper to "The Fast Tanker Design" and Mr. Armstrong well remembered the lengthy discussions and drafting of specifications for this project when he was, with the then Chief Engineer Superintendent of Commander Platt's present company, a fellow member of the Committee and Study Group. He was pleased to know that the two vessels built had given good results. How very true must be the statement near the bottom of page 400 "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". All too seldom did ship and engine builders have such opportunities to learn of the many fundamentals in ship operation which might influence design and performance.

The authors' remarks on the recent rate of progress in the design of the steam turbine for main propulsion, as compared with the 30 years of almost standstill from the days of Sir Charles Parsons, complimented the designers of early days as well as the present decade. It was, however, a pity that the science of "fitting out" should have advanced so little since the early days.

Whilst on page 396 of the paper the authors gave a list of possible reasons for turbine thrust failures, he could not help but suspect that a good proportion of those failures had been due to item (g), i.e. dirt in the lubricating oil. A recent series of tests on a typical thrust was reported to have allowed loading up to twice design loading without any sign of distress. Even when mal-alignment was simulated, failure did not occur. However, immediately a small quantity of foreign matter was added to the oil, failure took place. The difficulty of getting a lubricating oil system clean was only exceeded by that of getting the ship or engine builder sufficiently interested to try to achieve this necessary condition.

L.P. casing distortion resulting in blade rubs had been experienced in their fleet, but not to the extent indicated by the authors. The rubs his company had had, had taken place with single flow machines so that, unfortunately, the authors' assumption that a change over from double to single flow did not necessarily mean automatic release from that problem. On encountering the problem his company had instructed the operating staff that under no circumstances was excess exhaust steam to be dumped to the main condenser if the engines were stopped or on light load. The excess was dumped to the auxiliary condenser under those circumstances. Further precautions taken were to remove the lagging on the upper half of the outer casing and fit the dump internal pipe as low down in the condenser as possible. Since then no further rubs had been reported. Drawings had been seen where a radiation shield was placed between the bottom of the L.P. inner casing and the condenser tubes to maintain a more even temperature of the inner cylinder. Perhaps the authors would care to give their opinion of this approach.

The authors did not mention casing distortion relative to the H.P. cylinder and this made Mr. Armstrong wonder if this indicated a freedom from this problem. His company's experience of double cased H.P. turbines, where a fabricated envelope was used was one of unpredictable casing distortion, which resulted in some fairly heavy turbine rubs. The smaller frame sizes were more prone to this defect. The fact that bent rotors did not ensue spoke highly of the spring-backed glands.

Boilers were undoubtedly one, if not the major item, affecting the performance and economics of steam vessels. Mr. Armstrong agreed with the authors on the desirability of chemical cleaning of the boiler prior to commissioning. The calculated weight of debris removed during this process was measured in hundredweights. His company had carried out this practice in a fair number of their ships, now extending it to the main steam and condensate systems also. One of the main reasons for adopting this cleaning process was the dreadful mutilation of the H.P. nozzles and first stages of the turbine blading discovered when a turbine was opened up after a few hours run on preliminary trials. The opening up of subsequent machines after chemical cleaning had been applied had more than convinced them of the good sense of this procedure.

Once again he would make the point that no matter how much the designers strove for perfection much of their efforts would be wasted unless the builders were prepared to improve, on their own initiative, pre-commissioning procedures.

The selection of turbo-self-condensing sets against back pressure sets would not, in his opinion, be justified on the figures shown in Table III. The authors were extremely sensitive with regard to condenser design and appeared to be very much aware of the possibility of contamination of the feed system, but were quite happy to accept the fitting of a further condenser into the system for a theoretical gain of 0·001lb./s.h.p. hr. and a saving of £450 per year. Even then, to show this saving, it was necessary to accept one low cost, low efficiency set. The use of this low efficiency set for standby or even a few running hours whilst carrying out minor repairs to the main alternator set would rapidly evaporate this small saving.

Commenting on the selection of the heat balance, Mr. Armstrong would agree that this must be done on a realistic basis. The very figures used in the calculation must be carefully screened in the first instance for possible manufacturers' margins, etc. If equal allowances were made in the various heat balance calculations, the differential should not be too much affected. A recently completed class of ships of 22,000 s.h.p. which were now in his company's service, approximated to the heat balance shown in Fig. 16. The calculated fuel rate was 0·523lb./s.h.p. hr. whilst on trials 0·51lb./s.h.p. hr. was achieved and sea service records continued to show, so far as instrumentation provided, that this figure was being maintained.

It would be useful to know whether Table I could be read with Figs. 16 and 17. Could proposal A in Table I be looked upon as Fig. 16 and proposal C as the chosen cycle illustrated in Fig. 17?

Using the figures in Table III the impression was obtained that the cost of proposal C allowed for the fitting of one high efficiency and one low efficiency condensing turbo-generator, while proposal A allowed for two high efficiency back pressure generator sets.

There again was it not so that a case was made out in Table II that the single casing H.P. turbine cost £4,000 less than the double casing H.P. turbine and that proposal A in Table I was unfairly penalized by that amount, since the back pressure generator cycle could equally well use the single casing H.P. propulsion turbine?

It seemed to him important that those points be clarified since a difference of interpretation of the tables could easily convert an estimated overall saving into a loss.

The authors had chosen to ascribe the term "sophisticated" to their split feed, three bleed-point cycle.

Mr. Armstrong hoped they would not feel too roughly used if he observed that most dictionaries define "sophisticated" as "corrupted", "not genuine" or "rendered unsound".

The authors' reference on page 402 to the fact that they confidently expected that the high efficiency turbo-alternator would benefit rather than lose from continuous usage, prompted him to think that they might be interested to know that in the case of one of his company's steam turbine-propelled ships now nearing completion, they had only one steam turbo-alternator of 425 kW, which would be in use under all normal conditions. They also had on this ship a Diesel-driven alternator of 400 kW capable of maintaining the ship under full power conditions at sea. This Diesel set also did duty as an auxiliary set under shut-down conditions and they had been able to eliminate the usual auxiliary Diesel set of about 100/150 kW. This was one of the steps in the direction of reducing the number of auxiliary units in the engine room.

Mr. Bauer on page 433 of his paper referred to the avoidance of the use of unnecessary standby equipment bringing with it maximum requirements for reliable machinery. Maybe one day ships would have only one main alternator set, but he did not think that this standard of reliability had yet been achieved.

Mr. Armstrong agreed with many of the points put forward in the section Future Trends of paper No. 3. There were, of course, one or two points which intrigued him. One was the philosophy behind the hot water circulated evaporators. Since sensible heat only was being used, the weight of water which had to be circulated must consume a reasonable amount of power to achieve presumably a modulated boil within the evaporator. If people in the evaporating world were to be believed, modern separators currently being produced were quite capable of preventing carry over.

Packaged units could indeed effect a saving in installation, but there some caution was required. By packaged unit, was a standard unit of some manufacturer implied, which might eventually mean accepting something of less than optimum design for several components, resulting in a net loss of efficiency. It had been stated that by making a package of lots of items, the untidy pipework of the builder would be reduced. One of the prices which might have to be paid for that would be poor utilization of the available machinery space, resulting in wide open spaces around a monument, which was in reality a tight complex mass of machinery, not fully understood by the operator, and, furthermore, difficult to maintain. Would not the vetting of pipework drawings clear up the untidiness more readily and still leave open the choice of individual items?

With regard to Mr. Bauer's paper, this was again mainly an historic review of turbine and ancillary equipment advance which made interesting and informative reading. Particularly so since it was an account of the progress on the other side of the Atlantic.

There were naturally many points of interest in this paper on which he would like to comment:

1) It was noted that each model shown in the paper had the nozzle control valves and steam inlet situated on the upper cover. This arrangement must entail more work when opening up for inspection. It was admitted that this should not be a frequent chore, but it did add to the labour costs. No doubt the manufacturers had a perfectly good reason and it would be interesting to have some further information.
2) It was noted that in the 1960 era turbine the author had considered it necessary to fit a tilting pad thrust to the primary wheels. This was a more elaborate arrangement than utilized in the 1945 series. Was there any history attached to this? Some authorities in this country felt that a solid coupling of quill shaft to secondary pinion, thus eliminating the thrust entirely, was more satisfactory. Perhaps the author would be kind enough to give his views on this point.
3) It was noted that separate thrust collars were preferred in the turbine design. Would the author care to give his reasons for this against an integral collar, and perhaps comment on his experience with thrust failures and design.
4) The fitting of the emergency shut-off valve in the ahead steam line only, referred to on page 423 was a practice Mr. Armstrong's company had followed for some time. The combining of the manœuvring valve and the emergency stop, whilst making for a tidy arrangement, must deny one the possibility of checking the operation of the safety feature with steam on the valve. If this was taken care of, would the author indicate how this was done?
5) The feature of fitting the main thrust without journal bearing as shown on Fig. 16, was new to him and was very interesting.
   The fact that the alignment of shafting and gears was considered sufficiently important for a panel to be formed to study this feature was interesting and he would agree with this approach, which was referred to on page 429. It would seem from the paper that the initial alignment was carried out with the vessel still on the berth and was checked after launching. He wondered how far the alignment varied after launching. Could the author give some figure for this?
6) Lubrication and lubrication systems were of extreme importance and he would be interested to know if any flushing procedure was laid down to achieve cleanliness and what class of filtration the author considered necessary.
7) The author commented on the possibility of using bled steam for auxiliaries in future machinery. On one class of vessel in his company's fleet this very thing was done with a fair amount of success. This was by bleeding at cross-over pressure to a mixed pressure turbo-alternator with attached feed pump. These machines ran extremely well. A live steam nozzle connexion was of course fitted and they had tested the governor gear to the extent of closing the main engine throttle. The live steam took over without making even 1 c/s dip.
8) The possibility of leading back into the L.P. turbine, excess exhaust steam, was interesting and should not

prove difficult providing the necessary safeguards were were taken to ensure that steam admission was not possible when the engines were at rest. This would solve the main objection to the use of back pressure sets and the dumping of excess exhaust steam, as commented upon in the paper by Commander Platt and Mr. Strachan.

Dr. T. W. F. Brown, C.B.E., S.M. (Member) said that this was a most useful combined paper which was unusual in that it described the various designs emanating from Pametrada from 1948 to the present day, and gave extremely valuable results of experience with the turbine machinery in service. He was exceedingly pleased to know that the fast tanker design had been so successful in service. The feed system associated with these two-speed ships had been successfully applied to passenger liners operating both in the cruising and normal service conditions.

He felt that in the reference to the British ships and Italian ships, it should be made clear that the date of the British design was 1949 and that of the Italian ships 1957. It might be that the turbine designs were approximately contemporary, but it was certainly true that the feed systems, margins in auxiliaries, etc., which were allowed in 1949 were greatly improved in 1957 whatever the origin of the turbine design might be. The improvement in feed cycles was greatly assisted by Commander Platt allowing one of Dr. Brown's colleagues to make an investigation of the operation of the Company's ships which enabled an economic assessment of the appropriate feed cycle to be worked out on a scientific basis. The results of this appraisal were reported by Commander Bonny to the Institute of Marine Engineers (reference 2 in the paper). With reference to the table under Fig. 9, although the designed steam rates for both designs were the same, it was rather unfair to give the fuel rate for all purposes on trials with totally different feed systems and design of auxiliaries. The turbine was clearly not solely responsible for the all purposes fuel rates which were given to three decimal places, although the shaft horsepower was probably not determinable within 2 per cent.

The turbines shown in Figs. 13 and 14 were those on which full-scale trials were carried out, particularly the effect of warming up and rapid transient conditions due to change of load. At the end of these trials which were much more severe than could be given in service in a ship, there was scarcely any evidence of touching between fixed and moving parts of fine clearances. Could Commander Platt elaborate on this point as the feeling was generally that if no marks had been seen at all under these very extreme trials, then clearances would have been too large.

The drawings of H.P. and L.P. turbines shown in Figs. 18 and 19 were the new Pametrada standard for the range of 15,000 to 20,000 h.p. In seeing the improved effect between 1950 and 1961 shown in Figs. 22 and 23, it should be realized that the H.P. turbine with minor modifications in nozzle height and blading was capable of developing 26,500 h.p. as a maximum in this frame size, so that in 11 years the turbine was capable of twice the power in a shorter overall length and a smaller weight. This was as a result of an aggregation of a number of researches over the period. Similarly, the change in Fig. 23 from double flow L.P. ahead blading to single flow blading at twice the horsepower was a result of special investigations of the vibration characteristics of long, low pressure blades. Again the point was made that these improvements would not have been made without a very great deal of research work which was going to continue in the future.

In conclusion he wished the new Pametrada a very successful future, on his ceasing to be the Director of Pametrada and assuming responsibility as Director of Marine Engineering Research of the British Ship Research Association.

Referring to Mr. Bauer's paper, Dr. Brown said that the paper gave an interesting summary of marine turbine developments in the author's company over the last 30 years and corresponded broadly to similar developments in this country. The following comments applied particularly to the 1960 design of turbines shown in Figs. 14 and 15:

Panting plates (author's flexible plates) required careful design to give adequate flexibility in the direction of expansion with rigidity to resist pipe thrusts, etc., in other directions. Accurate setting up in the ship was required but given this, they were effective and reliable.

Elongation of the H.P. turbine foot was presumably designed to give the same length as the L.P. turbine. It was probably bolted to the hot casing and therefore might suffer from distortion above 850 deg. F. Certainly a rise in temperature would aggravate the distortion. Dr. Brown preferred a separate pedestal supporting rotor palms and running rotor in the same plane. This arrangement was suitable for 950 deg. F. and had been successful in important ships like *Oriana, Windsor Castle, Transvaal Castle,* etc.

In the matter of bearing design it was clear that De Laval had not progressed to a length/diameter ratio between ⅓ and ⅔ which could carry greater total loads than the longer bearings shown on the drawing. The shorter bearings were not only safer but had increased stability against oil whip. Referring to Fig. 17, page 423, it was stated that oil dams were added to the quill shaft flexible coupling sleeves to ensure adequate lubrication. His experience was that they acted as sludge traps and were best avoided.

