THE MARINE POWER PLANT

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The MARINE POWER PLANT

BY

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THE MARINE POWER PLANT

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To the Memory of My Son

ENSIGN CHARLES EDMUND CHAPMAN, USNR

LOST IN THE BATTLE OF MIDWAY ISLAND, JUNE 6, 1942

PREFACE TO THE SECOND EDITION

Since the first edition of "The Marine Power Plant" was written twenty years ago, the improvements in economy and changes that have taken place in marine engineering have necessitated a complete rewriting of the text and the introduction of practically a complete new set of illustrations. The plan and purpose set forth in the preface to the first edition have been adhered to in the present edition. This is intended as a first book in marine engineering and makes no pretense of being an exhaustive treatise. Fundamental principles and the thermodynamic and economic features have been stressed, and only enough descriptive matter and details have been included to make clear the principles involved.

More attention has been given to fuel oils, combustion, geared turbines, and feed-water systems than in the first edition. Most of the new types of marine machinery have been treated, although in some cases rather briefly. The space devoted to Diesel engines has been greatly increased. The power plant computations set forth in the last chapter are completely new, and the data used conform to present-day practice.

In writing the book, the point of view of the seagoing engineer and the shore operating staff has been kept in mind, as well as that of students of marine engineering, naval architecture, and marine transportation. It is hoped that the book will prove of value to these groups of readers.

Although the book deals largely with American practice, frequent references and a number of illustrations have been included dealing with the power plants of ships of other countries, especially of Great Britain.

The author wishes to thank his colleagues on the faculty of the Massachusetts Institute of Technology who have rendered assistance during the preparation of the manuscript. The following have read various parts of the manuscript and offered valuable criticism: Professors Joseph H. Keenan, Hoyt C. Hottel, James R. Jack, Frank M. Lewis, Henry H. W. Keith, Clifford E. Lansil, Evers Burtner, Carl L. Svenson, Theodore H. Taft, and Captain Charles S. Joyce, U.S. Navy (Retired). The author is especially pleased to acknowledge the assistance of his former student, A. S. Thaeler, scientific engineer of the Federal Shipbuilding and Dry Dock Company, who has read part of the manuscript and offered suggestions for its improvement.

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LAWRENCE B. CHAPMAN.

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LAWRENCE B. CHAPMAN.

PREFACE TO THE FIRST EDITION

The purpose of this book is to bring before the student the thermodynamics of the marine power plant and the types of machinery used for ship propulsion, and to give him a comprehensive idea of the layout and function of the various pieces of auxiliary machinery.

The book makes no pretenses at being an exhaustive treatise. It is intended as a first book in marine engineering. At Lehigh University the study of the marine power plant as presented in this book is preceded by a course in thermodynamics and followed by a summer at sea and by a more thorough and detailed study of marine engines, turbines and Diesel engines.

The thermodynamic and economic features of the power plant have been accentuated throughout the book. Very little attention has been given to mechanical details and all pure descriptive matter has been reduced to a minimum. Details can be better learned under actual operating conditions on shipboard than from the inadequate treatment in a text book.

A short chapter on thermodynamics has been added as a review for the engineering student and also as a foundation study for others who may study the book. Complete calculations for the sizes of the boilers and auxiliaries of a typical plant are given in Chap. XIX. It is believed that this is the first time such calculations have appeared in print.

A special feature of the book is the comparison of the various types of machinery used today for ship propulsion which is concluded with a table showing an unbiased comparison of seven types of propelling machinery.

While the book is intended primarily for the students of naval architecture, marine engineering, and ship operation, it is believed that it will bring before the sea going engineer and ship owner a better understanding of the many types of propelling machinery and auxiliaries used today.

L. B. CHAPMAN.

BETHLEHEM, PENN.,
June, 1922.

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CHAPTER I

INTRODUCTION

1. Types of Propelling Machinery.—In 1906 practically the only type of machinery installation used for propelling merchant ships was the reciprocating engine and Scotch boiler. Coal was the fuel universally used. Boiler pressures were around 200 lb per sq in. and superheat was practically never used. Fuel consumptions were between 1.5 and 2.0 lb of coal per ihp-hr. In 1941, 35 years later, marine engineering has undergone radical improvements in economy, and many new types of propelling machinery have been introduced. The reciprocating engine and Scotch boiler are still used to propel a large percentage of the world's merchant ships, but this type of machinery is being installed only to a very small extent in new ships. Oil has largely superseded coal. Boiler pressures of around 450 lb and initial steam temperatures of 750°F are common practice. In 1939 the U. S. Maritime Commission's *S.S. Red Jacket* attained a fuel consumption on the official trials of 0.545 lb of oil per shp-hr (18,500 Btu per lb).

Today a large proportion of the new steamships are propelled by geared turbines with steam supplied by water-tube boilers. In addition to the geared turbine, we have the following types of propelling machinery available: straight Diesel engine drive; geared Diesel engines, Diesel-electric drive, turbo-electric drive, reciprocating engines of greatly improved types, and reciprocating engines with exhaust turbines.

With all these various types available for ship propulsion, the marine engineer and shipowner have a more interesting and difficult task in deciding on the proper type of machinery than was the case 35 years ago. In the following pages, all these types will be fully treated, and the advantages and disadvantages of each type will be discussed.

2. Comparison of Land and Marine Installations.— The problems confronting the designer of a marine power plant are far more difficult than those confronting the designer of a power plant on shore. The marine engineer and stationary engineer have many problems bearing on efficiency, economy, and costs that are almost identical, and in many ways power plants on shipboard and on shore are much alike. The marine engineer, however, has limitations of weight and space that are not met by the engineer on shore.

For high-speed ships such as passenger and naval vessels it is vital, in order to reduce the resistance, that the displacement be kept as small as possible. This necessitates propelling machinery of light weight per horsepower. With reciprocating engines, the natural way to reduce the weight of machinery for a given horsepower is to increase the revolutions. High revolutions of the propeller, however, are not compatible with good propeller efficiency, and high propeller efficiency is necessary in order to keep the power of the ship for a given speed as low as possible. Thus the designer may be confronted at the outset with two restricting conditions that conflict with each other.

Much of the cargo carried by ships today is taken on a measurement basis. Hence it is essential to keep the length of the machinery space at a minimum in order to increase the capacity of the cargo holds. Thus, for cargo ships, a small machinery space is usually more important than a low machinery weight. In some trades, where a dead-weight ship is called for instead of a cubic ship, attention must be given to machinery weights. However, for cargo ships of low or medium speed-length ratios ($V/\sqrt{L}$), machinery weight is not so vital a factor as it is in high-speed passenger and naval vessels.

Although low steam and fuel consumption are just as important on shore as on shipboard, from the viewpoint of economy, the marine engineer has additional incentives for low fuel consumption besides the cost of fuel. A reduction in the fuel consumption of a ship's power plant reduces the weight and space occupied by fuel in the bunkers and thus reduces the displacement or allows increased space and weight for cargo with a given displacement, which, in turn, results in a greater earning capacity of the ship. If the steam consumption of the propelling machinery can be reduced, the size and weight of boilers, main engine, piping, and auxiliaries can be reduced, with a further saving in weight and space. The design of the power plant for a ship of high speed requires more attention than that for the slow-speed cargo ship; yet there are many features, such as economy, low propeller rpm, and space requirements, that need careful attention with the latter type.

The necessity for low fuel consumption, light machinery, and low propeller revolutions has been the cause of the introduction of high steam pressures and temperatures, water-tube boilers, geared turbines, geared Diesel engines, and electric drive. The low fuel consumption of the Diesel engine has caused it to be adopted for propelling many merchant ships, notwithstanding its somewhat higher initial cost than steam machinery.

The average fuel consumption and weights of the various types of machinery used today are given in the following table.

TABLE I

	Boiler pressure (gage)	Vacuum, in.	Propeller rpm	Total mach., lb per hp	Fuel, lb per shp-hr
Reciprocating engines, Scotch boilers, and coal. Saturated steam. . . .	200-250	25–27	75–85	500	1.95
Reciprocating engines, Scotch boilers, and oil. Saturated steam . .	200-250	25–27	75–85	500	1.2
Geared turbines, water-tube boilers, and oil. $t_1 = 750°F$	450	$28\frac{1}{2}$	90	200–300	0.60
Geared turbines, express-type boilers, and oil * $t_1 = 750°F$	450–600	28	350–450	40	0.65
Diesel engines..	.....		90	500–600	0 38
Geared Diesel engines. .. .	. .		90	300–400	0 38

* Naval installations.

All weights except those of the reciprocating engine installations are based on shp. The weights of the reciprocating engine installations are on an ihp basis.

3. The Elementary Steam Power Plant.—Practically all marine steam power plants are condensing, since it is necessary to save the condensed steam for feed water and surface condensers are used entirely for ships navigating in salt water. Ships operating in fresh water such as the Great Lakes of North America often use jet condensers.

A diagrammatic layout of a simple plant is shown in Fig. 1. The steam is generated in the boiler by absorbing heat from the

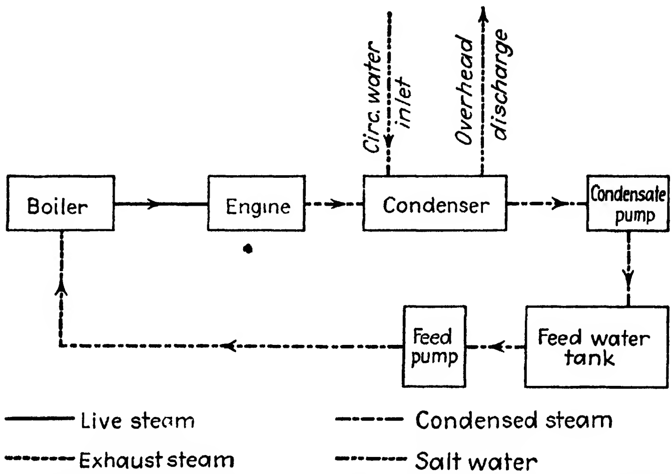

FIG. 1.—Diagram of elementary power plant. Many necessary auxiliaries have been omitted from this diagram for the sake of simplicity.

fuel; the steam leaves the boiler with the enthalpy h_{g_1} per pound and enters the engine or turbine through the throttle valve. In the engine, mechanical work is done at the expense of the enthalpy of the steam. The heat left in the steam after expansion in the prime mover is rejected to the condenser. Here the latent heat of the steam $x_2h_{fg_2}$ is absorbed by the water circulating through the condenser and is discharged overboard. The air or condensate pump removes the condensed steam from the condenser and discharges it with the enthalpy of the liquid h_{f_2} to the feed tank. The boiler feed pump takes the feed water that collects in the feed tank and delivers it under pressure to the boiler.

The object of the condenser is twofold. It saves the condensed steam so that it can be used again in the boiler, and it lowers the exhaust temperature. This lower final temperature allows a greater expansion of the steam in the prime mover. A larger amount of the enthalpy of the steam is then converted into mechanical work.

A complete diagram of a ship's power plant using steam auxiliaries, which shows the various auxiliaries, feed-water heaters, etc., is shown in Fig. 118. Diagrams of power plants with motor-driven auxiliaries and the feed water heated by steam extracted from the turbine are shown in Figs. 128, 130, and 141.

4. Over-all Plant Efficiency.—By using the fundamental expression for efficiency, $\frac{\text{output}}{\text{input}}$, and using for our units Btu and the horsepower hour, the expression for over-all plant efficiency becomes

$$\frac{2{,}545}{f \times \text{heating value of the fuel}}$$

where f = fuel consumption per shp-hour.

If we use the values for average fuel consumption given in the preceding table and assume the heating value of coal as 14,300 Btu per lb and fuel oil as 18,500 Btu, we have for the over-all efficiencies of the power plants the following values:

1. Reciprocating engine and coal $\frac{2{,}545}{1.96 \times 14{,}300} = 9.1$ per cent
2. Geared turbine and oil $\frac{2{,}545}{0.60 \times 18{,}500} = 23.0$ per cent
3. Diesel engine $\frac{2{,}545}{0.38 \times 18{,}500} = 36.2$ per cent

The foregoing figures show how small a percentage of the heating value of the fuel per shp-hour is utilized for useful mechanical work. For the steam installations, between 77 and 90 per cent of the available heat is lost, whereas with the Diesel this loss is greatly reduced, being only about 64 per cent. The following chapters will take up the study of these losses and the means adopted to increase the over-all plant efficiency and thereby reduce the fuel consumption.

5. Heat Balances.—The heat balances for the three installations of Art. 4 are approximately as follows:

1. Reciprocating engines, Scotch boilers, steam auxiliaries, and coal (f = 1.96).

	Btu per shp-hr	Per cent
1. Useful work. . .	2,545	9.1
2. Boiler losses. . .	8,960	32.0
3. Work done against mechanical friction (main engine)	255	0.9
4. Auxiliary work and ship's load	175	0.6
5. Rejected in main condenser	15,100	54.0
6. Pipe-line losses, auxiliary condenser loss, radiation, leakage, etc.	965	3.4
Heating value of fuel per shp-hr = 14,300 × 1.96	28,000	100

2. Geared turbine, water-tube boilers, and fuel oil; electric auxiliaries; extracted steam used for feed-water heating (f = 0.60).

	Btu per shp-hr	Per cent
1. Useful work	2,545	23.0
2. Boiler losses	1,432	13.0
3. Work done against mechanical friction (gears and bearings). . . .	105	0.9
4. Auxiliary work and ship's load	135	1.2
5. Rejected in main and auxiliary condenser	6,615	59.5
6. Leakages, radiation, and heating	268	2.4
Heating value of fuel per shp-hr = 18,500 × 0.60	11,100	100

3. Diesel engine installation (f = 0.38).

	Btu per shp-hr	Per cent
1. Useful work.	2,545	35.4
2. Rejected in the exhaust (main engine)	2,100	29.2
3. Heat to cooling water (main engine)	2,190	30.4
4. Auxiliary generator set (work and losses).	185	2.5
5. Radiation and unaccounted for	180	2.5
Heating value of fuel per shp-hr = 18,900 × 0.38.	7,200	100.0

Mechanical friction and engine driven auxiliaries are not included in the above balance as these are accounted for in items 3 and 2 and radiation.

6. The Generation of Steam.—It is assumed that the student beginning his study of marine engineering has made a study of thermodynamics of steam and gas engines and is familiar with the use of steam tables and diagrams. The following articles are included as a brief review and also for the assistance of those students who are not well versed in the use of the steam tables.

Steam tables have been prepared that give the various properties of saturated liquid, saturated steam, and superheated steam. In solving problems throughout this book, frequent use will be made of the properties of water and steam contained in the steam tables. The values used in this text have been taken from Keenan and Keyes, "Thermodynamic Properties of Steam."[1] The properties of water and steam most frequently referred to are pressure, temperature, and enthalpy. The latter property is expressed in Btu for 1 lb of fluid and is the sum of its internal energy plus the product of its absolute pressure and its specific volume. The terms "heat content" and "total heat" have been used frequently in the past for what is here termed "enthalpy."

The steam tables give for various pressures the enthalpy of saturated liquid h_f, the enthalpy of evaporation h_{fg}, the enthalpy of saturated steam h_g, and the enthalpy of steam for various degrees of superheat h. The steam tables are based on the enthalpy of saturated liquid at 32°F—in other words, saturated liquid at 32°F is assumed to have zero enthalpy.

For engineering purposes, steam is generated in a boiler from pure water by the combustion of coal or oil. The feed water is pumped into the boiler at some temperature above 32°F with the enthalpy of the liquid corresponding to the feed-water temperature h_{feed}. As soon as the boiler is under steady operating conditions, the pressure is practically constant and corresponds to the predetermined boiler pressure. The water in the boiler absorbs heat produced by the combustion of the fuel, and the enthalpy of the liquid is increased from that corresponding to the feed temperature to that corresponding to saturation at the boiler pressure P_1. The increase in enthalpy of the liquid is thus $h_{f_1} - h_{\text{feed}}$. As further heat is added, the water, which has now reached the boiling temperature, begins to boil at constant pressure, and steam is formed. The heat absorbed in

[1] John Wiley & Sons, Inc., New York, 1936.

changing from water to steam is the enthalpy of vaporization h_{fg_1}. If the boiling takes place in a violent manner or in too confined a space, a small part of the water, perhaps 1 or 2 per cent, is thrown off and does not form steam but is held in suspension in the steam. Obviously, if 2 per cent of every pound of water is thus thrown off with the escaping steam bubbles, this 2 per cent does not absorb the enthalpy of vaporization h_{fg_1} but contains only the enthalpy of the liquid h_{f_1}. The heat absorbed by the water per pound is therefore $0.98h_{fg_1}$. The enthalpy of vaporization is expressed as $x_1h_{fg_1}$, where x_1 represents the percentage of water actually turned into steam. This factor x is called the "quality" of the steam. When the steam contains moisture, it is known as "wet" steam; but if all the water in every pound is turned into vapor and the quality is 100 per cent the steam is said to be "dry" or "saturated" steam. The unit in all steam computations is the pound, and the values given in the steam tables are Btu's per pound above saturated liquid at 32°F.

If dry steam is withdrawn from the boiler drum and subjected to further heating in a heater in which no liquid is present, it absorbs further heat. Its temperature then rises above the temperature corresponding to the pressure for saturated steam. The steam is now said to be superheated. The amount that the temperature rises above that of saturated steam for a given pressure is known as the "degree of superheat." Thus, from the steam tables, the temperature of saturated steam at 300 lb per sq in abs is 417.33°F. If this steam were heated to a temperature of 617.33°F in a superheater, it would be said to have 200 degrees of superheat.

7. Entropy.—Entropy is a useful property of steam in solving many problems in steam engineering. All our work with this property is limited to changes in entropy, as is the case with enthalpy, rather than with absolute amount. Change in entropy can be defined as

$$dS = \left(\frac{dQ}{T}\right)_{\text{reversible}}$$

where dQ = heat received by the system.

T = constant absolute temperature at which this change takes place.

Therefore, the common definition of entropy becomes

$$S_2 - S_1 = \int \left(\frac{dQ}{T}\right)_{\text{reversible}}$$

As just pointed out, all our study with entropy applied to changes in entropy, yet we are accustomed to speak of the entropy of liquid and steam under various conditions, meaning thereby the entropy change above saturated liquid at 32°F.

An adiabatic change of gas or vapor is one under such conditions of insulation that no energy in the form of heat is given up and none received during the change. Thus, for a reversible adiabatic, $dQ = 0$ and, from our definition of change in entropy given above

$$S_2 - S_1 = 0$$

We can thus make the important statement that during a reversible adiabatic change (expansion or compression) there is no change in entropy. Such a change is frequently termed "isentropic."

A convenient way of thinking of entropy in the early stages and securing some tangible grasp of its significance is to consider entropy that property of steam which is constant during a reversible adiabatic expansion, in a similar way in which temperature (a familiar property) is constant during an isothermal expansion. This, of course, is not a true and complete definition of entropy, but it is often found helpful in the early study of thermodynamics. A true understanding of and familiarity with entropy can be acquired only by the solution of problems involving its application.

The steam tables give values of s_f, entropy of the liquid, s_{fg}, entropy of vaporization, and s_g, the total entropy of the steam. The total entropy of superheated steam can also be found from the steam tables.

8. The Temperature-entropy (Ts) Diagram.—One of the valuable uses of entropy is its application in the Ts diagram. Here absolute temperatures are laid off as ordinates and entropies as abscissas. Since entropy is zero for saturated liquid at 32°F or 492° abs, our starting point for liquid at 32°F is represented at 0 on the chart (Fig. 2). The entropy of the liquid for various temperatures is now plotted on the chart, giving the liquid line. In a similar manner, the entropy of saturated steam is plotted for various temperatures from the data given in the steam tables. This gives the line of saturated steam. Intermediate between these lines are various intermediated conditions of xs_{fg} at various

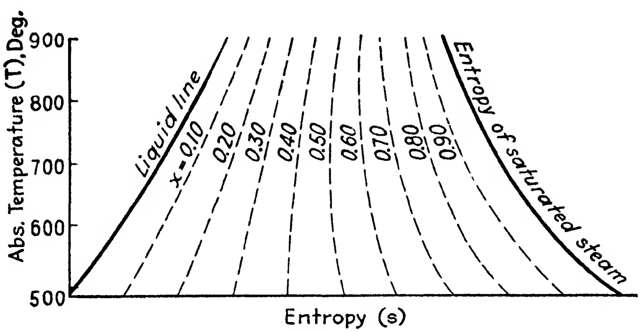

FIG. 2.

temperatures. Quality lines are constructed giving the quality from 0 to 100 per cent. The reader is referred to the Ts diagram in Keenan and Keyes, "Steam Tables."

From our equation for entropy change, $dS = (dQ/T)_{rev}$, we get $dQ_{rev} = T\,dS$. The right-hand member in the latter equation is at once recognized as the expression for an area. If we take some point, as point 2 in Fig. 3, at temperature T_1, the change in entropy between point 8 and point 2 is ds. The heat added, from point 2 to point 3 is

$$\int_2^3 T\,ds$$

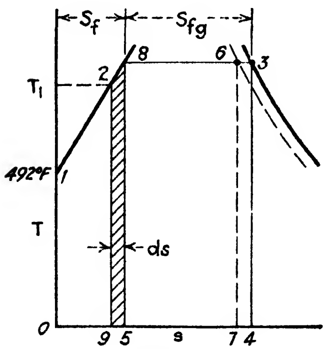

FIG. 3.

This expression is represented by the area 9-2-3-4 in Fig. 3. Therefore it is evident that the area on the Ts diagram represents heat as in a similar manner the area on the PV diagram represents work. This area 9-2-3-4 must extend down to absolute zero, for it is represented by $\int T\,ds$ where T is absolute temperature.

Obviously, area $\int_0^5 T\,ds = (0\text{-}1\text{-}8\text{-}5)$ must represent the enthalpy of the liquid h_f, and area 8-3-4-5 represents the enthalpy of vaporization h_{fg}. If the steam is not dry and saturated, h_{fg} becomes xh_{fg}, and s_{fg} becomes xs_{fg}. The area 8-6-7-5 represents xh_{fg} under these conditions.

9. Adiabatic Expansion.—As already mentioned (Art. 7), an adiabatic expansion or compression of steam or gas is one in which no heat is received from or given off to the surrounding media; during a reversible adiabatic change, entropy is constant. The most common reversible adiabatic expansion is the expansion of steam in the cylinder of an ideal reciprocating steam engine or turbine.

In Fig. 4, if dry steam at pressure P_1 corresponding to temperature T_1, is expanded adiabatically to pressure P_2, corresponding to temperature T_2, it will be represented on the Ts diagram by a vertical line extending downward from point 1 on the dry steam line.

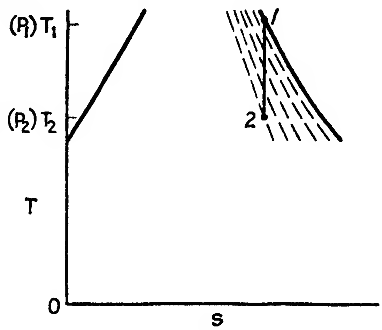

Fig. 4.

Entropy s is constant, and the temperature is reduced. Therefore the adiabatic expansion is represented by the vertical line 1-2. At condition 1, the steam was dry, but as it expands at constant entropy it becomes wet. This is clearly shown in Fig. 4, where the line 1-2 cuts lines of lower and lower quality in its expansion to P_2. The quality at point 2 is x_2. An example will make this clear. Suppose that 1 lb of dry steam at 275 lb per sq in. abs (409.4°F) were expanded adiabatically to 6 lb per sq in abs (170°F). Noting the initial condition on the Ts diagram in the steam tables, we have $s_1 = 1.52$, $t_1 = 409.4°$, $x_1 = 1.00$. Since the entropy does not change, $s_1 = s_2 = 1.52$. Running down along the line $s = 1.52$ to $t_2 = 170°\text{F}$, we have $x_2 = 0.803$. The same results can be obtained more accurately from the Mollier diagram (see Art. 10).

The quality after an isentropic expansion can be found by equating the entropies for conditions P_1 and P_2.

$$s_1 = s_2$$
$$s_1 = s_{f_1} + s_{fg_1} = s_{f_2} + x_2 s_{fg_2}$$

Here all the quantities can be obtained from the steam tables except x_2, which can readily be evaluated by solving the equation. A solution of a problem by means of the foregoing equation is given in Art. 13.

10. Mollier Diagram.—Another very useful steam diagram is the Mollier diagram,[1] in which the data from the steam tables are plotted with total enthalpy as ordinates and total entropy as abscissas. Lines of constant pressure, constant temperature, constant percentage of moisture, and constant degrees of superheat are plotted on this diagram. Thus all the properties of steam except specific volume are shown, although data cannot be read with the same degree of accuracy as from the tables. The Mollier diagram is very useful in solving problems dealing with expansion of steam, both at constant entropy and under conditions prevailing in actual turbines.

11. Constant Enthalpy Change.—Another expansion of great importance in steam engineering is a change in pressure without any change in enthalpy. This obviously is an adiabatic change, but it is an irreversible process in which entropy is not constant, as was the case in the reversible adiabatic just studied. No mechanical work is done during this change.

This expansion occurs most commonly when steam is reduced in pressure by a reducing valve and in the "wire drawing" through the valves and passages of an engine or turbine. In both cases the reduction in pressure is accompanied by an increase in velocity of the steam and a corresponding decrease in enthalpy, but as the steam is brought more or less to rest after the reduction in pressure the enthalpy is increased again by impact and friction.

This change can best be studied by means of the Mollier diagram. Suppose that steam at 200 lb abs and 98 per cent quality is reduced in pressure through a reducing valve to a pressure of 35 lb abs. Referring to the Mollier diagram, we find at the intersection of the 200-lb line and the 0.98-quality line the enthalpy of 1 lb of steam = 1182 Btu and the total entropy is 1.525. Since this is a reduction in pressure (expansion) without

[1] A Mollier diagram is contained in the steam tables as a folding plate.

any change in enthalpy, we follow across the enthalpy line of 1182 Btu until it crosses the line of 35 lb pressure. The steam is now superheated 30 degrees—in other words, the wet steam has become superheated during the reduction in pressure. This should naturally follow for a constant enthalpy change, since some of the enthalpy of the liquid at 200 lb has resulted in superheat at the reduced pressure of 35 lb where h_{f_2} is smaller. During this reduction in pressure the entropy has increased from 1.525 to 1.707. This expansion is adiabatic, but it is not isentropic.

This reduction in pressure without any loss in enthalpy is the principle on which is based Peabody's throttling calorimeter for measuring the quality of steam. Wet steam at a known pressure and unknown quality is expanded through a partly closed valve until it is superheated. The temperature and pressure after the throttling of the steam can be read and its enthalpy computed. The enthalpy before and after throttling are, of course, the same, and x_1 can therefore be found.

12. Engine Efficiency.—In order to compare the performance of an actual reciprocating steam engine or steam turbine with an ideal one (*i.e.*, one with isentropic expansion), an expression known as "engine efficiency" is used. This is defined as follows:

$$\text{Engine efficiency} = \frac{\text{useful work by actual engine per pound of steam}}{\text{isentropic enthalpy change between initial and final condition per pound of steam}}$$

$$= \frac{\dfrac{2{,}545}{w_a}}{h_1 - h_2} = \frac{2{,}545}{w_a(h_1 - h_2)}$$

where w_a = steam consumption of the actual engine per horsepower hour measured by a test of the engine.

h_1 = enthalpy of the steam at inlet conditions.[1]

h_2 = enthalpy of the steam after a reversible adiabatic expansion from initial conditions to condenser pressure.

For reciprocating engines, w_a is usually expressed in terms of indicated horsepower (ihp) and for turbines, in terms of shaft horsepower (shp). As we shall see later, the two steam rates are

[1] The usual practice today is to use inlet pressure and steam conditions at engine or turbine inlet, *i.e.*, just inside the throttle valve.

not comparable until a correction has been made for the mechanical efficiency of the reciprocating engine.

The foregoing expression for engine efficiency does not hold for a turbine working on the regenerative cycle where steam is extracted from the turbine for heating the feed water.

The expression given above was formerly termed the "efficiency ratio" or the "Rankine efficiency ratio," and it is still so defined in many books. The true definition of efficiency ratio is the ratio of the thermal efficiency of the actual cycle to the efficiency of the Rankine cycle. Because of the work done by the feed pump and because of friction effects in pipe lines and condenser, this expression is not exactly the same as the expression given above for engine efficiency. Engine efficiency is now the generally accepted term.

13. Steam Consumption.—The steam consumption of an actual engine or turbine is determined by a test of the engine where the steam used is carefully measured over a period of time and the average horsepower is also measured. The steam consumption or water rate for a turbine in pounds per shp per hour is

$$w_a = \frac{\text{steam used by turbine per hour}}{\text{shp}}$$

For a reciprocating engine the water rate is usually expressed in pounds per ihp per hour,

$$w_a = \frac{\text{steam used by engine per hour}}{\text{ihp}}$$

Since the shp of a reciprocating steam engine is equal to the ihp times the mechanical efficiency of the engine

$$\text{Shp} = \text{ihp} \times e_m$$

the steam consumption per ihp per hour will be lower than the steam consumption based on shp. This point is further discussed in Art. 14.

The theoretical steam rate of the ideal engine or turbine, where the steam is assumed to be expanded isentropically, is

$$w_t = \frac{2{,}545}{h_1 - h_2}$$

where h_1 = enthalpy of the steam at inlet conditions.
h_2 = enthalpy of the steam at condenser pressure after a reversible adiabatic expansion from inlet conditions.

Since

$$\text{Engine efficiency} = \frac{2{,}545}{w_a \times (h_1 - h_2)}$$

and

$$w_t = \frac{2{,}545}{h_1 - h_2}$$

it is apparent that engine efficiency is the ratio of the theoretical and actual water rates w_t/w_a. From Art. 12, it follows that

$$w_a = \frac{2{,}545}{(h_1 - h_2) \times \text{engine efficiency}}$$

Hence, if values of engine efficiencies are available from existing installations, the actual water rate w_a for a new design can be readily estimated as soon as the initial and final steam conditions are fixed, which will allow h_1 and h_2 to be determined.

Example.—The data from trials of the steamship that was fitted with a quadruple-expansion engine were as follows:

Throttle pressure (P_1) = 240 lb abs
Quality of steam at throttle = 99.5 per cent
Vacuum in condenser = 27.9 in.
w_a = 12.8 lb per ihp per hr
Barometer = 30.49 in.
P_2 = 30.49 − 27.9 = 2.59 in. of mercury = 1.275 lb per sq in. (110°F)
From the Mollier diagram

$$\begin{aligned} h_1 &= 1{,}196 & s &= 1.524 \\ h_2 &= \phantom{1{,}}863 & s &= 1.524 \qquad x_2 = 0.761 \\ h_1 - h_2 &= \phantom{1{,}}333 \\ w_t &= \frac{2{,}545}{333} = 7.64 \end{aligned}$$

$$\text{Engine efficiency} = \frac{2{,}545}{12.8 \times 333} = 0.597$$

By equating entropies at conditions 1 and 2

$$\begin{aligned} s_{f_1} + x_1 s_{fg_1} &= s_{f_2} + x_2 s_{fg_2} \\ 0.5631 + (0.995 \times 0.9667) &= 0.1471 + 1.8106x_2 \\ 1.8106x_2 &= 1.3779 \\ x_2 &= 0.761 \\ h_2 &= 77.94 + (0.761 \times 1{,}031.6) = 863 \end{aligned}$$

The foregoing expression for theoretical water rate, like the foregoing equation for engine efficiency, does not hold when steam is being extracted from a turbine for feed heating. The regenerative cycle is discussed in Art. 145.

In the Ts diagram (Fig. 5), the enthalpy h_1 in the foregoing expression for theoretical water rate, when superheated steam is used, is represented by the area 0-*a*-*f*-*b*-*c*-*g*-0, and h_2 is represented by the area 0-*a*-*e*-*g*-0. The area 0-*a*-*h*-0 represents the enthalpy of the liquid h_{f_2} at the condenser back pressure. If dry steam at the same initial pressure is used, h_1 is represented by the area 0-*a*-*f*-*b*-*k*-0 and h_2 by area 0-*a*-*m*-*k*-0.

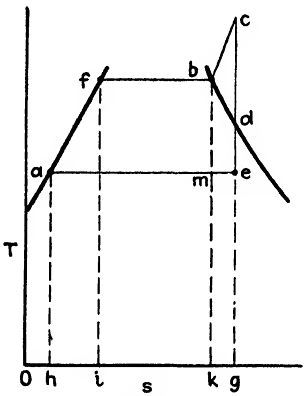

FIG. 5.

14. Shaft Horsepower and Indicated Horsepower.—It is customary to express the power of reciprocating engines in indicated horsepower (ihp) and that of turbines in shaft horsepower (shp). The power of Diesel engines and other types of internal combustion engines is generally expressed in brake horsepower (bhp), which is the same as the shp.

In the case of reciprocating engines, the ihp has been used because of the early adoption of the steam-engine indicator for measuring the power of an engine. This has continued as the basis because of the ease with which the ihp can be measured. Because of unsatisfactory results due to the uneven turning moment, the shp and bhp is not measured on the ship. The ihp is thus the power developed in the engine cyclinders and is greater than that delivered to the propeller shaft because of the losses due to engine friction.

With turbines, on the other hand, the internal horsepower developed by the steam within the turbine is not so readily determined; hence the shp delivered to the shaft has been used as the basis. The shp is readily measured by means of a torsion meter.

The steam consumption is usually expressed in pounds per ihp per hour for reciprocating engines and in pounds per shp per hour for turbines. These different methods of stating the steam consumption should cause no confusion, however, in working out engine efficiencies for engines and turbines. In comparing horsepowers and steam consumptions, the different bases should be kept in mind. The engine efficiencies of a turbine will run somewhat lower and unit steam consumption somewhat higher than those of a reciprocating engine that delivers the same power to the propeller with the same total steam consumption.

Example.—Suppose each of two sister ships requires 5,000 hp to be delivered to the propeller. One ship is fitted with a reciprocating engine and the other with a turbine, and each uses 65,000 lb of steam per hr at full speed. The shp of the turbine is 5,000 and the ihp of the engine is 5,420.

$$w_a \text{ (turbine)} = \frac{65{,}000}{5{,}000} = 13 \text{ lb per shp per hr}$$

$$w_a \text{ (engine)} = \frac{65{,}000}{5{,}420} = 12 \text{ lb per ihp per hr}$$

Assuming $w_t = 7.0$ lb per hp per hr,

Engine efficiency (turbine) = 0.538
Engine efficiency (engine) = 0.582

The foregoing is a strictly theoretical example for purpose of illustration. Actually, the performance of the turbine in the first ship would be better than that of the reciprocating engine in the second ship, resulting in a lower hourly steam consumption and a lower steam consumption per shp per hour and a higher engine efficiency.

The shp of a geared turbine is measured by means of a torsion meter placed on the line shafting aft of the reduction gears. Since the actual steam rate w_a is calculated by using this shp, it is obvious that the engine efficiency of a geared turbine includes the loss in the gears. In solving problems dealing with turbines, frequently the internal engine efficiency of the turbine is desired, *i.e.*, without including the losses due to bearings and reduction gears. By estimating the friction loss in the gears and main turbine bearings, this internal engine efficiency can be readily attained. Thus, if the loss in the gears and bearings is estimated as 3 per cent, the engine efficiency, excluding gears and bearings, is

$$\frac{2{,}545}{0.97w_a(h_1 - h_2)}$$

The engine efficiency of a turbine driving a generator in a turbo-electric drive installation cannot be compared with that of a geared turbine installation until a correction has been made for the loss in the reduction gears.

15. Initial Pressure and Temperature.—In order to investigate the influence of the initial pressure and superheat, we shall compare cycle efficiencies for simple cases where the feed enters the boiler at the condenser temperature. Pump work and losses between boiler and engine will be neglected. Under these conditions the efficiency of the cycle is,

$$e_{\text{cycle}} = \frac{h_1 - h_2}{h_1 - h_{f_2}}$$

where h_1 = enthalpy under initial conditions.

h_2 = enthalpy after an isentropic expansion to the condenser vacuum.

h_{f_2} = enthalpy of the liquid at the condenser temperature.

For a fixed value of h_{f_2} the cycle efficiency can be improved by increasing h_1 and decreasing h_2. However, since h_1 appears in the denominator as well as in the numerator, the largest gain can be obtained by a reduction in h_2.

Comparison cannot be made on the basis of the steam rate of the ideal turbine,

$$w_t = \frac{2{,}545}{h_1 - h_2}$$

An increase in h_1 would show a decrease in the steam rate, yet an increase in h_1 can be obtained only by burning more fuel in the boiler per pound of steam, and the decrease in steam rate might be offset by an increase in the fuel burned per shp-hour. Hence a true comparison should be based on cycle efficiencies or the fuel burned per shp-hour.

A study of the steam tables shows that after a pressure of 420 lb abs is reached, there is no increase in the value of h_1 for dry steam as the steam pressure is raised, and if the pressure is carried beyond 470 lb the value of h_1 begins to decrease as the pressure is increased.

Let us consider the case where the steam pressure is increased without any increase in h_1. From the steam tables we find that enthalpy of saturated steam at 280 lb abs is 1202.3 Btu per lb and for saturated steam at 660 lb abs it is 1202.1 Btu. A steam pressure of 280 lb is somewhat above the average pressure used on steamships 25 years ago, and 660 lb is about 200 lb above the pressure used on most modern merchant ships; but pressures approaching this are used on a few of the latest ships.[1]

Let us consider two ideal turbines operating under the two pressures mentioned above and a vacuum of 28.5 in.

	Case 1	Case 2
P_1, lb per sq in. abs.	280	660
Initial temperature t_1, °F	411	496 6
P_2, in. of mercury for 30-in. barometer	1 5	1.5
h_1, saturated steam	1,202 3	1,202 1
h_2, after an isentropic expansion	833	788
s	1 516	1.436
x_2 (from Mollier diagram)	0 742	0 698
h_{f_2}	59 7	59.7
$h_1 - h_2$	369	414
Cycle efficiency, per cent	32.3	36.2
Steam rate, w_t	6 90	6.15

The cycle efficiency for case 2 is 12.2 per cent higher than for case 1, and the theoretical steam rate for case 1 is 12.2 per cent higher than for case 2. It will be observed that, whereas the value of h_1 are the same for cases 1 and 2, the value of h_2 for case 2 is lower than h_2 for case 1. It will also be noted that the entropy and quality x_2 are lower for case 2 than for case 1. The result is that less heat is rejected to the condenser in the ideal cycle in case 2 than in case 1. Thus the raising of the boiler pressure, although resulting in no increase in h_1, did result in lower value of h_2 after an isentropic expansion.

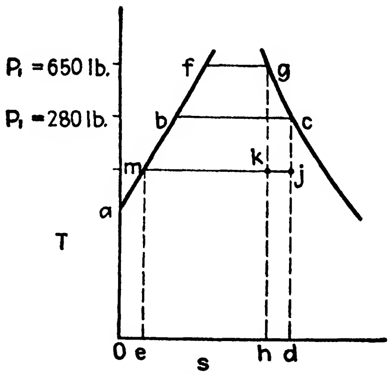

Fig. 6.

[1] The turbo-electric tanker *J. W. Van Dyke* operates with a throttle pressure of 600 lb gage and a temperature of 825°F. See *Mar. Eng. & Shipping Rev.*, 1938, p. 548, and *Trans. A.S.M.E.*, 1938.

On the Ts diagram in Fig. 6, h_1 for case 1 is represented by the area 0-*a*-*b*-*c*-*d*-0 and h_2, by area 0-*a*-*m*-*j*-*d*-0. For case 2, h_1 is represented by the area 0-*a*-*f*-*g*-*h*-0 and h_2, by the area 0-*a*-*m*-*k*-*h*-0. The areas *e*-*m*-*j*-*d* and *e*-*m*-*k*-*h* represent the heat rejected in the condenser for cases 1 and 2, respectively. The amount of heat rejected in the condenser in case 2 is less than that for case 1 by the area *k*-*j*-*d*-*h*, because the final quality after an isentropic expansion is lower in case 2(0.698) than in case 1(0.742).

Let us now consider a third case, where the pressure is held at 280 lb abs, as in case 1, and the steam is superheated 200 degrees.

	Case 3	Case 1
P_1, lb per sq in. abs	280	280
Initial temperature t_1, °F	611	411
Initial superheat, °F	200	0
P_2, in. of mercury at 30-in. barometer	1.5	1.5
h_1	1,322	1,202.3
h_2 (after isentropic expansion)	902	833
s	1.641	1.516
x_2	0.808	0.742
h_{f_2}	59.7	59.7
$h_1 - h_2$	420	369
Cycle efficiency, per cent	33.2	32.3
Steam rate, w_t	6.07	6.90

The increase in the cycle efficiency of case 3 over case 1 is only 2.8 per cent, yet the reduction in steam rate is 12 per cent. Obviously a greater cycle efficiency can be obtained by increasing the pressure than by increasing the steam temperature by superheating.

The foregoing discussion has been confined to the ideal turbine, in which the expansion is isentropic. In actual engines and turbines the expansion is not isentropic but slopes to the right from the constant entropy line, resulting in an increase in both entropy and quality of the steam at the condenser pressure P_2 over those for the ideal engine. In later articles we shall see that there are practical gains due to superheat that increase the engine efficiency and make the reduction in actual steam rate, w_a, greater than that of the reduction in the theoretical steam rate, w_t. We shall also see that increasing boiler pressure without an increase in superheat tends to lower the engine efficiency. It is pointed out in Chap. VI that, because of the necessity of protecting the low-pressure blading of turbines from erosion, the moisture in the steam at condenser pressure should not be allowed to exceed 11 or 12 per cent. We have already seen in Fig. 6 that an increase in P_1 tends to make the steam at exhaust wetter; hence there is a practical limit to which we can raise the initial pressure and still operate on saturated steam. In actual practice, both the initial steam pressure and superheat are increased so that a high cycle efficiency and high engine efficiency can be attained without too high a percentage of moisture in the low-pressure blading of the turbine.

16. Vacuum.—From our expressions for cycle efficiency and theoretical steam rate, it is apparent that a low value of h_2 will increase the cycle efficiency and also lower the steam rate w_t. For a given initial condition of the steam, h_2 can be decreased by lowering the back pressure P_2. Thus, for an isentropic expansion from saturated steam at 280 lb abs to atmospheric pressure (14.7 lb per sq in. abs), $h_2 = 990$ Btu, and if the expansion is continued to 0.736 lb abs (1.5 in. of mercury), $h_2 = 833$. The cycle efficiency in the first case is 20.7 per cent and in the second case, 32.2 per cent.

It is thus obvious that engines and turbines should be operated with as low a back pressure as it is economical and practical to use. In later articles these economical and practical limits are discussed for both reciprocating engines and turbines.

When the back pressure is below atmospheric pressure, it is customary to express it as a vacuum in inches of mercury below atmospheric pressure for a standard barometer of 30 in.

Thus an absolute back pressure of 0.736 lb per sq in., which is equivalent to 1.5 in. of mercury, is expressed as a 28.5-in. vacuum, *i.e.*, 30 − 1.5 in. Thus a *low* back pressure (0.736 lb) becomes a *high* vacuum (28.5 in.).

The most definite and satisfactory way to express a back pressure is in pounds per square inch, for the pressure is then independent of barometer reading and temperature. It can also be expressed in inches of mercury, but since mercury changes volume with a change in temperature, a back pressure expressed in inches of mercury must be at some standard temperature. The usual temperature employed, and that used in the steam tables, is 32°F.

Condenser pressures are practically always measured by a vacuum column (a U tube using mercury) that measures the vacuum or the difference between the atmospheric pressure and the pressure in the condenser. Since the pressure of the atmosphere is not standard but changes from day to day, it is necessary to refer such vacuum readings to a standard barometer.

The standard atmospheric pressure at sea level and 45 deg latitude is 760 mm or 29.92 in. of mercury, with the mercury in the barometer at 32°F. The corresponding pressure in pounds per square inch is 14.696.

As pointed out above, it is customary to state vacua referred to a 30-in. barometer instead of a 29.92-in. barometer. For the barometer to read 30 in. for the foregoing standard pressure, the mercury in the barometer would have to be at 58.4°F. Hence the vacuum gage and barometer readings should be reduced to 58.4°F in order to obtain the absolute pressure that is to be subtracted from 30 in. Also, absolute pressures obtained by subtracting a quoted vacuum from 30 in. will give absolute pressures in inches of mercury for mercury at 58.4°F, and hence such absolute pressures are not comparable to those given in the steam tables for mercury at 32°F. However, for the vacua ordinarily used on shipboard, such absolute pressures can be used without appreciable error for obtaining data from the steam tables.

Example:

Vacuum column reading (corrected to 32°F) = 27.45 in. of mercury
Barometer reading (corrected to 32°F) = 29.55 in. of mercury
Absolute pressure in condenser = 2.10 in. of mercury (mercury at 32°F), which is equivalent to 1.03 lb per sq in.
Vacuum column corrected to 58.4°F = 27.52 in.
Barometer corrected to 58.4°F = 29.63 in.
Absolute pressure in condenser (with mercury at 58.4°F) = 29.63 − 27.52 = 2.11 in. of mercury.
Vacuum referred to a 30-in. barometer = 30 − 2.11 = 27.89 in.

CHAPTER II

FUELS

17. General.—Fuels used on shipboard are almost entirely limited to two kinds: bituminous and semibituminous coal and oil. Other fuels such as anthracite and lignite coal, wood, gasoline, and gaseous fuels, which are used to a considerable extent ashore, find very little use on shipboard. Anthracite coal is used to a very limited extent in a few harbor craft, wood is used in some of the inland water craft in foreign countries, and gasoline is, of course, used in internal combustion engines of low-powered craft and small pleasure boats.

Practically all the seagoing merchant ships and all the naval ships in the United States now use oil fuel either under boilers (bunker *C*) or in Diesel engines (Diesel fuel oil). However, many of the merchant ships on the Great Lakes (both United States and Canadian) and many of the older foreign merchant ships and a good percentage of the newer ships in foreign countries that have large coal reserves still burn coal. Many of the craft on the inland waters of the United States also burn coal. On Jan. 1, 1938, the ratio of the number of coal-burning ocean-going merchant ships of the world to the oil-burning steamships (excluding all ships below 2,000 gross tonnage) was 1.48. Obviously coal is still an important marine fuel despite the present practice in the seagoing merchant ships of the United States.[1]

Today coal can be burned on shipboard by the use of mechanical stokers and in the pulverized form, and we are not limited to hand firing, which was practically the only method in use 20 years ago.

[1] The percentage distribution of the gross tonnage (all boats of 100 gross tons and over) of the ships recorded in "Lloyd's Register" as of June 30, 1939, was as follows: 44.67 per cent coal burners, 29.63 per cent oil-burning steamships, and 24.36 per cent motorships. In 1914, 88.84 per cent of the world's gross tonnage used coal fuel. In 1938, the percentage of coal burners was 46.54.

18. Comparison of Coal and Oil.—In making a comparison between coal and fuel oil for generating steam, we notice that there are some features connected with use of fuel on shipboard that have little or no influence on the use of fuel on shore. Some of the points to consider are cost, weight, bunker space required, availability and future supply, method of firing (*i.e.*, hand or mechanical), method of bunkering coal. In making a study of the costs of the two kinds of fuel, the comparison should be based on the amount of fuel necessary to produce equal amounts of steam under the same conditions of feed temperature and boiler pressure. In order to make this comparison, certain assumptions must be made regarding heating value of the fuel and boiler efficiency. Fair average figures for calorific values are 14,300 Btu per lb for semibituminous coal and 18,500 Btu for fuel oil. The boiler efficiency in operating with (hand-fired) coal has been assumed as 68.5 per cent and in operating with fuel oil, as 85 per cent. These efficiencies are for operation under the best of conditions; trials will often show higher values, and at times efficiencies under working conditions will be lower. However, the figures given are comparable, regardless of whether or not they satisfy a particular case.

With the foregoing figures, the ratio of the heating value of a pound of oil to a pound of coal becomes

$$\frac{18{,}500 \times 0.85}{14{,}300 \times 0.685} = 1.60$$

Coals with lower calorific value will give a larger ratio than 1.60. When coal is burned by a mechanical stoker or in the pulverized form, the boiler efficiency will be around 80 per cent instead of the 68.5 per cent assumed for hand firing. With this improved boiler efficiency, the ratio becomes 1.37.

If we assume that when coal is used on shipboard in the future it will be burned by mechanical stokers or in the pulverized form, the price of oil per ton can be about 37 per cent higher than coal for equal fuel costs in producing steam. With coal selling at $5.10 t.i.b., the shipowner can afford to pay $7 a ton for his fuel oil and still have the fuel costs equal. If we allow 8 lb per gal as the weight of fuel oil, $7 a ton would be $1.05 per bbl or 2.5 cents per gallon. Some twenty years ago Walter M. McFarland gave the following rule: When the cost of coal in dollars per ton is double that of oil in cents per gallon the fuel costs of producing steam are equal.

Since McFarland stated this rule, the efficiency of burning oil under a boiler has increased appreciably. However, if we assume that coal is to be burned by some mechanical method with a boiler efficiency around 78.5 per cent, McFarland's rule still holds.

Other factors, however, must be taken into account that will make the burning of oil on shipboard advantageous even if the price of oil is appreciably greater than given by the above ratios. (1) Coal cannot be stowed in the double bottom as oil can, and hence the space required for coal bunkers will reduce the volume of the cargo holds. This is an important factor, since most of the world's cargoes today are carried on a measurement basis. (2) Bunkering of coal today is a slow and dirty process. (3) Wide variation exists in burning characterisitcs of coal obtained at bunkering ports throughout the world. (4) Fireroom force is increased. (5) Fixed charges caused by mechanical stokers and pulverized-coal-burning equipment are increased.

19. Composition and Classification of Coal.—Coal is classified in various manners according to its physical or chemical characteristics. The common method is to classify it according to the percentage of volatile matter or hydrocarbons that it contains. There are no definite boundaries between the various classes. The following table presents a general grouping of coal by this method:

Kind of Coal	Per Cent Volatile Matter
Anthracite	3–8
Semianthracite	8–12
Semibituminous	15–25
Bituminous (Eastern)	25–40
Bituminous (Western)	35–50
Lignite	50 and over

Chemically, coal consists mainly of carbon, hydrogen, oxygen, nitrogen, sulphur, ash, and moisture. The exact nature in which these elements are combined is not clearly known. The analysis of coal into the elements mentioned but without regard to their chemical combinations is known as the "ultimate analysis."

Another form of analysis, known as the "proximate analysis," is also used. This is a much simpler analysis to make and is of great importance to the engineer. In this analysis the percentages of moisture, volatile matter, fixed carbon, and ash are determined. The hydrocarbons and other gaseous constituents of the coal that distill off upon heating are termed "volatile matter"; the uncombined carbon is termed "fixed carbon"; and the residue left after burning the volatile matter and fixed carbon is the ash.

Table II gives the ultimate analysis and proximate analysis and heating values of a few representative coals. Since the proximate and ultimate analyses are made both on "coal as received" and "dry coal," care must be used in comparing analyses to make certain which basis is used.

The heating value or heat of combustion of coal is the number of Btu that are given up in burning 1 lb of coal. The heating value is expressed in Btu per pound of coal "as received," per pound of "dry coal," and per pound of "combustible."[1] The heating value of coal varies roughly between 13,500 and 15,800 Btu per lb. The highest values are for semibituminous coal, where the percentage of fixed carbon is between 75 and 85. The anthracites with higher percentages of fixed carbon and the bituminous coals with lower percentages have lower heating values. Semibituminous coal is found in the United States in the Appalachian Range in Pennsylvania and West Virginia. It is a peculiar fact that the heating value of coal found in the United States decreases roughly as the distance of the mine from these semibituminous mines mentioned above.

The heating value of coal can be found approximately from the ultimate analysis by Dulong's formula:

$$\text{Btu per lb} = 14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}050S$$

For the ordinary run of coals used on shipboard, this formula will give values within 1½ per cent or 200 Btu. Unless care is taken to obtain a good representative sample, the Btu value found by means of the bomb calorimeter is not good within about 100 Btu. In the preceding formula, C, H, O, S are the decimal parts by weight of the carbon, hydrogen, oxygen, and sulphur, respectively, in 1 lb of coal.

[1] Fixed carbon + volatile matter.

The heating value of fuel discussed above is the higher heating value and is the commonly accepted one in use in the United States. In Great Britain and some other countries, a "lower" heating value is frequently used. The commonly accepted definition of the lower heating value is the higher heating value minus the enthalpy of vaporization at the initial calorimeter temperature (usually 60°F) of the steam formed by the combustion of the hydrogen in the fuel. Obviously, if the lower heating value is used in calculating plant efficiencies and boiler efficiencies, higher values will be obtained than when the standard higher heating value is used.

Coals vary widely in burning characteristics, percentages of moisture, ash, and volatile matter, temperature at which the ash fuses, percentage of lumps, and heating value. The characteristics of coals vary not only among the various regions of the world but sometimes within the same general coal field, because of variation in age and initial composition.

A higher percentage of sulphur in a coal is usually indicative of a low fusing temperature of the ash. Coals with low fusing temperatures of the ash are bad for hand firing and especially bad for use under water-tube boilers in the pulverized form.

Coal is usually sold in the following grades: run of mine, screened by sizes, and slack. The latter grade is finely broken up coal with practically no lumps. The British refer to this grade as "duff." It has been found by experience that best results with hand firing are obtained when the coal is of uniform-sized lumps, *i.e.*, screened by sizes.

Bituminous coal is extensively distributed and furnishes by far the largest percentage of steaming and bunker coal. Bituminous coals are distinguished by the very high percentage of volatile matter they contain and hence give forth large volumes of smoke unless carefully fired. A large variety of grades is included under the name "bituminous" and includes dry and caking coals and long-flame and short-flame coals. The dry bituminous coals are much to be preferred for use under boilers, since they burn freely without fusing and with a short flame. The caking coals, on the other hand, swell up and fuse together in the furnace and burn with a long flame. Bituminous coal is used extensively in foreign ships.

TABLE II.—REPRESENTATIVE COALS

Kind of coal	Location or name of coal	Proximate analysis				Ultimate analysis						Heating value dry coal, btu per lb
		Moisture, per cent	Volatile matter, per cent	Fixed carbon, per cent	Ash, per cent	C, per cent	H, per cent	N, per cent	O, per cent	Ash, per cent	S, per cent	
1. Anthracite	Lehigh	2.51	5.75	84.72	7.02	87.44	1.97	0.77	2.03	7.20	0 59	13,730
2. Semibituminous	Pocahontas	0.65	17.13	76.06	6.16	82.72	4.63	1.31	4.42	6.19	0.72	14,817
3. Semibituminous	George's Creek	4.03	18.79	69.24	7.94	79.19	4.85	2.12	4.63	8.27	0.94	14,386
4. Bituminous	Pittsburgh, Steaming	1.92	36.63	54 24	7 21	76.59	5.21	1.59	7.70	7.35	1.56	13,889
5. Bituminous	Hocking, Ohio	6.13	35.35	50.93	7.59	73.12	4.94	1.51	10.96	8.06	1.41	12,633
6. Brown lignite	North Dakota	12.89	46.96	30.41	9 74	58.33	4 33	0 81	23.40	11.18	1.95	9,657
7. Bituminous	Yorkshire	3.26	33.46	60.45	2.83	...	...	...	...	...	0.78	14,380
8. Bituminous	Scotland	6–10	24–37	48–74	2–5		...	...	...	...	6–1.5	14,600
9. Bituminous	Wales Admiralty	0.72	15.22	80.6	3.46	...		...	...	...	0 90	15,200
10. Bituminous	German (Ruhr)	2–8	18–35	50–70	4–12	...	...	...	...	...	...	11–14,000
11. Bituminous	Natal, South Africa	0.50	15.2	71.5	11 5	...	...		...	...	1.3	
12. Bituminous	Sumatra	7–10	35 0	55 0	2 0	79 6	5 6	1.7	12.6	...	0 5	12,600

Semibituminous coal is intermediate between anthractie and bituminous. The fixed carbon runs around 70 to 75 per cent and the volatile matter around 15 to 25 per cent. Semibituminous coal is very high in heat value and low in ash and moisture content. It burns with a comparatively short flame and is an excellent burning and coking coal. Because of its low percentage of volatile matter, it burns with much less smoke than bituminous coal. This coal is the most desirable of all for marine purposes.

The length of flame is dependent not necessarily on the percentage of volatile matter but on the rapidity with which the volatile matter is given off during combustion. This is well illustrated by the Sumatra and Borneo coals (No. 12, Table II). These coals are not higher in volatile matter (35 per cent) than most of the bituminous listed in Table II, yet great difficulty has been experienced in burning them because of the rapidity with which the volatile matter is given off.[1]

Considerable experience was necessary before these coals could be burned successfully. Large furnace volumes were found necessary, and higher percentages of excess air were required when the coal was freshly fired. They were successfully used in water-tube boilers fitted with mechanical stokers.

There are enormous reserves of coal in the world, and coal is widely distributed, although the percentage of the world's coal fields within the Southern Hemisphere is small. Coal is found in South Africa and eastern Australia, but only very small amounts are found in South America. The world's leading producers of coal are the United States, Germany, and the United Kingdom. Very little coal is produced, however, on the West coast of the United States. At the present time only the better grades of coal and those accessible to industry and the seaboard are mined.

20. Fuel Oil for Steamships.—Fuel oil began to replace coal in merchant ships rather generally around 1914. It was used earlier than this in the Navy and several merchant ships on the West Coast were using oil as early as 1904. Since then it has been gradually replacing coal as a boiler fuel at sea and has

[1] H. T. Baker, "The Combustion of Tertiary Coal on Ships of the Royal Dutch Packet and Navigation Co. in the Indian Archipelago," and D. J. L. Westenberg, "Methods of Burning East Indian Coal," World Power Conference, London, 1928.

practically replaced coal in the seagoing merchant ships of the United States and in practically all naval vessels of the world.

Two general types of fuel oils are used on shipboard today: those used under boilers of steamships, commonly referred to as "bunker *C*," and those used in Diesel engines, usually referred to as "Diesel fuel oils." Both are obtained from crude petroleum. Here we are concerned only with those fuel oils used by steamships; Diesel fuels are discussed in a latter article. Fuel oils used by steamships may be grouped under the three following headings: (1) residuals from straight running of petroleum distillation to remove the gasoline, kerosene, gas oils, and lubricating oils; (2) residuals from the cracking process; and (3) blends of residuals from the cracking process and lighter distillate gas oils.

Fuel oil has been pretty much a by-product of gasoline production, and its comparatively low price during the past few years in the United States is largely due to the enormous gasoline production. The tendency today at the refineries is toward a controlled fuel oil, and hence fuel oil is becoming a coproduct rather than a by-product of gasoline production.

Large quantities of petroleum are being produced today throughout the world, and it appears now that there will be an adequate supply of fuel oil for many years to come. However, as discussed later, as the demands for gasoline increase, more of the straight-run residual oil will be treated by the cracking process, which will result in poorer grades of fuel oil being available for steamships.

In 1939 the world's production of crude petroleum was approximately 2 billion barrels. The percentages produced in the eight leading countries were approximately as follows:[1]

	Per Cent
United States	61.0
U.S.S.R	10.4
Venezuela	9.9
Iran	3.8
Netherland East Indies	3.0
Rumania	2.2
Mexico	2 1
Iraq	1.5

[1] Figures are compiled from "Petroleum Facts and Figures," 1941, published by the American Petroleum Institute. The original source of the data is given as the U.S. Bureau of Mines. The publication contains a wealth of statistical data on petroleum and petroleum products.

The average yield of principal petroleum products per barrel of crude oil (42 gal) in 1939 at United States refineries was as follows:

	Per Cent
Gasoline*	45 0 (18 9 gal)
Kerosene	5.5
Gas oil and distillates	13 1
Lubricating oil	2.8
Residual fuel oil	24 7 (10 4 gal)

* In 1920 the gasoline yield per barrel of crude was 26.1 per cent or 11.0 gal.

21. Properties of Fuel Oil.—One of the most important physical characteristics of fuel oil for use under boilers, and especially for Diesel engines, is its viscosity. Fuel oil must be pumped from the bunkers through pipe lines and atomized before burning under a boiler or in the cylinders of an oil engine. The viscosity must be low enough to allow this to be done in a satisfactory manner.

The viscosity, or internal friction of an oil, is measured by the rate of flow in seconds of a fixed quantity through the orifice of a standard viscosimeter at a stated temperature. Several viscosimeters are in current use, but the two standard ones used in the United States are the Saybolt Universal and the Saybolt Furol. The former is used for the lighter oils, *i.e.*, lubricating oils and fuels used by Diesel engines, and the latter for the heavier fuels used under boilers. The standard temperatures used for marine fuels with the Saybolt Universal instrument is 100°F and with the Saybolt Furol, 122°F (50°C). The viscosities of boiler fuels when heated up to the low viscosities required for atomizing are usually measured in seconds Saybolt Universal. The symbols used are SSU for seconds Saybolt Universal and SSF for seconds Saybolt Furol. Except at the low ranges, the viscosity at a given temperature by the Saybolt Universal scale is ten times that of Saybolt Furol scale.

It is generally agreed that the viscosity of oil when pumped from the bunkers should not exceed 500 SSF. The longer the pipe line and the smaller its diameter the lower should be the viscosity. A viscosity of between 250 and 300 SSF is usually considered the upper desirable limit for oil at the pumps.

In order to reduce the viscosity of fuel oil in the tanks in contact with cold sea water, some heating is necessary for the heavy viscous oils before they are pumped to the settling tanks. Heating is always required before the oil is pumped to the burners.

To accomplish this heating, steam heating coils are installed in the bunkers of steamships and especially at the suction end of the fuel piping in the bunkers. Heating coils are also fitted in the settling tanks, and a fuel oil heater is placed in the line between the settling tanks and the burners. It is this pumping requirement from the double bottoms that is largely responsible for fixing the maximum viscosity of steamship fuel oils at 300 SSF at 122°F. Oils with higher viscosities would require considerable heating for ships operating in cold waters. Oil temperatures of over 130°F in the bunkers are not recommended.

With the oil pressures generally used on shipboard, a viscosity of around 150 SSU at the burners seems to give the best results for combustion and high boiler efficiency. Changes from this viscosity will result only in small gains. The viscosity required at the burners will, of course, vary with the oil pressure used. When higher oil pressures are employed, the same particle size can be obtained with higher viscosities.

Murdaugh[1] gives the following table of atomizing temperatures as the approximate correct ones for obtaining a viscosity of 150 SSU with present-day fuels:

Viscosity SSF at 122°F	Atomizing Temperature, °F
50	168
100	189
150	202
200	210
300	221

The gravity of fuel oils and other petroleum products is now expressed in the United States in degrees A.P.I. (American Petroleum Institute) instead of according to the former Baumé scale.

The formula for the A.P.I. scale is

$$\text{Degrees A.P.I.} = \frac{141.5}{\text{specific gravity at } 60/60°\text{F}} - 131.5$$

The values by the A.P.I. scale differ only slightly from those of the Baumé scale. The degrees A.P.I. for water is 10; oils heavier than water have values less than 10, and oils lighter than water have values greater than 10.