The author criticised the arrangement of the astern turbine in a separate casing as requiring a greater length of turbine and higher cost. (See previous note on the length of H.P. turbine foot.) Increase in length was of no consequence as the H.P. turbine was still shorter than the L.P. The important point was to provide enough blading to develop adequate astern power at steam conditions of 950 deg. F. (high heat drop). Astern losses varied as the fifth power of the diameter and were actually less with an H.P. astern than without, due to the reduction in diameter of the remaining stages. Thermal shock was also reduced. Had the author experience of astern turbines wholly within the L.P. turbine operating at 950 deg. F. without attemperation? Compound astern turbines overcame these difficulties and also balanced the astern power on the two gear trains.

Plugs fitted in the casing adjacent to the balancing strips on the rotor were noted. Had this facility proved of value in practice?

The L.P. turbine was fitted with a thrust block remote from the inlet end. Would the author please explain the differential expansion between rotor and casing on a long astern run.

A bar life nozzle valve simplified operating gear when a number of nozzle control valves were used to give good efficiency over a large power range. At low powers a very high standard of tightness was required if efficiency was not to suffer. There appeared to be nothing to keep the valves tight except their own weight, before pressure difference established the seal.

Studies carried out at Pametrada had indicated consistently that higher steam conditions than the current American 600lb./sq. in. gauge 850 deg. F. standard were justified. This was no doubt due to the difference between the economic factors involved (e.g. in U.S.A. fuel cost represented only about one fifth of the overall cost of tanker operation, whereas in Europe this figure was nearer one third).

How far conditions could practically be raised would depend ultimately on solution of superheater corrosion problems. At present 950 deg. F. was a certain workable upper limit of temperature but he would disagree with the pressure of 1,200lb./sq. in. gauge which seemed to be almost automatically associated with 950 deg. F. in the paper.

Pametrada had found that the gain in overall fuel rate, at 950 deg. F., through raising the superheater outlet pressure from 800lb./sq. in. gauge to 1,200lb./sq. in. gauge was only 1¼ per cent for a four heater cycle at 15,000 s.h.p. and 1½ per cent at 30,000 s.h.p. (This took account of feed pump work and the effect of high pressures on the turbine efficiency.) On this basis their conclusions were that 800lb./sq. in. gauge was the highest pressure that could economically be justified at 950 deg. F. below 30,000 s.h.p., and the Pametrada standard frames were designed on that basis.

MR. L. PARODI (Member) whose contribution relating to the paper by Commander Platt and Mr. Strachan was read, congratulated the authors on their very interesting paper, and also expressed his thanks for the favourable comments they had made about the work in Italy.

Remarking upon the boiler superheat temperature control, Mr. Parodi said that in the modern design of tankers, where the difference between the normal and the maximum evaporation was considerable, the choice of superheat steam control range was very important. That point was not always well understood. It was required that the steam control unit should not operate during normal ship navigation, without cargo heating. That condition in a tanker corresponded to about 70 per cent of the maximum continuous boiler load. He thought that a good boiler design required that, when the boiler was steaming at 100 per cent load, that was, normal navigation with cargo heating or Butterworth heating—the superheat control must be working to its maximum capacity, leaving only the small margin required by a wise design. Should that condition match the service results, he considered that the steam temperature control was sound even though there might be some control of superheat taking place during the ship's normal navigation, without cargo heating.

The main reasons for that were as follows: first, it represented a good margin for keeping the design steam temperature when the boiler was dirty; second, it was also possible to have the design superheated steam temperature at reduced power, giving a temperature at the lower turbine speeds and better steam conditions inside the L.P. turbine, with correspondingly better fuel oil ratings.

Fig. D.1 showed two curves. The solid line referred to

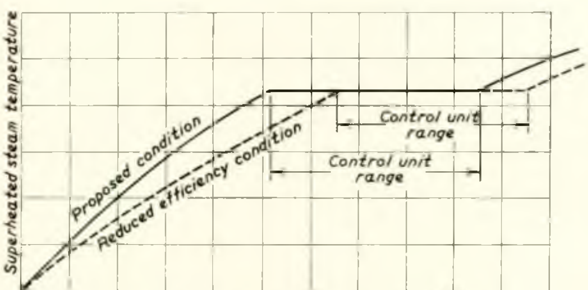

FIG. D.1—*Control unit operation range—for boilers fitted on turbo-tanker*

the suggested condition, while the dotted one referred to a condition of reduced efficiency.

Another point that he would like to raise related to the split economizers that his organization was adopting in one ship that it was building.

From the figures shown on the heat balance for 68,000-d.w.t. tankers he had tried to find the calculated gain, using the split economizer, but he was compelled to make some assumptions because that heat balance did not show all the necessary figures. Within those limits he found that the improvement reached was below 0·5 per cent. Should that figure be correct, he thought that that gain did not justify the cost of installation and the more complicated circuits. He would like to have the comments of the authors about that point.

One more question related to the boiler tube failures due to metallic oxides inside the fire row tubes. He supposed that those phenomena were also connected with flame impingement.

In the paper recent progress in combustion techniques was mentioned, and he had heard that the authors' companies had experience, in some ships, of new types of burner. He would be very pleased to know whether tube failures were also experienced in the boilers where those newer burners were fitted.

MR. G. KAUDERN said that he wished to make a few comments on the paper presented by Commander Platt and Mr. Strachan, which had been extremely interesting to him, particularly with regard to the machinery for the 50,000-d.wt. tanker class. That was of particular interest to his company since it was very similar both in layout and power to the machinery which it had installed in 11 tankers since 1955.

All those plants were rated at 15,000 normal h.p. Steam conditions were 600lb./sq. in. gauge and 865 deg. F. at the superheater outlet. Back pressure turbo-alternators were fitted, operating on superheated steam and exhausting to a de-aerator and to steam air heaters.

The turbo-alternator size chosen for the first vessel was for 700 kW, which, incidentally, was the same as the authors had chosen. However, soon after delivery, service reports indicated that the alternators were far too large. The turbo-alternator size in later vessels was consequently reduced to supply 575 kW. Reports had proved that those alternators, under normal service conditions for a 50,000-d.w.t. vessel, gave ample margins.

Electric power consumption of that magnitude, coupled with the higher efficiency of the back pressure turbo-alternators and the main feed pump, formed a good starting point for an economical heat balance. In the case of his company, they used a turbo-alternator with a steam consumption of 18·7lb./kWh. as compared with the authors' figure of 20·2.

Trial fuel rates at normal power for the latest installations ranged from about 0·529 to 0·508lb./s.h.p. hr. All fuel rates were corrected to normal operating conditions—28·5 in. vacuum, and so on.

One thing that was very important in that connexion was the method that was used for measuring the h.p. It would be very interesting to hear the comments of the authors in that respect and to know what methods they had used and what, in their view, might be the estimated accuracy of the methods.

It would lead too far to go into all the details of the authors' forecast of the machinery of the near future. He would, however, like to add a few words to their comparison between the back pressure and self-condensing alternators. The authors had compared four installations with the following alternators installed: two back pressure, high efficiency; two back pressure, low efficiency; two self-condensing, high efficiency; and two self-condensing, one high and one low efficiency. They had come to the conclusion that the last combination gave the best overall economy, resulting in an annual saving of £5,050.

There could be a fifth combination, using one high efficiency and one low efficiency alternator, both of the back pressure type. According to his calculations, an annual saving of about £6,000 instead of £5,000 would then be obtained.

MR. ELVINO DERDINI said that he wished to express his appreciation as a representative of the Cantieri Riuniti Dell'Adriatico, for the very interesting papers by Commander Platt and Mr. Strachan and by Mr. Bauer, which gave concisely a complete picture of the progress of steam turbine tanker machinery from 1930 to the present time. He also wished to thank Commander Platt for the assistance which he had given in connexion with the ships which had been built in Italy for his company.

As ship and machinery builders and licensees of Mr. Bauer's company, his organization had experience in their new plant of practically all the latest ideas concerning turbine propelling machinery, as far as both the construction of the main geared turbines and the design of the whole plant was concerned. By that he meant the boilers, the steam flow design, the lubricating system, the design of the main condenser, and so on.

In their opinion, it was difficult to suppose that much more could be done on those steam cycles with regard to economy, except in increasing the steam temperature and pressure, in view of the few ship services requiring a certain energy in the form of both heat and power.

They were having an easier job in that field on the passenger liner field, where they had, for example, associated the production of electric power and the production of distilled water for ship service by using pass-out turbo-alternators supplying bled steam at constant pressure to the big low pressure sea water evaporators.

A small improvement in the steam cycle with regard to fuel economy could be achieved by the adoption of two-stage steam air heaters instead of using the higher pressure to reach the established air temperature.

What his organization considered to be an important improvement was the adoption of main condensers, expanded in both tube plates. In fact, they had never had any trouble, in about 20 ships which had been fitted with that type of condenser.

Rear-Admiral J. G. C. Given, C.B., C.B.E. (Member) said that it was very encouraging to have a paper with the co-authorship on the one side of a designer-builder and on the other side an owner-operator and maintainer. He thought that the progeny of that marriage was a wonderful paper. It was also very encouraging to him to be able to listen to and comment upon Mr. Bauer's paper. It was twenty years since he was at Trenton, New Jersey, U.S.A. What a progressive firm the De Laval Steam Turbine Company had always been!

He wished first to refer to the paper by Commander Platt and Mr. Strachan, and in particular, to the comparison between British and Italian ships. He felt that the heterogeneous nature of the British marine turbine industry had something to do with some of the back log. It was to be hoped that in future this situation might be improved.

He entirely endorsed what Commander Platt had said about the double flow L.P. turbine. He felt that inherently they were always liable to distortion. In the early part of 1939 he was in a cruiser where they had 24,000 h.p. sets with double flow L.P. turbines, and those were somewhat of a worry to him. The vibration they had produced was sometimes rather like starting a car with a fierce clutch in top gear.

With regard to future development, he could not agree with Commander Platt that the Diesel and the turbine should lie down like the lion and the lamb. Competition was the spur to progress. He wondered whether Commander Platt, when he had to consider 25,000 and 35,000 s.h.p. installations, would have a second look at the turbo-electric drive. The rotational speeds might well go up. The voltage might go up to 12 kV. and there were a number of other potential advances.

Finally, he wished to put a point to Mr. Bauer. He was very interested in the Michell Connisbury type of thrust block on the primary gear wheel spindle, which seemed to him to provide an opportunity to measure the transient thrusts that occurred when manœuvring in a gear train of that nature. He suggested that this in a way would give a measure of the real actual flexibility of the couplings.

Captain R. G. Raper, R.N. (Member) said that the paper by Commander Platt and Mr. Strachan was full of interest both technically and on the design philosophy which was developed in it. From purely financial considerations, the authors had had to face problems in development which were parallel to many of the problems which faced the designer of naval machinery in his struggle to reduce the weight and space occupied in a ship by the propulsion machinery and its fuel. The authors concluded that complexity in the form of additional heat exchangers referred to as "inanimate objects" could be achieved without loss of reliability. It would be interesting to know whether the inference from this was that the "animate objects", presumably rotating machinery, could not be accepted as reliable. He asked this question particularly in view of the fact that in the summary of defects which had been suffered by the machinery installations discussed, there was no mention at all of trouble with rotating auxiliaries. The inference, therefore, did not seem to be entirely supported by the facts presented in the paper.

On the other hand, at the bottom of the first column on page 409 it was stated that the economic advantage of more complex systems must not be dismissed altogether through fear of breakdown, for really good designs should always be able to eliminate such risks. This was a statement of an ideal situation which, in naval experience, was seldom achieved. In theory, of course, it was indisputable; in practice, it had been found that human failings were such that a period of stringent testing was required before any new design could be called reliable. Nor could the blame for this be laid entirely at the door of the designer of the machines concerned, since it was often difficult to foresee and define all the operating conditions for which the designer must allow, and, while it was generally possible to specify what a given system should do, it was frequently difficult to produce an adequate analysis of every condition which any component in the system had to meet. From experience in the field of naval design he would therefore personally agree with the conclusion of the authors that the most profitable path lay in choosing a propulsion system which had a development potential and raising its efficiency and reliability, stage by stage, from experience with the plant at sea.

To change from the general to the particular, the only part of the paper which came as something of a surprise was that on the trouble with boiler tubes. The types of corrosion quoted as leading to tube failure, on the face of it, bore some resemblance to the "scab pitting" experienced in the past in naval boilers. It was not stated whether the corrosion was confined to the hottest parts of the tubes and it would be of interest to know more about the distribution of the scabs or barnacles within the fire row, if they extended to other tubes than those which actually failed and to other positions than the position of failure. It would also be interesting to know whether there was any actual evidence of hydrogen embrittlement which would appear to be an unexpected phenomenon in boilers of this nature. While he realized that the conditions under which boilers were operated in the Royal Navy were not the same as those applying in tankers, particularly in respect of the much more frequent shut-downs that were normal routine in naval ships, it might be of interest that the use of standard naval boiler compound and the maintenance of boiler water within the following limits had completely eliminated any serious form of internal corrosion in naval boilers:

a) Alkalinity between 0·25 per cent and 0·5 per cent normal (0·35 per cent preferred)

b) Salinity of not more than 70 p.p.m. of chlorine.

Following on from this, naval experience had not shown that the exclusion of metallic oxides was the only cure, although it was fully agreed that oxides should be kept out of the boilers. Naval experience was that the cure for such troubles consisted of three parts. Firstly, care in design to avoid excessive heat transfer at local points. To this end, the characteristics of the flames in high heat release rate furnaces of prototype boilers in post war boiler designs were carefully analysed and generous clearances were allowed between the registers and the fire row and water wall tubes. Secondly, a careful commissioning of the boiler and the whole plant to ensure complete cleanliness and the establishment of a protective film on the metal surfaces. This would undoubtedly be helped by carefully controlled chemical cleaning. Thirdly, careful operation to ensure that the protective film was maintained or, if it was broken, that it was re-established as quickly as possible.

The mention of the need for cleanliness during installation must be referred to the whole process of installation as well as to feed systems. The success of a design, and often the designer's reputation, when he had done all that he could himself, still hung on a thread consisting of the manufacture, inspection, and installation of the machinery in the ship and he would like to support Mr. Armstrong's plea that a great deal of thought should be given to the improvement of these parts of the process. Naval experience was that it was often extremely difficult to dis-entangle the causes of failure in a new design because the dirt built into the various systems in a new ship frequently obscured the issue of whether a failure was inherent in the design or accidental. Quite apart from the serious implications on the operation of a new ship, the energies of those involved in the design work were too often dissipated in studying failures which should never have occurred.

It was also interesting to read the satisfactory report on the "Fast Tanker" machinery. Some work was done in the Admiralty on the possibility of a two-speed tanker for Fleet Auxiliaries which led to a study by the Yarrow-Admiralty Research Department of a machinery plant somewhat similar to that in the fast tankers. It was very gratifying to know that with such a concept the performance of the tanker at the design point for normal operation was not adversely affected by the built-in reserve of speed.

He was very grateful for all the information included in the paper and for the presentation of the history of the development of the machinery and the design philosophy which had been evolved from it.

Captain N. J. H. D'Arcy, R.N. (Member) added his congratulations to the authors on their excellent papers.

His company had been privileged to work with the B.P. Tanker Co. in the development of the 16,000 s.h.p. machinery installation for the 50,000-d.w.t. class described by Commander Platt and Mr. Strachan on pages 393-395 of their paper. It was particularly interesting therefore to see their developments in the comparatively short period since that design was prepared. This was exemplified by the H.P. turbine, which had progressed from the double casing design shown in Fig. 13 for the 50,000-d.w.t. class to the special single casing design shown in Fig. 18 for the 68,000-d.w.t. class, in the space of only 3-4 years.

Dr. Brown had already referred to the full scale shore trials carried out at Pametrada on one of the 16,000 s.h.p. machinery sets with the double cased H.P. turbine, built by Captain D'Arcy's company. The machinery was put through a very severe trials programme, covering nearly every aspect of operation at sea, and even including some deliberate mishandling. The detailed results were still being worked out, but he was sure members would be interested to hear that the turbines had opened up in an extremely satisfactory condition at the end of the trials, in spite of the abnormally severe handling. He thought this disposed of certain criticisms that had been heard of the double casing design of H.P. turbine.

He was glad to note the very proper emphasis placed by the authors on the importance of the boiler design. This was undoubtedly one of the main limitations to the advance of steam turbine machinery at the present time.

In the considerable number of steamships they had built, his company had had no reported cases of early boiler tube failures as described by Commander Platt and Mr. Strachan, although they had not had the boilers chemically cleaned. He was inclined to attribute this immunity to care in keeping the pipe systems clean during erection. It would be interesting if the authors could say whether their boiler tube failures had been less prone to occur in ships built where the machinery contractors concentrated more on maintaining standards of pipe cleanliness.

An important item which had emerged from both papers was the necessity to cut out unnecessary margins on every part of the machinery installation. In carrying out the design studies for the 50,000-d.w.t. B.P. tanker class and other machinery installations, his own company had been able to eliminate many traditional margins which were wholly unnecessary and wasteful. He felt sure there was still ample scope for further refinements in this direction.

There was one other aspect of these papers, to which he thought it worth drawing the attention of shipowners in particular. Both papers had clearly shown the advantages to be gained from comprehensive preliminary design studies before deciding the machinery installation. But neither had pointed out the time required for this to be done satisfactorily. It was useless for the shipowner to come along to the builder at the last minute and want it done in a week. In the case of the 50,000-d.w.t. class already mentioned, it had taken nearly a year of combined work by his company, Pametrada and the owners to evolve the final installation. It could be done more quickly of course. But 4 months was about the minimum time required, if the shipowner was to get the best installation for his particular service.

He concluded by thanking the authors again for their most valuable and interesting papers.

Captain W. S. C. Jenks, O.B.E., R.N. (Member) said that the meeting had had two admirable papers. It was a matter of great regret that there was not time for the various contributors to discuss them more fully.

He proposed to confine his remarks to one aspect only, and that was the suitability of the installations for operation and maintenance by seagoing personnel.

In the paper by Commander Platt and Mr. Strachan, the design study for the 50,000-ton tanker was of particular interest, for it seemed to have resulted in as simple a heat cycle as could be hoped for, combined with a fuel rate which was within measurable distance of the best obtainable for the steam conditions selected.

It was noticed that the objectives for the 68,000-ton tanker design study were given as "complete reliability, simplicity of operation and maintenance, and the best possible fuel rate"—in that order. Later on in the paper the order had however, been changed. Item a) of the basic design requirements on page 398 required "maximum economy combined with simplicity". There seemed to be a change of emphasis. In any event, a considerably more complex design had been chosen than that for the 50,000-ton tanker with a penalty in weight and first cost and only a marginal overall gain. It should be appreciated that a fuel cost had been assumed which appeared to be high and, if the fuel was in fact cheaper, the theoretical saving would be substantially reduced. He could not help wondering whether the type of installation selected for the 50,000-ton design was not, in fact, nearer the objectives originally postulated for the 68,000-ton design.

With regard to future trends, there was an acute problem facing all who operated ships at sea, and that was to get enough men of the right calibre to take up seagoing engineering as a career or, having taken it up, to persist in it. This problem was not peculiar to this country, but it was so important that he suggested shipowners should lay down as a paramount objective in every design, that it should be suitable for operation and maintenance by the absolute minimum number of seagoing personnel of a calibre no higher than that likely to be available in the foreseeable future.

The authors had referred to that factor on page 409 of the paper in the paragraph on automation, but he would like to take the problem a little further.

Inherently a steam plant must consist of a number of major units built into the ship together with a number of auxiliary units which could, if so desired, be made self-contained and replaceable. All these units must be interconnected by a number of more or less permanent pipe systems which, even in quite a simple plant, were extensive and complex. The whole system was controlled by the operation of valves, electric controllers, etc., which could be operated automatically (by adding yet further pneumatic, hydraulic or electric systems), but must in emergency be capable of manual operation. Furthermore, the setting up of the system prior to operation, lighting up and warming through procedures, filling of tanks, etc., would be carried out manually. It followed that if designers and manufacturers were to provide some solution to the acute personnel problem, to which he had referred, it was not enough just to add automation to present systems: there must be a much more fundamental approach and he wished to suggest some ways in which he thought they could help in solving the problem.

First of all there were the basic systems. They should be kept as simple as possible, compatible with an efficiency which was within measureable distance of the best obtainable. Secondly, there were the main units—turbines, boilers, gearing and so on—built into the ship. There emphasis should be placed on the avoidance of too frequent introduction of new designs. The aim should be rather to get the basic design on sound principles and then develop it to get all the bugs out and keep it standard long enough to reap the benefit. The target should be absolute reliability and freedom from maintenance over long periods, at least for the four years between surveys. Thirdly there were the auxiliary units and components. In many cases they should be made completely self-contained and designed for very easy replacement as complete units. Also important, were absolute reliability and absence of maintenance between overhaul periods, which could be much shorter than for the main units. Manufacturers should develop, for suitable cases, schemes for factory overhaul of units, thus enabling working units to be replaced by guaranteed factory reconditioned units on an economic basis.

It was believed that there was much scope for improvement in the field of auxiliaries, and many troubles in existing plants stemmed from units which were either of inadequate standard originally or else had become so by the attentions of personnel less skilled and less careful than the original manufacturers. Again, all too few manufacturers provided drawings and handbooks which were adequate for ensuring proper overhaul.

Another aspect was piping systems. Much greater care was needed in both the design and the engineering of the various piping systems. The reliability of the plant was dependent upon the integrity of those systems and upon the freedom from contamination of the fluids which they contained. In working out piping arrangements, he suggested that the following rules should be observed. Firstly, permanent interconnexion between systems containing incompatible fluids should be avoided at all costs. Devices like blind and open flanges provided no real safeguards. Secondly, multiplicity of valves and of cross-connexions designed to meet every conceivable situation should be avoided. It was much better not to have a facility and have some emergency measure available in the unlikely event of something unforeseen going wrong. Thirdly, the physical layout of pipes themselves and the positioning and labelling of the valves were tremendously important. Careful planning and scale model techniques should be used to get the best possible arrangements providing good access and easy comprehension by personnel.

If the aims he had mentioned led to real improvement in the reliability of system components and a marked reduction in the likelihood of human error, then the adoption of automatic controls on a comprehensive scale would be really worthwhile, providing they themselves had a high degree of inherent reliability and a built-in system for diagnosing their own defects. He ventured to suggest to the authors that future trends in design must take account of the personnel problem either by the means he had suggested or by some other method if any further significant advantage were to be obtained in practice from the advances in steam conditions or more efficient heat cycles that might become available.

Mr. E. G. Hutchings, B.Sc. (Member) said that both papers were very interesting. The paper by Mr. Bauer gave one a great deal to think about. From the point of view of boilers, he agreed with what Mr. Bauer had said about superheaters.

With regard to the paper by Commander Platt and Mr. Strachan, he noted that the E.S.D. II boiler was chosen for the 68,000-ton class of tankers on the grounds of the improved control method and reduction in weight and cost. It was to be presumed that the comparison drawn by the authors was with the E.S.D. I, since other boilers were available using a similar type of control to the E.S.D. II, giving a quicker response, and even wider range of control and were backed by several years of seagoing experience. Also, they were usually more compact and lighter in weight. It was unwise to generalize on price of boilers as local conditions could very often affect the price of boilers between two competitive designs.

With regard to feed cycles, the authors came to the conclusion at one point in their paper that a simple cycle with feed at 280 deg. F. and mild steel economizers was the most attractive. Earlier on they had implied that a choice of machinery with a 240 deg. F. feed was more attractive. Elsewhere they said that a split feed system was desirable. He suggested that those conclusions must be wrong, and that this was due to the fact that the authors had not optimized the boiler efficiency on each scheme. Had that been done, the simple feed cycle would have been more attractive in all cases. Also, the simple feed cycle had been made to appear less attractive than it was owing to the double cased turbine.

A cycle including economizers and steam air heaters was undoubtedly very sound, but he considered that gas air heaters were still very worthy of more consideration. A tanker operator with a large fleet had consistently persevered with gas air heaters for many years in an attempt to minimize the maintenance of the air heaters and get the advantage of the high efficiency theoretically offered by such a scheme with four stages of feed heating. That had been a long process due partly to the ignorance of boiler designers about the aerodynamic effects on air heater tubes. However, they seemed to have got over the problem. They had two pilot plants at sea which appeared to be working satisfactorily, and a full scale plant was being installed that week. It would, however, take about a year before they could be sure about it.

The same company had been operating at 850lb./sq.in. pressure for many years, and the troubles which Commander Platt expected with regard to tube failures had undoubtedly proved to be true, but they appeared to have been overcome in recent years, and operating boilers at such pressures no longer presented a problem.

He agreed with what the authors had said about steam pressures. The suggestion made by Commander Platt about increasing the furnace volume was a good one provided that owners were prepared to pay the extra cost.

Commander V. M. Lake, R.N. (Member) said that he wished to draw attention to one point in connexion with the excellent paper by Commander Platt and Mr. Strachan. He wished to pay tribute to the contribution which the B.P. company had made to the industry over the years by the continuous development which was shown here.

The paper implied by the number of diagrams of turbines and feed cycles that that was all that had been involved. However, practically every auxiliary had been developed, as the result of B.P.'s requirements, to such a pitch that it was extremely interesting to see that, for instance, the evaporator was now quite calmly put in as an integral part of the feed system. Only a few years ago people would have been very chary indeed about doing that because of the risk involved.

That led automatically to the boiler failures. He would echo Captain Raper's remarks with regard to Admiralty experience. Perhaps there was a difference there, and he would like to draw attention to it. It had generally been agreed that boiler corrosion took place while boilers were idle, yet the authors were saying that they were suffering severe trouble when boilers were not idle but in continuous operation. That was a totally different view.

He suggested that the cause was largely due to the dirt, not necessarily in the naughty sense of the word, but in the sense of normal corrosion products which occurred in a normal feed system and in a boiler where concentration naturally occurred. Because the boilers were operating more or less continuously, these were not being removed. That showed the difference between merchant service and naval operation. In the Navy, boilers were normally pumped up at regular intervals between short trips, and the amount of feed run out of the boiler in raising steam after such an operation, eliminated very large quantities of the corrosion products from the boilers. They could not, therefore, collect in the fire tubes. He suggested that perhaps there might be in the operation of tankers some reluctance on the part of the chiefs to waste water from the boilers in running down or blowing out those corrosion products.

As a final word with regard to failures, he thought that it was time that there was a revision of the method of rating boilers. With the modern combustion equipment where an extremely intense flame was utilized, to call a furnace highly rated was not presenting a true picture. The temperatures that were achieved in those intense flames were such that overheating appeared to occur in spite of the fact that there was no flame impingement. They had had that experience in the Admiralty, and he believed that research was continuing to establish a better parameter for safety from burn-out in fire tubes.

	Fuel rate lb./s.h.p.hr.		
	Design	Service	
		Minimum	*Maximum*
British ships (table on page 391)	0·59	0·567	0·634
Basic 14,000 s.h.p. ships (Figure 4)	0·577	0·585	0·625
Fast tankers (Figure 7)	0·571	—	—

However, scrutiny of the heat balance diagrams suggested that it was the basic 14,000 s.h.p. installation which had the same main engine non-bleed water rate as the Italian ships, and presumably therefore the comparison drawn by the authors was between the Italian ships and the basic 14,000 s.h.p. U.K. design.

As a matter of interest, the non-bleed main engine water rate in the fast tankers appeared to be about three per cent greater than that in the 14,000 s.h.p. designs, and this seemed a small enough price to be paid for an economical overload capacity of almost 60 per cent.

From the three heat balance diagrams for 14,000 s.h.p., Figs. 4, 7 and 10, it had been noted with some surprise that, despite the differing design fuel rates quoted, the heat to be transferred from fuel in the boilers was virtually the same in all cases, i.e.