[1] A. C. Murdaugh, "A Method of Determining the Correct Burning Temperature for Fuel Oils," *Jour. Am. Soc. Naval Eng.*, 1936, p. 377.

The heat value of fuel oil is expressed in the same way as for coal—in Btu per pound at 1 atm pressure, with water as liquid. This is the gross or high heat value, and it is adequate for comparing fuel oils, because the hydrogen constituent of fuel oils is nearly constant. However, the net or low heat value is more significant and forms a better basis for comparison of various kinds of fuels, as oil, coal, and gaseous fuels. Heat values are also frequently quoted in Btu per gallon, and since fuel oil is bought on a barrel basis and not by weight, the latter method of expressing heat value is more satisfactory. The heat value of cracked fuel oils is expressed very closely by the following formula:

$$\text{Btu per pound} = 17{,}780 + (54 \times \text{A.P.I. gravity})$$

The unit of fuel oil is the barrel of 42 U.S. gal.

Other characteristics of fuel oils for steamships often included in the specifications are the minimum allowable percentage of sediment and water, the minimum flash point, and the pour point. The percentages of carbon, hydrogen, sulphur, and oxygen in the fuel are useful in obtaining the theoretical amount of air required to burn a pound of fuel (Art. 41).

Although the foregoing physical characteristics of the fuel are useful factors in purchasing fuel oil, they do not give an indication of the burning characteristics of an oil in a boiler or the amount of solids that may be deposited in tanks, heater, atomizer, and boiler or their influence on boiler brickwork.

Previous to about 1927 most of the fuel oil used by steamships was a residue of crude petroleum after the gasoline, kerosene, Diesel and gas oils, and lubricating oils had been removed by the distillation process. This straight-run residual fuel oil met the U.S. Navy's specifications for bunker *C*. It had a gravity of around 16 A.P.I. at 60°F (specific gravity, 0.96). The specifications fixed the upper limit of the viscosity at 300 SSF at 122°F, and the viscosity of most of the fuels was near this limit. The heat value was around 18,800 Btu per lb, 150,000 Btu per gal, and its weight was approximately 8.0 lb per gal. The percentage of hydrogen varied generally between 11.5 and 13.5 per cent. These straight-run residual fuels were very satisfactory fuels and were burned in merchant and naval vessels with comparatively little difficulty.

The increasing demands for gasoline and distillate fuel oils (furnace and Diesel oils, etc.) have caused a larger and larger quantity of the straight-run residual oils and the heavier gas-oil distillates to be treated by the cracking process to produce gasoline and the lighter distillate fuel oils. In 1918 only about 14 per cent of the gasoline produced in the United States was obtained by the cracking process; in 1940 the percentage had risen to about 53 per cent. The result is that today practically all the fuel oils used by steamships are cracked residua or cracked residua blended with distillate fuel oils.

These cracked residual oils have a higher specific gravity, a lower percentage of hydrogen, a lower viscosity, and a higher percentage of ash than the former straight-run residual fuels. Because of the fact that a much larger proportion of the crude petroleum is now going into gasoline and the distillate fuel oils, there is greater concentration of ash and other objectionable impurities in present-day residual fuel oils.

The cracking process has caused changes in fuel oil, with the result that some of the carbon in the oil is in an especially refractory form, being made up of high-molecular-weight compounds of carbon, hydrogen, and oxygen, known as "asphaltenes," which are very resistant to oxidation in the boiler furnace. When burning cracked residua, these compounds leave the furnace incompletely burned and form deposits on the ends of the burners, hard deposits on the boiler and superheater tubes, and granular soot deposits in the cooler parts of the boiler and on deck. These latter granular particles are referred to as "stack solids." Some refineries are maintaining a control of the qualities of their products by suitable blending that reduces the tendency for stack solids to form. Some of these cracked fuels also have a tendency to deposit sludges in the bunker tanks.

The ash in some of the present-day fuels also causes serious erosion of the furnace brickwork. Fuels that show vanadium oxide in the ash are particularly hard on the brickwork, for the vanadium oxide has a property of entering the pores of the brickwork and causing rapid disintegration (see Art. 38).

The present-day fuel oils are still referred to as "bunker *C*," although the term has little significance. The only specifications set for this so-called "bunker *C*" are:

Minimum flash point, 150°F.
Maximum viscosity, 300 SSF at 122°F.
Maximum water and sediment, 2 per cent.

The specifications used for bunker *C* are the same as federal specifications (National Bureau of Standards) for Grade 6 fuel oil. The federal specifications have the additional requirement of a minimum viscosity of 45 SSF at 122°F.

The physical characteristics of these cracked residual fuels and their behavior in boilers vary greatly, depending on the field from which the crude oil has come, the degree of cracking practiced by the refineries, their treatment during production, and the amount of blending they receive. Obviously, the more distillate fuel blended into the fuel the higher the price. The urge to use a low-priced fuel naturally tends toward the use of lower priced and unblended fuels.

Tests carried out with 18 samples of bunker fuels by the U.S. Navy at its boiler laboratory[1] showed the following range in characteristics for cracked residual fuels, some blended and some unblended:

A.P.I. gravity	6.7–15.25
Flash point, °F	160–260
Pour point, °F.	−10–+65
Carbon residue, per cent	6.6–17.6
Ash, per cent	0 005–0.13
Carbon, per cent	86 28–89.91
Hydrogen, per cent	9 42–11.08
Sulphur, per cent	0 33–2.56
Water by distillation, per cent	0.1–0 9
Btu per pound	18,000–18,700
Btu per gallon	150,275–154,044

Ten analyses published by M. J. Hanlon[2] for cracked residual fuels purchased under contract specifications from a number of oil refineries in 1934 show the following range in characteristics:

Viscosity, SSF at 122°F	93–132
A.P.I. gravity	2 4–3.7
Btu per pound	17,753–17,944
Sediment, per cent	0.15–0.63
B.S. & W, per cent	0.2–6.0
Sulphur, per cent	0.86–1.05
Flash, °F	194–202
Pour point, °F	10–15

[1] C. A. Jones and J. E. Hamilton, "Bunker Fuel Oil Problems," *Trans. Soc. Naval Arch. and Mar. Eng.*, 1934.

[2] *Trans. Soc. Naval Arch. and Mar. Eng.*, 1934, p. 351.

The following specifications are for a cracked residual fuel sold in large quantities as a marine bunker fuel at North Atlantic ports before the outbreak of the war in 1939. It represents a carefully controlled fuel and is probably a fair average of bunker *C* fuels in gravity and viscosity.

A.P.I. gravity	10.5
Viscosity, SSF at 122°F	175
Flash point	200°F
Water, per cent	0.3
Sediment, per cent	0.2
Ash, per cent	0.1
Carbon residue (Conradson), per cent	13.0
Pour point	40°F
Ultimate analysis	
Carbon, per cent	86.2
Hydrogen, per cent	10.3
Oxygen, per cent	0.7
Nitrogen, per cent	0.4
Sulphur, per cent	2.3
Heating value (Btu per lb)	18,500
Origin of crude	South America
Weight per gallon	8.186 lb

The present-day cracked residual fuel oils have an appreciably lower viscosity than the former bunker *C* fuels. Thus, little or no heating is required in the bunkers, and less heating is required in the fuel oil heaters, factors that are favorable in operation. Whereas the Btu value per pound is around 5 per cent lower than formerly, the specific gravity is around 6 per cent higher, and since fuel is purchased on a barrel basis the heating value per barrel is about the same as or a little higher than in 1925. These cracked fuels also have a low pour point, an important characteristic for ships frequenting cold waters.

Uncracked residual fuels are still available at some bunkering ports.

22. Colloidal Fuel.—A fuel that has been discussed as a possible marine fuel and that has possibilities as a future fuel is a coal-oil mixture of 60 per cent fuel oil and 40 per cent very finely ground coal. This fuel has frequently been referred to as "colloidal" fuel, since it is theoretically supposed to be a colloidal mixture of oil and coal. So far, however, it has not been possible to grind the coal fine enough to make a stable colloidal mixture. A much finer grind is required than that used for pulverized coal (Art. 55), and fine grinds make the fuel expensive. Stabilizing agents have been used with some success to keep the coal from precipitating. This fuel was tried out in several boilers of the *S.S. Scythia* of the Cunard Line in 1932,[1] but it was apparently not a commercial success at that time. The advantages of this fuel are: (1) it can be bunkered easily, compared with coal; (2) it can be stowed in double bottom tanks; (3) it can be burned in the same manner as fuel oil; (4) it is heavier than water; (5) low-grade coals can be used; (6) it has a higher heat value per gallon than fuel oil. Besides the difficulty of securing a colloidal mixture and the expense of grinding the coal to fine particles as mentioned above, another objection is that the fuel will have a much higher percentage of ash than oil, and difficulties may be encountered in boiler furnaces and the cleaning of burners. For nations having no oil reserves and large coal supplies, colloidal fuel offers interesting possibilities.

[1] *The Shipbuilder and Marine Engine Builder*, vol. 39, pp. 361–362; vol. 40, p. 287; *Fairplay*, Nov. 30, 1933, p. 20.

CHAPTER III

MARINE BOILERS

23. Classification.—Marine boilers can be divided into three main groups: (1) cylindrical internally fired fire-tube or Scotch boilers; (2) water-tube boilers such as are used on merchant ships; and (3) the small-tube water-tube boiler of the so-called "express" type used on high-speed naval vessels and, to some extent, on fast passenger ships. There is no sharp line of demarcation between the last two types. Practically all the boilers used on shipboard today employ natural circulation. However, marine boilers using forced circulation have been used to a very limited extent and may be employed to some extent in the future on shipboard.

24. Scotch Boilers.—The Scotch boiler was used almost universally in the world's merchant ships up to about 1914. It is still being fitted in a great many foreign merchant ships, especially British. In the United States it has been completely superseded by the water-tube boiler in all seagoing merchant ships constructed during the past 10 years. The navies of the world gave up the Scotch boiler many years ago because of its excessive weight. However, most of the existing merchant steamships of the world still produce their steam by means of Scotch boilers. On Jan. 1, 1938, 90 per cent of the gross tonnage of the world's merchant *steamships* of 2,000 gross tons or over used this type of boiler. The percentage of the existing gross tonnage of United States ocean-going merchant steamships on this same date fitted with Scotch boilers was 71.

Scotch boilers still find a wide use on the bulk carriers and other ships on the American Great Lakes, but the recent bulk cargo ships constructed on these lakes have been fitted with water-tube boilers.

The construction of the Scotch boiler is clearly shown in Fig. 7 and in Figs. 8, 9, and 10. These figures should be studied carefully in conjunction with the following description. It consists of a cylindrical plate shell and two flat plate ends that are flanged and riveted to the cylindrical shell. The furnaces, which are of corrugated steel, are fitted internally and hence are entirely surrounded by water. Each boiler is fitted with two or more furnaces.

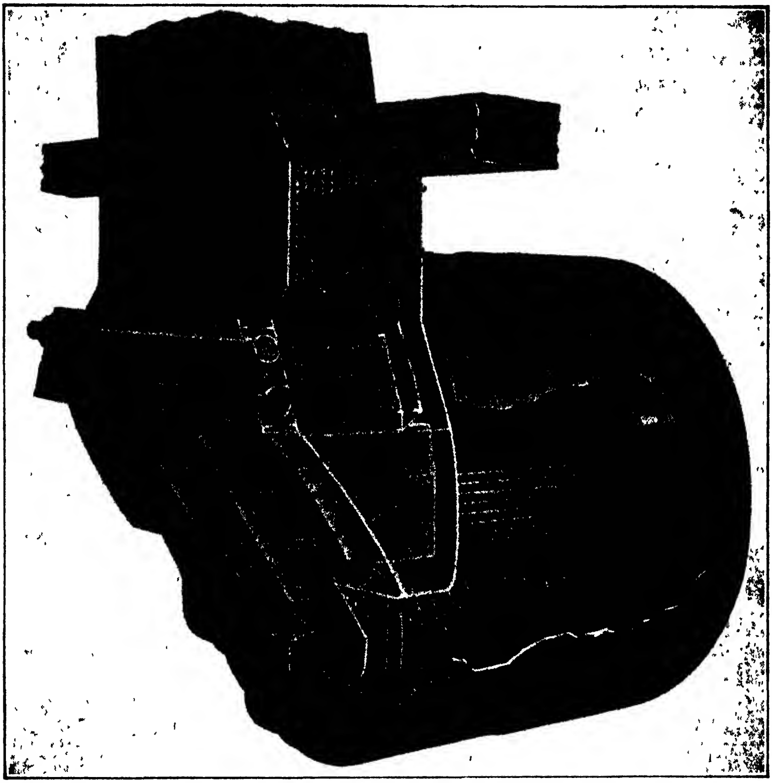

Fig. 7.—Scotch boiler with waste-heat type superheater and Howden air heater in the uptake.

The furnaces are attached to combustion chambers located in the back of the boiler; and the boiler tubes lead from the upper part of the combustion chamber and extend through the boiler to the front head. As shown in the drawings and figures, the furnaces, combustion chamber, and boiler tubes are completely surrounded by water. From the furnace, the products of combustion pass into the combustion chamber and thence through the boiler tubes to the uptake on the front of the boiler.

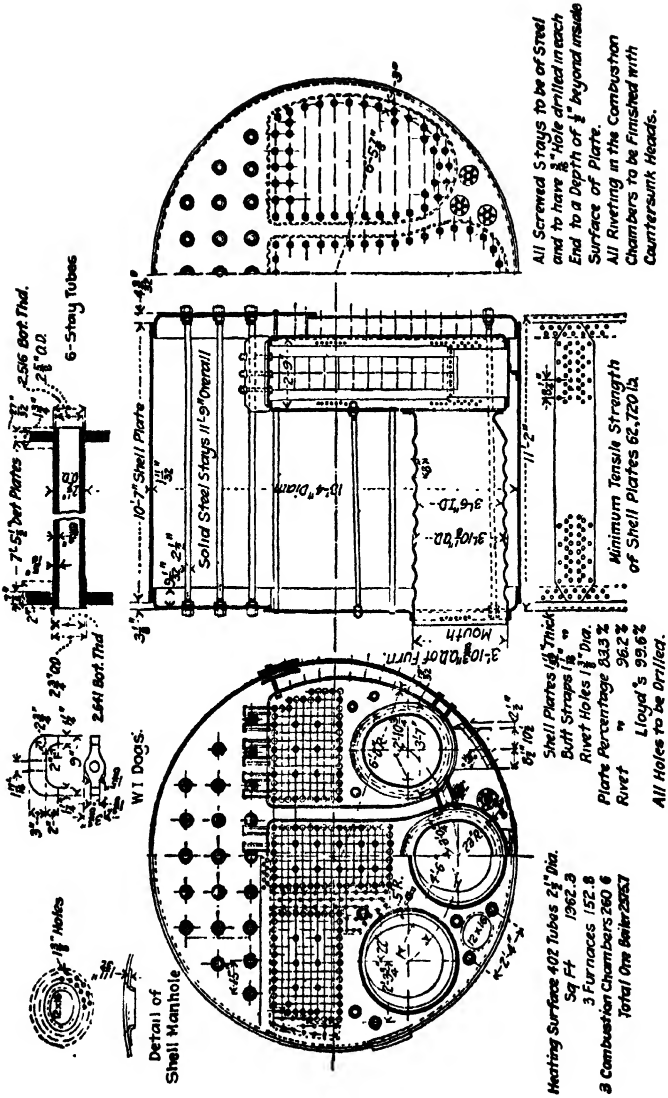

FIG. 8.—Scotch boiler with separate combustion chambers.

All the flat surfaces of the Scotch boiler subject to pressure, which include the front and back heads, the front, back, side, and top surfaces of the combustion chamber, must be thoroughly stayed to prevent collapse. In the steam space, solid steel stays extend from the front of the back head. These extend through the heads and are secured with nuts on both sides of the plate.

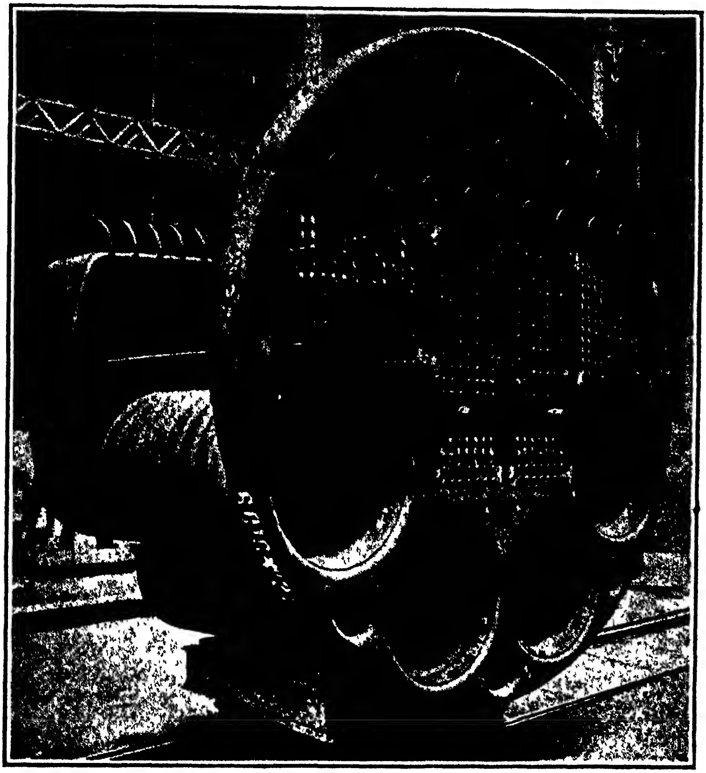

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Fig. 9.—Scotch boiler under construction—front view.

The stays in the steam space must be placed far enough apart to allow for inspection and cleaning. The boiler tubes are expanded into the front head plate and the combustion chamber tube plate but are not secured well enough to stiffen these two flat plates. To stiffen these flat plates, stay tubes, which are extra-thick boiler tubes with screwed ends, are placed at regular intervals among the boiler tubes.

The back head and the back sheet of the combustion chamber are fastened together by screwed stay bolts. The top of the combustion chamber is supported by means of longitudinal girders (Figs. 8 and 10), which are attached to the flat plate by screw bolts. These girders bear on the front tube plate and

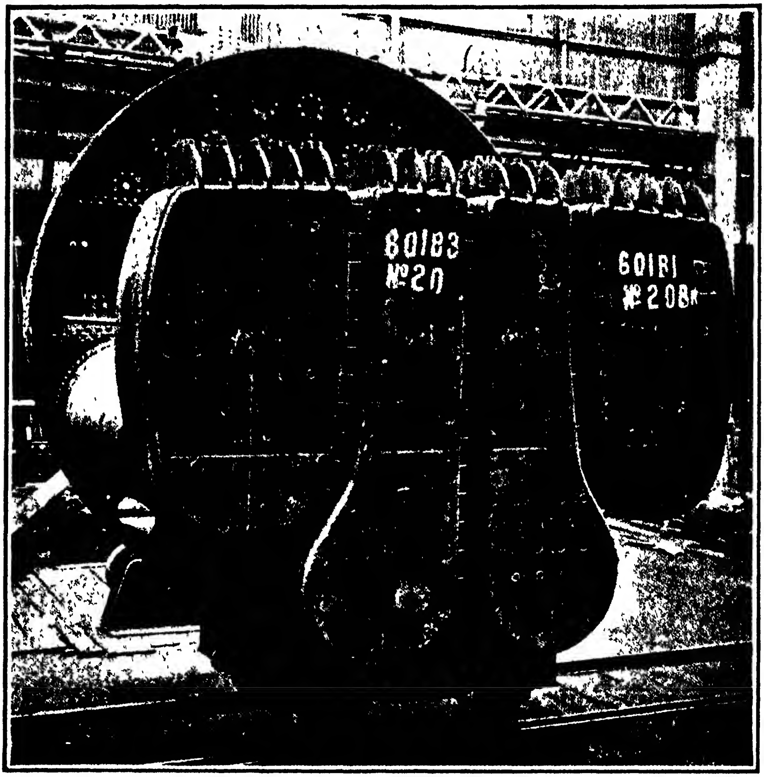


Fig. 10.—Scotch boiler under construction—rear view. Showing the four combustion chambers.

back plate of the combustion chamber and transmit the load to these members. The furnaces are corrugated both to strengthen them against collapse and to provide for longitudinal expansion.

Many Scotch boilers are fitted with separate combustion chambers for each furnace, as shown in Fig. 10. However, to reduce the cost, some ships have boilers fitted with a common combustion chamber. Scotch boilers fitted in tugs and small coastwise ships often have only two furnaces, and these are usually attached to a common combustion chamber.

Scotch boilers run from around 10 ft in diameter and 10 ft in length, with two furnaces and 1,000 sq ft of heating surface, up to 17 ft in diameter and 13 ft in length, with four furnaces and 4,000 sq ft of heating surface. Furnaces are generally between 42 and 45 in. in diameter and 7½ to 8½ ft in length. The weight of a Scotch boiler with water up to the steaming level varies between 65 and 70 lb per sq ft of heating surface.

Double-ended Scotch boilers have been fitted on many ships. Such boilers are fired at each end and the combustion chambers are placed back to back in the center of the boiler.[1]

A modification of the Scotch boiler known as the "gunboat boiler" was used on gunboats and shallow-draft passenger ships 30 years ago. In the gunboat type of boiler the combustion chambers are about midway between the two ends, and the fire tubes extended out from the back end of the combustion chamber to the back head. The uptake is on the back instead of the front, as in Scotch boilers (Fig. 7). The gunboat boiler is thus a long boiler of small diameter suitable for boats of small depth. The boiler of this type fitted on the *S.S. City of Philadelphia* in 1910 had a heating surface of 2,100 sq ft, a diameter of 9 ft 4 in., and a length of 22 ft. The reader should compare these dimensions with those shown for the Scotch boiler in Fig. 8.

The merits of the Scotch boiler are that (1) extreme care does not have to be taken, as with water-tube boilers, to have pure feed water; (2) the Scotch boiler has a large water volume and therefore is safer in the hands of inexperienced and inferior firemen; (3) repairs can be readily made in most ports of the world.

The possibilities of operating Scotch boilers with feed water that contains (1) a certain amount of salt from condenser leakage, (2) small amounts of lubricating oil, carried over from the auxiliaries (and the main engine, when reciprocating engines with superheat are used), and (3) moderate amounts of air in solution in the feed water have been strong arguments advanced for retaining Scotch boilers in cargo ships in certain trades.

[1] The *S.S. Aquitania* had 21 double-ended Scotch boilers 17 ft 8 in. in diameter and 22 ft in length. See *Engineering* (London), May 22, 1914, Fig. 174.

The defects and disadvantages of the Scotch boiler are many: (1) Because of its large water volume, it is extremely heavy. (2) It has poor water circulation. (3) Steam must be raised slowly and the boiler cooled down slowly or unequal stresses will be set up that may cause leaks to develop. (4) Scotch boilers cannot be forced as water-tube boilers can, and evaporation rates higher than 5 or 6 lb of water per sq ft of heating surface per hr are not to be recommended. The great drawback of the Scotch boiler and the main factor that is causing it to be replaced by water-tube boilers in most modern ships is the limited steam pressure that it can carry. Because of its large diameter and flat surfaces, the thick plating that would be required at high pressures has limited the pressure of Scotch boilers to around 220 lb per sq in. Today, with the average pressure carried on modern cargo and passenger liners in the neighborhood of 450 lb and with steam pressure of 600 lb and higher also being used to some extent, it is obvious that the Scotch boiler is doomed for modern steamships of high economy where high pressures are demanded.

The *S.S. City of Roubaix*, of the British Ellerman Line (1928),[1] was fitted with Scotch boilers (diameter, 16 ft 3 in.) carrying a pressure of 300 lb, but the author knows of no other ships where pressures as high as this have been used with this type of boiler. No doubt the reluctance of some shipowners to adopt water-tube boilers has held steam pressures down in many merchant ships.

Although the Scotch boiler cannot carry high steam pressures, high steam temperatures are possible, and a number of recent ships fitted with Scotch boilers and reciprocating engines are using steam temperatures in the neighborhood of 750°F (Art. 60).

Scotch boilers have found almost universal use in tramps fitted with steam machinery, and because of the nature of tramp service they will probably be used in tramps, for many years to come, when steam machinery is adopted. Tramps seldom have powers over 2,000 ihp, and, as pointed out later on, the reciprocating steam engines have special advantages over the turbine for powers lower than 2,000. However, steam machinery of small horsepower having a fuel consumption appreciably above that of the modern ship using high pressure steam will find it difficult to compete with the Diesel engine, and the tramp powered with

[1] *Shipbuilding and Shipping Record*, vol. 31, p. 244.

oil engines is in many cases the logical ship for this service (Art. 117).

When fitted with an air heater and operated at low rates of evaporation, the Scotch boiler has a good efficiency. Table III gives some test data for Scotch boilers operating on oil where over-all boiler efficiencies of 84 and 85 per cent were attained (see also Fig. 23).

Practically no seagoing ships of the United States merchant marine have been fitted with Scotch boilers since the completion of the First World War program in 1920. The last transatlantic liner fitted with these boilers was the *S.S. Aquitania* (1914). However, 60 of the ships constructed in the United States for Great Britain in 1941–1942 were fitted with Scotch boilers. The *S.S. Ile de France* (1927), of the French Line, was fitted with 32 boilers of a modified Scotch type that had water tubes attached to the rear head to improve the circulation. These water tubes were placed in an external combustion chamber. The usual type of internal combustion chamber was not fitted.[1]

25. Water-tube Boilers.—Water-tube boilers, as the name implies, are distinguished from the Scotch or fire-tube boiler in that the water circulates through the boiler tubes instead of surrounding them on the outside. Water-tube boilers are further characterized by the small volume of water they contain and have a weight of one-half to one-fifth of that of a Scotch boiler of the same amount of heating surface.

Water-tube boilers are made up of one or more drums and one or more banks of water tubes (see Figs. 11 to 18). The water level under normal steaming conditions is at about the middle of the upper drum. The steam collects in the upper part of this drum. The upper drum is usually referred to as the "steam" drum, although, as mentioned, it also contains a certain amount of water. The feed water is usually introduced into the lower part of the upper drum. The boilers are designed so that part of the tubes or tube headers act as downcomers to carry the cooler water downward, and part or all of the tubes act as risers to carry the steam and hot water to the steam drum. Several types of boilers will be described in detail in later articles.

[1] Prudhon-Capus boilers, see *The Shipbuilder,* vol. 35, p. 25; vol. 34, p. 148.

TABLE III.—PERFORMANCE OF SCOTCH BOILERS

1. Type of boiler	Scotch	Scotch	Scotch	Scotch	Scotch	Scotch	Scotch	Scotch
2. Fuel	Oil	Oil	Oil	Oil	Coal[a]	Coal[e]	Coal[a]	Oil
3. Heating surface, sq ft, H.S.	3,022	3,022	3,022	3,022	2,722	2,555	910	
4. Furnace volume, cu ft, F.V. (furnace and combustion space)	590	590	590	590	467	470	165	
5. Superheating surface, sq ft	774	774	774	774	774	0	....	[f]
6. Air heater surface, sq ft	1,500[h]	1,500[h]	1,500[h]	1,500[h]	812	638		
7. Boiler pressure, lb abs	185	189	190	193	181	190	200	224
8. Quality or superheat	24°F	20°F	45°F	45°F	47°F	99.8%	98.0%	372°F
9. Draft, in. of water	Forced	Forced	Forced	Forced	0.18	1.63	0.80	
10. Heating value of fuel, Btu per lb, as fired	18,200	18,300	18,300	18,200	13,500	14,305	13,228	18,400
11. Temperature of gases leaving boiler, °F	533	538	624	665	487	551	583	
12. Temperature of gases leaving air heater, °F	340	346	414	428	337	452	233	380
13. Temperature of air entering air heater, °F	61	65	65	72	79	88	83	
14. Temperature of air entering furnace, °F	193	202	204	229	172	217	436	237
15. Temperature of oil entering burner, °F	232	235	224	242				
16. Temperature of feed entering boiler, °F	188	208	180	211	155	209	160	279
17. Fuel burned per hr, lb	835	1,080	1,390	2,010	1,180	1,383	632	1,768
18. Water evaporated per hr, lb	12,100	16,500	19,200	29,100	11,500	14,980	6,982	22,670
19. Actual evaporation per lb of fuel	14.6	15.3	13.8	14.5	9.6	10.81	11.00	12.85
20. Equivalent evaporation per lb of fuel	...	...	...	...	...	11.36	12.00	
21. Actual evaporation per sq ft, H.S.	4.02	5.45	6.36	9.62	4.23	5.86	7.67	
22. Equivalent evaporation per sq ft, H.S	...	...	...	...	...	6.15	8.33	
23. Excess air, per cent	40.0	24.0	34.0	20.0	57.0	10.3	83.5	
24. Fuel per sq ft H.S. per hr, lb.	0.28	0.36	0.46	0.67	0.42[b]	0.54	0.0	
25. Fuel per cu ft F.V. per hr, lb.	1.42	1.83	2.36	3.40	1.53	2.94		
26. Efficiency of boiler, superheater, economizer, and air heater	84.5	86.5	81.0	83.0	78.0	77.1	88.1	80.75
27. Heat lost in dry stack gases, per cent	7.0	6.5	8.0	8.0	7.5	7.5	5.2	
28. Heat carried away by steam in gases, per cent	7.0	6.5	7.0	7.0	3.5	3.9		
29. Heat lost by CO in stack gases, per cent	0.0	0.0	0.0	0.0	0.5	0.0	0.0	
30. Radiation and unaccounted for, per cent	1.5	0.5	4.5	2.0	6.5[c]	6.1[d]	4.3[g]	
	Test	Test	Test	Test	Test	Test	Test	Service performance
31. Reference	[i]	[i]	[i]	[i]	[i]	[j]	[k]	[l]

[a] Hand-fired.
[b] Coal per square foot G.S. per hr = 19.2.
[c] Unburned carbon in ash = 4.0 per cent.
[d] Unburned carbon in ash and flue gas = 5.4%.
[e] Pulverized coal.
[f] Combustion-chamber superheater.
[g] Unburned carbon = 2.4 per cent.
[h] Approximate.
[i] *U.S. Bur. Mines Bull.* 214.
[j] Jefferson, *Trans. Soc. Naval Arch and Mar. Eng*, 1937.
[k] Owen, Institute of Engineers and Shipbuilders in Scotland. vol. 67.
[l] *Trans. Inst. of Naval Arch.*, 1938, p. 70.

The early water-tube boilers had furnaces lined with brickwork (see Fig. 17), but the tendency in design today is to enclose the furnace either partially or completely with water tubes. This change in design is partly to secure a lighter and more efficient boiler and partly to do away with the brickwork because of the serious erosion of brickwork that has taken place with present-day fuels (see Art. 38).

All burners in most marine water-tube boilers are usually located in one end or at one side of the boiler, but some of the water-tube boilers fitted on German ships are double-ended; *i.e.*, burners are located at each end.[1]

The drums of water-tube boilers are placed fore and aft in the ship whenever possible so that the water in the drums will not be influenced by the ships rolling. When drums are placed athwartship, baffles must be installed in the drums.