Design	Steam flow lb./hr.	Heat reqd. per lb. of steam B.t.u.	Total heat required B.t.u./hr.	Heat supplied from air B.t.u./hr.*	Heat to be supplied from fuel B.t.u./hr.
Figure 4 Basic 14,000 s.h.p. U.K. ship	111,160	1224·7	$136{\cdot}1\times10^6$	$3{\cdot}9\times10^6$	$132{\cdot}2\times10^6$
Figure 7 Fast tankers	121,150	1127·8	$136{\cdot}6\times10^6$	$4{\cdot}8\times10^6$	$131{\cdot}8\times10^6$
Figure 10 Italian ships	111,680	1225·4	$136{\cdot}8\times10^6$	$3{\cdot}9\times10^6$	$132{\cdot}9\times10^6$

* Based on 16·5:1 air/fuel ratio, an ambient temperature of 85 deg. F., and an assumed boiler efficiency of 87 per cent. in each case.

MR. J. NEUMANN, B.Sc.(Eng.) (Member) said that the authors referred to a superiority in performance of the Italian-built tankers as compared with their U.K. counterparts. The figures quoted for the design fuel rate were 0·56 as compared with 0·59lb./s.h.p. hr., i.e. an advantage of more than 5 per cent. Such an improvement was most significant: for example, it was greater than could be expected from an increase in initial steam temperature of 100 deg. F. In the comparison quoted, the cycle steam conditions were identical and the gain in performance was put down to various listed features. Unfortunately only two of those lent themselves to a quantitative analysis, which furthermore showed disappointingly small results:

i) The low pressure drop in the main steam piping, 21 instead of 40lb./sq. in., showed a theoretical fuel rate improvement of only about 0·25 per cent, and in practice the improvement could be expected to be even less than this.

ii) The use of a cavitating extraction pump, which was attractive when feed surges could be accommodated in a de-aerating feed tank, showed a saving in overall fuel consumption rate at the service power of less than 0·1 per cent.

The British ship fuel rates quoted in the comparison table on page 391 corresponded neither with the fuel rates previously given for the basic 14,000 s.h.p. installation, nor with those quoted for the fast tankers:

It appeared therefore that a fundamental reason for the low overall design fuel rate of 0·56lb./s.h.p. hr. in the Italian ships must be a very good boiler efficiency. Assuming that the calorific value of the boiler fuel was 18,500 B.t.u./lb., and that the fuel was heated to 230 deg. F., the fuel rate of 0·56lb./hr. was indeed found to correspond with a boiler efficiency of about 91 per cent, coupled with a funnel gas temperature of about 200 deg. F. He was therefore forced to the conclusion that the designers had either managed to steal a march on the second law of thermodynamics since the funnel gases would be cooler than the feed water entering the boiler; or that the calorific value of the fuel used in the design was greatly in excess of 18,500 B.t.u./hr. It would be of interest to hear the authors' comment on this matter.

It was noted that the major difference between the cycle used in the fast tankers and that selected for the 1963 68,000-ton tankers was the replacement of a surface type intermediate pressure feed heater by a direct contact de-aerating feed heater and an associated de-aerator extraction pump. It would be valuable to learn whether the authors considered locating the de-aerator at a sufficient height relative to the feed pump to pressurize its suction by static head rather than by reliance on an additional pump. Further, could the authors comment on the record of reliability of the de-aerator extraction pumps fitted in the Italian-built ships and in the 42,000-49,000-ton tankers.

Referring to the development of the 20,000 s.h.p. design, the authors had given a concise comparison between the cycles they considered, including back-pressure and self-condensing steam turbo-alternators. It would be of interest to learn whether attention was also given to the possible replacement of one or both turbo-alternators by Diesel-driven alternators and if so, what were the reasons for rejecting this alternative.

In Table V, comparing main condenser designs with ¾-in. and 1-in. outside diameter tubes, it was suggested that an additional factor militated against the adoption of the larger tubes. Any water drawn into the ship must be accelerated to the ship speed, and appropriate power must be supplied at the propeller to do this. Use of 1-in. outside diameter tubes called for an increase in cooling water flow of 5,300 g.p.m. which at a ship speed of 16 knots represented an additional power requirement of almost 50 horsepower if 100 per cent intake and propulsive efficiencies were assumed. In practice both these efficiencies would be less than 100 per cent and the additional power to be supplied to the propeller would be correspondingly greater.

Mr. L. J. Culver, B.Sc.(Eng.) (Member) referring to paper No. 3, page 398, said that there was a reference to the influence of naval prototypes on the geometry of the modern marine boiler. Since there were a number of marine boiler designs produced specifically for the merchant service it was assumed that the authors' reference was mainly directed towards the furnace sizes and the post-war trend to higher furnace ratings. While they mentioned the reduced furnace volume required for complete combustion with improved register designs, they also recommended the installation of the maximum furnace volume that could be accommodated within the contours of the ship. The margins provided by this policy should be more practicable in the larger tankers than in the immediate post-war tonnage. The provision of remote operation of air registers would also enable the fires to be spread more uniformly over the furnace, since scope for manual operation would not be important. In the past burners were often located too close to the furnace floor because the position of the firing floor was not always arranged to suit the ideal burner positions. The needs for improved flame clearance to bare brickwork was a lesson that had been learnt in the earlier programmes described.

The authors also referred to the reduction in design margins but not to the lower level of steam temperature often specified to suit manœuvring or emergency conditions. This temperature margin added to the boiler equipment. Mr. Bauer might be able to comment on this point since the United States practice did not require a reduction from 850 deg. F. superheat for astern running. Would this practice be maintained with merchant marine turbines designed for 950 deg. F. or would the United States turbine designer call for a potential reduction of superheat to be available?

In paper No. 4 Mr. Bauer mentioned the efforts towards automated equipment. Were the studies being applied to conventional or austerity ships, or was the problem being tackled from first principles with possibly simplification of the heat balance or main components to minimize the controls and instrumentation?

Again in paper No. 4, page 424, the boiler illustrations and text might suggest that the use of two-drum or bent tube boilers was largely in the post-war period whereas it was introduced in the 1930's, was used extensively in the 1940's and was well established in both the United States and the U.K. by the end of the war. These units often had closely pitched bare tubes for the water walls, a feature which again was credited to the 1960's.

The use of floor tubes in American-built boilers was mentioned. In many cases there was a considerable downward slope of these tubes immediately on leaving the water drum before they reached the horizontal section of the floor. In the author's experience had this led to circulation trouble or had there been failures of floor tubes due to wastage of the floor brickwork?

The previous paper referred to the use of steel surface in economizers with feed temperatures of 280 deg. F. whereas Mr. Bauer suggested feed temperatures of 277 deg. F. and the use of copper bearing steels. The trend upwards from 240 deg. F. to 280 deg. F. for the initial economizer inlet water temperatures originated in American installations, but was it not true to say that cast iron extended surface was still used for the "cold" section of most economizers?

It was noted from the flow diagrams that the heat recovery equipment on the boilers was as follows:

Fig. 21	1930 era:	Gas air heater Economizer, if fitted, feed temperature 310 deg. F.
Fig. 22	1945 era:	Gas air heater Economizer, if fitted, feed temperature 310 deg. F.
Fig. 23	1960 era:	Steam air heater, air rise to 100 to 265 deg. F. Economizer, feed temperature 281 deg. F.
Fig. 31	Advanced Cycle:	Gas air heater Economizer, if fitted, high temperature feed.

Was the author satisfied that gas air heater corrosion or fouling could be prevented by a good air heater design or should a limited amount of preheat in the air, by a bleed steam heater upstream, be anticipated.

Commander E. Tyrrell, R.N. (Member) said that, in his opinion, the authors of both papers had made an important omission.

Some five years ago, he had drawn the Institute's attention to the need for regulations or some form of authoritative guidance for carrying out marine heat balances. He had followed this with a letter to the Institute suggesting that the Institute might like to set up a committee to investigate this matter, and publish rules on how heat balances should be carried out. This suggestion was rejected. Since that date, he had found that, if all the assumptions and allowances necessary for making a heat balance were kept within recognized limits, but chosen to give the best possible fuel rate, this would be between 4 per cent to 6 per cent better than that obtained from a heat balance where the assumptions and allowances (although still within recognized limits) tended to give a higher fuel rate. Contracts for marine steam turbine machinery were often obtained because of a difference of 1 per cent or even less on the fuel rate, which was much less than the difference obtainable between different firms using different rules for calculating their heat balances.

Heat balance calculations were therefore useless to the shipowner as a basis of comparison between the products of different companies unless the owner could be sure that they had been made on the same basis. In both papers given that day, the authors had published heat balances, but gave no reference to the basis on which they were calculated. The balances therefore indicated only one person's opinion of what the fuel rate was likely to be, and had little or no value when compared one with the other.

Today, when all shipbuilders were short of orders, it was possible for the less scrupulous to give heat balance figures which were better than those obtainable in practice. The owner could protect himself from this in two ways. First, he could ask for a guaranteed fuel rate under financial penalty. Most shipbuilders were reluctant to give this, but many would do so under pressure. Secondly, he could ensure that he got a direct comparison between the heat balances of different firms by insisting that the heat balances be carried out in accordance with the recommendations of a recently published paper on this subject by the Society of Naval Architects and Marine Engineers in the United States. Commander Tyrrell recommended that members of the Institute and owners should in future ask for heat balances to be carried out in accordance with S.N.A.M.E. Technical and Research Bulletins Nos. 3-11, "Recommended Practices for Preparing Marine Steam Plant Heat Balances", published November, 1961.

To give this matter absurdity, the Institute some years ago published in its Transactions a heat balance where the heat in the steam at the turbine inlet was greater than that at the boiler outlet. While this might commend itself to people who still thought perpetual motion was possible, it could hardly find favour with those who considered that heat balances provided a reliable mathematical guide on the fuel to be consumed by a new ship.

Commander Tyrrell was staggered by the work done by the authors in preparing the two papers, and pointed out that the electronic digital computer had long since been programmed to do heat balance calculations. It was therefore now possible to draw a number of curves relating to a standard thermodynamic cycle, and to adjust the fuel rates caused by any changes made to the standard cycle by simple addition or subtraction.

Mr. C. W. Hayes (Member) said that a point on which he wished to comment was that of integral versus separate thrust collars on turbine rotors. In the paper on *Canberra* which was read on the previous day, a remark was made in the discussion which he took to indicate a necessity for integral thrust collars. Mr. Bauer also made a reference to this point when he showed a sectional drawing of a turbine which had a separate thrust collar. Mr. Hayes's view was that the turbine shaft with a separate thrust collar was sound practice. There was certainly no need to excuse it. In fact, solid rotors should preferably be designed so that later in the life of the turbine a separate thrust collar could be fitted if required, because it was well known that repairs to turbine rotors were sometimes necessary.

The separate thrust collar allowed a free choice of materials. The demands of rotor stress and temperature influenced the selection of the rotor steel. A steel which met the requirements of high stress and temperature, as was now well known, might not be the best material for the turbine thrust collar. This was because some steels were less compatible with white metal than others and were liable to "pick-up."

Another point, already referred to at some length by a number of speakers, concerned the cleanliness of oil systems. The thrust bearing was particularly vulnerable to dirt because the clearance between the trailing edge of a tilting pad and the thrust collar was very small. Dirt particles collected at the trailing edge of the pad and damage occurred to the thrust pads or collar, sometimes to both at the same time, depending on the steel and the amount of lubrication.

Four other points on which he would like to make brief comment were as follows:

1) He was sure that all turbine builders welcomed continuous running of their turbines. The best way to make sure that a turbine ran properly was to run it continuously.
2) In his opinion, there was a need for more use of turbine supervisory equipment on marine turbines. This kind of equipment was extensively used in the power stations of the Central Electricity Generating Board for monitoring essential information, such as rotor position, both axial and radial. When a shipping company built a new class of vessel, the first ship in particular should be monitored for perhaps twelve months in order to get accurate and precise information regarding the performance of both the main machinery and important auxiliaries.
3) Mr. Bauer, in his paper, mentioned wet steam conditions. In turbines for operation with steam from nuclear reactors, use had been made of chromium plating on the surface of the rotor to protect it from the erosive effects of wet steam.
4) Finally, his view was that the lead given by the Royal Navy, in discussing in detail machinery design with manufacturers, could be followed with advantage in the Merchant Navy. The result of this policy meant that the machinery was not only designed to be efficient but maintenance was considered as well. This eliminated the "plumber's nightmare" referred to by a previous speaker.

Mr. F. D. Roberts congratulated the authors upon the very frank way in which they had recorded not only their achievements but their disappointments.

On page 396 of the paper by Commander Platt and Mr. Strachan there was reference to turbine defects in manufacture and service. Mr. Hayes had just mentioned turbine thrust bearing failures. In the paper it was stated that the cause had not been fully explained. A possible explanation was given as errors in the squareness of the thrust collar. He suggested that such errors should be considered in conjunction with errors in squareness of the thrust pads themselves. The use of a spherically seated thrust bearing would ensure that all the thrust pads were taking the axial thrust under all positions of the shaft, but, of course, the initial cost would be greater.

With regard to blade failures, mention was made in the paper of blade failures in the last row of the H.P. turbine. He wondered whether they could possibly have occurred due to insufficient area or obstruction in the exhaust from the cylinder causing an unsymmetrical pressure distribution downstream of the blade. The phenomenon of a pressure pattern in the exhaust casing was often even more pronounced in the L.P. cylinder and could produce harmonics which might coincide with some of the last blade frequencies.

He had been very interested in the remarks on the batching of blades on the last two L.P. stages, and he would like to ask the authors what was the relative stiffness, both circumferentially and axially, of the junction wires and the lacing wire so that the blades could act as if they were batched in large continuous groups and also reduce the danger from nodal diameter modes of vibration.

With regard to the question of cylinder distortion, Mr. Bauer had mentioned that it was the practice of some firms to provide astern turbines at both ends of a double flow cylinder in order to avoid unsymmetrical thermal conditions. He would like to hear the views of the authors in that respect.

With regard to the table on page 391 showing the comparison between British and Italian turbines, it was stated that for the same designed steam rate, the designed fuel rate of the Italian machines was five per cent better, whereas on actual trials the fuel rate was between five per cent and 11 per cent better. Dr. Brown had mentioned that there was probably a difference in the date of design of the turbines. Mr. Neumann had said that it might be due to boiler efficiency. He wondered whether it could be that the turbines themselves were anything up to six per cent more efficient.

It was also stated that on trials the fuel rates on the British ships covered a range of 11 per cent between minimum and maximum, whereas that range was reduced to five per cent on the Italian machines. He wondered whether that could mean that the Italian manufacture was confined within far closer limits. Improvements in manufacturing standards, closer tolerances and more rigorous inspection could be applied here but the cost of the product would necessarily be increased.

He agreed with the authors that the foreseeable gains in turbine efficiency were relatively small compared with those to be attained in the heat cycle of the complete installation. However, he would point out that experimental work was still producing results, particularly in the development of blading which could maintain a high degree of efficiency over the whole range of propeller speeds.

Correspondence

MR. F. A. MANNING wrote that Commander Platt and Mr. Strachan were to be congratulated on their very informative paper which undoubtedly would prove of great value to anyone contemplating steam machinery for a new tanker installation.

The authors' experience with the various turbine units and feed systems was very interesting and one small point needing comment concerned the ships with the system shown in Fig. 4, with an inlet water temperature of 197 deg. F. to the gland steam condenser. The associated condensate temperature was 90 deg. F. and related to 28·5in. Hg. vacuum for temperate waters. It seemed likely from the diagram that in the tropics the gland steam condenser coolant would exceed 212 deg. F. and the unit would fail to function as a condenser. Would the authors state whether this occurred in service?

It was noted the authors stated that failure to achieve design fuel rates in the 32,000-35,000-d.w.t. vessels was the fault of the manufacturing errors in blading.

Variation from design in first stage nozzle area was easily corrected by blanking off surplus area. Variations in downstream blading could hardly be attributed to poor performance unless the aerofoil section was deformed. In fact the whole basis of any standard turbine range was the ability of both impulse and reaction blading to accommodate itself to changes in steam mass flow, thus implying that blade opening variation was acceptable by design.

Excessive gland and reaction blade clearances naturally resulted in poor performance, but it was not to be expected that fuel rate variations, such as stated in the paper, were solely due to this source.

Throughout the paper the main theme seemed to be the improvement in efficiency with each successive class of vessel. The final conclusion that turbine machinery had increased in reliability and efficiency to a degree, not thought possible in 1950, did not appear to have been quite proven in the paper.

The measure of turbine efficiency was the turbine water rate. Fuel rate was an interesting factor and since it concerned operating costs it was of prime importance to a ship owner. However, this figure of fuel rate could almost be varied at will by the choice of terminal steam conditions and the ratio of auxiliary and hotel load to propulsion load, all of which were independent of the marine turbine designer's ingenuity.

In a study of the progress of turbine efficiency it seemed fairer to consider a pre-war unit of 13,000 s.h.p. with terminal steam conditions of 640lb./sq. in., 840 deg. F. and 28·5in. Hg and to compare its water rate with comparable units detailed in the paper.

The result was tabulated as follows:

Steam conditions 600lb./sq. in., 850 deg. F., 28·5in. Hg.

Date	1938	1957	1959
Power s.h.p.	13,000	14,000	16,000
Water rate lb./s.h.p. hr	6·29	6·20	6·15
	corrected to 600lb./sq. in.		

The improvement in efficiency was seen to be two per cent which was very hard won, considering the effort that had been applied by the marine industry throughout the world in the post-war years.

This figure was understandable when one considered where improvements could be made. Mechanical losses, normally about one per cent, had been reduced, as also had gearing losses which were nominally three per cent; improvement had also taken place in blading forms but blade and gland leakage reached a safe minimum 20 years earlier.

Turbine reliability was improving throughout the world, although from time to time, cases still occurred of the best designed turbines coming to grief owing to mal-operation.

The foregoing illustrated that the success and progressive improvement in overall plant efficiency was due to careful choice of feed cycle, auxiliary plant, and the selection of steam conditions consistent with the operators capability.

Did the authors agree that technical effort was best employed in improving reliability and cycle efficiency, and reducing manufacturing costs of the installation as a whole rather than seeking a very elusive further ½ per cent improvement in the main turbine units?

Mr. Manning also wrote that Mr. Bauer was to be thanked for his very informative paper on the development of geared turbines for tanker propulsion.

Referring to the 1930 units, the courage of the designers in producing a symmetrical H.P. turbine with its complicated assembly, could only be admired.

It was relevant that even today theoretical studies still contemplated designs without the heavy section horizontal joint flanges and their adverse distortion to the cylinders.

Would the author indicate whether these early units still retained the heavy flanges and also whether they were able to operate with fine gland clearances?

In the L.P. turbine of the 1930 era (Fig. 4) it was noticed that claw couplings were used. Was the service performance of the couplings satisfactory, particularly in relation to blade vibration and misalignment?

Referring to the 1960 units, it was noted that the so-called pressure equalizing holes were fitted in all H.P. wheels—even where partial admission occurred. Would the author agree that these holes served no useful purpose, increased stresses and also adversely affected the turbine efficiency?

Turbine thrust bearings always provoked very interesting discussions. In most impulse turbines today, the axial thrust was mainly consequent upon the piston load, due to the shaft diameter in way of the glands being less than the shaft interstage diameter.

Until recently the impression was that this change in diameter was essential, but on making a few calculations the following conclusion was formed:

For turbines in the 10,000-20,000 s.h.p. range, by making those two diameters equal, the loss of power due to extra gland steam leakage was offset by a reduction of high friction pressure in the thrust pads.

Since the oil film thickness was a function of pad mean pressure, greater reliability and increased protection against dirt contamination resulted; a minor advantage was a saving in gland labyrinth spares. The author's view as to the feasibility of this suggestion would be appreciated.

In conclusion the statements by the author with reference to commercial application of nuclear power were noted with interest and were in general supported by the facts available in U.K.

COMMANDER J. H. D. MIDDLETON, R.N. (Member) wrote that while he greatly enjoyed the admirable and most interesting paper by Commander Platt and Mr. Strachan, he felt that it only presented the fruit of much painstaking work and investigation which, if it could be revealed, would be even more interesting and informative to marine engineers.

The paper was confined, in the main, to the technical aspects of the problem, but the underlying economic aspects were at least equally important, and would be of even greater interest if only because so little informed matter had been published concerning them.

The fundamental problem was, presumably, the design of a machinery installation which would result in the lowest possible cost per ton/mile of cargo carried and discharged. This figure must comprise components representing, *inter alia,* capital cost, fuel costs, salaries and wages, and maintenance and overhaul costs. Presumably extrapolation from historical analysis in a large fleet would form the basis on which to rest a forecast, and it was such an analysis, together with a comparison of actual with predicted cost performance, which he felt would be of very great interest to all concerned with similar work.

It could be inferred from the paper that the fuel rate was of paramount importance, but he was doubtful if this was so, or so intended, since it was well known in land practice that reductions in capital cost could justify fuel rates significantly higher than the best potentially possible, especially when plant mean load factors were considerably below the design point. Again, it must be well known to many marine engineers that with some designs, service performance sometimes fell far short of trials results, perhaps because the operation of the plant at its maximum efficiency was too difficult for those in whose charge it must work, or because its maintenance requirements exceeded the facilities available.

The authors were undoubtedly well qualified to discuss such points and to illustrate their opinions from their experience. It might be that the firms which they represented would be reluctant to disclose such details of their working costs, but if they could be prevailed upon to do so, he was sure that a paper of very wide interest and importance could be written.

Mr. F. T. Randell (Associate Member), in a written contribution, began by congratulating Commander Platt and Mr. Strachan on a very interesting and unbiased paper: interesting to him because it gave an invaluable record of the thinking which had led to the present-day design of the 68,000-ton tanker; unbiased because it had left little unsaid.

In particular, the comparison between British and Italian/American machinery gave food for thought, although it would be interesting also to have a few more relevant facts concerning the relative efficiencies of the boiler, turbines, turbo-generator sets and feed pumps, in order that the major discrepancy could be pinpointed.

In particular, the summary of features leading to the superiority of the Italian-built vessels did require some comment, if only to arrange in order of importance their contribution to the efficiency of the cycle.

He would place high manufacturing standards, particularly in respect of turbine nozzle plates and blade paths, first and foremost. This had a direct bearing on the overall efficiency of the main turbine, a unit which after all played a major, if not the most important, part in the overall performance of the vessel.

Secondly, no-one would dispute that excessive margins on auxiliary equipment did waste power but the feed pump, for instance, must be designed for the maximum continuous rating of the boiler, and if this latter unit was grossly oversized in the first instance, there was very little that the feed pump manufacturer could do about it.

Another major loss was represented by the pressure drop between the superheater outlet and the H.P. turbine nozzle chest, which could, in some British equipment, be as much as 45lb./sq. in.

Again, the maximum use of L.P. bled steam in a properly designed heat balance was well understood, but the system designer was in the hands of the main turbine manufacturer who, in order to supply a basic standard turbine, could not offer bleed points for optimum system efficiency.

His final point concerned the condenser extraction pump. The use of cavitating extraction pumps in themselves had little or no bearing on the overall fuel consumption of the vessel. For a 20,000 s.h.p. installation, a cavitating main extraction pump would have a motor rating of 20 h.p. An extraction pump with a discharge controller, i.e. a controller fitted for the sole purpose of preventing cavitation at the pump suction, would at the most have another 2 h.p. due to the friction through the controller. Obviously the authors of this paper were aware of this point, and in mentioning the cavitating extraction pump it was felt that by association they were comparing two basically different types of feed systems, i.e. one with internal feed storage capacity, which used cavitating extraction pumps and another, with external feed storage capacity, which could not have cavitating extraction pumps.

In the case of the former, this, of course, was present day practice, whereby hot feed storage was retained in the de-aerator at the feed pump suction and was ready for immediate demand by the feed pump. This then allowed the extraction pumps feeding the de-aerator to be rated individually on the quantities of water which would be presented to their respective suction branches. On the other hand, the older type of feed system did not normally incorporate a de-aerator but even if it did, there was no feed storage between the de-aerator and feed pump. As such it was necessary for the main extraction pump to be capable of pumping the same quantity of water as the main feed pump. As one could not depend entirely on the necessary quantity being obtained by condensation of steam in the condenser, the difference had to be obtained through additional supplementing devices to the main condenser, i.e. from the main feed tank. The liquid level in the condenser being taken as the appropriate signal to supplement or otherwise, a self-regulating cavitating pump could not therefore be used. In this case, for a 20,000 s.h.p. installation, the main extraction pump would now have a 30 h.p. motor or even more. While this increase was still a very small percentage of the main propulsion power, it was appreciably more than was necessary and was, of course, associated with a much more complicated feed system.

Finally, reference to packaged machinery was most gratifying as this had been his company's policy for some considerable time. Apart from engine rooms where their packaged feed systems and evaporators had been fitted, it was their intention, in the case of tankers, to extend this to other sections of the machinery. It was felt that this aspect offered as much, if not more, scope for improvement than radically advanced super-performance machinery.

Mr. R. F. Rimmer, referring to the paper by Commander Platt and Mr. Strachan, wrote that during the discussion, Commander Tyrrell had made reference to the recently published S.N.A.M.E. Technical and Research Bulletin No. 3-11; "Recommended Practices for Preparing Marine Steam Power Heat Balances". Armed with this document, superheater outlet conditions of 600lb./sq. in. gauge, 850 deg. F. and a condenser vacuum of 28·5in. Hg. the writer had prepared flow diagrams comparable to those illustrated in Figs. 4, 7 and 10 of the paper for a normal shaft horsepower of 14,000. Those diagrams were shown in Figs. D.2, D.3 and D.4.

The final feed and combustion air temperatures quoted in Figs. 4, 7 and 10 had been retained as had the bleed point pressures. In all other respects the heat balances had been

Figure	Main turbine, N.B.W.R.	Boiler efficiency, per cent.	Designed fuel rate, lb./s.h.p.hr.
4	6·2	Not quoted	0·577*
D.2	6·2	88·3	0·547
7	Not quoted	Not quoted	0·571
D.3	6·2	88·4	0·531
10	6·2	Not quoted	0·572*
D.4	6·2	88·3	0·545

* Stated by Mr. G. R. Strachan during the discussion.

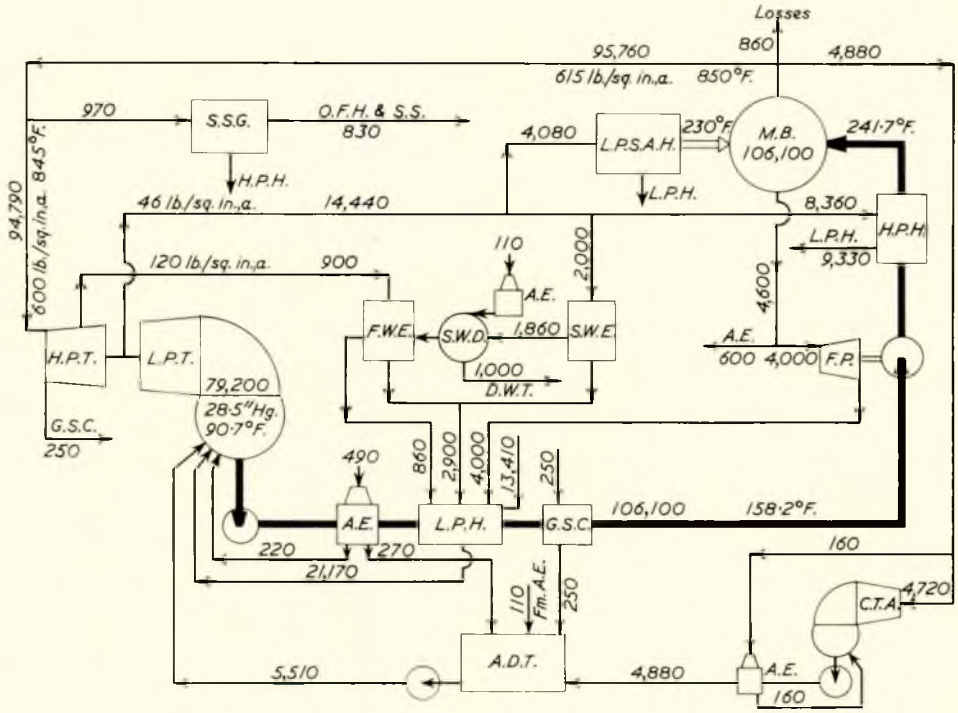

FIG. D.2

prepared from the data contained in Bulletin No. 3-11 with the assumption of a total complement of 62 and a funnel temperature of 320 deg. F.