Many of the marine water-tube boilers being built today (1942) are for steam pressures varying between 400 and 600 lb per sq in. Most of the existing water-tube boilers in the older ships, however, are carrying steam pressures ranging from 250 to around 375 lb. A number of merchant ships in service are carrying pressures higher than 600 lb,[2] and one of the cargo ships of the American Export Line has boilers carrying a pressure of 1,200 lb per sq in. and uses the reheat cycle.[3]

26. The Babcock and Wilcox Cross-drum Boiler.—The latest development of this make of boiler is shown in Figs. 11 and 13 (see also Fig. 21). The straight water tubes forming the larger part of the boiler heating surface are placed above the furnace at an angle of about 15 deg. Vertical sinuous headers, as shown in Fig. 12, extend along the front and back of the boiler. The front headers are connected directly to the steam and water drum that extends across the front of the boiler, and they act as downcomers. The back sinuous headers that act as risers are con-

[1] *The Shipbuilder and Marine Engine Builder*, vol. 43, p. 59.

[2] *S.S. Potsdam*, 1,500 lb per sq in., *The Shipbuilder and Marine Engine Builder*, vol. 43, p. 295; *S.S. Scharnhorst* and *S.S. Gneisenau*, 710 lb, *The Shipbuilder and Marine Engine Builder*, vol. 42, pp. 413 and 457, and *Schiffbau*, Mar. 15, 1936; *S.S. Tannenberg, The Shipbuilder and Marine Engine Builder*, vol. 43, p. 56.

[3] B. Fox and R. H. Tingey, "A 1200-Pound Reheat Marine Installation," *Trans. Soc. Naval Arch. and Mar. Eng.*, 1941.

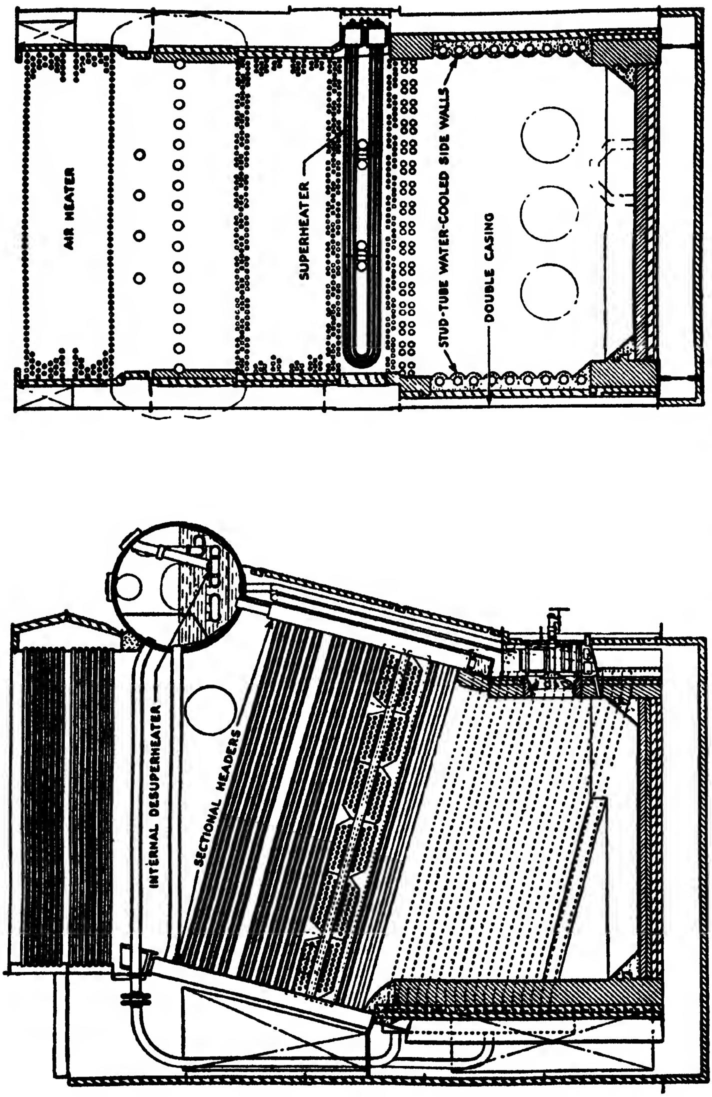


FIG. 11.—Babcock and Wilcox single-pass cross-drum boiler.

nected to the steam drum by horizontal tubes, as shown in Fig. 11. The boiler tubes are thus divided into vertical sections, and the shape of the headers causes the tubes to be staggered. This staggering of the tubes causes the hot gases to be retarded in their passage through the boiler and thus to give up their heat to the water in the tubes. The circulation in the boiler is down the front headers, up the sloping tubes and back headers, and through the horizontal tubes to the steam drum.

As shown in Fig. 12, the tubes that are expanded into the headers are arranged with a handhole for every 9 tubes. These handholes are for cleaning and replacing the tubes. The size of the tubes used in boilers of this type varies from 1 to 4 in. in diameter. The number of tubes per handhole varies with the diameter from 1 tube per handhole for 4-in. tubes to 14 tubes per handhole for 1-in. tubes.

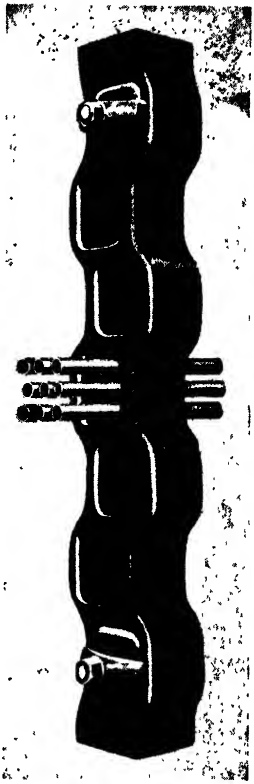


FIG. 12.—Header for Babcock and Wilcox boiler with nine-tube arrangement.

This boiler is of the single-pass type, the gases passing straight up around the staggered tubes, no baffle plates being used.

The superheater is shown in the lower part of the tube bank just above the two rows of furnace tubes; an air heater is located on top of the boiler. The present practice, when this type of boiler is fitted on merchant ships, is to use tubes of $1\frac{1}{4}$ to 2 in. in diameter; 4-in. tubes are used in the lowest row, just above the furnace. The Babcock and Wilcox boilers constructed 20 years ago for the merchant service used 4-in. tubes entirely, but very few, if any, boilers with the main tubes of this diameter are constructed today.

The boiler shown in Fig. 11 has special water-cooled furnace side walls. The two ends of the furnace are fitted with fire brickwork. The tubes at the side of the furnace are of the patented Babcock and Wilcox stud type shown in Fig. 13.

These stud tubes are covered with plastic chrome ore refractory. This construction gives a water-cooled furnace yet provides a refractory surface to protect the water tubes and assist combustion by maintaining a high furnace temperature. This construction allows a smaller furnace volume than would be possible if

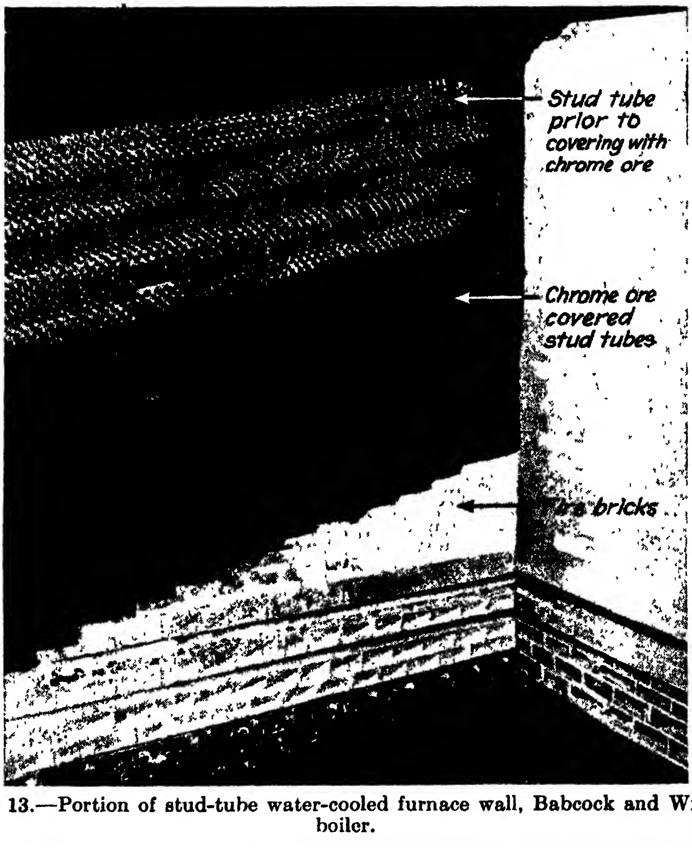


Fig. 13.—Portion of stud-tube water-cooled furnace wall, Babcock and Wilcox boiler.

bare tubes surrounded the furnace. The plastic chrome ore is very durable and is easily replaced. As shown in Fig. 11, the furnace water tubes are connected with the steam drum by downcomers in the front and by risers at the back. Some of the latest boilers of this type have the stud water tubes on the front and rear walls, also. The furnace floor of this boiler is protected by brickwork and is not water-cooled.

The location of the fuel oil burners is shown clearly in Fig. 11. Boilers of this type are usually built for pressures of around 450 lb and steam temperatures of around 750°F. They have been widely used in American merchant ships. Some have been built for pressures of 600 lb and steam temperatures of 835°F.[1]

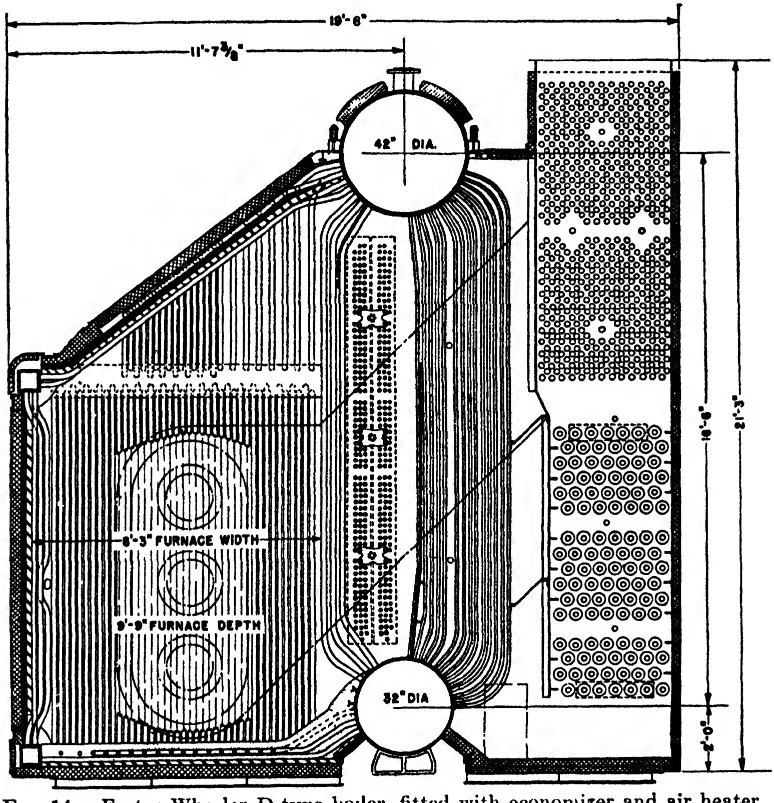


Fig. 14.—Foster Wheeler D-type boiler, fitted with economizer and air heater.

27. The Foster Wheeler Boiler.—Another boiler used extensively in modern American merchant ships, is the Foster Wheeler two-drum D type shown in Fig. 14. This boiler has two drums: a water drum at the bottom and a steam and water drum at the top. The steam generating tubes are vertical and straight for most of their length; they are bent at the ends in order to allow them to enter the drums normally. The furnace is completely

[1] See footnote on p. 19.

water-cooled, and the furnace tubes are bare except on the floor of the furnace. An interior view of this boiler, giving a good idea of the furnace tubes, is shown in Fig. 15. It will be noted that all four furnace walls are water-cooled. The superheater is placed between the two drums and is protected from the radiant heat of the furnace by three rows of water tubes. This boiler is fitted with an economizer (Art. 144) and an air heater above the economizer. On leaving the furnace, the gases pass through the

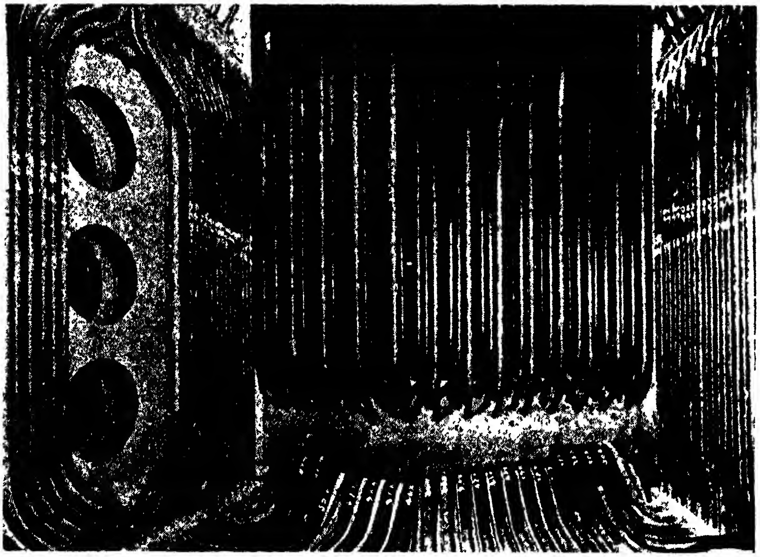


Fig. 15.—Interior view of furnace of Foster Wheeler D type boiler. Openings for burners are shown at the left.

water tubes next to the furnace, then through the superheater tubes, and are then deflected by a vertical baffle downward through the main bank of boiler tubes and thence up through the economizer and air heater before going up the stack. This type of boiler fitted on the U. S. Maritime Commission's C-type ships had a designed efficiency of 88 per cent for normal operating conditions, with a steam pressure of 465 lb at superheater outlet and a steam temperature of 750°F. The three circles in Fig. 14 show the location of the oil burners.

28. Combustion Engineering Company's Boiler.—Figure 16 is a very instructive illustration of the Combustion Engineering Company's marine boiler, which is fitted on a large number of the U. S. Maritime Commission's C-2 type ships. It is of the D type, with a completely water-cooled furnace, a superheater protected by three rows of water tubes, an economizer, and an air

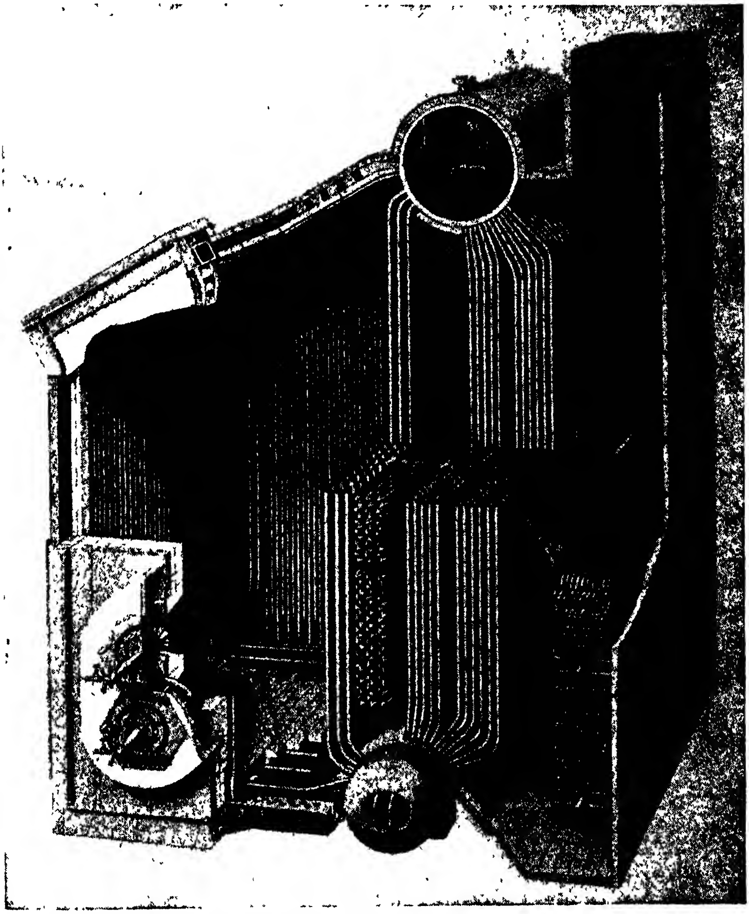


FIG. 16.—Sectional view of Combustion Engineering Company's boiler.

heater. The brickwork protecting the horizontal furnace floor tubes, the tube headers at the corners of the furnace, and the fuel oil burners are clearly shown in Fig. 16.

29. Three-drum Express-type Boiler.—Boilers commonly referred to as the "Yarrow" or "express" type are shown in

Figs. 17 and 18. Figure 17 shows a present-day Yarrow boiler fitted on a destroyer of the Portuguese navy.[1]

The Yarrow-type boiler was originally of the three-drum type similar to that in Fig. 18. In the earlier types of Yarrow boilers, which had three drums, the superheater was located at one side, above the sloping water tubes. However, the present-day demands for high degrees of superheat have necessitated placing the superheater closer to the furnace, where it will be in the high-temperature gases. This demand has caused present-day Yarrow boilers to be built with four drums, as shown in Fig. 17. Boilers of this general type but of heavier construction have been fitted on a large number of passenger liners and on a few cargo ships.

The boiler shown in Fig. 17 has a complete brick-lined furnace. It is also fitted with air heaters of small heating surface on each side of the boiler. The eight circles in Fig. 17 show the location of the burners.

A lightweight Babcock & Wilcox three-drum boiler is shown in Fig. 18. This boiler has the furnace divided into two parts, with a water-cooled division wall composed of stud tubes and covered with chrome ore, as described for the boiler in Art. 26. The purpose of the divided furnace is to allow the temperature of the superheated steam to be controlled (Art. 33). It will be noticed that superheater tubes are fitted on only one side of the boiler. The boilers fitted on the former transatlantic passenger liner *S.S. America* (1940)[2] were of the general type shown in Fig. 18, but without the furnace division wall. Figure 20 shows another three-drum boiler in which no furnace brickwork is fitted except at the ends. Like the Foster Wheeler D type boiler, the furnace floor of this boiler is water-cooled. This boiler is discussed in Art. 31.

30. Superheaters.—As pointed out in Art. 6, steam is superheated by being heated above the temperature corresponding to that of saturated steam. Steam cannot be superheated while it is in contact with a boiling liquid, and hence the saturated steam is withdrawn from the boiler drum and heated in a separate heater known as a "superheater."

[1] Taken from a paper, "Trial Performances of a Torpedo-boat Destroyer," by H. E. Yarrow, *Trans. Inst. Eng. and Shipbuilders in Scotland*, vol. 78, 1934–1935, pp. 131–152.

[2] See *Trans. Soc. Naval Arch. and Mar. Eng.*, 1940, p. 34.

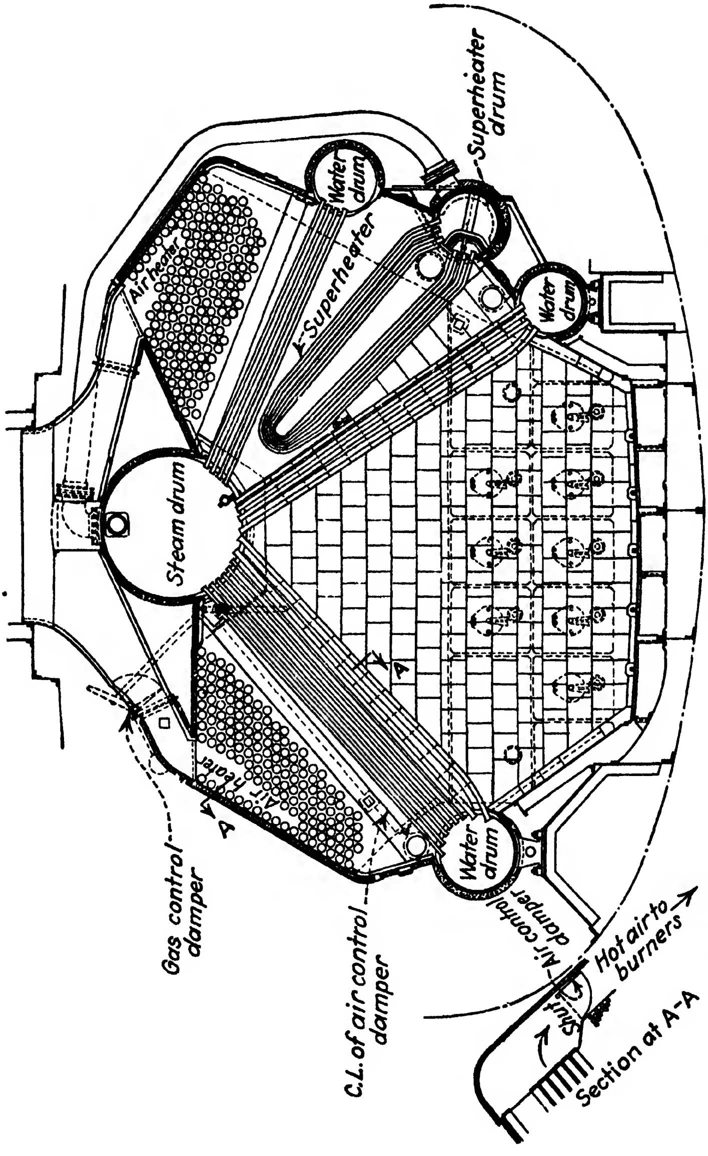


FIG. 17.—Yarrow boiler fitted on destroyer. (*By permission of Inst. Eng. & Shipbuilders in Scotland.*)

Superheaters may be divided into four main types: (1) the waste-heat type installed in the boiler uptakes where the steam is superheated by the hot gases leaving the boiler, (2) the convection (interdeck) type installed within the boiler heating surface in the direct passage of the hot gases; (3) the radiant type installed in the boiler furnace, receiving direct radiant heat from the burning fuel; and (4) separately fired superheaters.

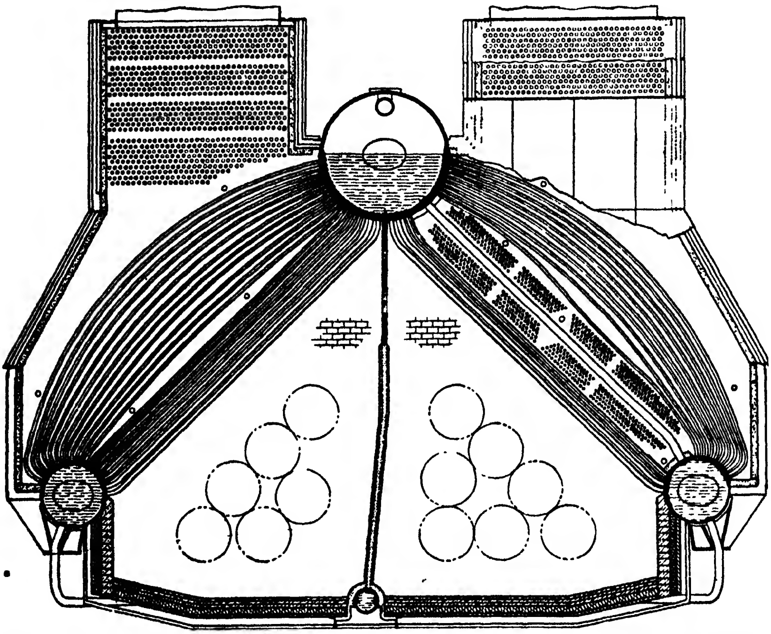


FIG. 18.—Babcock and Wilcox three-drum express boiler with divided furnace.

The degree of superheat attained depends upon the location of the superheater and the amount of superheating surface installed. The higher the degree of superheat desired the closer must the superheater be located to the furnace.

Superheaters of type 1 have been widely used in connection with Scotch boilers. An illustration of a waste-heat superheater is shown in Fig. 7. Here the superheater is shown in the uptake just above the level of the boiler tubes. Many of the superheaters of this type, especially those installed in American ships, were of the Foster type, consisting of steel tubes with cast-iron gilled rings on the outside to increase the heating surface, similar to those shown on the economizer tubes in Fig. 125. In the superheater shown in Fig. 7, the steam makes four passes through the gases. The dry-steam header and superheated steam header are clearly shown in the illustration. Superheaters of this type can produce a superheat of only 50 to 60°F, with steam pressures around 200 lb and uptake temperatures of around 500 to 600°F, but when used in connection with Scotch boiler and reciprocating engine installation, this amount of superheat results in an appreciable gain in economy.

The superheat in a waste-heat superheater is obtained from the heat rejected from the boiler in uptake gases, hence there is no increase in fuel consumption to obtain the superheat. All the other types of superheaters require the expenditure of additional fuel to obtain the superheat. Tests with Scotch boilers with and without waste-heat superheaters show an increase of 2.5 points in the over-all boiler efficiency when a superheater is fitted. Waste-heat superheaters have never been used with water-tube boilers to the author's knowledge.

When higher degrees of superheat are desired with Scotch boilers, fire-tube superheaters are used, (Fig. 19). This type of superheater has two vertical headers located on the front of the boiler within the uptake casing. Steam leaves the dry-steam header through a series of small U tubes that extend into the boiler tubes. These tubes extend partway through the boiler tubes and are fitted with return bends so that one tube extends in and out of about six tubes before entering the superheated steam header. The superheating tubes are supported by small carriers inside the boiler tubes so that they are not in contact with the heating surface. The degree of superheat obtained is fixed by the distance that these units extend into the boiler tubes. Superheat as high as 250 degrees can be obtained with this type of superheater. Mechanical soot blowers can be fitted with this superheater; hand cleaning would be difficult and not thorough.

A third type of superheater fitted to Scotch boilers has come into use rather recently. This is the combustion-chamber superheater. As the name implies, it is located wholly within the combustion chamber. The temperature within the combustion chamber is in the neighborhood of 1200 to 1500°F, compared with a mean temperature of around 700 to 800°F in the fire tubes; hence high degrees of superheat can be obtained with this type of superheat with a small amount of superheating surface. Some recent British installations using Scotch boilers, combus-

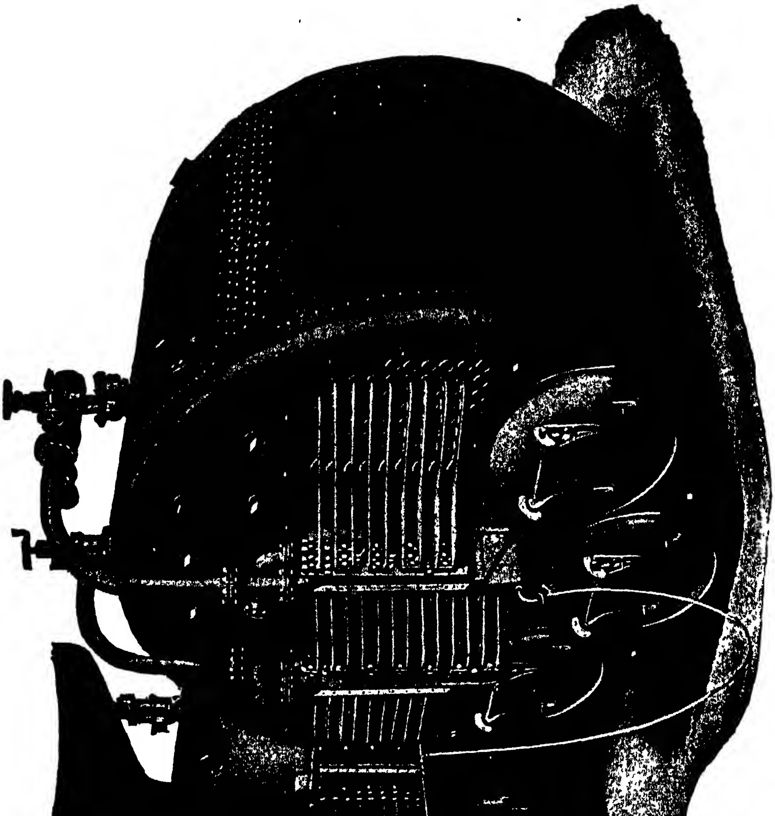


FIG. 19.—Fire-tube superheater installed in Scotch boiler.

tion-chamber superheaters, and reciprocating steam engines attained temperatures of 750 to 775°F at the superheater outlet (Art. 60). The combustion-chamber superheater has an advantage over the two former types in that it does not cause a reduction in draft and hence it can be used at times with natural draft.

The convection superheaters fitted in water-tube boilers are clearly shown in Figs. 11, 14, 16, 17, 18, and 21. With the present demand for steam temperatures of between 650 and 850°F, it is necessary to locate these superheaters close to the furnace, where the gases are very hot. In order to protect the superheater from the radiant heat of the furnace, one or more rows of boiler tubes are placed between the superheater and the furnace. When only low degrees of superheat are required, more boiler tubes are placed between the furnace and superheater. These superheaters are usually of the U-tube construction, with the dry-steam header and superheated-steam header located side by side (see Fig. 11). The headers are often constructed with diaphragms so that the steam can make several passes through the superheater. Provision must be made for withdrawing the complete superheater or the tubes for inspection and tube renewal; hence sufficient length must be allowed for this purpose in locating the boilers in the ship.

Radiant superheaters located in the boiler furnaces have been used on shipboard only in a few installations. High degrees of superheat can be obtained with a small amount of heating surface. The possibility of securing dangerously high steam temperatures with careless operation has tended to discourage the use of radiant superheaters on shipboard.

31. Separately Fired Superheaters.—A few installations of separately fired superheaters have been made on shipboard.[1] The advantage of this type of superheater is that it allows steam temperature to be always under control. A boiler with an attached separately fired superheater is shown in Fig. 20. The gases leaving the superheater pass through the boiler furnace and boiler tubes along with the hot gases produced in the main furnace. Figure 18 shows a modification of the separately fired superheater. Here the superheater is installed on one side of the divided furnace, and the degree of superheat can be increased or reduced by burning more or less fuel in the right-hand side of the divided furnace. The separately fired superheater is especially valuable in ships that run at variable speeds; where

[1] See *Mar. Eng. & Shipping Rev.* Aug. 1938, for an account of the tanker *Associated.*

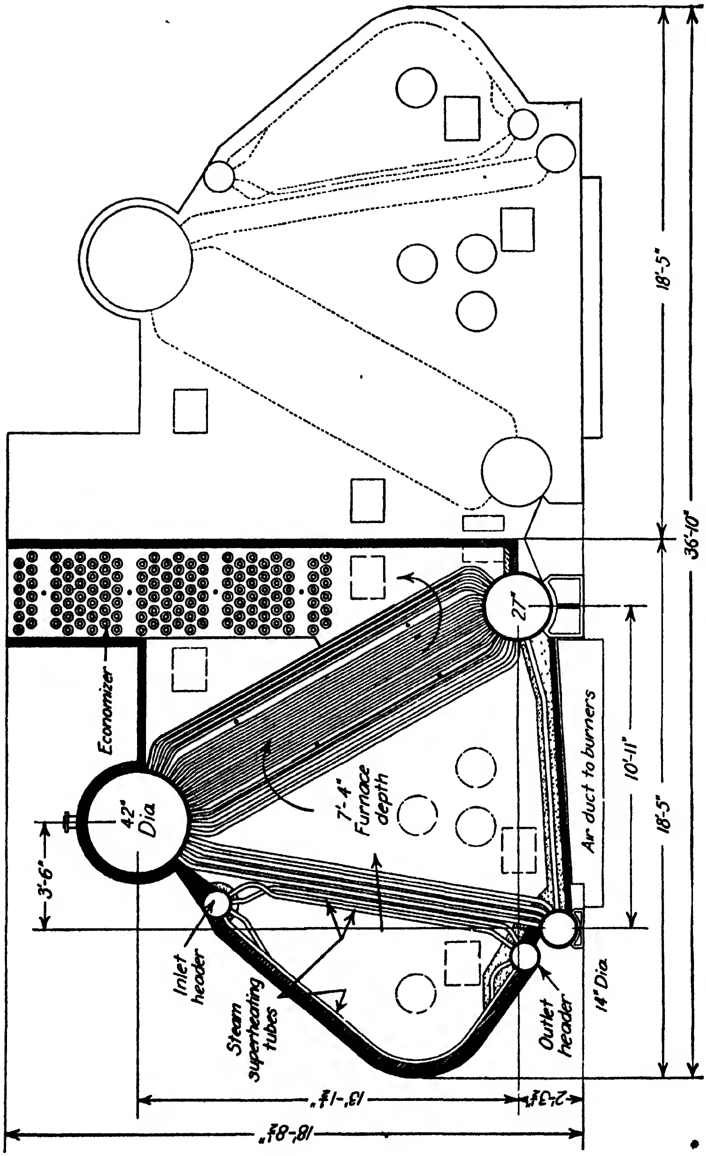


FIG. 20.—Arrangement of two Foster Wheeler boilers with separately fired superheaters.

reversing for long periods may be necessary; and for use in port where superheated steam is not desired for auxiliaries and steam winches.