Since the basic efficiencies of the components were the same and all allowances identical, the calculated fuel rates should give a reasonable assessment of the relative worth of the cycle employed. The preceding table summarized the main factors:

It was interesting to note that the designed fuel rate in Fig. D.2 was 5·2 per cent less than that in Fig. 4; that in Fig. D.3 it was 7 per cent less than in Fig. 7 and that in Fig. D.4 it was 4·7 per cent less than in Fig. 10. While some of

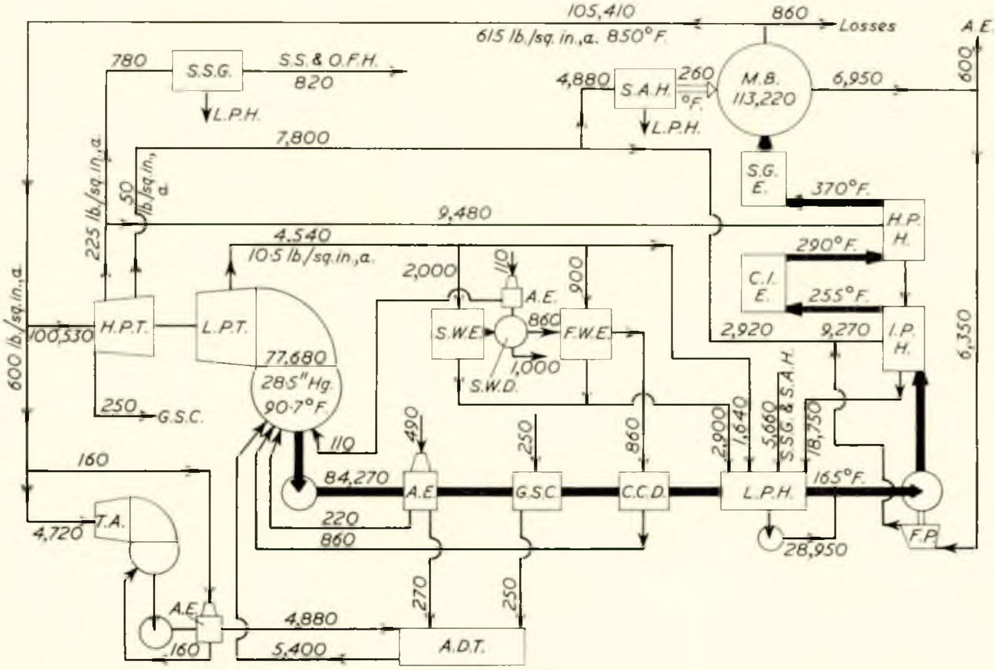

FIG. D.3

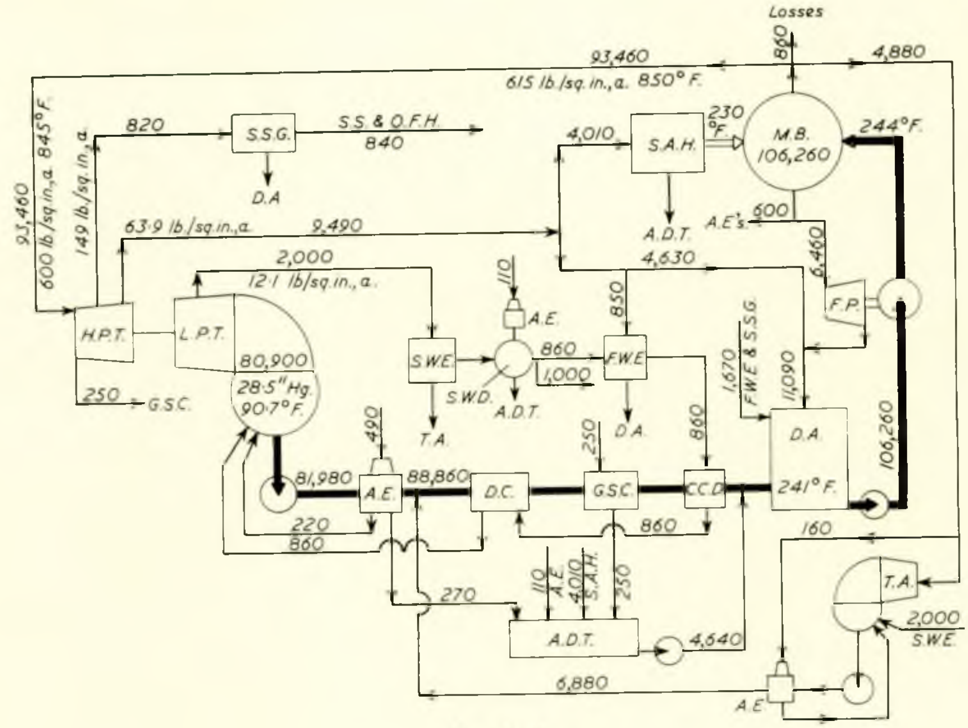

FIG. D.4

these differences could be accounted for in boiler efficiency and, in the case of Fig. 7, in main turbine performance, the main conclusion was that it arose from the different standards used in respect of losses, air ejector steam consumptions, evaporator loads, domestic steam loads, electrical loads, etc.

The futility of comparing cycles which might contain differences of such magnitude was obvious. Mr. Rimmer made a strong plea for the preparation of a standard code of practice along the lines of the S.N.A.M.E. Bulletin to be used in the preparation of marine heat balances and could think of nobody better suited to such a task than this Institute.

MR. G. R. STRACHAN, M.A. (Associate Member) wrote that he wished to comment on the section dealing with the considerations leading to the 68,000-d.w.t. vessels with which he himself had had the privilege of being closely concerned.

TABLE D.I.—COMPARISON OF COSTS AND CONSUMPTIONS FOR ECONOMIZER AND GAS AIR HEATER CYCLES.
68,000-D.W.T. CLASS

System	"As fitted" Split economizer	Rotary regenerative air heater			
Funnel temperature, deg. F.	317	317	297	277	257
Boiler efficiency, per cent. Fuel rate, lb./lb. of fuel	88·17 0·0689	87·64 0·0690	88·12 0·0686	88·60 0·0683	89·09 0·0679
Total feed to boilers, lb./hr.	148,710	149,150	149,150	149,150	149,150
Fuel consumption, tons/day Percentage difference	109·78 Basis	110·26 +0·36 per cent.	109·63 −0·14 per cent.	109·14 −0·58 per cent.	108·50 −1·17 per cent.
Fuel cost differential for 270 days per annum at £7 10 0 per ton	Basis	+£1,000	−£300	−£1,300	−£2,600
Initial costs of installations:					
Cast iron economizers	£13,000	—	—	—	—
Gas air heaters	—	£12,000	£12,500	£14,100	£16,600
Steam air heaters	£3,500	£500	£500	£500	£500
Feed heaters	£3,500	£6,100	£6,100	£6,100	£6,100
Additional for F.D. fans	—	£500	£500	£500	£500
Piping and valves difference	£1,000	—	—	—	—
Total	£21,000	£19,100	£19,600	£21,200	£23,700
Initial cost differential	Basis	−£1,900	−£1,400	+£200	+£2,700
15 per cent. capitalization and amortization	Basis	−£300	−£200	—	+£400
Overall annual saving or loss	Basis	£700 loss	£500 saving	£1,300 saving	£2,200 saving

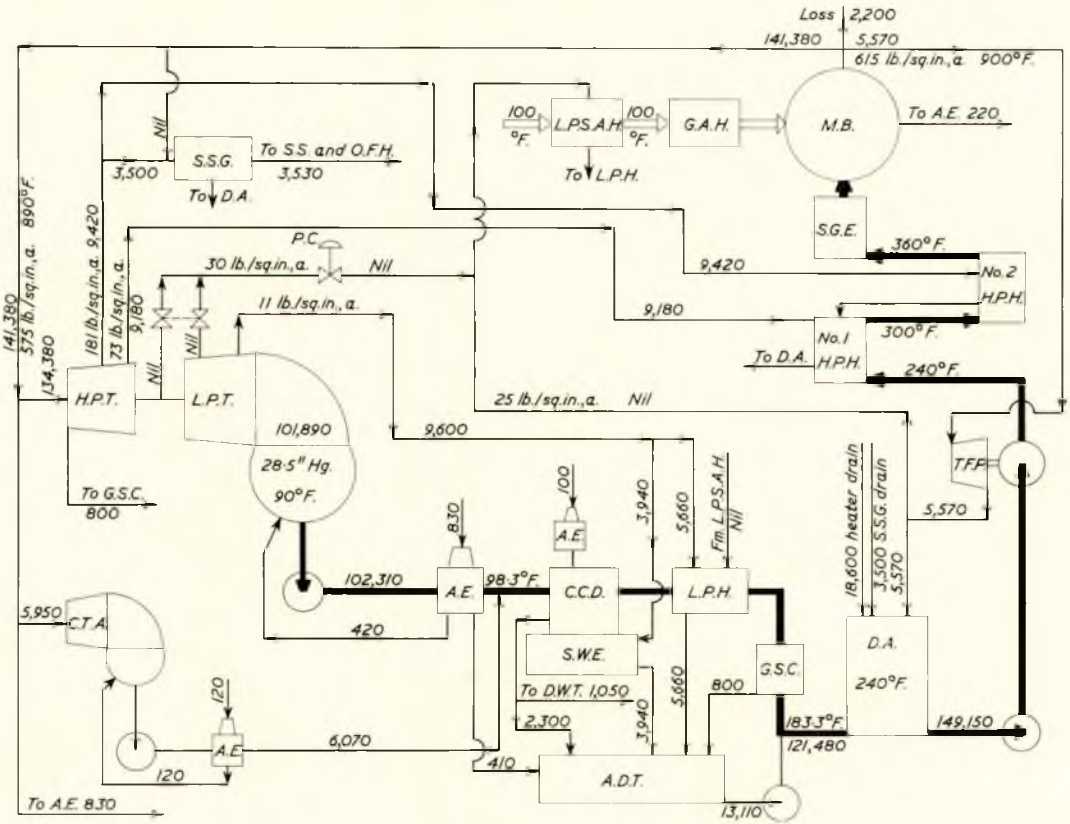

FIG. D.5

It could well be seen from the paper just how hard won were additional improvements in fuel consumption and thermodynamic efficiency. From the detailed work carried out in determining the optimum feed cycle it would be apparent that for machinery of this power and steam conditions the economic limit had now been reached in respect of the amount of regenerative feed heating that could be employed.

In addition, increases in steam pressure and temperature beyond the conditions employed today required considerable expense in research and development before they could be adopted generally throughout the marine industry.

It would be noted that on page 401 the question of gas air heating had been somewhat lightly dismissed as being inferior from overall consideration of first costs and fuel consumption to a system incorporating split economizers and bled steam air heaters. It should be borne in mind however that since this design study was carried out there had been considerable advances both in the design of gas air heaters from corrosion aspects and also in associated artifices for counteracting corrosion and chokage due to prolonged steaming at reduced powers.

It would now appear that the one major factor remaining which would show a reasonable economic improvement for a comparatively small expenditure would be that of reducing the funnel temperature.

Table D.I showed comparative fuel consumptions and costs using a rotary regenerative type of gas air heater in place of the cast iron section of the economizer.

The basis for the comparison had been taken as the heat balance for the 20,000 s.h.p. class as shown in Fig. 17. Figures of cost and fuel consumption were given for various sizes of rotary regenerative air heaters for a range of funnel temperatures from 317 deg. F. down to 257 deg. F. The feed cycle and heat balance was as shown in Fig. D.5 from which it would seen that the changes required from Fig. 17 were as follows:

a) Gas air heater replaces cast iron economizer.
b) The two H.P. feed heaters which were arranged in parallel are replaced by two heaters in series.
c) The first of these H.P. heaters is supplied with steam bled from an additional point in the H.P. turbine at 73lb./sq. in. abs.
d) The two-stage steam air heaters are replaced by a small single-stage steam air heater used only at low boiler evaporation.
e) The electrical load is increased by 10 kW. due to the requirement for additional fan power necessary to overcome air heater leakage and slightly higher overall system resistance.

It would be seen that at a funnel temperature between 317 and 297 deg. F. the systems were of equal merit financially. It was when the funnel temperature was reduced below 297 deg. F. that the advantages became apparent and whilst it was not suggested that the funnel temperature should be designed as low as 257 deg. F. immediately, there appeared to be a good case for reducing it at least to between 270 and 280 deg. F.

Problems formerly associated with corrosion and chokage due to prolonged operation at low powers could then be overcome firstly by the employment of enamel or special steels and epoxy resin coated elements and secondly by the inclusion of the small steam air heater.

Such an arrangement could be incorporated so that the size of the gas air heater was not prejudiced, as through the medium of automatic control, steam air heating would only be employed at low boiler loads when the funnel temperature fell to some preset figure.

It would be most interesting to have the authors' comments on this suggestion and to learn whether they considered that a rotary regenerative type of air heater could now be incorporated in a tanker installation of the 20,000 s.h.p. class.

Authors' Replies

Reply by COMMANDER PLATT and MR. STRACHAN

The authors expressed their gratification at the volume of discussion and the interest shown in the paper. Several interesting points had been brought out in the discussion and they would endeavour to answer them as fully as possible.

The authors agreed with Mr. Armstrong that too little attention had been paid in the past to fitting out procedures. In recent years considerable thought had been given to this problem and it might be of interest to mention some current practices which it was hoped would show that attention was being given to improving fitting out conditions.

Steam and lubricating oil pipes after pickling and water testing were dried with hot air and sealed with tape or blanks. During erection on board it was an offence at any time to leave an exposed end to a pipe. Orders to sub-contractors contain instructions to send their supplies fully protected. When this had not been done photographs had been sent back to firms, with excellent results, and it was fair to say that the standard was now good.

Main and auxiliary steam piping above 3in. diameter was now butt welded and the installation planned to allow large lengths to be installed leaving a minimum of closures which might be flanged or site welded.