32. Desuperheaters.—Superheated steam is practically never used in the auxiliary engines of steamships except in turbine-driven feed pumps, and when superheated steam is used at the main throttle provision must be made for a supply of nonsuperheated steam for the auxiliary steam line. The only exception occurs when the auxiliaries are driven by electric motors from an auxiliary generating set. Under these conditions the auxiliary generating set usually operates with the same initial steam conditions as the main turbines. However, even in this case there are large demands for saturated steam in the engine room.

Formerly, when only a low degree of superheat was used and the superheaters were placed in the boiler in positions of relatively low gas temperature, a certain amount of dry steam could be taken from the boiler drum and sent to the auxiliary steam line. However, with the high steam temperatures now used, the superheater is located close to the boiler furnace, and it is essential that as much steam as possible be circulated through the superheater when the fires are burning to protect the superheater from damage. Such a case, of course, would arise when the main engines were shut down and all the auxiliaries were in operation. To meet this situation it is the practice to pass all the steam generated in the boiler through the superheater and desuperheat the steam for the auxiliaries before sending it to the auxiliary steam line. The common practice today on shipboard is to pass the steam through a desuperheater consisting of a number of coils or a series of tubes located under the water surface in the upper steam drum of the boiler (Figs. 11, 16, and 21). In the desuperheater the steam is at a temperature above that of the water in the drum, and hence heat passes from the superheated steam to the boiling water. With this type of desuperheater the steam enters the auxiliary line with around 20 degrees of superheat instead of in saturated condition, but by the time the steam reaches the auxiliary engines it has been reduced nearly to the saturated condition. Other means of desuperheating steam are discussed in the next article.

33. Control of Superheat.—As indicated in Art. 31, it is very desirable to control the degree of superheat, *i.e.*, to prevent

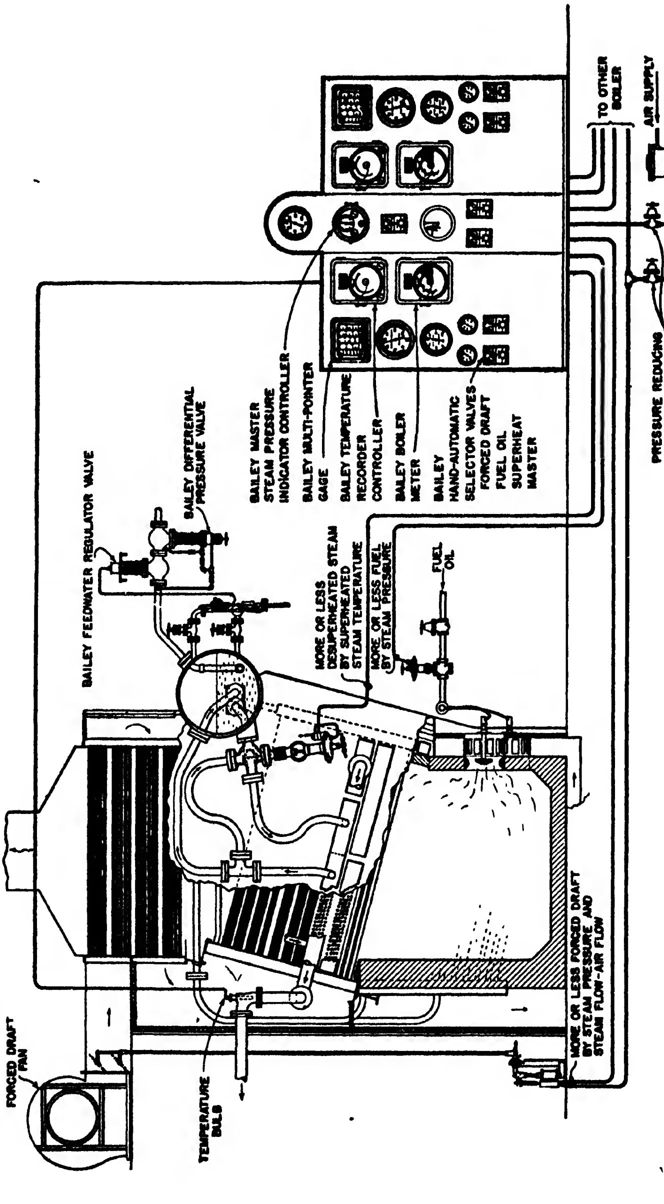


FIG. 21.—Diagrammatic layout of Bailey boiler control system, fitted to Babcock and Wilcox boiler.

excessive steam temperatures at reduced speed, in maneuvering, in reversing, or in operating in port, where no superheat is desired. Besides these changes in steam demands, the degree of superheat will be influenced by the rate of combustion, the feed-water temperature, the amount of saturated steam sent to auxiliaries, the percentage of moisture in the steam leaving the boiler, and the amount of excess air supplied to the boiler.[1]

The following means are employed for controlling superheat:

1. Desuperheaters; usually located beneath the water surface in the upper boiler drum. Desuperheaters sometimes operate with water spray or by use of the boiler feed water as a cooling medium.
2. Dampers in the boiler uptakes (see Fig. 17), in boilers with two uptakes.
3. Separately fired superheaters.

Boilers are now being fitted with a complete automatic control that regulates combustion (*i.e.*, fuel and air supply), feed-water supply, and the temperature of the steam entering the main steam line. A diagram of the Bailey boiler control is shown in Fig. 21. A full explanation of this control apparatus cannot be given here, but a study of Fig. 21 will show that there is a valve automatically governed by steam temperature that controls the amount of main steam to be desuperheated between two passes of the superheater, and in this way the temperature of the steam leaving the boiler is controlled.

34. Boiler Performance.—The rate of evaporation of a boiler is expressed in pounds of water per square foot of boiler heating surface per hour. The service rate of evaporation for a Scotch boiler seldom exceeds 5 or 5.5 lb. Rates of evaporation for water-tube boilers in merchant ships average somewhat higher than for Scotch boilers; in naval vessels rates of evaporation average around 15 and 16 lb per sq ft of heating surface per hr. Rates appreciably higher than this have been attained for short periods. Yarrow, in 1935,[2] gave a maximum value for cruisers of 24.0 lb per sq ft of heating surface per hr.

These rates of evaporation are the average for the whole heating surface. Obviously the rate of evaporation varies through

[1] For a complete discussion of superheat control and the factors influencing superheat, the reader is referred to a paper, "Control of Superheat," by Thomas B. Stillman, *Trans. Soc. Naval Arch. and Mar. Eng.*, 1935.

[2] *Trans. Inst. Eng. and Shipbuilders in Scotland*, vol. 78, pp. 131–152.

TABLE IV.—PERFORMANCE OF WATER-TUBE BOILERS
(All Data for One Boiler)

Make of boiler	B & W	Yarrow	Foster Wheeler D type	Foster Wheeler D type	Yarrow	3-drum express	Yarrow	Yarrow mechanical stoker	B & W[c]	B & W pulverized[e] coal
Date	1930	1931	1940	1939	1930	1934	1934	1928	1934	1932
Heating surface, sq ft, generating tubes	7,160	12,444	2,395	3,065	5,700	4,753	5,990	3,150	4,211	2,574
Furnace volume, cu ft	1,030	2,685	......	....	1,320	589	....	760	360	550
Superheating surface, sq ft	1,680	6,576	....	.....	1,820	814	1,980	....	None	1,045
Air heating surface, sq ft	7,160	14,629	None	None	5,700	None	2,540	....	None	2,400
Economizer surface, sq ft	None	None	1,656	2,640	None	3,700	None	None	None	None
Boiler-drum pressure, gage	425	420	450	335	373	450	386	286	232	499
Steam temperature, °F, superheater outlet	646	740	750	645	690	670	660	652	399.7	755
Temperatures, °F										
Air entering air heaters			.	.....	54		. .	71		
Air entering furnace	.....	. .	100	100	276		307	323	82.3	325
Gases leaving boiler	....	550	670	690	530	. .	... .	....	519.4	587
Gases entering stack	280	240	330	343	322	.. .	.	354	519.4	399
Feed water	300	300	240	225	279	......	195	167	211.4	112
Type of fuel	Oil	Oil	Oil	Coal	Oil	Oil	Oil	Coal	Oil	Pulv. coal[g]
Heating value of fuel, Btu per lb	18,400	18,500	18,500	14,150	18,700	19,000	19,000	11,150	18,321[f]	13,159
Fuel burned per hr, lb	2,782	3,022	1,840	2,340	1,800	.....	7,810	2,100	1,403	2,438
Water evaporated per hr, lb	42,080	55,000	25,000	25,000	26,390	. .		15,962	21,075	19,700
Actual evaporation per lb of fuel	15.13	18.20	13.6	10.68	14.65			7.60	15.01	8.08
Equivalent evaporation per lb of fuel	.					16.23			15.8	
Actual evaporation per sq ft boiler H.S. per hr[d]	5.9	4.43	10.44[j]	8.15[h]	4.62		16.0[a]	5.05	5.0	8.08
Equivalent evaporation per sq ft boiler H.S. per hr[d]		.				24.34			5.26	9.93
Excess air, per cent		...	14% CO_2	14% CO_2	14.5% CO_2				20% CO_2	18.0

Over-all boiler efficiency, per cent..	87.4	88.0	87 7	86.75	86 0	82 8	76 5[b]	83.0	83.68	80.6
Fuel burned per sq ft boiler H.S. per hr	0.388	0.304	.767[j]	763[k]	0 316	1 5[c]	1.3	0.657	0.334	0.95[h]
Fuel burned per cu ft boiler F.V. per hr	2.70		..		1.36	12 32	.	2 76	3.89	4.42
	Service	Service	Test	Test		. .		Official trials	Test	
Source of data	*S.S. Statendam*	*S.S. Empress of Britain*	Merchant type	Merchant type Mechanical stoker	*S.S. Viceroy of India*[e]	Design values	Portuguese destroyer	*S.S. Beaverburn*	Army dredge, *Col. P.S. Michie*	
Reference	l	l			o	m	n	p	q	r

[a] Approximate, reported as "about 16."
[b] Approximate. Based on 16.0 lb evaporation per sq ft H S. per hour
[c] 0.843 lb per sq ft boiler H.S. and economizer H.S.
[d] Does not include economizer H.S.
[e] From tests of four large boilers.
[f] Oil analysis: C = 85.65 per cent; H = 11 78 per cent; O = 0 71 per cent, N = 0.29 per cent; S = 1 57 per cent
[g] Yorkshire coal. Fusing temperature of ash = 2264°F; 9.7 per cent ash.
[h] 58,211 Btu (4.44 lb of coal) liberated per hr per cu ft F.V
[i] This boiler had no air heater. See Table VI for test on similar boiler fitted with air heater.
[j] Evaporation ÷ H.S. + econ. H.S. = 6.18 lb. per sq ft per hr. Oil per hr ÷ H.S. + econ. H S. = 0.455 lb per sq ft per hr. Feed leaving economizer = 350°F.
[k] Evaporation ÷ H.S. + econ. H.S. = 4.38 lb per sq ft per hr. Coal per hr ÷ H S + econ. H S = 0.410 lb per sq ft per hr.
[l] *Trans. Inst. Naval Arch*, 1936, p. 45.
[m] *Trans. Soc. Naval Arch. and Mar. Eng.*, 1934, p. 284.
[n] H. E. Yarrow, "Trial Performance of a Torpedo Boat Destroyer," *Trans. Inst Eng. and Shipbuilders in Scotland*, vol. 78, 1934–1935.
[o] *The Marine Engineer & Motorship Builder*, vol. 53, p. 18.
[p] *Shipbuilding and Shipping Record*, vol. 31, p. 309
[q] *Mar. Eng. and Shipping Age*, vol. 40, p. 320.
[r] North East Coast Institute of Engineers and Shipbuilders, vol. 49, p. 220.

a boiler, being highest for the furnace tubes and lowest for the tubes in the upper part of the boiler. When economizers are fitted, the question often arises whether rates of evaporation and rates of combustion should be expressed in terms of boiler heating surface alone (main boiler surface) or for the combined heating surface of boiler and economizer. Since economizers are fitted for feed-water heating and are not actually part of the main boiler heating surface, the best practice is not to include the economizer surface in calculating rates of evaporation. The surface of the superheater should not be included as part of the heating surface.

The Foster Wheeler boilers in the *S.S. Red Jacket* had a designed normal rate of evaporation of approximately 8.9 lb of water per sq ft of main boiler heating surface and 5.7 lb per sq ft of combined boiler and economizer surface. The latter figure is more nearly in line with the evaporation rates given in Table IV for boilers that have no economizers and shows clearly how the economizer has relieved the boiler tubes of much of the function of feed-water heating.

In order that the performance of boilers that are operating under different initial conditions and feed-water temperatures may be compared with one another, it is customary to reduce all performances to a common basis. The common basis chosen is the equivalent evaporation from feed water at 212° into steam at atmospheric pressure. This is spoken of as "the equivalent evaporation from and at 212°F," meaning that the evaporation takes place "from" feed water at 212° into steam "at" 212°F, or at atmospheric pressure. To reduce an actual evaporation to an equivalent evaporation, we divide the Btu absorbed by the boiler per square foot of heating surface per hour under actual operating conditions by the enthalpy necessary to evaporate water from and at 212°F, *i.e.*, the enthalpy of evaporation at 212°F. An example will make this clear

Example.—Suppose that a boiler test showed that 5.8 lb of superheated steam per sq ft of heating surface per hr was produced under the following conditions: boiler pressure, 450 lb gage, steam temperature, 640°F, and feed-water temperature, 300°F. The equivalent evaporation is

$$\frac{5.8(1{,}325.1 - 269.6)}{970.3} = 6.30 \text{ lb per sq ft per hr}$$

where 970.3 = enthalpy of evaporation at 212°F.

The general procedure is to calculate what is known as the factor of evaporation and to multiply the actual evaporation by the factor of evaporation to obtain the equivalent evaporation. The factor of evaporation is the ratio of the heat required to evaporate 1 lb of water into steam under actual conditions to that required from and at 212°.

For dry or wet steam, the factor of evaporation

$$\text{F.E.} = \frac{x_1 h_{fg_1} + h_{f_1} - h_{f_2}}{970.3}$$

Boiler performance from the economic viewpoint is expressed as the number of pounds of water evaporated per pound of fuel. In order to bring this value to a common basis, it is also reduced to the equivalent evaporation from and at 212° per pound of coal or oil.

The rate of evaporation of a given boiler will naturally depend on the amount of fuel burned per hour, which is expressed by the rate of combustion. For oil-burning boilers, the rate of combustion is expressed in terms of pounds of oil burned per square foot of heating surface per hour and for coal-burning boilers, in terms of the pounds of coal burned per square foot of grate surface (G.S.) per hour. For designing data, rates of combustion are also expressed in fuel burned per cubic foot of furnace volume (F.V.) per hour.

For hand-fired coal-burning boilers, rates of combustion are limited by the capacity of the fireroom force to handle coal over long periods of time. For continuous service in merchant vessels, the rate of combustion seldom exceeds 20 to 28 lb of coal per sq ft of grate surface per hr. When naval vessels burned coal, rates as high as 65 lb were attained for short periods during high-speed runs.

For oil-fired boilers, rates of combustion in the past have been held down to around 0.30 to 0.35 lb of oil per sq ft of heating surface per hr in merchant ships. In the navy, rates of 1.0 and 1.1 have been the standard practice for maximum speed conditions, and test rates of 1.5 lb have been attained for short periods.

When boilers are not fitted with economizers no confusion arises as to the method of expressing rates of combustion, but since the introduction of economizers, rates of combustion have been expressed in terms of boiler heating surface and in terms of combined boiler and economizer surface. Sometimes the surface of the superheater is also included. In using and expressing rates of combustion, one must take care to make certain how the rate of combustion is expressed. As pointed out on page 66, the most logical procedure would be to express rates of combustion as well as rates of evaporation in terms of main boiler heating surface.

Rates of combustion and evaporation for merchant ships have been increased appreciably in the past few years. This has been due partly to the introduction of economizers, which relieve the main boiler tubes of much of the feed-water heating, and partly to the elimination of furnace brickwork by the introduction of water-cooled furnaces. Service rates of combustion for the U. S. Maritime Commission's new cargo ships run as high as 0.50 and 0.60 lb of oil per hr per sq ft of main boiler heating surface (0.35 – 0.40 lb, including economizers).

Other methods frequently used in expressing rates of combustion, especially for purposes of boiler design, are oil burned per hour per cubic foot of furnace volume and heat release in Btu per cubic foot F.V. per hour. Ratios of furnace volume to heating surface for various types of boilers are given in the following table:

Stationary plants using pulverized coal	0 45–0 50
Scotch marine (furnace and lower half of combustion chamber)	0.122
Marine water tube (hand-fired, coal)	0.10–0.12
Marine water tube (oil)	0 09–0 24

For oil-fired boilers, the lower values are for naval express boilers, and the higher values are for merchant-ship boilers fitted with water-cooled furnaces. For brick-lined furnaces, the upper value is about 0.15.

Ratios of heating surface to grate surface for coal-fired boilers are as follows:

Scotch	30–35
Water tube	25–45

Heat releases in Btu per cubic foot of furnace volume per hour cover the following ranges:

Stationary practice	12,000–35,000
Merchant practice	35,000–50,000
Navy practice	200,000+
Velox boiler	500,000–1,000,000

It is obvious that the higher the rate of combustion the smaller, lighter, and cheaper will be the boiler installation. High rates of evaporation, however, require that pure feed water be assured.

The evaporation per square foot of H.S. for a given rate of combustion will vary with the initial steam conditions, the feed temperature, the boiler efficiency, and the heat value of the fuel.

Evaporation per square foot H.S. per hour

$$= \frac{\text{Btu per pound of fuel} \times e_b}{h_1 - h_f} \times \text{fuel per square foot H.S. per hour}$$

$$\text{Evaporation per pound of fuel} = \frac{\text{Btu per pound of fuel} \times e_b}{h_1 - h_f}$$

where e_b = boiler efficiency.
h_1 = enthalpy of steam at superheater outlet.
h_f = enthalpy of the liquid at feed temperature.

$$\text{Rate of combustion} = \frac{\text{rate of evaporation} \times (h_1 - h_f)}{\text{Btu per pound} \times e_b}$$

35. Boiler Efficiency.—The boiler efficiency generally used is the combined over-all efficiency of the furnace, burners, boiler heating surface, and superheating surface (and economizer and air heater when fitted). The boiler efficiency does not include the power necessary to operate the feed pump, fuel pump, and forced-draft blower or the heat necessary for the fuel oil heater.

Over-all boiler efficiency in percentage is

$$\frac{\text{Pounds of water feed to boiler per hour} \times (h_1 - h_f)}{\text{Btu per pound of fuel (as fired)} \times \text{fuel burned per hour}} \times 100$$

where h_1 = total enthalpy of the steam leaving the superheater.
h_f = enthalpy of the feed water entering boiler (or economizer, when fitted).

Example.—The test of a water-tube boiler gave the following results:
Water fed to boiler per hour = 22,800 lb
Fuel burned per hour = 1,611 lb
Steam pressure at superheater outlet = 465 lb abs
Temperature of steam at superheater outlet = 720°F
Btu per pound of fuel = 17,950
Temperature of feed water entering economizer = 304°F
Boiler heating surface = 3,050 sq ft
Economizer heating surface = 1,800 sq ft

$$\text{Over-all boiler efficiency} = \frac{22{,}800 \times (1{,}370.1 - 273.7)}{17{,}950 \times 1{,}611} \times 100$$
$$= 86.5 \text{ per cent}$$

$$\text{Actual evaporation per pound of fuel} = \frac{22{,}800}{1{,}611} = 14.15$$

Actual evaporation per square foot of boiler heating surface per hour
$$= \frac{22{,}800}{3{,}050} = 7.48 \text{ lb}$$

Oil burned per square foot of boiler heating surface per hour
$$= \frac{1{,}611}{3{,}050} = 0.528 \text{ lb}$$

Oil burned per square foot of heating surface of boiler and economizer
$$\text{per hour} = \frac{1{,}611}{4{,}850} = 0.332 \text{ lb}$$

$$\text{Factor of evaporation} = \frac{1{,}370.1 - 273.7}{970.3} = 1.13$$

Equivalent evaporation per square foot of boiler heating surface per hour
$$= 7.48 \times 1.13 = 8.47 \text{ lb}$$

The largest loss in boiler operation is that due to the dry stack gases.

$$\text{Loss} = 0.24W_g(t_g - t_a) \text{ Btu per pound of fuel}$$

where W_g = weight of dry stack gases per pound of fuel.
0.24 = specific heat of stack gases.
t_g = temperature (degrees Fahrenheit) of gases leaving boiler unit. If an air heater is fitted, t_g is the temperature of the gases leaving the air heater.
t_a = temperature of air for combustion entering the boiler unit. When no air heater is fitted, this is the temperature of the air entering the furnace. If an air heater is fitted, t_a is the temperature of the combustion air entering the air heater.

This loss is discussed more fully in connection with other boiler losses in the next chapter.

It is obvious from the foregoing equation that for equal values of W_g, the lower the temperature t_g the less will be this loss and the greater will be the boiler efficiency. Heat-recovering apparatus placed in the path of the gases after they leave the boiler proper is now used with most marine boilers to reduce this uptake temperature. The waste-heat superheater mentioned in Art. 30 is such a unit. Other heat-recovering devices are feed-water heaters known as "economizers" (Art. 144) and air heaters, discussed in the next article.

Another factor influencing boiler efficiency is the rate of combustion (oil per square foot of heating surface or per cubic foot of furnace volume per hour). As the rate of combustion is increased and more Btu are released per cubic foot of furnace volume per hour, the larger volume of gases passing through the boiler cannot give up as much heat to the heating surface, and

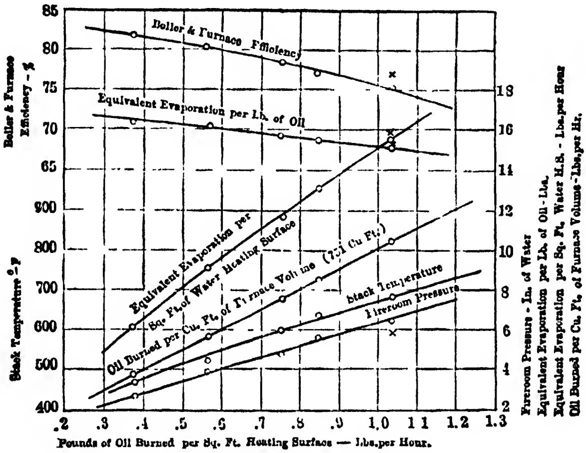


FIG. 22.—Performance curves of White-Forster boiler.

hence more Btu escape up to the stack. This is well illustrated by the curves in Fig. 22, which shows the performance of an older type of navy water-tube boiler not fitted with economizer or air heater. Figure 23 shows how the efficiency of an oil-fired Scotch boiler falls off with the increase in rate of evaporation. Here the efficiencies are plotted against the rate of evaporation, but since the rate of combustion increases with the rate of evaporation the curves are comparable to those shown in Fig. 22. Other factors influencing boiler efficiency besides the two mentioned are:

1. The weight of gases going up the stack, W_g, in the equation on page 70. This depends upon the amount of excess air used (Art. 42).
2. The completeness and rapidity of combustion as determined by combustion air temperature, viscosity of the oil, burner-tip sizes, etc.
3. The extent and arrangement of the heating surface.
4. The cleanliness of the boiler heating surface.

The size, weight, and cost of a boiler—all important characteristics for a marine installation—depend directly on the amount of heating surface. Where weight and space are important the amount of heating surface is kept down by using a high rate of combustion. The introduction of water-cooled furnaces, economizers (for heating the feed water), and air heaters have tended to make the heating surface more effective and thus allow high efficiencies to be obtained with reasonable weights and costs. It is interesting to note in this connection that in a boiler operating under 450 lb gage pressure, a steam temperature of 700°F, and a feed-water temperature of 300°F, the absorption of heat is 70.1 per cent for evaporation, 15.7 per cent for heating the liquid to boiling temperature, and 14.2 per cent for superheating the steam 240°F.

The temperature of the gases leaving a Scotch boiler are much higher than for water-tube boilers under similar conditions. This is because of the short length used with Scotch boilers. The temperature of the stack gases can be reduced and the efficiency of the boiler increased by adding more heating surface at the place where the gases leave the boiler. This means increasing the length of the boiler tubes and consequently the length of the boiler. Retarders are frequently fitted in the tubes of Scotch boilers to increase the rate of heat transmission by agitating the hot gases and bringing them in better contact with the tube heating surface. Retarders consist of flat pieces of steel twisted into the form of screw threads of large pitch. These retarders are about ¼ in. smaller than the diameter of the tubes and extend the whole length of the tube. Retarders are used frequently with coal-fired boilers and almost universally with oil-fired boilers. The amount of agitation of the gases is controlled by the number of turns given to the steel bar that forms the retarder.

The common practice with Scotch boilers is to use a short boiler and fit air heaters, waste-heat superheaters, or both in the uptake. (Fig. 7.) These uptake auxiliaries absorb the heat in the waste gases and reduce the temperature at the base of the stack to 400° or even lower. Without these additional auxiliaries the efficiency of the Scotch boiler in service conditions would be considerably below that of the water-tube boiler.

Table III (page 46) gives the efficiencies of Scotch boilers from a number of different sources. It will be observed that efficiencies as high as 86.5 per cent have been attained and that uptake temperature have been reduced to 350°F for normal rates of combustion. Tests of oil-fired Scotch boilers reported in the *U.S. Bureau of Mines Bulletin* 214 showed that the use of retarders increased the efficiency by about 4 per cent (decreasing uptake temperature by about 100°F); waste-heat superheaters increased the efficiency between 1.0 and 3.0 per cent, and air

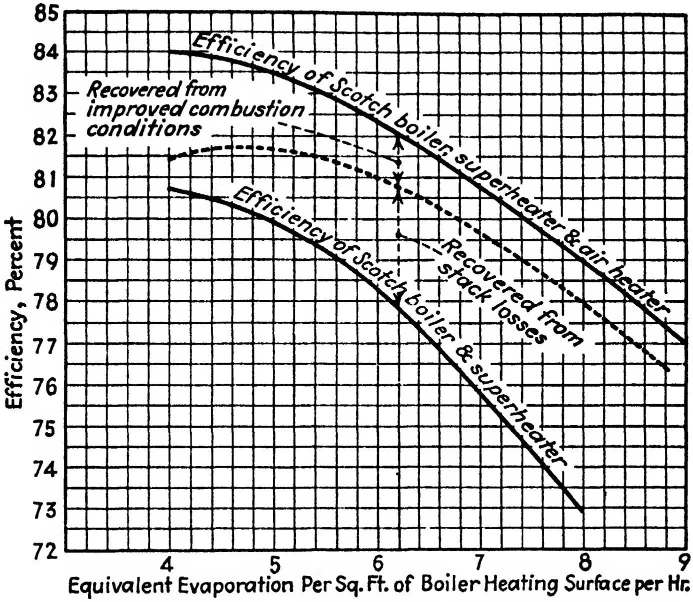


FIG. 23.—Results of tests on oil-fired Scotch boiler, with and without air heaters, as reported in *U.S. Bureau of Mines Bulletin* 214. (*Taken from a paper by Thomas B. Stillman*, "Air Heaters and Economizers for Marine Service," *Jour. Amer. Soc. of Naval Eng., May*, 1935.)

heaters increased the efficiency by about 3 per cent through the heat recovered in the uptakes and around 1.25 per cent through better combustion (see Fig. 23).

Figure 24 shows in an interesting manner the effect of dirty boiler tubes on uptake temperature and hence on boiler efficiency. Practically all marine boilers in modern ships are equipped with mechanical soot blowers to remove the soot from the boiler tubes, the superheater, the economizer, and the air heater.

Efficiencies of modern oil-fired water-tube boilers are high, varying between 85 and 88 per cent for rates of combustion used in the merchant service and around 80 per cent for naval vessels operating at high rates of combustion (see Table IV). Twenty years ago boiler efficiencies averaged much lower than this. These high boiler efficiencies are due to better boiler design and the use of economizers and air heaters. Table IV shows that stack temperatures have been reduced to around 300° in

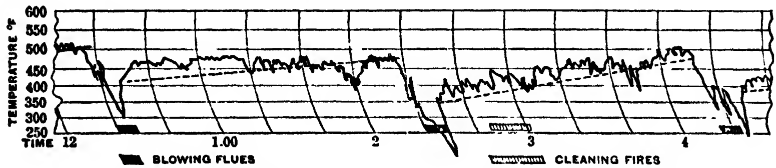


Fig. 24.—Reduction in stack temperatures by blowing soot from tubes.

some cases. The data in Table IV are worthy of careful study, since they indicate how the installation of air heaters has reduced the uptake temperatures and how the boiler efficiency falls off with high rates of combustion.

Some of the new ships designed by the U.S. Maritime Commission are operating with uptake temperatures around 300° and boiler efficiencies of over 85 per cent. The rate of combustion for operating conditions is around 0.55 lb of oil per sq ft of boiler heating surface and around 0.35 lb per sq ft of boiler and economizer H.S.

The efficiencies of hand-fired coal-burning boilers are much lower than oil-burning boilers because of the higher percentage of excess air used and the generally poorer combustion. Boiler efficiencies in service range between 60 and 70 per cent. Coal-burning boilers fitted with mechanical stokers have shown efficiencies as high as 86.5 per cent.

36. Air Heaters.—Tubular air heaters have long been used with Scotch boilers, but only within recent years have they been used with marine water-tube boilers. Not only do air heaters cut down the dry-stack-gas loss by recovering some of this heat to heat the air used for combustion but the hot air entering the furnace improves combustion and increases the boiler efficiency (see Fig. 23).

Three types of air heaters are in use: (1) tubular heaters (Figs. 7, 11, 14, 16, 17, and 21); (2) corrugated plate heaters; and (3) rotating regenerative heaters (Ljungstrom).

The tubular heater is the common type: The combustion air passes through the interior of the tubes, and the hot gases pass over the outside of the tubes in some installations (Figs. 11, 14, and 16); in other installations the hot gases pass through the tubes (Fig. 7). In installations where large air heating surfaces are installed, the air makes several passes through the heater.

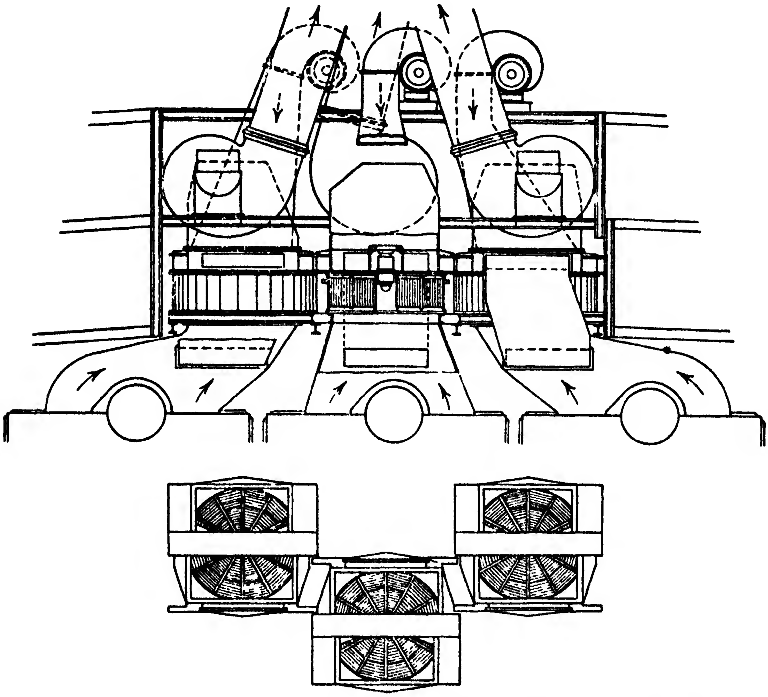


FIG. 25.—Ljungstrom air heater fitted on the Johnson boilers of the *Asturias* and *Alcantara*. (*From The Marine Engineer.*)

Corrugated plate heaters have been used in a few instances, notably on the *S.S. Berwindlea*, a ship burning pulverized coal.[1]

The Ljungstrom heater (Fig. 25) is made up of grids of corrugated steel plates. These plates are placed in the uptake above the boiler and revolve very slowly. The heater is so arranged that the hot gases are passing up through one side of the revolving plates while the draft air is passing down the other half. Thus half of the plate area, which is heated up to the temperature of the uptake gases, has revolved a few seconds

[1] See *Shipbuilding and Shipping Record*, Oct. 31, 1929, p. 531.

later so that the draft air is passing down through the plates. A small amount of leakage takes place from the draft air into the uptake gases, and the draft air is diluted slightly by the gases remaining in the heater plates as they are revolved into the current of draft air. Tests on the heaters show very low uptake-gas temperature and high temperature of combustion air. One test showed the following results:[1]

Air entering heater	82°F
Air leaving heater	435°F
Gases entering heater	574°F
Gases leaving heater	209°F

The Ljungstrom heater is able to obtain a large air heating surface in a small space.[2] They have not been used extensively on shipboard. Four ships of the British Ellerman Line,[3] the *S.S. Asturias* and the *S.S. Alcantara*,[4] and four converted ships of the Union Castle Line[5] are known to have this type of heater.

Because of the small space available in high-speed naval ships, air heaters have seldom been used on ships of this class. Two examples, however, are given in the following table. Table V gives the ratio of air heating surface to boiler heating surface for a number of marine installations.

TABLE V.—RATIO OF AIR HEATING SURFACE TO BOILER HEATING SURFACE

Scotch boiler practice	0 25–0.30
H.M.S. Acheron (destroyer)	0 183
Portuguese destroyer (Yarrow boilers)	0 425
S.S. Manhattan	0 78
S.S. Mariposa	0 60
S.S. Red Jacket (Foster Wheeler boiler)	0.83
S.S. Statendam (B & W boilers)	1 00
S.S. Tannenberg (Wagner boilers)	2 00
S.S. Empress of Britain (for one Johnson boiler)	3.5
Ljungstrom	3–4

37. Boiler Weights.—In the past it has been the custom to express boiler weights in terms of boiler heating surface. Weights

[1] W. H. Owen, "Tests of a Cylindrical Marine Boiler with and without Preheated Air," *Trans. Inst. Eng. and Shipbuilders in Scotland*, vol. 67, pp. 444–508.

[2] *Shipbuilding and Shipping Record*, Apr. 9, 1931, p. 459.

[3] *Shipbuilding and Shipping Record*, vol. 31, p. 244.

[4] *The Shipbuilder and Marine Engine Builder*, vol. 41, p. 565.

[5] *The Shipbuilder and Marine Engine Builder*, vol. 44, p. 570.

including water ranged from 65 to 75 lb per sq ft H.S. for Scotch boilers down to from 10 to 12 lb for the express-type water-tube boilers used in naval vessels. The weight of water carried in the various types of boilers is approximately as follows:

Scotch . . .	20–25 lb per sq ft of heating surface
Large-tube, water-tube boilers . .	5–7 lb per sq ft
Express type	1.5–3.0 lb per sq ft

Expressing boiler weights in terms of heating surface is unsatisfactory, for navy boilers, besides being lighter than merchant boilers per square foot H.S., are operated at much higher rates of combustion. The better way to express weights of boilers is in pounds per pound of steam the boiler evaporates per hour. Present-day boiler weights on this basis are approximately as follows:

Naval-express type	0.9–1.0 lb per lb of evaporation per hr
Merchant-ship type water-tube boilers.	3.5–8.0 lb per lb of evaporation per hr
Scotch boilers	13–15 lb per lb of evaporation per hr

The preceding weights are for the complete boiler and its water and include economizers and air heaters when fitted.[1]

38. Boiler Furnaces.—Until quite recently the furnaces of practically all marine water-tube boilers have been lined with refractory firebrick backed up by the insulating bricks. This brickwork acted as a furnace insulation and also, by maintaining a high furnace-wall temperature, assisted combustion.

Because of the limited space on shipboard, marine boilers have had a much smaller ratio of furnace volume to heating surface than has been the practice on land. Expressed in another way, the heat release per cubic foot of furnace volume per hour is higher for marine boilers, especially for naval boilers, and the refractory brick has assisted combustion at these high heat releases. In merchant-ship boilers, the refractory brickwork on the sides and ends was around 9 in. thick and the insulating brickwork, from 2 to 3 in. Furnace brickwork has always been a source of trouble and maintenance expense. With the intro-

[1] See tables of boiler weights, *The Marine Engineer and Motorship Builder*, April, 1931, p. 153.

duction of poorer grades of fuel oils that are residua from the cracking process, erosion, slagging, and spalling of the brickwork became severe.

The noncombustible ash contained in fuel oils impinges on the refractory bricks, enters the pores of the bricks, and fuses with the brick, forming slag. This slag runs down the furnace side, and the bricks are gradually eroded away. The ash of some fuel oils contains a high percentage of vanadium oxide (V_2O_3), which melts at a comparatively low temperature and causes very severe erosion of the bricks.

To obviate this brickwork trouble, water-cooled furnace walls have been introduced. During the past few years a great many of the marine boilers have been built with completely water-cooled furnaces (see Figs. 14, 16, and 20). It should be noted that Scotch boilers have always had completely water-cooled furnaces and have shown good efficiencies when burning oil. In order to keep the flames from impinging on the water walls and to reduce somewhat the heat release per cubic foot of furnace volume, the sizes of furnaces with water-cooled walls have been increased over previous marine practice.

The Babcock & Wilcox stud-tube wall, covered with a thin layer of chrome ore as a refractory over the studs (Art. 26 and Fig. 13), makes an excellent furnace-wall construction, for it combines the advantage of a water-cooled furnace with the advantage of the refractory furnace wall, which assists combustion and allows high furnace temperatures. Furnace walls of this type are constructed with fully studded furnace tubes covered with refractory, as in Fig. 13, or with partially studded tubes. The partially studded tube is bare on the furnace side, and the space between the tubes is filled with the chrome ore.

39. Boiler Feed Water.—Water used for boiler feed with water-tube boilers must be of an extremely high degree of purity, be free from lubricating oil, and contain no oxygen in solution. The rate of heat transmission through the furnace tubes of water-tube boilers, especially those operated at high rates of combustion, is so high that the formation of a film scale on the water side due to impurities in the water will eventually cause the tubes to overheat and fail. The formation of scale will necessitate frequent mechanical cleaning of boiler tubes with the consequent expense and increased time in port. Today, when the boiler feed water of merchant ships is kept at the proper degree of purity, the boilers are not opened up for cleaning oftener than once in 3 or 9 months. Scotch boilers are usually opened up for cleaning at less frequent intervals.

In order to prevent corrosion of the boiler and economizer tubes, it is necessary that the feed water contain no dissolved oxygen through air in feed water and that the water in boilers always have an alkaline reaction.

Lubricating oil introduced into the boiler with the feed will form a coating on the inside of the tubes that eventually will cause the tubes to fail. Whereas small amounts of oil can be allowed in Scotch boilers, even the smallest quantity cannot be tolerated in a water-tube boiler. When a ship is propelled by turbines and the auxiliaries are driven by turbines or electric motors, there is little possibility of lubricating oil getting into the boilers. However, when the ship is propelled by a reciprocating steam engine using superheat (Art. 60) and reciprocating steam auxiliaries are used, there is always danger of a small amount of lubricating oil entering the boiler with the feed water. This is one reason why Scotch boilers are so widely used when ships are propelled by reciprocating engines.

Direct-contact feed-water heaters (Art. 143) and a closed feed-water system, now widely adopted on all new ships, keep air out of the boiler feed and thus ensure that the water entering the boiler is practically free from dissolved oxygen.

When all the reserve feed taken on board in port is put through an evaporator before it is sent to the boiler, the impurities in the water are eliminated. Also, make-up feed water obtained from the evaporation of sea water is free from scale-forming impurities if the evaporator is properly operated. However, boiler feed water is always in danger of contamination from salt-water leakage in the condensor and from evaporator "carry-over," and means must be provided for taking care of such situations when they arise.

Boiler feed water is now almost universally given routine tests and treated with boiler compounds that precipitate the scale-forming impurities in the form of sludge. The proper treatment of the boiler water also maintains proper alkalinity of water and thus prevents corrosion in boiler and economizer tubes. The boiler compound should be such that excessive priming does not result from the chemicals used, with the resultant possibility of moisture being carried over into the superheater and perhaps into the turbine itself.

Boiler compounds must always be used when raw water, taken on board in port, is used for boiler make-up feed without being put through an evaporator before being introduced into the boiler.[1]

No boiler compounds are available that will eliminate the harmful effect of lubricating oil in the feed water.

40. Forced-circulation Boilers.—Practically all marine boilers depend upon the natural circulation of water in the boiler tubes. Boilers using forced circulation have been used to some extent on shore and probably will be used to a limited extent on shipboard where high steam pressures are used and where space and weight consideration are important. Forced circulation was used in the boilers of the German merchant ship *Potsdam*, with a Benson-type boiler operating at 1,500 lb steam pressure.

In forced-circulation boilers, the boiler water is pumped through the tubes and emerges at the end of the tubes as steam. The amount of water circulated is usually greater than that evaporated. An example of a forced-circulation boiler is the Velox boiler.[2] This boiler uses a combustion pressure of 35 lb per sq in. in the furnace and very high heat releases per cubic foot of furnace volume per hour. The volume of water in the boiler is very small, and the water circulated through the tubes is ten times the amount evaporated. The furnace is completely water-cooled, and very high gas velocities are used. Several ships in Europe have been fitted with this boiler,[3] but so far none has been so fitted in the United States.

[1] For further information on the treatment of boiler water, see A. C. Purdy, "Water Conditioning and Related Problems of Boiler Operation," *Trans. Soc. Naval Arch. and Mar. Eng.*, 1933.

[2] A. Meyer, "The Velox Steam Generator for Merchant and Naval Vessels," *Trans. Soc. Naval Arch. and Mar. Eng.*, 1936.

[3] *The Shipbuilder and Marine Engine Builder*, vol. 42, p. 601. See also *Engineering*, London, Feb. 5, 1937, p. 143.

CHAPTER IV

COMBUSTION

41. Chemistry of Combustion.—Combustion in a boiler is the rapid chemical union of the combustible elements in a fuel and the oxygen in the air with the resultant liberation of heat for the generation of steam. Oxygen, as well as carbon and hydrocarbons, is necessary for this combustion; hence an adequate supply of combustion air is essential in boiler operation.

In coal combustion, the fuel must first be brought to the ignition temperature by the application of heat. The hydrocarbons are driven off and ignited; the remaining fixed carbon unites with the oxygen of the air, forming carbon dioxide.

Oil is burned in a boiler by pumping the oil into the furnace in a fine atomized spray. The air for combustion is admitted at high velocity through openings around the burner and mixes thoroughly with the fine particles of the oil in the furnace, resulting in oxidation of the fuel and the liberation of heat. The finer the oil particles the more rapid the combustion; hence burners of correct design are required, and the oil must be under pressure and be reduced to a low viscosity by heating before it is sent to the burner.

During the process of burning, the proper amount of oxygen must be brought in contact with each particle of the fuel so that complete combustion of the fuel will result. The exact chemical combinations of the hydrocarbons and elements in the fuel are rather complicated, and a complete analysis of the reactions taking place during combustion is outside of the scope of this book. At the completion of the combustion the carbon has united with oxygen, forming CO_2, and the hydrogen has united with oxygen, forming H_2O. Thus

$$C + O_2 = CO_2$$
$$H_2 + \frac{1}{2}O_2 = H_2O$$

The elements unite in proportion to their atomic weights as follows:

$$12 \text{ lb C} + (2 \times 16) \text{ lb O} = 44 \text{ lb } CO_2$$
$$(2 \times 1) \text{ lb H} + 16 \text{ lb O} = 18 \text{ lb } H_2O$$

Dividing the foregoing by 12 and 2, we have

$$C + 2.667\ O = 3.667\ CO_2$$
$$H + 8O = 9H_2O$$

Thus each pound of carbon unites with 2⅔ lb of oxygen, forming 3⅔ lb of CO_2, and each pound of hydrogen unites with 8 lb of oxygen, forming 9 lb of water.

If sufficient air is not supplied to give enough oxygen to unite with the carbon and form CO_2, carbon monoxide (CO) is formed. If CO_2 is formed the combustion is said to be "complete"; if CO is formed the combustion is said to be "incomplete." In the latter case the reaction is

$$12\ C + 16\ O = 28\ CO$$

or

$$C + 1.334\ O = 2.334\ CO$$

Thus, if only 1⅓ lb of oxygen is supplied per lb of carbon, the carbon unites with the oxygen, forming 2⅓ lb of CO.

During the foregoing reactions, heat is liberated as follows:[1]

$$C + O_2 = CO_2 \qquad 14{,}600 \text{ Btu per lb C}$$
$$C + O = CO \qquad 4{,}500 \text{ Btu per lb C}$$
$$H_2 + O = H_2O \qquad 61{,}950 \text{ Btu per lb H}$$

If the CO is burned to CO_2 by the application of more oxygen we have

$$2CO + O_2 = 2CO_2 \qquad 10{,}100 \text{ Btu per lb C}$$

In the total process of combustion of C to CO and thence to CO_2, we have 4,500 + 10,100 = 14,600 Btu or the same heat liberated when carbon is burned to CO_2 directly. The foregoing clearly shows the great loss of the heat in the fuel if sufficient oxygen is not supplied for complete combustion. When CO is formed, only 4500 Btu is liberated per pound of carbon against 14,600 Btu when CO_2 is formed.

[1] These are the higher heat values discussed in Art. 19.

The sulphur in the fuel combines with oxygen, forming sulphur dioxide (SO_2):

$$32S + (2 \times 16)O = 64SO_2$$

Dividing by 32, we have approximately

$$S + O = 2SO_2$$

The oxygen for combustion is supplied by the atmospheric air, which consists of a mechanical mixture of oxygen, nitrogen, small amounts of water vapor and carbon dioxide, and very small amounts of other elements. Air is made up approximately of 23 parts of oxygen and 77 parts of nitrogen (including the other constituents mentioned above) by weight. The ratio of the weight of air to oxygen can be taken as $100/23 = 4.35$ as sufficiently accurate for combustion problems. Hence $4.35 \times 2.667 = 11.6$ lb of air must be supplied for the complete combustion of 1 lb of carbon.

The nitrogen in the fuel takes no part in the combustion process and merely dilutes the uptake gases. The products of combustion formed by the burning of 1 lb of carbon will be 3.67 lb CO_2 and $(11.6 \times 0.77) = 8.93$ lb of nitrogen, making 12.6 lb of gases per lb of carbon.

The *theoretical* amount of air required for the combustion of coal or fuel oil can be obtained from the following formula:

$$\text{Pounds of air per pound of fuel} = 11.6C + 34.8\left(H - \frac{O}{8}\right) + 4.35S$$ [1]

where C, H, O, and S = decimal parts by weight of the carbon, hydrogen, oxygen, and sulphur in 1 lb of fuel.

The expression $\frac{O}{8}$ is to allow for the hydrogen in the fuel, which is assumed to be combined with oxygen in the form of water.

The weight of the dry uptake gases resulting from the burning of 1 lb of fuel with just the theoretical amount of air supplied is

$$12.6C + 26.8\left(H - \frac{O}{8}\right) + 5.35S + N$$

[1] The exact percentage of oxygen in the air is 23.15. With the use of this value, the preceding equation is, $11.52C + 34.56(H - O/8) + 4.32S$. The equation used in the text is usually accurate enough for engineering computations.

where

C, H, S, and N = parts by weight of the elements in the fuel.[1]

The theoretical amount of air required by fuels will vary widely, depending upon percentages of carbon, hydrogen, and oxygen in the fuel.

Consider the three fuels with analyses given below:

	Bituminous coal	Semibituminous coal	Fuel oil
Carbon, per cent.	65 41	82 72	86 4
Hydrogen, per cent	5 06	4 63	12 25
Oxygen, per cent......	13 41	4 42	0 03
Sulphur, per cent......	3 39	0 72	0 74
Nitrogen, per cent...	1 35	1.31	0 58
Btu per lb	11,661	14,817	18,800

The theoretical air requirements per pound by the preceding formula are

Bituminous coal = 8.90 lb
Semibituminous coal = 11.03 lb
Fuel oil = 14.32 lb

42. Excess Air.—In the actual combustion of fuel it is essential that the theoretical required amount of air be present for all the small particles of fuel at the time of combustion, *i.e.*, the correct amount of air at the right place and at the right time. In order to meet the preceding requirements, more air than that theoretically required for combustion is supplied to the furnace. The amount by which the actual quantity supplied exceeds the theoretical amount determined by the formula in the foregoing article is termed the "excess air."

When boilers are operated on hand-fired coal, large amounts of the excess air are required because of the thickness and unevenness of the fuel bed and the unequal distribution of the combustion air. Excess air with hand-fired coal usually runs between 40 and 75 per cent and may go to 100 per cent and even

[1] Since each pound of hydrogen requires 8 lb of oxygen for combustion, the nitrogen in the flue gases due to the combustion of the hydrogen will be $8 \times {}^{77}\!/\!_{23} = 26.8$ lb per lb of hydrogen burned.

higher. With oil-fired boilers the percentage of air with a good burner and careful management may not exceed 10, and 30 per cent should be the maximum ever required for good combustion.

When coal is burned by mechanical stokers or in the pulverized form, lower amounts of excess air are possible than with hand-firing, and percentages approaching oil-fired boilers can be attained.

All the air supplied to the furnace that is not used for combustion is discharged up the stack with the products of combustion at the uptake temperature. This stack-gas loss represents the largest amount of heat lost in boiler operation; hence it is essential that the excess air be kept to a low value if a high boiler efficiency is to be attained (Art. 43).

From the foregoing discussion, it is seen that the uptake gases will contain CO_2, superheated water vapor (from the burning of the hydrogen), sulphur dioxide, nitrogen, oxygen (in the excess air supplied), and possibly small amounts of carbon monoxide (from incomplete combustion), unburned hydrocarbons, as smoke and soot, and ash dust.

In actual boiler operation the weight of dry stack gas per pound of fuel fired can be calculated from a chemical analysis of the flue gases by the following formula:

$$W_g = \left\{ \frac{11CO_2 + 8O + 7(CO + N)}{3(CO_2 + CO)} \right\} \frac{C}{100}$$

where CO_2, CO, O, and N are the percentages by volume of the carbon dioxide, carbon monoxide, oxygen, and nitrogen in the flue gases. C is the percentage (by weight) of the carbon in the fuel (less that in the ash). Such an analysis is usually made with an Orsat apparatus.

The weight of dry air supplied per pound of dry fuel can be expressed closely by the following formula:

$$W_a = 3.03 \left(\frac{N}{CO_2 + CO} \times \frac{C}{100} \right)$$

If an ultimate analysis of the fuel is available, the theoretical weight of air required per pound of fuel can be calculated from the formula on page 83, and the amount of excess air can then be obtained.

When the combustion is complete the *weight* of CO_2 in the uptake gases is constant, no matter how much excess air is used. The volume percentage of CO_2 in the gases will decrease, however, as the amount of excess air is increased. The volume percentage of CO_2 in the uptake gases can thus be used to determine the amount of excess air being supplied. The CO_2 percentage can be obtained by an analysis of the flue gases with an Orsat apparatus or by a CO_2 meter. By the use of a CO_2 meter, the fireman can thus keep the air supply at the correct amount and hence control the loss due to dry stack gases.

The weight of CO_2 and nitrogen produced by the burning of any fuel can be readily computed by the preceding equations when the ultimate analysis of the fuel is known. The CO_2 per cent by volume can then be calculated for complete combustion with just the theoretical amount of air and various percentages of excess air. A curve can then be drawn for any particular fuel that will show the correct relationship of CO_2 and excess air.

Example.—The fuel oil mentioned on page 36 has the following analysis:

$$\begin{aligned} C &= 86.2 \text{ per cent} \\ H &= 10.3 \text{ per cent} \\ O &= 0.7 \text{ per cent} \\ N &= 0.4 \text{ per cent} \\ S &= 2.3 \text{ per cent} \\ \text{Btu per pound} &= 18{,}300 \end{aligned}$$

The theoretical amount of air required for combustion of 1 lb of fuel is

$$(11.6 \times 0.862) + (34.8 \times 0.102) + (4.35 \times 0.023) = 13.66 \text{ lb}$$

The weight of the dry uptake gases per pound of fuel for complete combustion and 0 per cent excess air is

$$W_g = (12.6 \times 0.862) + (26.8 \times 0.103) + (5.35 \times 0.023) + 0.004 = 13.73 \text{ lb}$$

If we assume complete combustion (*i.e.*, no CO formed) with 0 per cent excess air, the burning of the carbon gives the following products:

$$\begin{aligned} CO_2 &= 3.667 \times 0.862 = 3.16 \text{ lb} \\ N &= 11.6 \times 0.862 \times 0.77 = 7.70 \text{ lb} \end{aligned}$$

The burning of the hydrogen gives

$$\begin{aligned} N &= 26.8 \times \left(0.103 - \frac{0.007}{8}\right) = 2.75 \text{ lb} \\ H_2O &= 9.0 \times 0.103 = 0.927 \text{ lb} \end{aligned}$$

The burning of the sulphur gives

$$SO_2 = 2 \times 0.023 = 0.046 \text{ lb}$$
$$N = 4.35 \times 0.023 \times 0.77 = 0.078 \text{ lb}$$

The products of combustion in the dry gas per pound of fuel are

$$\begin{aligned} CO_2 &= 3\ 16 \text{ lb} \\ N &= 10\ 528 \text{ lb} \\ SO_2 &= 0\ 046 \text{ lb} \\ \text{Total } W'_g &= 13\ 734 \text{ lb} \end{aligned}$$

DRY UPTAKE GASES PER POUND OF FUEL OIL. COMPLETE COMBUSTION WITH NO EXCESS AIR

	Weight, lb	Specific weight, at 68°F	Volume at 68°F and atm pressure, cu ft	Per cent by volume
CO_2	3 16 ÷	0 1142 =	27 6	15 95
N	10 528 ÷	0 0727 =	145 0	83 88
SO_2	0 046 ÷	0 1662 =	0 3	0.17
Total	13 734		172 9	100.00

The weight of the dry uptake gases per pound of fuel as calculated above is 13.734, and the water vapor produced by the combustion of the hydrogen in the fuel is shown to be 0.927 lb. Thus the total weight of the products of combustion is 14.66 lb. The air supplied per pound of fuel was 13.66 lb; thus the weight products of combustion that must equal the weight of air and fuel = 13.66 + 1.00 = 14.66 lb. This checks the weight obtained from a detailed calculation.

With 10 per cent excess air, the increase in volume of the gases would be

$$(13.66 \times 0.10) \div 0.0753 = 18.2 \text{ cu ft}$$

$$\text{Per cent } CO_2 = \frac{27.6}{172.9 + 18.2} = 14.42 \text{ per cent}$$

With 20 per cent excess air

$$CO_2 = \frac{27.6}{209.3} = 13.18 \text{ per cent}$$

A curve of CO_2 percentages for various percentages of excess air is shown in Fig. 26 for the fuel used in the preceding example. This curve applies only to the fuel for which it was calculated. Fuel oils and coals having different percentages of carbon and hydrogen will have different values. Coals with a low percentage of hydrogen will have higher percentages of CO_2 for a given percentage of excess air than fuel oils.

For the fuel used in the preceding example the weights of the dry stack gases W_g for complete combustion are as follows:

	Pounds
0 per cent excess air.	13.73
10 per cent excess air = 13 73 + (13.66 × 0.10)	15 10
20 per cent excess air = 13.73 + (13.66 × 0 20)	16 46
30 per cent excess air	17 83

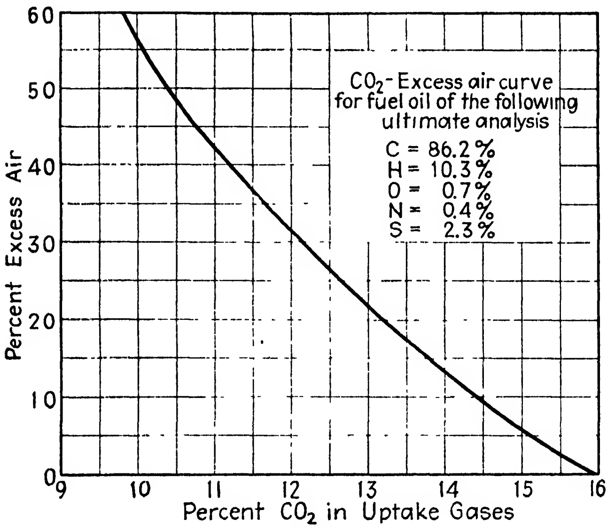


FIG. 26.

43. Boiler Losses.—We have seen that the efficiency of a modern marine water-tube boiler is in the neighborhood of 86 per cent; thus about 14 per cent of the heat supplied in the fuel is lost in the process of generation of steam. The losses in a boiler can be grouped as follows:

1. Dry-stack-gas loss.
2. Loss due to water vapor produced by the burning of hydrogen.
3. Loss due to moisture in fuel and combustion air.
4. Incomplete combustion.
5. Radiation and unaccounted for.
6. Combustible in the ash (coal-burning).

The largest loss in a boiler is that due to the heat carried away by the hot gases going up the stack. This loss is usually divided into two main divisions: (1) the loss caused by the water vapor that has entered with combustion air and fuel and that produced by the burning hydrogen, which go up the stack as superheated steam (items 2 and 3); and (2) the loss caused by the dry products of combustion going up the stack at a high temperature (item 1).

1. *Dry-stack-gas Loss.*—This loss in Btu per pound of fuel is

$$W_g \times 0.24 \times (t_g - t_a)$$

where W_g = weight of the dry uptake gases, as determined by the formula on page 85.

0.24 = specific heat of the uptake gases.

t_g = temperature of gases leaving the boiler. If economizer and/or air heater is fitted, t_g is the temperature of the gases leaving the last heat-recovery device.

t_a = temperature of the air entering the boiler. If an air heater is fitted, t_a is the temperature of the air entering the air heater, not the temperature of air entering the furnace.

From the preceding equation, it is evident that the loss up the stack can be kept down by reducing the temperature of the gases at the base of the stack or by reducing W_g. The temperature at the base of the stack depends upon the amount and arrangement of the boiler heating surface for absorbing the heat in the gases, the rate of combustion, and the use of heat recovery apparatus, *i.e.*, economizers and air heaters that are installed between the boiler proper and the uptakes.

The weight of dry gases going up the stack are made up of (1) the products of combustion of the carbon and sulphur with the oxygen of the air, (2) the inert nitrogen left after the oxygen in the air has combined with the carbon, hydrogen, and sulphur in the fuel, and (3) the excess air. The weight of the CO_2, SO_2, and the nitrogen resulting from complete combustion of 1 lb of the fuel with 0 per cent excess air for the example in the foregoing article was 13.73 lb.

If we assume that t_a = 80°F and that the temperature of the gases leaving boiler t_g = 530°F, the loss due to dry stack gases with no excess air is

$$13.73 \times 0.24 \times (530 - 80) = 1350 \text{ Btu per lb of fuel}$$

With 40 per cent excess air, the additional loss would be (if the air is assumed to be dry and no change in uptake temperature with an increase in excess air is assumed)

$$(13.66 \times 0.40) \times 0.24 \times 450 = 588 \text{ Btu per lb of fuel}$$

This makes a total dry-stack-gas loss of 1938 Btu per lb or 10.5 per cent of the heat in each pound of fuel burned. If the uptake temperature is reduced to 280° by the installation of an air heater and the excess air is cut down to 20 per cent, the loss becomes $660 + 131 = 791$ Btu per lb or 4.3 per cent.

Example.—A boiler test using the fuel of the analysis given in the example in Art. 42 gave the following results:

ANALYSIS OF UPTAKE GASES

CO_2 = 13.18 per cent by volume
CO = 0 by volume
O_2 = 3.62 per cent by volume
N_2 = 83.2 per cent by volume
t_g = 280°F
t_a = 80°F
Carbon in fuel = 86.2 per cent by weight

Find the dry-stack-gas loss.

$$W_g = \frac{\{(11 \times 13.18) + (8 \times 3.62) + 7(83.2)\} \times 0.862}{3 \times 13.18} = 16.46 \text{ lb}$$

$$\text{Loss} = 16.46 \times 0.24 \times (280 - 80) = 791 \text{ Btu per lb}$$

It will be observed that this loss is the same as that calculated above for 20 per cent excess air. According to the curve shown for this fuel in Fig. 26, the excess air for a CO_2 of 13.18 per cent is 20 per cent.

2. *Loss Due to Water Vapor Produced from the Combustion of Hydrogen.*—The uptake gases are made up of CO_2, air, superheated steam, and other products of combustion. The total pressure P_t is approximately atmospheric and is the sum of the partial pressure of the gases P_g and the partial pressure of the superheated steam P_s.

Since the steam in the uptakes is superheated, its partial pressure, and hence the degree of superheat, are not readily evaluated. The empirical formula usually used for the enthalpy of saturated or superheated steam at low vapor pressures is

$$h = 1{,}059 + 0.46t_g \text{ Btu per lb}$$

where t_g = temperature in degrees Fahrenheit of the uptake gases.

The loss due to hydrogen is

$$\frac{9H}{100}\{(1{,}059 + 0.46t_g) - h_f\} \text{ Btu per lb of fuel}$$

where H = percentage of hydrogen in the fuel.
h_f = enthalpy of water at the temperature of the fuel entering the furnace.

3. *Loss Due to Moisture Contained in the Fuel.*—This loss is equal to

$$\frac{\text{Per cent moisture in fuel}}{100} \times (1{,}059 + 0.46t_g - h_f) \text{ Btu per lb}$$

4. *Loss Due to Moisture in the Combustion Air.*—This loss is very small and in heat balances is usually included in radiation and is unaccounted for.