Cheap and simple scale models ½in. to the foot were now employed to give proper evaluation of usage of space and suitability of main pipe leads, platforms, gratings and ventilation trunks.

Packaged units had been referred to in the paper. Attempts were now made to mount and pipe duplicate auxiliaries with their inter-connexions as a "shore job" rather than on board.

To facilitate fitting out, the practice for the past ten years had been to leave access holes in the side of the ship, which were not closed until a few days before sailing. In many cases machinery was shipped before launch.

Mr. Armstrong and Mr. Hayes both attributed turbine thrust failures to be due mainly to dirt in the lubricating oil and had suggested a certain amount of disinterest in obtaining a clean lubricating oil supply. No one was more aware than the engine builder of the necessity for removing all extraneous matter from the lubricating oil system before both shop and sea trials. A rigorous procedure was laid down whereby oil was circulated for at least a week before commencement of shop trials. A temporary combined valve and filter fitted at the inlet to each bearing or set of sprayers allowed complete effectiveness of the circulating period, each filter being cleaned in turn without stopping the circulation. These filters were only removed immediately before the sea trials.

In a recent naval installation the gearing was assembled inside a polythene cubicle with a sliding roof kept closed at all times except when lifting apparatus was in use. After shop trials it was observed that the journals and bearings were in a most satisfactory condition compared with previous contracts constructed in open air conditions. The stage may now also have been reached when the use of cast iron in lubricating oil systems should be reconsidered.

The authors agreed with the practice of avoiding dumping exhaust steam to the main condenser although the problem was more acute with a back pressure system where the quantity of dumped exhaust was larger.

Radiation shields had been used on both merchant and naval turbines but generally only in single casing designs where it was agreed that it helped to maintain a more even temperature of the cylinder. In double casing designs where the main distortion came from the outer casing, it was felt that the effect of such a shield would be small.

The authors had no experience of distortion in H.P. turbines. They were aware of the cases to which Mr. Armstrong referred and during the full scale testing of the 16,000 s.h.p. set at Pametrada efforts were made to reproduce this form of distortion without anything untoward occurring. They now felt confident, therefore, that the changes of detail design, particularly in the use of radial locating keys, had overcome this problem.

The authors would comment further on pre-commissioning cleaning of boilers in the reply to Captain Raper but would agree with Mr. Armstrong that the first attack was to avoid putting the dirt into the boilers.

Mr. Armstrong had criticized the figures in Table III. Although the authors had endeavoured to present the facts without bias Table III could not tell the whole story. Comparison of fuel costs and capital costs were not the total saving, which could only be assessed after setting out all the economic factors involved. As Commander Middleton had pointed out, the object was to carry oil at the lowest cost per ton/mile, which involved working out the complete freighting calculation for the ship. This, of course, was outside the scope of this paper. Suffice it to say that substantial hidden economies in the selected cycle were due to the carriage of more cargo at lower cost than any of the alternatives considered.

The suggestion that the high efficiency turbo-alternator would not run continuously and would therefore result in a heavy fuel loss if the low efficiency set had to be run was inconsistent with Mr. Armstrong's own proposal, echoed by Mr. Kaudern, to install one turbo set only and rely upon a full capacity Diesel set as standby. This alternative had been considered but despite its attraction was rejected on the grounds that logically one must assume the worst case, i.e. that sufficient Diesel oil should be carried to supply the Diesel set for a complete round voyage. This resulted in loss of deadweight, which was unacceptable.

Mr. Armstrong also asked whether Table I could be read with Figs. 16 and 17. Fig. 16 may be read with Table I but Fig. 17 was not exactly comparable; the Pametrada Standard Frame design was introduced after the study on which Table I was based was completed. The revised figures given in Table VI for the number of turbine stages, speed of rotation, gearing and turbine water rate applied to Fig. 17.

The figures given in Table III all referred to the split economizer system. System 4 represented the "as fitted" cycle for the 68,000 d.w.t. class. Table I was not therefore comparable with Table III but the costs in Table I included two high efficiency turbo-alternators. Other alternatives such as a single casing H.P. turbine for Proposals A and B had not been presented in order to limit the number of possible permutations.

Mr. Armstrong had taken the authors to task over their use of the term "sophisticated". While acknowledging that

Mr. Armstrong's argument was etymologically correct, they contended that words changed their meaning with time. While the word objected to came into common usage with the meaning "worldliwise" or "artificial" it was commonly today taken by engineers to mean "refined" or "advanced". That such a meaning was now applied to a word which originally meant "corrupted" or "falsified" was perhaps a reflection on our society, but the subject was hardly appropriate to the present discussion. It was perhaps worth recording that originally an "engine" was an instrument of war though that meaning was seldom applied to the word today.

The term "packaged unit" did not imply that one was forced to buy a standard set of units but rather that each unit would be optimized and then packaged. It could be agreed that in tankers, where adequate space was available, a packaged unit tended to create wide open spaces, but there was a cost saving in installation time. The cost of valves and piping in machinery installations today equated with that of the main engines and gearing and a careful study of all arrangements was essential. The authors welcomed Mr. Randell's statement that more types of packaged unit were to be put on the market.

Dr. Brown had suggested that as the steam rates of the Italian and British designs were identical it was unfair to compare fuel rates where cycles and auxiliaries differed. The point which the authors wished to bring out, however, was that good design could be nullified by inadequate attention to manufacture and to design margins. Admiral Given's comment on the heterogeneous nature of the British marine industry undoubtedly was relevant here but in the authors' view the primary factors affecting the improved performance of the Italian-built ships were improved manufacturing techniques, closer inspection and reduction of unnecessary margins. The authors would like to emphasize once again, however, that they had no intention of implying that the Italian-American turbine design itself was any more efficient than their U.K. counterparts.

Incorrect figures in the table on page 391 and in the heat balance in Fig. 10 had created some confusion, for which the authors wished to apologise. The design fuel rate for the British installation should have been given as 0·577lb./s.h.p./hr. (as stated correctly in Table VI). The heat balance was confused by the fact that the Italians also appeared to lack a standard approach and in their case they over-estimated the total flow. In practice the ships improved on design and the total flow at the design service power was 104,000lb./hr. This resulted in a lower fuel rate since the boilers in both the Italian and British ships were in fact identical.

When carefully analysed the superiority of the Italian design was basically due to the feed system and illustrated, as stated on page 391 of the paper, the importance of applying equal attention to all details of the installation so that gains made in one feature were not offset by a lack of attention to another.

The authors were glad that Dr. Brown had drawn attention to Figs. 19 and 20. At a time when particular turbine designs had come under heavy but usually ill-informed criticism, it was well to take stock and to show what had indeed been achieved.

Mr. Parodi's comments on superheat control referred only to the E.S.D. Mark I type boiler and the authors would agree with his conclusions provided that the maximum evaporation included a large margin over the normal service power. Where, however, this margin had been reduced to a minimum a condition could arise where all combustion air heating was effected in the air attemperator and bled steam extraction was reduced, leading to an overall fall in thermodynamic efficiency, even though the boiler efficiency was unaffected.

Mr. Parodi was unable to find the calculated gain in fuel consumption between Fig. 16 and Fig. 17 and after some assumptions had estimated this to be 0·5 per cent. With respective boiler efficiencies of 87·94 per cent and 88·17 per cent based on a funnel temperature of 317 deg. F. for both systems the design fuel consumptions were 0·525lb./s.h.p./hr. for Fig. 16 and 0·512lb./s.h.p./hr. for Fig. 17, giving an improvement of some 2½ per cent in favour of the split economizer system.

In the tube failures referred to there was no evidence that the flame actually struck the tube metal but it was believed that where the combustion chamber was small local hot spots could occur due to the proximity of the flame and the small clearances. New designs of burner enabled the oil to be burnt in a smaller volume so, assuming the combustion chamber volume was unchanged, flame clearances were increased. The result must be beneficial but there was no evidence as yet that these new types of burner had any significant effect on tube failures of this kind.

The authors were interested in Mr. Kaudern's brief description of a 15,000 s.h.p. installation built by his company in Sweden, which was of similar type to that installed in the 50,000 d.w.t. class.

The choice of alternator size required considerable study and co-operation between builder and owner. Too often generator capacity had been too generous and considerable gains could be made by pruning in this direction. The authors would agree, as stated in the paper, that where electrical loading was low and the turbo-alternator efficiency was high this gave a good heat balance. In practice, however, if the owner had an expressed preference for electric galleys and if a high air conditioning load was to be expected, then the average electrical load would be high, the size of turbo-alternators would be increased, and the back pressure arrangement would show to disadvantage.

The alternative of one high efficiency and one low efficiency back pressure alternator would still show to disadvantage as compared with condensing turbo-alternators in the 68,000 d.w.t. class when the complete freighting calculation was carried out.

The measurement of horsepower on trials was normally carried out by means of an integral type electric torsionmeter on a calibrated shaft giving an accuracy of ±2 per cent.

The authors were inclined to agree with Admiral Given that the heterogeneous nature of the British marine turbine industry was not helpful; he would no doubt approve of the recent steps to unify design in the set-up of the new Pametrada.

Rivalry between steam and Diesel protagonists would undoubtedly continue unabated in the future. Each type of propulsion had its application and it was the duty of the owner's superintendent to adopt that most suited to the size and trade of the projected ship. In the higher power field Diesel engines were now in a position to compete with steam turbines and a fine choice frequently resulted. It was in this context that the authors had stated that they expected Diesel and steam to advance side by side, although in the period reviewed by the paper the turbine had on the whole maintained a slight lead in the higher power range.

Captain Raper had read more into the reference to inanimate objects than was intended. It was merely stated that additional complication in the form of feed heaters, etc. did not reduce reliability since these units functioned without manual intervention. The authors agreed that additional complication in the form of rotating auxiliaries increased repair costs due to wear and tear and the possibility of breakdown.

The economic justification of the complex feed system had been discussed in the reply to Mr. Armstrong.

The type of internal corrosion referred to in the paper would appear superficially to be similar to "scab pitting" in naval boilers but it was postulated here that the chemistry was different. These failures were confined to the hottest parts of those tubes, subjected to the most intense radiant heat and the incidence of "barnacles" was therefore highly localized.

The boiler water conditions suggested by Captain Raper were generally similar to those adopted in the Company's fleet and did in fact eliminate internal corrosion once clean conditions had been established. In this connexion the more frequent cleaning of naval boilers undoubtedly assisted. It had been shown that once clean conditions in boilers and steam and condensate systems had been established correct boiler water conditioning could enable boilers to steam for up to two years without cleaning, although annual cleaning was normal practice. The authors entirely agreed with Captain Raper's conclusions regarding flame clearances, boiler commissioning and maintenance of correct water conditions.

Captain D'Arcy had also raised the question of tube failures. As far as the authors were aware, the ships to which he referred were cargo or cargo/passenger ships in which boiler cleaning was probably carried out more frequently than in a tanker whose boilers were expected and required to operate for 350 days per annum without shut-down. Thus, as in naval vessels, even if the system was not entirely clean before commissioning, the boilers were probably cleaned before damage occurred. In tankers this could only be done by cleaning, preferably by chemical means, before the vessel entered service.

Captain D'Arcy had rightly drawn attention to the fundamental changes in H.P. turbine design between the 16,000 s.h.p. design with which his Company was associated and that for the 20,000 s.h.p. 68,000 ton design.

The evolution of this new turbine design was influenced by the concept of a turbine of sufficient flexibility to cover a range of power from 15,000 to 25,000 s.h.p. and a range of steam conditions to suit individual owners' requirements. With appropriate L.P. turbines the H.P. turbine design described fulfilled this requirement.

Secondly, the 16,000 s.h.p. design was built round a simple feed cycle whereas present-day thinking and the advent of automatic controls permitted of more complicated cycles being adopted.

Captain Jenks had given some useful thoughts on the application of control engineering to future trends. The authors would have expanded on this theme in the paper had space been available but now commented on the points raised.

It was felt that one should not take too gloomy a view of the calibre of seagoing engineers of the future. It was true that we were faced with an acute shortage of men of the right type and this shortage was likely to continue. There were two ways of finding a solution to this problem. The first had already been tackled and the marine industry as a whole and the Institute of Marine Engineers in particular, had done a great deal to develop alternative training schemes for engineers. The shipping companies had developed engineer apprentice training schemes which were producing young engineers of high practical and academic standard. Allowing for inevitable wastage one expected to retain enough of these young men to provide a few high quality engineers competent to take charge of future types of machinery.

On the other hand it must be accepted that engine rooms of the future should be designed to be operated by men of their more sophisticated (*pace* Mr. Armstrong) mentality. Thus they looked not to gradual diminution of standards among engineers, as the more gloomy prognosticate, but to an eventual raising of the standard accompanied inevitably by a reduction of numbers. It was here that they expected control engineering to enter into its own.

Captain Jenks had stated the problem correctly in saying that a fundamental approach was required rather than to add "automation" to existing systems. His comments on the design of systems, reduction of cross connexions, maintenance by replacement, etc. were in line with the authors' views.

The important point to remember was that control engineering could be made reliable only by rendering each control system simple and by careful study of the characteristics of each component in the system. The actual number of systems was unimportant from the point of view of reliability but naturally affected the economics.

It was hoped that a considerable degree of standardization and rationalization in steam plants as suggested by Captain Jenks would ultimately be achieved by the industry employing the facilities of Pametrada in this field.

Replying to Mr. Hutchings the authors confirmed that the comparison given was between E.S.D. Marks I and II boilers and would not necessarily apply to boilers of other makes.

In their comments on page 408 the authors had intended to make it clear that a 280 deg. F. feed temperature was attractive only if a mild steel economizer was employed, giving a simple system with good boiler efficiency at lower cost than the split economizer arrangement. Its disadvantage lay in the possibility of corrosion of the steel surface since there was very little margin to ensure that the gas temperature would not fall below the dew point at low loads. For this reason the split economizer system with 240 deg. F. feed and cast iron surface in the low temperature zone was adopted for the 68,000 tonners.

It was undoubtedly true that the measures taken to overcome tube failures had been sufficiently successful to enable a further increase in boiler pressure to be contemplated with confidence.