5. *Loss Due to Carbon Monoxide.*—If the carbon in the fuel is not completely burned to CO_2 because of insufficient air or improper distribution of the air, CO will be formed, and only 4500 Btu per lb will be liberated instead of 14,600 Btu. If CO is present in the uptake gases, the loss is

$$10{,}100\,\frac{C}{100} \times \frac{CO}{CO + CO_2} \text{ Btu per lb}$$

where C = weight percentage of carbon in the fuel.
CO = volume percentage of CO in flue gas.
CO_2 = volume percentage of CO_2 in flue gas.

6. *Loss Due to Combustible in the Ash.*—It is customary to assume that all the combustible in the ash is carbon; hence the loss is

$$A \times \frac{\text{per cent combustible in ash}}{100} \times 14{,}600 \text{ Btu per lb}$$

where A = pounds of ash per pound of fuel.

44. Heat Balance.—In a complete boiler test the various losses mentioned in the preceding article are calculated with the exception of that due to moisture in the air. These losses, together with heat absorbed by the boiler, constitute what is known as a "heat balance." The difference between the heat value per pound of fuel and the sum of the preceding losses and the heat absorbed by the boiler per pound of fuel are recorded as "radiation and unaccounted for." Table VI gives values for a few typical heat balances from tests on marine boilers.

TABLE VI

	B & W "SX"* express boiler	B & W cross drum	B & W cross drum	Oil-fired† Scotch boiler	Hand-fired‡ Scotch boiler
Fuel	Oil	Oil	Oil	Oil	Coal
Pressure at superheater outlet, lb abs	469.4	262.4	407 3	189	173
Steam temperature, °F	716	472.4	658 5	398	408
Rate of combustion, fuel per sq ft boiler H.S. per hr	1.0	0.334	0 277	.	0 42
Heat absorbed by boiler, per cent	82 2	83.60	88.05	86 5	76 0
Loss due to dry stack gases, per cent	10.07	8.24	4.14	6 5	9 5
Loss due to hydrogen in fuel, per cent	6 56	6.16	5 82	6.5	9 5
Loss due to moisture in fuel, per cent	0	0	0	0	0
Loss due to incomplete combustion, per cent	0	0	0	0	0
Loss due to combustible in ash, per cent					3 5
Radiation and unaccounted for, per cent	1.17	2.00	1.99	0 5	7.5
Excess air, per cent	28.57	20.0	12.5	24.0	52.0
Temperature of gases at base of stack, °F	539	452.6	294 0	346	396
Coal per sq ft grate surface per hr				. .	18.8
Heat-recovery equipment	None	None (1934)	Air heater (1937)	Air heater and waste-heat super	Air heater and waste-heat super

* *Jour. Am. Soc. Naval Eng.*, 1931, p. 538.
† *U.S. Bur. Mines Bull.* 214. Test 39, marine boiler.
‡ *U.S. Bur. Mines Bull.* 214. Hand-fired Scotch boiler. Test 11, marine boiler.

45. Oil Burning.—Oil is burned in marine boilers by injecting the oil into the furnace under pressure at high velocity. The oil burners have several small orifices at the tip that are designed to give the oil a whirling motion as it leaves the burner and enters the furnace in a cone of fine atomized spray. In stationary plants steam is sometimes used in the burner to assist in atomization of the oil as it enters the furnace, but steam is practically never used with marine burners because of the loss of feed water it would entail. When oil is atomized by oil pressure alone, it is known as "mechanical atomization." The mechanical atomizing burner was developed for use on shipboard in order to avoid the loss in feed water that would result if a steam atomizing burner were used. Air for combustion is admitted around the burner in a whirling and turbulent state and mixes with the oil spray as it enters the furnace.

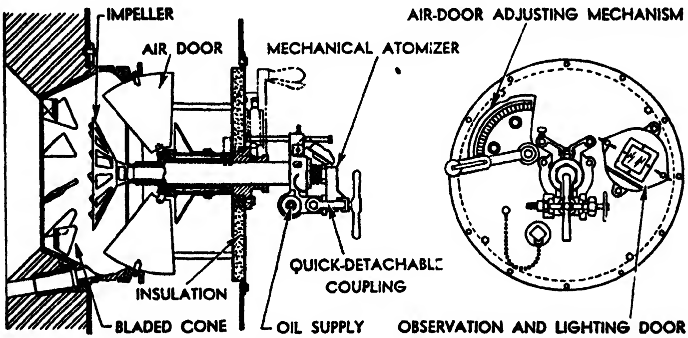


FIG. 27.—Babcock & Wilcox mechanical atomizer in register having radial doors; for double-front operation.

A complete oil burner (Fig. 27) is made up of a register with doors for controlling the admission of air, the burner barrel or atomizer, and usually some sort of impeller plate at the burner tip, which imparts a whirling motion to the air. The burner barrel or atomizer is fitted with a sprayer plate, which is held in place by a burner tip screwed on the end of the barrel. The burner barrel can be readily removed for cleaning or changing sprayer plates. The capacity of the burner can be varied by using sprayer plates with openings of different sizes. When natural draft, the closed fireroom, or induced draft is used, the register, with its doors for admitting the air around the burner, projects out into the fireroom. When force draft is used, a false boiler front is fitted that acts as an air duct (Fig. 11), and the register and air doors are placed between the two boiler fronts as shown in Fig. 27.

The finer the oil particles can be made the more rapid the combustion and the less will be the amount of excess air required. Fine oil particles depend on the design of the burner and the viscosity and the pressure of the oil.

Usually two or more burners are fitted to each furnace, and where high rates of combustion are used boilers have been equipped with seven or more burners. The maximum capacity of oil burners in the merchant service is from around 400 to 1,000 lb per hr. Stationary boilers use burners that have maximum capacities up to 2,500 lb per hr.

The fuel oil is carried in the ship's bunkers, usually located in the double bottom. From here it is pumped by the fuel-oil transfer pump to the settling tanks in the engine room, where it is usually heated a small amount by steam coils in the settling tanks. The fuel-oil service pumps take the oil from the settling tanks and deliver the correct amount to each burner. A fuel oil heater in which steam is the heating medium is located in the oil line between the settling tanks and the burners. The oil is heated to lower its viscosity so that it can be readily sprayed into the furnace through the burner orifices. Oil pressures vary from 100 to 300 lb per sq in., and oil temperatures at the burner, from 100 to 250°F, depending upon the viscosity of the oil at 122°F SSF when purchased. The amount of fuel oil of a given viscosity burned per burner per unit of time depends upon the size of the orifices at the burner tip and the oil pressure. The size of orifices can be varied if desired by removing the burner and changing the burner tip. For a fixed burner tip and oil temperature, the rate of burning oil in a boiler furnace and hence the amount of steam generated can be varied by changing the oil pressure and varying the number of burners in operation (*i.e.*, by "cutting-in" or "cutting-out" burners). The amount of air supplied can be controlled by the doors in the register as well as by the speed of the forced-draft fan.

Ships operating in cold waters and using a fuel oil of high viscosity may have to heat the oil in the bunkers. Steam coils are usually installed in the bunkers for this purpose.

Recently a number of types of burners have been introduced known as "wide-range burners" that have a wide range of capacity. Formerly, as pointed out above, the capacity of a burner was varied by changing the oil pressure, changing the size of the burner tip, or cutting-in or cutting-out burners. The first method gives unsatisfactory results, since a burner tip designed for a given oil pressure does not give satisfactory results for other oil pressures, and large changes in oil pressure are not feasible. The second method requires a considerable amount of time to change sprayer plates for a number of burners. The third method gives satisfactory results but allows the capacity to be changed only by a series of steps, depending on the number of burners installed. Further, it requires manual operation.

The wide-range burners are finding considerable use on shipboard; they allow the use of automatic devices for controlling the rate at which fuel is burned. One of these burners[1] obtains this variable capacity by using a return flow. The burner is fitted with an oil-delivery pipe and a return-flow pipe to the oil service pump. A constant pressure is maintained on the delivery line, and the rate of burning oil is controlled by varying the pressure in the return line from around 100 lb for maximum capacity to 20 lb for minimum capacity.

46. Area of Stack.—Various empirical formulas have been devised for the area of stack, based upon boiler horsepower and height of stack. These are far from satisfactory and are not applicable to marine boilers. Calculations for stack area can readily be made for a fuel of given ultimate analysis and an assumed percentage of excess air. Gas velocity is generally assumed at 1,200 ft per min, but velocities higher and lower than this are permissible. Since the theoretical air requirements vary from one fuel to another and the percentage of excess air, uptake temperatures, and gas velocities are not fixed quantities, exact calculations are not usually warranted.

Marine practice is as follows:

1 sq ft of stack area for each 150 lb of coal burned per hr.
1 sq ft of stack area for each 200 lb of oil burned per hr.

In naval vessels the area of the stack is generally smaller than given by the preceding allowances. The general practice is 1 sq ft of stack area per 300 lb of oil burned per hr, and still further reductions in area can be made when necessary.

[1] G. P. Haynes and S. Letvin, "A New Fuel-oil Burner," *Trans. Soc. Naval Arch. and Mar. Eng.*, 1938.

47. Method of Supplying Coal with Air.—If all the air were supplied from below the fuel bed, carbon dioxide (CO_2) would be formed in the lower part of the bed, and then later this CO_2 would be reduced to carbon monoxide (CO) as it came in contact with the incandescent carbon at the top of the fuel bed. Moreover, as the hydrocarbons were distilled off from the top of the fire, there would be little or no oxygen present to unite with these gases and liberate heat by the formation of CO_2 and H_2O. Under these conditions there would be a large loss of available carbon in the form of CO, and, also, smoke would be produced.

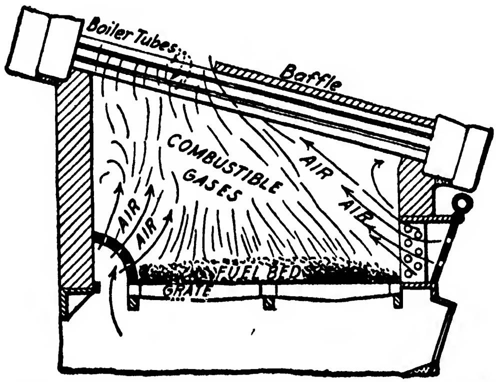


FIG. 28.—Wager bridge wall fitted with water-tube boiler.

To secure proper combustion it is necessary that a sufficient amount of air be distributed over the top of the fuel bed. This air must be brought to a high temperature before coming into contact with the volatile gases; otherwise the gases will be chilled, and smoke will result. It is also essential that air supplied at the top of the fuel bed should be thoroughly mixed with the gases so that complete oxidation will take place. This air supply for the top of the fire is sometimes passed up through the fuel bed, but more often it is admitted through openings in the fire door or at the bridge wall in the rear. The Wager bridge wall, shown in Fig. 28, is an excellent means of admitting air to the top of the fire, and as the air comes in through small slots proper mixing will take place. The heating of the air can be accomplished by means of an air heater or by passing the air and gases over incandescent portions of the fire. Air is especially needed at the top of the fire after firing, when the volatile gases are being distilled off; hence the practice of covering part of the fire at one time with fresh coal leaves the incandescent part of the fuel bed to assist in the combustion of the gases given off from the fresh coal.

48. Draft.—In order that the air required for combustion may enter the boiler and the products of combustion pass through the boiler tubes, superheater tubes, heat-recovering devices, and uptakes, there must be a difference in air pressure or draft between the furnace front and the base of the stack. The draft may be produced by building up a positive pressure at the entrance to the burners or in the ash pit for coal-fired boilers (forced draft); by reducing the pressure in the uptakes below that of the fireroom (induced draft); or by a combination of forced and induced draft (balanced draft). Draft may be either natural or mechanical. Natural draft is devoloped by the suction produced by a chimney or stack, and mechanical draft is an artificial draft, usually produced by fans or blowers. The total draft required is the sum of the following pressures and resistances: (1) resistance in the passages between the fan and the boiler; (2) air resistance in the air heater when fitted, (3) resistance offered by the boiler casing if the combustion air is passed around the boiler (Fig. 11); (4) air pressure to give the necessary velocity for air entering the registers of the oil burner or to overcome the resistance of the fuel on the grate; (5) resistance of the boiler heating surface; (6) resistance of the superheater; (7) resistance of the economizer; (8) resistance of the hot gases through the air heater; (9) resistance offered by the uptakes; and (10) pressure to provide the necessary velocity of flow through the uptakes. Since the required draft for a boiler is small, it is usually expressed in inches of water instead of in pounds per square inch.

49. Natural Draft.—Natural draft is produced by a stack and is the difference in weight between the column of hot gases in the stack and a column of cold air of the same height as the stack. Natural draft varies with the height of the stack, the temperature of the stack gases, and the temperature of the outside air.

Consider an apparatus similar to that shown in Fig. 29. Let A be a long pipe H ft high, filled with cold air at temperature t_a, and let B be a pipe of the same height and diameter, filled with hot gases at temperature t_s. At C a flexible diaphragm is fitted as shown. The weight of 1 cu ft of air at 32°F and at sea level is 0.0807 lb; the weight of 1 cu ft of stack gas at the same temperature and pressure is approximately 0.085 lb. P, V and R are constants in the equation $PV = WRT$; hence the weight, or pressure exerted, by the column of air A is

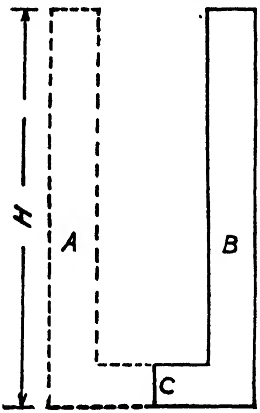


FIG. 29.

$$P_a = \frac{0.0807 \times 492}{(460 + t_a)} \times H \text{ lb per sq ft}$$

The pressure exerted by the column of hot gases B is

$$P_s = \frac{0.085 \times 492}{460 + t_s} \times H \text{ lb per sq ft}$$

The difference in pressure on the diaphragm C, due to the heavier column of air, is

$$P = P_a - P_s = \frac{(0.0807 \times 492)H}{460 + t_a} - \frac{(0.085 \times 492)H}{460 + t_s}$$

$$P = H\left(\frac{39.6}{460 + t_a} - \frac{41.8}{460 + t_s}\right) \text{ lb per sq ft}$$

If this is converted to inches of water, we have to divide by $62.5/12 = 5.2$ lb (1 in. of water pressure being equal to 5.2 lb per sq ft).

$$P = H\left(\frac{7.63}{460 + t_a} - \frac{8.05}{460 + t_s}\right) \text{ in. of water}$$

If the diaphragm at C in Fig. 29 is replaced by a boiler furnace that will maintain the stack B at a temperature t_s, higher than the outside air, we have an example of draft created by a stack. The pipe A can be removed, for the column of cold air equal in area and height to column B will still exist.

The preceding formula can now be used to calculate the difference in pressure or draft measured in inches of water created by a stack.

Suppose, for example, that air is at 70°F, gases leave the boiler at 500°F, and the height of the stack is 100 ft. The draft under these conditions is

$$P = 100\left(\frac{7.63}{530} - \frac{8.05}{960}\right) = 0.60 \text{ in. of } H_2O$$

The formula may be transposed to give the height of stack H in feet necessary to produce a required draft:

$$H = P \div \left(\frac{7.63}{460 + t_a} - \frac{8.05}{460 + t_s}\right)$$

Thus, if ¾-in. draft were necessary with stack gases at 550 and outside air at 60°F

$$H = \frac{0.75}{\frac{7.63}{520} - \frac{8.05}{1010}} = 114 \text{ ft}$$

The height given by the preceding formula is the theoretical height without friction. It is customary in using the formula to take the temperature of the gases at the base of the stack, which is known, instead of the mean temperature in the stack, which is generally unknown. In practice the height of the stack is increased about 25 per cent to allow for the friction losses in the chimney and to allow for the fact that the mean stack temperature is lower than that at the base of the stack. Likewise, for a given height of stack, the actual draft obtained would be about 80 per cent of that obtained by the formula. The formulas now become

$$\text{Actual } P = 0.80H\left(\frac{7.63}{460 + t_a} - \frac{8.05}{460 + t_s}\right) \text{ in. of } H_2O$$

$$\text{Actual } H = 1.25P \div \left(\frac{7.63}{460 + t_a} - \frac{8.05}{460 + t_s}\right)$$

The movement of the air across the top of the stack, because of the speed of the ship or a head wind, will increase the draft appreciably; a following wind may reduce the movement of air across the top of the stack to zero and hence give a draft due only to the height of the stack. Likewise, a low outside temperature t_a will increase the draft as shown by the formula. However, it is not wise to rely too much on these factors, for the full required draft should be obtained on a summer's day with a following wind.

The intensity of natural draft, therefore, is dependent on the height of the stack above the grate and the temperature of the gases leaving the boiler. Since high boiler efficiency requires a low exit temperature for the gases, we see that a strong draft is created only at the expense of boiler efficiency.

Since the intensity of natural draft is fixed for good operating conditions by the height of the stack, for an ordinary cargo ship the draft is thus limited to about ½ in. of water. For higher drafts, or where any flexibility is desired, mechanical draft must be used.

50. Forced Draft.—Forced draft consists in creating a positive pressure greater than that of the atmosphere in the ashpit in coal-burning boilers or around the registers in oil-burning boilers. For coal-burning boilers the draft may be produced either by the closed-ashpit method or by the closed-fireroom method. In the former, the ashpit is sealed tight and put under pressure by the blower; in the latter the whole fireroom is sealed and put under pressure. With oil-fired boilers a double boiler front may be used (Fig. 11) and a positive air pressure built up in the space between the two fronts; or the fireroom may be put under air pressure as described above. The latter is known as the "closed-fireroom system." The closed-fireroom system is used almost universally in naval vessels but very seldom in merchant ships. It has been used to a very limited extent on passenger vessels. This system has the advantage that no air ducts are required but does offer some disadvantages when air heaters are used and has the serious disadvantage that the approach and exit from the fireroom must be made through air locks.

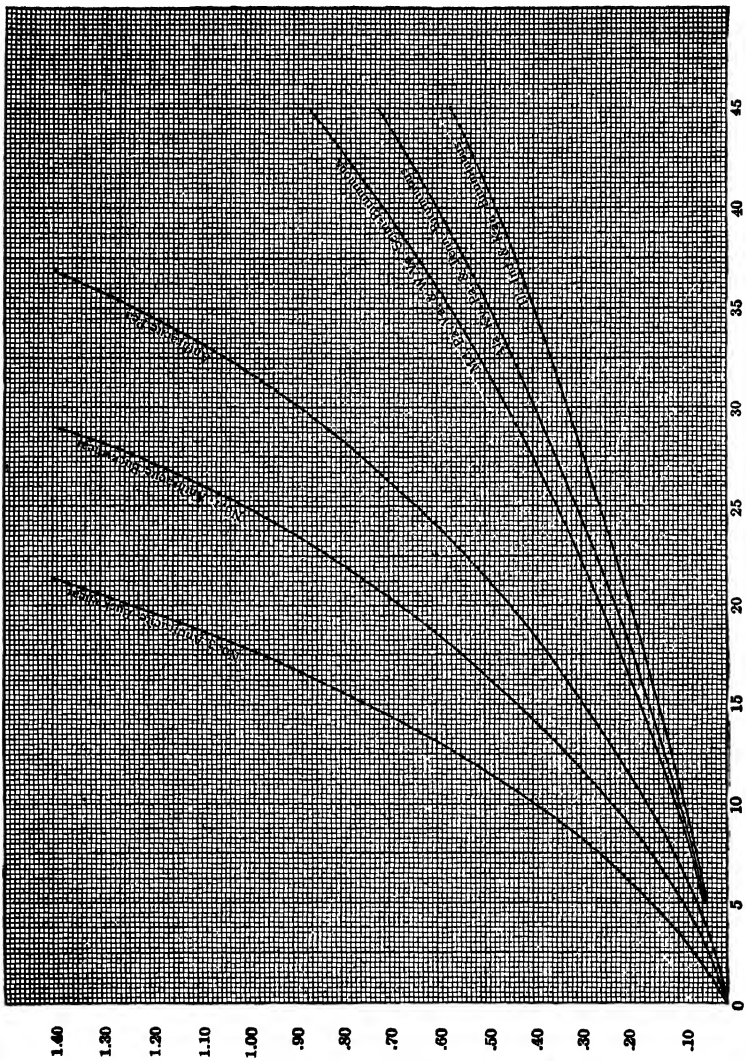
In the boiler shown in Fig. 11, the combustion air is sent by the blower into the upper left-hand end of the air heater (see Fig. 21) and after making two passes through the air heater passes down through the boiler casing (see Fig. 11) to the space under the boiler. It then passes up behind the false front of the boiler and enters the registers of the burners. An arrangement of burners and false front is shown in Fig. 16.

51. Induced Draft.—With induced draft the fans are located above the boilers and draw the hot gases from the uptakes and deliver them under pressure to the base of the stack. Induced draft can be considered as an intensified natural draft. Because of the large volume of gases handled by the fans due to their higher temperature, induced-draft fans must be much larger than forced-draft fans. One serious objection to induced-draft fans is that they are located over the boilers, where the temperatures are high and repairs are difficult, and these fans deteriorate more rapidly than forced-draft fans because of the hot combustion gases handled. With induced draft the pressure in the furnace is less than atmospheric, and the tendency is for air to leak into the furnace; with forced draft the pressure in the furnace is above atmospheric. Balanced draft requires two fans, and the pressure in the furnace is about atmospheric. Induced and balanced draft have been used only to a limited extent on shipboard. When Ljungstrom air heaters are installed (Art. 36), two fans, and hence balanced draft, are necessary.

Because of the large total resistance of modern boilers, mechanical draft is required, since stack heights are insufficient to give a natural draft that is strong enough. Natural draft still finds wide use with many Scotch boilers, but when fire-tube or waste-heat superheaters are installed it is necessary to use mechanical draft.

52. Draft Requirements.—The total draft required with an oil-burning water-tube boiler is made up of the sum of the items listed in Art. 48. The largest of these is usually the air resistance through the burners, which increases with the temperature of the combustion air. The other resistances vary with the make of the boiler and must be obtained from the boiler manufacturers. Draft increases with rate of combustion (see curve of fireroom pressure, Fig. 22). The resistances through the coal beds of coal-burning boilers for various rates of combustion are given in Fig. 30.

53. Automatic Boiler Control.—In the past the control of the fuel and air supply to the boilers has always been regulated by the fireman by manual control. The rate of supplying feed water to the boiler has also been more or less under manual control. Some of the new merchant ships recently constructed in the United States are now fitted with automatic boiler control, which supplies water, fuel oil, and combustion air to the boiler in proportion to the demands for steam. A complete control, such as the Bailey boiler control shown in Fig. 21, controls (1) the rate of boiler feed by a feed-water regulator that operates by the water level in the steam drum; (2) the combustion by regulat-



Courtesy Babcock & Wilcox Co.

Draft Required between furnace and ashpit—inches of water

Fig. 30.—Draft required with various kinds of coal.

ing the fuel supply to the burners and the combustion air supply; (3) the temperature of the steam leaving the superheater. The fuel and air supply to the burners are controlled by slight changes in boiler steam pressure; an increase in steam pressure cuts down the supply of both fuel oil and combustion air. The percentage of excess air can be kept constant by such a control regardless of the rate of burning fuel. A full explanation of automatic boiler control is beyond the scope of this book, but the reader is referred to the notes shown in the diagrammatic layout in Fig. 21. The introduction of wide-range burners (Art. 45) has made automatic combustion control possible.

54. Mechanical Stokers.—Mechanical stokers have been fitted on a fairly large number of ships in lieu of hand firing for burning coal. Most of the installations have been with water-tube boilers because of the difficulty of fitting the earlier types of stokers and removing the ashes and clinkers from the small-diameter furnaces of Scotch boilers.

Stokers offer the following advantages over hand firing:

1. Continuous and uniform firing and uniform distillation of hydrocarbons.
2. Uniform thickness of fire; hence uniform draft, with no holes in the fuel bed.
3. No chilling of volatile gases or dilution by frequent opening of furnace doors.
4. Higher rates of combustion.
5. Reduced fireroom force; less arduous duties for firemen.
6. Higher boiler efficiency (less excess air required).
7. Personal element of firing eliminated.
8. Smokeless combustion.

Three types of stokers have been used on shipboard; the underfeed, the chain-grate, and the thrower or spreader type. Many of the stoker installations with water-tube boilers, especially abroad, have been of the underfeed type of stoker; chain-grate stokers are installed on two car ferries on Lake Michigan, and the spreader (overfeed) type of stoker has been installed on ships on the Great Lakes in connection with both water-tube and Scotch boilers. The spreader or thrower type of stoker is the only one that has been used successfully with Scotch boilers. A spreader-type stoker built by the Hoffman Combustion Engineering Company, which is applicable to both water-tube and Scotch boilers, is shown in Fig. 31. In this type of stoker the coal is spread continuously and automatically on top of the fire. The coal is fed into the hoppers on the front of the boilers by chutes from overhead bunkers. Figure 31 was drawn for a stationary boiler; in a marine installation the deep ashpit shown in Fig. 31 would not be used. With water-tube boilers, dumping grates are fitted as shown in Fig. 31. Because of the

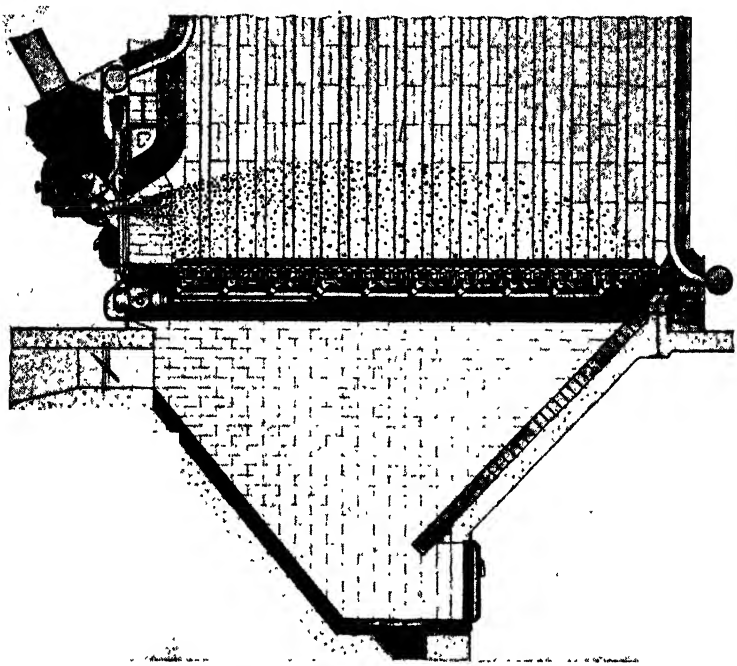


FIG. 31.—Hoffman stoker fitted to a water-tube boiler.

small depth of furnace in the Scotch boiler, dumping grates are not used with this type of boiler. The grates used with the Hoffman stoker are fitted with patented tube tuyères (see Fig. 31) which cause the combustion air to be introduced at an angle to the fuel bed and at a comparatively high velocity, thus creating turbulence and assisting combustion. A thin fuel bed is used that makes the boilers responsive to changes in demands for steam during maneuvering and also reduces the possibility of clinkering and slagging. A full-sized firing door is fitted directly under the coal-feeding mechanism, as shown in Fig. 31. This allows the fuel to be hand-fired if the stoker mechanism should fail. In a Scotch boiler this door is used for removing the ashes from the grate. Figure 32 shows a photograph of the Hoffman stokers fitted to the Scotch boilers of the Great Lakes ship *S.S. Reiss Brothers.* In the Hoffman stoker the impeller operates at constant speed, throwing the coal on the fire. The amount of fuel fed is varied by the length of the stroke of the feeder box.

FIG. 32.—Hoffman stoker installation on *S.S. Reiss Brothers.*

Hoffman stokers are installed on the Great Lakes car ferry *S.S. City of Midland* (1941) in conjunction with Foster Wheeler D-type boilers. Spreader-type stokers of both the Hoffman (Firite) and the Detroit "Roto" type are installed on the four Great Lakes bulk carriers (1937) *S.S. Governor Miller, S.S. John Hulst, S.S. Ralph W. Watson,* and *S.S. William A. Irwin* in conjunction with Babcock and Wilcox and Foster Wheeler water-tube boilers.[1]

[1] "Geared Turbines for Lake Freighters," *Mar. Eng. & Shipping Rev.,*

Some of the foreign makes of thrower-type stokers are designed to throw successive units of coal with three different intensities so that it lands on the fire bed in three different longitudinal postions, thus, successively, covering the grate. A number of foreign makes of thrower-type stoker have been fitted on Scotch boilers. Among the ships so fitted may be mentioned the *S.S. Leuna*[1] and the *S.S. Manchester Port.*[2]

Among the ships fitted with underfeed stokers and water-tube boilers are the four "Beaver" class ships of the Canadian Pacific Railway[3] and 24 small ships belonging to the East Indian service of the Koninklijke Paketvaart Maatschappij (K.P.M.).[4]

55. Pulverized Coal.—Coal in the pulverized form was first used in stationary plants about 1916, but because of the large furnace volumes required the methods of burning pulverized coal used ashore were not suitable for marine boilers. Research was carried out in 1927–1929 at the U.S. Fuel Oil Testing Plant at Philadelphia by the U.S. Shipping Board and the U.S. Navy in burning pulverized coal in a Scotch boiler. The result of this research was the development of a burner that allowed the pulverized coal to be burned with a short flame in the small furnace of a Scotch boiler. The first seagoing ship to operate on pulverized coal was the *S.S. Mercer*, belonging to the U.S. Shipping Board, which was converted for using pulverized coal in 1927.

Ships using pulverized coal grind the coal to a fine powder on shipboard by means of a ball mill or a high-speed pulverizer. The coal is sent from the pulverizer to the burners through pipes about 6 in. in diameter with a small supply of carrier (primary) air that floats the coal dust along the pipes. The coal dust is blown into the furnace under slight pressure. The secondary air for combustion is admitted around the burner

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1937, p. 586. These ships are fitted with geared turbines and have a boiler pressure of 450 lb and an initial steam temperature of 750°F. f = approximately 0.86 lb of coal per shp-hr. See also *Mar. Eng. & Shipping Rev.*, 1938, p. 262.

[1] *The Marine Engineer & Motorship Builder*, January, 1929.

[2] *Shipbuilding and Shipping Record*, vol. 48, p. 424.

[3] *Shipbuilding and Shipping Record*, Mar. 15, 1928.

[4] "Lentz Engines, Water-tube Boilers and Mechanical Stokers," *The Marine Engineer & Motorship Builder*, 1927, pp. 47–55.

much in the same way the air is admitted in burning oil. The coal particles burn in suspension in the furnace in the same manner as oil is burned. The fires are started with a torch, as in burning oil. The fires are shut off by stopping the pulverizer; they cannot be shut off by closing the pipe at the burner; if they were, the fuel pipe would be filled with compact coal.

The advantages of pulverized coal over hand firing are much the same as with mechanical stokers, *i.e.*, reduced fireroom force, improved boiler efficiency because of the smaller amount of excess air required, and much more flexible control. In addition, pulverized coal has certain important advantages over mechanical stokers, namely, the fires can be started and shut off in the same manner as oil fires; a shift can be readily made from coal to oil and oil to coal as fuel prices warrant; cheaper slack coals can be used, and there is no ash handling, since most of the ash leaves by the stack.

The installation on the *S.S. Mercer* was highly successful in every way except that the method of distributing the coal to the three furnaces of the Scotch boilers was not entirely satisfactory. Several other American ships were fitted with pulverized-coal-burning equipment since the *Mercer* was so fitted, but because of the prevailing comparatively low price of fuel oil and other factors mentioned later, no American seagoing ships have been fitted with pulverized burning equipment since 1930.