The authors agreed with Commander Lake that the less frequent cleaning of tanker boilers resulted in accumulation of normal corrosion products in boilers but modern methods of feed treatment, notably the use of volatile amines or hydrazine, could arrest this corrosion process. They concurred in his comments on boiler ratings and believed that increase in combustion space and radiant surface could be achieved at comparatively little cost.

Replying to Mr. Neumann the authors felt that there would appear to be little to choose between high level and low level deaerators, as the extra cost of the additional pumps for the low level system equated with the extra cost of supporting steel work for the high level system.

The use of Diesel driven alternators had been discussed in reply to Mr. Armstrong as had also the discrepancies in the heat balance between British and Italian installations.

The authors were indebted to Mr. Neumann's remarks on the additional horsepower required for circulating large bore condenser tubes.

The authors agreed with Mr. Culver that the adoption of remote operated air registers would enable fires to be spread more uniformly over the furnace and full advantage should be taken of this in future boiler designs.

The authors felt that with current designs of turbines, both of U.K. and foreign manufacture, the requirements for the control of temperature when manœuvring might shortly become a thing of the past. What would then be required was a boiler with a sensibly flat superheat characteristic from about 40 per cent full load onwards.

In reply to Commander Tyrrell, whilst there was much to be said in favour of authoritative guidance in the compilation of marine heat balances, the authors would not go so far as to advocate the incorporation of a set of regulations drawn up in another country, since the relevant conditions might be quite unsuited to British owners' requirements. The best practice was for the shipowner to advise the engine designer of the relevant factors such as losses, make up feed requirements, steam heating requirements, etc. from his service records so that the engine builder might then submit his heat balance and guaranteed fuel consumption in confidence that the figures quoted could be obtained.

Replying to Mr. Hayes, research work into thrust failures, while considering the effect of incompatible rotor materials, had shown that under loading conditions greatly in excess of design and even under misalignment the designs were adequate. The possibility of collection of dirt particles at the trailing edge of the thrust pads existed but the general indications showed that the failures reported had occurred due to excessive transient loads of a nature which had not yet been determined.

It was agreed that the use of supervisory equipment on the turbines would tend to grow, particularly as control engineering advanced.

The authors' experience confirmed that the practice of discussing machinery design in detail with the manufacturers was extremely valuable and was to be encouraged.

In answer to Mr. Roberts, as stated in the reply to Mr. Hayes, experimental work indicated that thrust bearings would operate satisfactorily even when a considerable amount of misalignment existed. However, they agreed that the extra cost of the spherically seated thrust bearing would not be great and this type of bearing would provide additional security.

As stated earlier in the reply, there was no difference in the designed efficiency between the British and Italian turbines.

Replying to Mr. Manning, as previously stated, the authors agreed that the measure of turbine efficiency was the water rate.

The authors point was that further advances in the turbine efficiency were hard to win and that the best effort was now applied to reliability, cost and cycle efficiency.

The authors sympathized with Commander Middleton's plea for more information to enable the machinery to be designed at the lowest possible cost per ton/mile. Such information would involve a lengthy paper in itself and was outside the scope of the present discussion. However, to those interested the authors recommended reference to MacMillan and Ireland's paper "The Economic Selection of Steam Conditions" in which the various factors involved were clearly described.

The table provided by Mr. Rimmer in which he had recalculated the 14,000 s.h.p. heat balances (Figs. 4, 7 and 10) on the basis of the S.N.A.M.E. recommendations, illustrated the point made in the reply to Commander Tyrrell. It would be seen that the originally quoted fuel consumptions had been improved between 4·7 and 7 per cent, whereas the same heat balances calculated on the basis of owners' records would provide a more reliable guide to the performance which could be expected from the ships in service.

The authors would agree with the order of importance in which Mr. Randell had listed the factors involved in a comparison between British and Italian turbines. Most of the points raised by Mr. Randell had already been dealt with in reply to other speakers. On cavitating extraction pumps the main advantage would appear to be in the reduction of surges in the system, since the cavitating pump was inherently self-regulating and lent itself to better control.

The authors were indebted to Mr. G. R. Strachan for Table D.I. which gave a useful comparison between the split economizer and the gas/air heater cycles. As mentioned in the reply to Mr. Hutchings, it was accepted that advances had been made in air-heater design and the air heater cycle might well prove attractive for future installations provided that the corrosion problem could be overcome. Past experience indicated, however, that a reduction of funnel temperature below 310 deg. F. could lead to appreciable maintenance difficulties and for this reason it was stated in the paper that the Company had retained the use of cast iron economizers as a prime design consideration.

Reply by Mr. Bauer

Mr. Bauer, who said that he was very grateful for the interest shown by all the contributors to the discussion, replied first to Mr. Armstrong.

In the author's experience, his company had not suffered from turbine casing distortion except in the initial development of some rather lightweight naval turbines and, even in these, the distortion had not resulted in rubbing of the blading. These were double flow turbines of a symmetrical design having an astern turbine on each end which, in their opinion, were easier to deal with from the standpoint of temperature distortion and were a much preferred design. The difficulties were easily overcome by properly divorcing the inner cylinder from the outer cylinder. The company had had no experience with distortions in double flow turbines having an astern turbine on only one end.

Insofar as the location of the nozzle control valves and steam inlet was concerned, removal of the nozzle valves was not required when opening up and, depending upon the piping layout, the main steam line could usually be left in position.

The use of tilting pad thrust on the primary wheel had been a logical development with the increased power of the 1960 era units. While the author's company had used both the solid coupled and flexibly coupled quill, the flexibly coupled quill had distinct advantages insofar as the alignment of the gear train was concerned.

It had been the company's practice to utilize separate thrust collars in turbine design to permit freedom in the choice of collar materials as well as to permit replacement if the collar was damaged.

The manœuvring valve could be tested under steam while under way with only a momentary loss of revolutions, with the precaution of closing the valve manually and re-opening by hand to prevent too sudden a flow of steam from the boilers.

The fitting of the main thrust without journal bearings permitted the thrust to be mounted adjacent to the gear and the journal bearings to be at the optimum location for the proper distribution of bearing loadings on both the second reduction gear wheel and line shafting.

It had not been the practice of the author's company to insist on any specific flushing procedure or class of filtration, but to develop with the shipbuilder in each case, a prudent procedure consistent with the shipbuilder's practice and the protection of the main unit. Mr. Armstrong's thoughts with respect to precommissioning cleanliness in the oil system were fully shared by the author.

The author had known and respected Dr. Brown for many years and was very proud of the fact. Dr. Brown said that he liked a turbine which just rubbed. The author ventured to suggest that in an impulse turbine the only places where close clearances would be important to good performance would be in the interstage and gland packing and at the top of the bucket, but here only in a radial direction and certainly not in an axial direction.

Dr. Brown in his contribution had referred to the comparison which Commander Platt had made between the British and Italian ships, bringing out two points: first, the date of the design of the respective units and, second, that it should not be thought that the turbine made all the difference in the fuel rate, as certain differences in the cycle might also account for the difference in the fuel rate. It was the author's opinion that the date of the basic design was nowhere near as greatly different as Dr. Brown had indicated, as the author's company had built the basic design of Commander Platt's Italian ships as far back as 1950. The author believed that there was not enough difference in the cycles employed between the British and Italian ships to account for the obvious difference in fuel rate. It should be noted that the maximum fuel rate for the Italian ships of 0·565lb./s.h.p. hr. had only been shown on one of the ships and that all others had performed well below the expected and designed fuel rate. A comparison between the designed and actual fuel rate given in Commander Platt's paper would be of considerable interest.

In regard to Dr. Brown's other comments, Mr. Bauer stated that panting plates (author's flexible plates) had been found eminently satisfactory and had been used over many years. They had not been found to create any difficulties with respect to resisting pipe thrusts and they did not require particularly sensitive treatment during installation by the shipyard.

As Dr. Brown surmised, the elongation of the turbine foot was only done to simplify the shipbuilder's foundation on the forward end of the unit.

With respect to bearing design, De Laval used length/diameter ratios in the range mentioned by Dr. Brown if permitted by specifications, which frequently were so conservative as to prohibit the use of modern bearings. Extremely little trouble had been experienced over the past ten years with the flexible couplings used by the company because of the greater attention paid to materials, accuracy and lubrication.

It had been found, through very satisfactory experience, that the arrangement of the astern turbine within the low pressure turbine worked out very well. The company had had extensive experience with astern turbines of this design operating at 950 deg. F. without attemperation of any kind. The unit shown in Fig. 15, where the astern turbine consisted of one double and one single row wheel had given perfectly satisfactory astern power and manœuvrability for a large number of vessels, operated to the demands of a number of different owners. The astern power in this arrangement was transmitted through the low pressure first reduction pinion which, due to its lower speed and therefore larger diameter, was less highly loaded than would be the high pressure first reduction pinion.

The balancing of turbine rotors after the original installation was an extremely rare occurrence but had been carried out successfully on shipboard.

The low pressure turbine was fitted with a thrust bearing at the inlet to the astern turbine, as this was subjected to higher changes of temperature than the inlet to the low pressure turbine.

The author's company had found the bar lift nozzle valve arrangement extremely satisfactory. At low powers the valves were kept tight by their own weight and the very considerable pressure differential, without injury to the sealing surfaces due to any fine pieces of foreign matter which might occur with direct mechanical closure.

Regarding future steam conditions, the author wished to refer Dr. Brown to page 430 of his paper under the heading of "Future Trends", as it seemed that Dr. Brown's opinion was not much at variance with that of the author.

Mr. Kaudern, who represented a well known Swedish company, had said a few kind words about back pressure turbine generator sets. The author's company had studied the application of back pressure turbine generators, as referred to on page 428 of the paper, but was not enthusiastic about it as yet. However, any comments made there need not detract from the fact that the author thought that Mr. Kaudern was using good turbines at generally acceptable steam rates.

The author was particularly pleased by the kind remarks of Mr. Derdini which indicated that Mr. Derdini's firm had used the designs of the author's company with considerable success. He also noted their success with condenser tubes expanded in both sheets.

Mr. Bauer was pleased with the comments given by Rear-Admiral Given, and he had already replied to his comments regarding turbine distortion. With respect to the pivoted pad type of thrust bearing, it was perfectly true that this type of bearing lent itself to measuring thrust. This had been done on many occasions in the propeller thrust and there was probably no reason why this could not be done on the intermediate speed elements.

With respect to the comments by Captain Jenks on Commander Platt's paper, Mr. Bauer wished to state that he fully agreed that the designer should always keep in mind not only the conditions under which the equipment would have to be operated, but also the available personnel required to operate, maintain and service it.

He agreed in the scope of the areas for improvement outlined by Captain Jenks; that is, the basic concept of the plant as a whole, the main units, and the auxiliaries. This was the approach being followed by the author's company with endeavours in automation as well as in the design of a more austere power plant, both of which were being given continuous attention.

With respect to the comments by Mr. Culver, the author had already stated that there was no attemperation of steam for astern running nor was it required for emergency steaming; that is, operation on either the high pressure or low pressure turbine alone.

In answer to the second point brought up by Mr. Culver, the author was not certain that he was fully familiar with all the various studies on automated or austerity ships which might be in progress at the present time in the U.S.A. However, the studies on automated ships with which the author was familiar began with the basic concept of the cycle followed by a choice of, possibly but not necessarily, the redesigned components and a thorough review of the design parameters, number of components and the arrangement thereof, in order to obtain the optimum result for a given task in accordance with the present state of the art.

The author was grateful to Mr. Hayes for his comments about thrust bearings, with which he thoroughly agreed. The reason for his apologetic manner in that respect was that he still felt himself to be a guest in the United Kingdom and he did not like to be critical of practices which had been successfully employed over many years. They were perfectly good and perfectly proper; however, they were different practices. Actually, it was American practice to have a separate thrust collar when it was easily applied, such as with the propeller thrust being arranged forward. As to Fig. 16, this thrust bearing was arranged aft and consisted of a stub shaft with an integrally forged collar. Both his own company and its United States competitors had thrust collars which were separate, for about the same reason as Mr. Hayes brought out, and they had been very successful.

Regarding protection against erosion from the rather adverse steam conditions in the n.s. *Savannah,* this was dealt with by embodying in the diaphragm design a rather effective water trap or drain and by fitting a moisture separator between the high and low pressure turbine. Furthermore, the usual protection of stellite strips fastened to the last rows of the low pressure turbine, which was standard practice in the turbines manufactured by the author's company, of course was followed in the n.s. *Savannah.* His company had not gone to the length of chrome plating the rotor.

A number of points made by Mr. Roberts had been answered in connexion with comments by other contributors. However, the author was particularly interested in the statements Mr. Roberts made with respect to the possibility that, in the comparison between the British ships and the Italian ships made by Commander Platt, the main turbines might have been a contributory factor to the better fuel rate shown by the Italian ships.

In reply to the written contribution of Mr. Manning, the author wished to state that the design of joint flanges for modern steam conditions were, of course, somewhat different to the requirements for the 1930 units; and that higher pressures today also resulted in comparatively heavy sections of horizontal joint flanges accompanied by adequate bolting. In his experience, the joint flange design, if properly carried out, did not seem to have much effect on casing distortion—from which, incidentally, his company was fortunately free today, as in the past.

With respect to claw couplings, their performance was satisfactory but their cost was prohibitive; hence, the introduction of the fine tooth type coupling.

The author could not agree with Mr. Manning's query regarding the use of pressure equalizing holes in the H.P. rotor discs. These had been found to substantially eliminate excessive thrust forces and, when properly carried out, they did not result in excessive stress concentrations. They had no adverse effect on the efficiency of an impulse type high pressure turbine.

In general, the author referred to various comments made throughout the discussion of the two papers which had to do with comparisons between steam and Diesel, and comparisons between one type of design and another. He expressed the opinion that, in order to make a fair and equitable comparison, it was necessary to be in possession of all the facts. This, it had been found, was extremely difficult in a complex subject such as a complete marine power plant. It seemed to the author that both Commander Platt and he had endeavoured to point out the fact that the steam turbine still had a future and probably would hold its own, compared with other prime movers, for a long time to come. This was true because of its generally inherent potential for future development, which would make it even more efficient, reliable and able to be installed, operated and maintained with a minimum of manpower.