After the conversion of the *Mercer* to pulverized coal burning, the development was taken up in Great Britain and to some extent in Germany. Clarke-Chapman brought out a very successful system that overcame the distribution problem that caused difficulties on the *Mercer*. The Clarke-Chapman system uses a high-speed paddle pulverizer (3,000 rpm) instead of the heavy, slow ball mill used on the *Mercer*. A number of British ships were converted to burning coal by the Clarke-Chapman system, and several new ships were fitted with this equipment, two of them Japanese.[1]

With the exception of the American ship *S.S. Berwindvale*, all new and conversion installations of pulverized coal have been on Scotch boilers. The total number of new and converted ships fitted with pulverized-coal equipment numbers around 25; to

[1] *S.S. Nagoya Maru* and *S.S. Jahore Maru*, *Engineering*, London, vol. 135, Apr. 7, 1933, p. 377.

the author's knowledge none has been fitted with this equipment since 1932.

The present handicaps to the burning of pulverized coal that have prevented its wider adoption on shipboard appear to be:

1. Inadequate methods of bunkering coal, making it a slow and dirty process.
2. Space occupied by coal on shipboard; it cannot be stored in double bottom and peak tanks as oil can.
3. Unavailability of cheap slack coals at bunkering stations.
4. Present relatively low price of oil at many bunkering ports.
5. High fixed charges of pulverized-coal-burning equipment and coal-conveying equipment that do not allow enough saving in operating costs over hand firing for low-powered installations.
6. High power required by pulverized-coal equipment.
7. Lack of sufficient experimental work with pulverized coal in marine water-tube boilers with small furnaces.

Considerable difficulties have been experienced in the burning of pulverized coal in brick-lined furnaces of marine water-tube boilers because of the slagging and erosion of the brickwork caused by the ash in the coal when the furnace temperature is above the fusing temperature of the ash. Lowering furnace temperatures by the use of an increased amount of excess air will, of course, lower the boiler efficiency. When coals such as Pocahontas were used with a fusing temperature of the ash around 2700°F, no difficulties with slagging were experienced.[1]

Tests were carried out with a Babcock & Wilcox marine boiler at the experimental plant at Renfrew, England, burning pulverized coal.[2] This boiler had a ratio of furnace volume to heating surface of 0.20. It was fitted with a water-cooled furnace (Bailey water walls) and had a water screen on the sloping bottom of the furnace. Coals with fusing temperature of the ash as low as 2264°F were burned successfully with 18 and 20 per cent excess air. The boiler efficiencies were in the neighborhood of 80 per cent; the clinkers were chilled by the tubes on the bottom of the furnace and removed through a special door in the front of the furnace (see Table IV, column 10).

[1] Thomas B. Stillman, "Pulverized Coal Tests of a Marine Water-tube Boiler," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1927.

[2] Whayman and Gregson, "The Water-tube Boiler and the Cargo Ship," *Trans. Northeast Coast Inst. Engineers & Shipbuilders*, 1932, vol. 49.

If the price of oil increases sufficiently over that of coal, no doubt pulverized coal will be used on shipboard with water-tube boilers. It is doubtful if serious problems will arise in water-tube boilers with water-cooled furnaces of fairly large volume.

Despite the handicaps of pulverized coal mentioned, it seems reasonable that a wider use should be made of this method of burning coal in Scotch boilers of foreign cargo ships where hand-fired coal is still the practice.

Results of the service performance of the *S.S. Recorder* (fitted with Clarke-Chapman system) showed a fuel-cost saving of 25 per cent over the average of eight sister ships on the same trade route, where the coal was hand-fired. A lower grade (slack) coal was used on the *Recorder* than on the sister ships. It was reported that this ship was a commercial success, since the saving in fuel and crew's wages was more than the increased fixed charges and upkeep of the pulverized-coal-burning equipment.[1] Burners are available that have an oil-burner atomizer in the center of the pulverized-coal burner so that a ship fitted with pulverized-coal-burning equipment can shift rapidly from coal to oil when fuel prices warrant.

[1] W. G. Gibbons and M. Arthurson, "Pulverized Coal Firing in Marine Scotch Boilers," *Trans. Inst. Mar. Eng.*, 1933, vol. 45, pp. 25 and 43.

CHAPTER V

THE RECIPROCATING STEAM ENGINE

56. General.—Up to about 1914 nearly all the world's seagoing merchant ships and large naval vessels were propelled by the reciprocating steam engine. The geared turbine, the turbo-electric drive, and the Diesel engine made their appearance in commercial vessels almost simultaneously around 1913 and by 1917 the triple-expansion and quadruple-expansion steam engine began to feel the competition of the Diesel engine and geared turbine. The direct-connected turbine had been used in many of the large passenger vessels, in some smaller passenger vessels, and in many naval ships between 1905 and 1913, but practically no direct-connected turbines had been used for the propulsion of cargo ships.[1]

The geared turbine and the Diesel engine have gradually replaced the reciprocating engines in new construction so that for a number of years no new ships of any size have been added to the American Merchant Marine with triple-expansion steam engines.[2] However, a large percentage of the existing merchant ships of the world are equipped with reciprocating steam engines, and many new foreign cargo ships, especially British, are being fitted with this type of prime mover. In 1939, 64.1 per cent of the world's merchant tonnage classified by Lloyd's had reciprocating steam engines, and 28.8 per cent of the new tonnage added in 1939 was propelled by this type of machinery. Many improvements have been made in the reciprocating steam engine in recent years, so that today its performance is greatly improved over that of the engines built in 1917. For the foregoing reasons it is essential that the marine engineer be familiar with the

[1] The first merchant ship propelled by direct-connected turbines was the *S.S. King Edward* (1901). The first geared-turbine ships were the *S.S. Vespasian* (1910), the *U.S.S. Neptune* (1911), and the *S.S. Normannia* (1911).

[2] See footnote, Art. 121, Chap. IX.

reciprocating engine and be able to compare its merits with other types of propelling machinery.

57. Reasons for Multiple-expansion Engines.—One of the largest losses in reciprocating steam engines is that due to cylinder condensation. This is due to the large temperature difference between the entering steam and the cylinder walls that have just been in contact with the exhaust steam. With high steam pressures and high vacua, this range in temperatures becomes very great. To reduce this temperature difference in the cylinder, compounding has been resorted to, the steam passing from one cylinder to another as it expands. Thus, if two cylinders are used, the temperature range in each cylinder is cut in half, for the steam now leaves the first cylinder at a temperature much higher than that which would have resulted if only one cylinder had been used. Consequently, the cylinder walls are at a higher temperature when the steam enters, and cylinder condensation is reduced. In addition to reduction in the temperature range, the loss due to the clearance volume is reduced by compounding, for with a smaller high pressure cylinder the high pressure steam fills a smaller clearance volume than would be the case if one large cylinder were used.

As would be expected from the foregoing, the steam consumption of an engine is reduced by compounding, and greater ranges of expansion are possible. Compounding has the further advantage in that a more uniform turning moment is obtained on the propeller shaft.

58. Triple- and Quadruple-expansion Engines.—A compound engine consists of two cylinders, a high pressure (H.P.) and a low pressure (L.P.). As steam pressures and vacua were increased, the expansion was divided into three steps in the triple-expansion engine. The cylinders in this engine are high, intermediate, and low. Finally came the quadruple-expansion engine, with four cylinders, high, first intermediate, second intermediate, and low.

Another type of multi-expansion engine used in marine work is the four-cylinder triple-expansion engine. Here, instead of using one large and cumbersome low pressure cylinder, the latter has been divided into two equal smaller sized cylinders. The engines used in the *Delaware* (Table VII) were of this type. Instead of using one large 107 ½-in. L.P. cylinder, two 76-in. cylinders have been substituted; the steam, on leaving the intermediate cylinder, divides, half flowing to the forward L.P. cylinder and half to the aft L.P. cylinder. This has the advan-

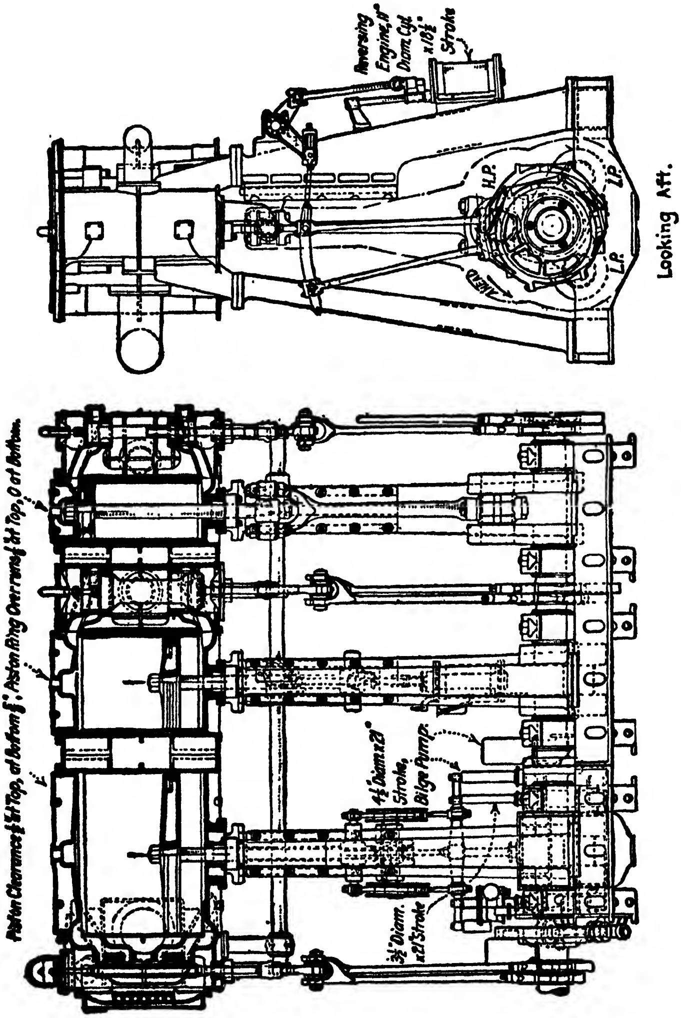

FIG. 33.—Triple-expansion engine. (*Newport News Shipbuilding and Dry Dock Company*, 1920.)

tage of reducing the size of the cylinder casting, piston, valve gear, etc., and also of giving a more uniform turning moment.

Figure 33 shows a drawing of a triple-expansion engine, and Fig. 34 shows a quadruple-expansion engine. Both of these are typical merchant engines with cast-iron columns or housings

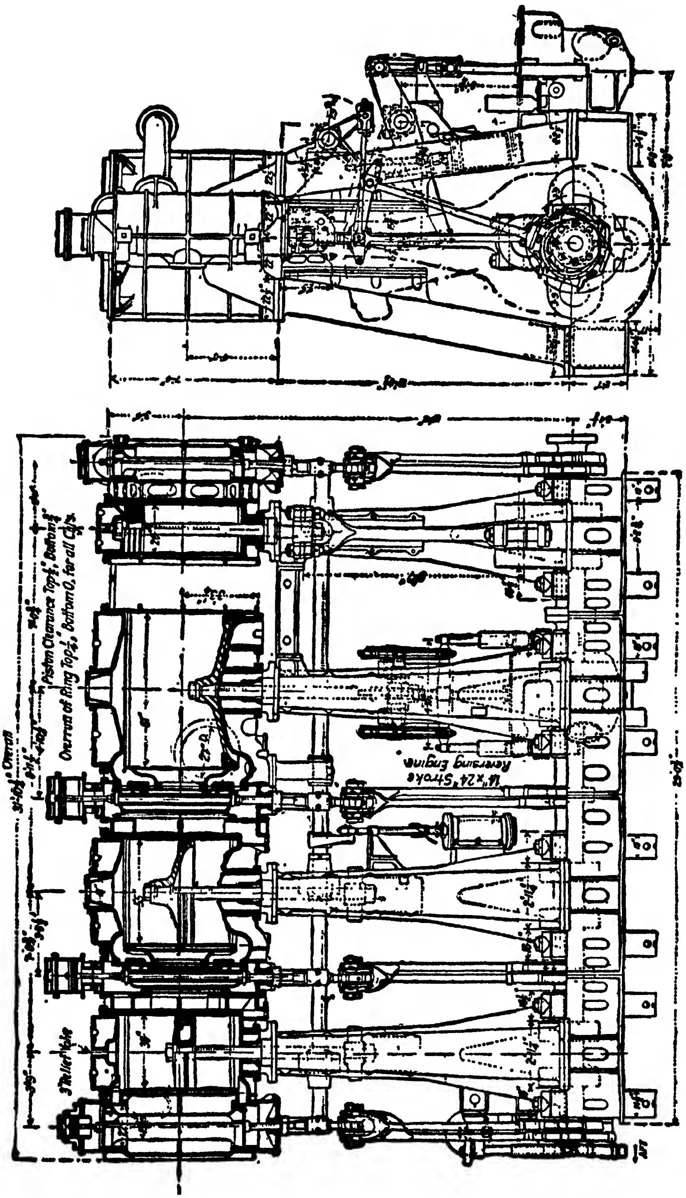

FIG. 34.—Quadruple-expansion engine. (*Newport News Shipbuilding and Dry Dock Company, 1920.*)

of box section. The ratio of the crank to the connecting rod is 1:4.5.

59. Determination of Engine Dimensions.—The L.P. cylinder in a multiple-expansion engine must be of sufficient size to handle all the steam used by the engine, and consequently this cylinder is of the same size that would be used in a simple one-cylinder engine working between the same initial and back pressures. Compounding does not reduce the size of the L.P. cylinder over that required for a simple engine, except insofar as compounding reduces the steam consumption.

In determining the proper dimensions for an engine, the size of the L.P. cylinder is first fixed and then the sizes of the other cylinders are determined so as to divide the work equally among the cylinders.

The piston speed (P.S.) is generally settled first, and this should conform with good practice. For merchant engines the P.S. is between 600 and 1,000 ft per min; the higher the P.S. the lighter will be the engine. Naval engines have used P.S. as high as 1,200 ft per min. The stroke is determined after the revolutions have been decided. The revolutions of the engine are governed by the propeller efficiency, and for the slow-speed merchant ship the rpm used is generally between 70 and 90. The revolutions should be settled on by a careful study of the speed of the boat, wake coefficient, and propeller efficiency.

$$\text{P.S. in feet per minute} = \text{rpm} \times 2 \times \text{stroke in feet}$$

$$\text{P.S.} = 2LN$$

$$\text{Ihp} = \frac{PLAN}{33{,}000} \text{ for single-acting engine}$$

where P = mean effective pressure acting on piston in pounds per square inch.

L = stroke, in feet.

A = area of the L.P. cylinder, in square inches.

N = rpm.

P.S. = piston speed.

For a double-acting engine with steam acting on both sides of the piston the Ihp becomes approximately twice the preceding or

$$\text{Ihp} = \frac{P \times A \times \text{P.S.}}{33{,}000} \text{ for double-acting engine}$$

In working up the design of a multiple-expansion engine, it is assumed that all the work is done in the L.P. cylinder and hence the P used in the preceding expressions is the mep (mean effective pressure) referred to the L.P. cylinder or, as commonly written, mrp (mean referred pressure) or referred mep. This mrp is used only in designing and is not the same as the mep in the L.P. cylinder of the actual engine. It is the mep that would result if all the expansion took place in the L.P. cylinder instead one-third or one-fourth of the expansion, as is the actual case.

The theoretical mrp can be quickly obtained when the initial pressure (P_1), the back pressure (P_B), and the ratio of expansion (R) have been settled upon. The ratio of expansion is the number of times that the steam admitted per stroke to the H.P. cylinder expands in volume. In other words, it is the ratio of the volume of the H.P. cylinder at cutoff to the volume of the L.P. cylinder. The proper ratio of expansion to use can be fixed only after some knowledge of engine design has been obtained by actual experience. Theoretically, the higher the ratio of expansion used the greater will be the thermal efficiency of the engine, but other considerations such as size, weight, and engine friction fix the ratio of expansion that it is advisable to use. In naval vessels this ratio is lower than in merchant vessels in order to cut down the weight of the engine and the size of the L.P. cylinder (see Table VII). If a hyperbolic expansion (PV = constant) instead of an adiabatic expansion is assumed, then

$$\text{Mrp} = P_1\left(\frac{1 + \log_e R}{R}\right) - P_B$$

For a given pressure at cutoff, the ratio of expansion fixes the pressure at release P_2. Thus, for the card shown in Fig. 35, the pressure $P_2 = P_1V_1/V_2$. $P_2 = 300/8.71 = 34.5$ lb abs (see Table VII for data). The back pressure P_B is the pressure existing in the L.P. cylinder at exhaust and is somewhat higher (usually 1.5 to 2.0 lb) than the pressure in the condenser.

The mrp found by the preceding formula is the theoretical value that would be obtained for an expansion between the limits of P_1 and P_B under ideal conditions. Actually, the mrp is much less than found by this expression because of cylinder condensation, leakage, transfer of steam from one cylinder to another, wire drawing, incomplete expansion, and clearance losses. The mrp, as found by preceding formula, is multiplied by a coefficient obtained from practice. This coefficient is known as the "card effect" or "mep factor." Values obtained from engine trials are given in Table VII. The mep factor increases as engine efficiency

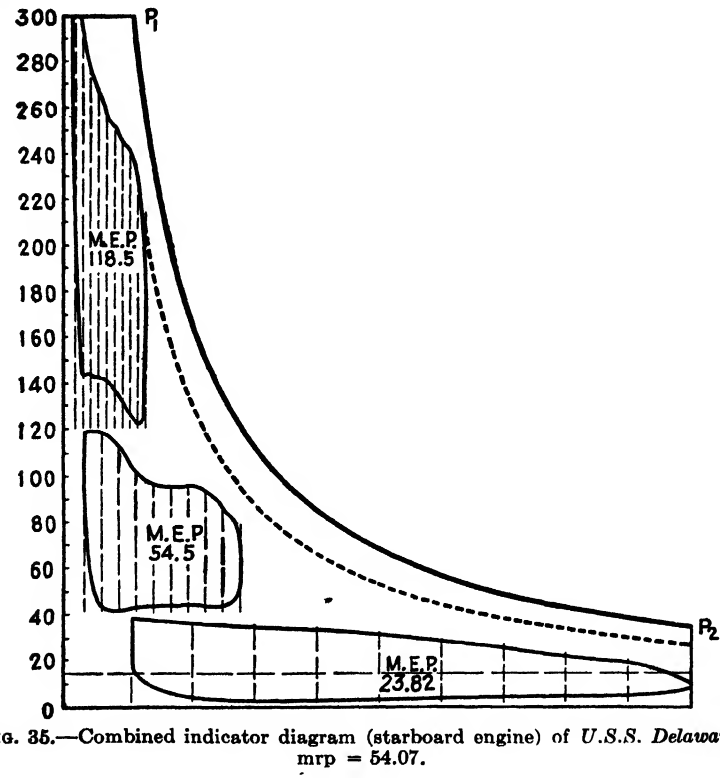

FIG. 35.—Combined indicator diagram (starboard engine) of *U.S.S. Delaware*, mrp = 54.07.

increases and as water rate decreases. Figure 35 shows the theoretical and actual cards of the *Delaware*, from a paper by R. T. Hall in the *Journal of the American Society of Naval Engineers*, 1909.

After the area of the L.P. cylinder has been obtained, the area of the H.P. cylinder can be obtained from the ratio of expansion

TABLE VII.—PERFORMANCE OF RECIPROCATING ENGINES

Ship or location	Type	Reference	Cylinder dimensions	Horsepower and rpm, 1 engine	Pressure (abs) at engine	Superheat	Vacuum, in.	Ratio of expansion	Mrp	Steam per ihp-hr, main engines	Mep factors	Engine efficiency	Coal per ihp-hr
Land—pumping	Quadruple*	*Eng. News*, May 4, 1899	$\frac{19.5\times29\times49.5\times57.5}{42}$	712 at 36.5	200	0	28	.	35.5	12.26	.	0.742	1.12
Land—pumping	Triple‡	C.H. Peabody's Thermodynamics	$\frac{13.7\times34\frac{3}{8}\times39}{72}$	575 at 50.6	190.4	0	27.25	...	....	11.10	...	0.74	1.15
Land—pumping	Triple	*Eng. News*, Aug. 23, 1900	$\frac{30\times56\times87}{66}$	801 at 17.2	185	0	28.2	.....	23.4	10.33	....	0.705	1.09
Land—pumping	4-cyl. triple	*Eng. News*, Oct. 2, 1902	$\frac{32\times47\times58}{59}$	2,860 at 85	173	230	27.9	....	.....	8.97	...	0.735	
Land—high speed	Triple	*N.E.C.I.E.&S.*, Morrison, 1910	$\frac{18.5\times27\times40}{20}$	1,080 at 250	202.7	91	27.9	11.7	...	12.7			
U.S.S. Delaware†	4-cyl. triple	*S.N.E.*, 1909	$\frac{38.5\times57\times76\times76}{48}$	14,290 at 128.5	300	61.6	26.3	8.71	50.9	13.38	0.545	0.56	1.89
U.S.S. Birmingham	4-cyl. triple	*S.N.E.*, 1910	$\frac{28\frac{1}{4}\times45\times62\times62}{36}$	7,738 at 191	221	0	26.5	11.5	36.3	17.4	0.546	0.455	1.93
U.S.S. South Carolina	4-cyl. triple	*S.N.E.*, 1912	$\frac{32\times52\times72\times72}{48}$	8,825 at 121	256	47	26.24	12.1	37.0	....	0.526	.	1.40
U.S.S. Michigan	4-cyl. triple	*S.N.E.*, 1912	$\frac{32\times52\times72\times72}{48}$	8,000 at 119	260	85.6	26.95	12.6	34.2	...	0.493	.	1.51
U.S.S. Michigan	4-cyl. triple	*S.N.E.*, 1912	....	1,745	88	51	27.35	12.3	12.35	..	0.527		
U.S.S. Cyclops	3-cyl. triple	*S.N.E.*, 1912	....	3,368	192	0	27.0	12.04	33.75	...	0.634		
U.S.S. New York	4-cyl. triple	*S.N.E.*, 1915	$\frac{39\times63\times83\times83}{48}$	15,190 at 128.5	280	3	26.1	10.9	45.0	14.92	0.433	.....	1.57
U.S.S. New York	4-cyl. triple	*S.N.E.*, 1915	$\frac{39\times63\times83\times83}{48}$	9,311 at 110.5	225	8	28.1	10.9	32.3	13.54	.....	....	1.75
S.S. Ramapo	Quadruple	*S.N.E.*, 1919	$\frac{24\times35\times51\times75}{51}$	2,990 at 74.5	232	0	27.9	14.55	34.7	12.8	0.582	0.597	1.11§
S.S. Lehigh (tug)	Triple	*Mar. Eng.*, Feb., 1922	$\frac{16\frac{9}{16}\times25\frac{1}{4}\times43\frac{1}{4}}{30}$	633 at 87.7	188	0	23.5	.....	32.3	.....	.....	.....	2.29
S.S. Lehigh (tug)	Triple	*Mar. Eng.*, Feb., 1922	....	803 at 92.3	188	200–250	23.4	.....	38.6	....	.....	.....	1.90
S.S. C. M. Schwab	Triple	*Mar. Eng.*, May, 1924	$\frac{24.5\times41\times65}{42}$	2,400 at 90	183.5	43.6	23.7	.....	37.6	14.5	.....	0.62	1.59
S.S. Britannia	Quadruple	*S.B.S.R.*, Oct. 11, 1928	$\frac{29.5\times42.5\times61\times87}{54}$	5,026 at 85.5	209	41.5	26.8	...	36.2	...	.....	...	1.04§

* Replaced because of high maintenance cost. † 12.5 lb per ihp at two-thirds power. ‡ Steam jackets and reheaters. § Oil fuel.

after the cutoff of that cylinder is decided on. If clearance volumes are neglected

$$\text{Ratio of expansion} = \frac{\text{volume of L.P. cylinder}}{\text{volume of H.P. cylinder} \times \text{per cent cutoff}}$$

Corrections must be made for area of the piston rod and clearance volumes as shown in the following example. The area of the intermediate piston is obtained from the L.P. and H.P. by a ratio from practice, so that the work done in each cylinder is the same.

Example of Engine Design.—Determine the dimensions of a three-cylinder triple-expansion (double-acting) engine for the following conditions:

Ihp = 2,900
Rpm = 90
Boiler pressure = 210 lb gage
P_1 (at throttle) = 200 lb gage
Superheat at throttle = 60°F
Vacuum in condenser = 27 in. (1.47 lb per sq in.)
Estimated steam consumption = 13.6 lb per ihp per hr (engine efficiency 0.56)
Back pressure in engine cylinder (P_B) = 3.0 lb per sq in.
Ratio of expansions (R) assumed as 11.5 (Table VII)
P.S. (assumed) = 900 ft per min

$$\text{Theoretical referred mep} = P_1 \frac{(1 + \log_e R)}{R} - P_B$$
$$= (215 \times 0.30) - 3 = 61.5 \text{ lb per sq in.}$$

Card effect or diagram factor = 0.55 (Table VII)
Actual referred mep = 61.5 × 0.55 = 33.8 lb per sq in.

$$\text{Net area L.P. cylinder} = \frac{\text{ihp} \times 33{,}000}{\text{mrp} \times \text{P.S.}} = \frac{2{,}900 \times 33{,}000}{33.8 \times 900} = 3{,}140 \text{ sq in.}$$

Diameter L.P. cylinder = 63¼ (uncorrected for rod diameter)

$$\text{Stroke} = \frac{\text{P.S.} \times 12}{2 \times \text{rpm}} = \frac{900 \times 12}{2 \times 90} = 60 \text{ in.}$$

Diameter L.P. cylinder (corrected for rod diameter) = 63.5 in.
H.P. cutoff (assumed) = 0.70
H.P. clearance (assumed) = 15 per cent
L.P. clearance = 10 per cent
Piston-rod diameter (assumed) = 5 in.

$$\text{Ratio of expansion} = \frac{(2A_L - \text{P.R.})(1 + c_3)}{(2A_H - \text{P.R.})(H_C + c_1)}$$

where A_L = L.P. cylinder area.
A_H = H.P. cylinder area.
P.R. = piston-rod area.
c_3 = per cent clearance L.P. cylinder.
c_1 = per cent clearance H.P. cylinder.
H_c = H.P. cylinder cutoff.

$$11.5 = \frac{(6{,}320 - 20) \times 1.10}{(2A_H - \text{P.R.}) \times 0.85}$$

$$(2A_H - \text{P.R.}) = \frac{6{,}300 \times 1.10}{11.5 \times 0.85} = 704$$

$$2A_H = 724$$

$$A_H = 362$$

$$\text{Diameter of H.P. cylinder} = 21.5 \text{ in.}$$

$$\text{Ratio } \left(\frac{A_L}{A_H}\right) = \frac{(63.5)^2}{(21.5)^2} = 8.70$$

The diameter of the intermediate cylinder is often fixed by assuming cylinder ratios from good practice. To obtain equal power in all the cylinders, the following equation is used:

$$A_I = \frac{\sqrt{A_H \times A_L}}{C}$$

where C = a factor slightly greater than unity.
Hence

$$A_I = \frac{\sqrt{A_H \times A_L}}{1.12} = \frac{\sqrt{3{,}160 \times 362}}{1.12} = 956 \text{ sq in.}$$

$$\text{Diameter of intermediate cylinder} = 35 \text{ in.}$$

For an existing engine the combined horsepower output of all the cylinders is governed by the cutoff in the H.P. cylinder; the manner in which the work is distributed among the cylinders is controlled by the cutoffs in the I.P. and L.P. cylinders. Thus increasing the cutoff in either the I.P. or L.P. cylinder decreases the pressure in the receiver before that cylinder and hence increases the power developed in the preceding cylinder and decreases it in the cylinder in question.

The usual practice is to make the horsepower outputs of the three cylinders of a triple-expansion engine about equal by adjusting the cutoffs as discussed above. C. J. Jefferson[1] gives the following distribution of power in the cylinders for the best economy and quiet running of triple-expansion engines that have cylinder ratios in accordance with usual practice: H.P. = 28 per cent; I.P. = 32 per cent; L.P. = 40 per cent.

60. Value of Superheat.—One of the largest losses in a reciprocating steam engine is caused by the entering steam giving up

[1] "Instructions for Inspectors on the Preparation of Voyage Reports," *Marine Engineering & Shipping Age*, 1926, p. 160.

heat to the relatively cool cylinder walls that have just been in contact with exhaust steam at a lower temperature. This results in the lowering of the quality of the steam and a consequent condensation of steam within the cylinder, which is known as "initial condensation." This condensation continues up to the point of cutoff and results in the loss of available steam. During the exhaust stroke the cylinder walls are at a higher temperature than the steam, and part of the condensed steam on the cylinder walls is re-evaporated and carried over to the next cylinder or to the condenser.

If superheated steam is used, a certain amount of heat can pass from the steam to the cylinder walls before the enthalpy of the steam is lowered below the saturation point and condensation takes place. If the amount of superheat is high enough, the enthalpy of the steam will not be lowered to the saturated condition, and hence no condensation will take place; the loss in steam due to condensation will be eliminated and the actual steam consumption of the engine decreased. Superheat has the further advantage that condensation is eliminated in the main steam line and valves and valve passages, and steam friction is reduced. Leakage of steam by the valves and piston rings is less with superheated steam than with wet steam.

Superheat increases the initial enthlapy h_1 and consequently reduces the theoretical steam consumption w_t, as pointed out in Art. 15. However, the greatest gain caused by superheat is due to the increase in engine efficiency $\left[\frac{2,545}{w_a(h_1 - h_2)}\right]$ by the elimination of condensation, which brings the actual steam consumption w_a closer to the theoretical w_t. The gain from superheat is thus more actual than theoretical. The higher the degree of superheat carried at the boiler the further it will be carried through the engine. It is possible to have so high a degree of superheat at the throttle that condensation will be eliminated in the H.P. and I.P. cylinders and also in the L.P. cylinder up to the point of cutoff. Thus, with 160 lb gage initial pressure and 250 degrees of superheat, an adiabatic expansion will not cross the dry steam line until it is reduced to about 15 lb gage. A superheat as high as this, of course, will greatly reduce the steam consumption and prevent loss due to condensation and leakage in all three cylinders, valves, and receivers. With proper lagging, radiation losses need be no higher than with dry steam. However, superheat may be carried so high that the cost of producing it plus the greater first cost and maintenance charges will offset the reduction in steam consumption. Neither steam nor fuel consumption is the true criterion for measuring the gain. The gain should be measured in reduced fuel costs minus increased capital and maintenance charges.

When a reciprocating engine is operated with saturated steam or with only 50 or 60 degrees of superheat at the throttle, sufficient moisture is present on the cylinder walls to act as a lubricant for the pistons. However, when a superheat of much over 60 degrees is used, lubricating oil must be introduced into the cylinders to lubricate the cylinder walls. This requires special attention to the extraction of oil from the feed water—a problem not present when superheat is used with turbines or when saturated steam is used. Further, high superheat requires special attention to valves and ports and passage because of the erosive effect of superheated steam. The volume occupied by a pound of steam increases as the degree of superheat is increased; consequently a larger sized cylinder will have to be fitted when superheated steam is used. This may result in a slightly increased first cost of the engine.

A large number of existing ships fitted with Scotch boilers and reciprocating engines carry a superheat of around 60 degrees. This amount of superheat can be obtained with a waste-heat type of superheater (Art. 30) at practically no increase in fuel consumption. Published performance data for a number of ships using this amount of superheat indicate a reduction in fuel consumption of the order of 12 to 15 per cent over the same or similar installations using saturated steam. With this amount of superheat, internal cylinder lubrication will not be necessary, and condensation and leakage can be eliminated in the steam line and high pressure valve chest and part of the way through the high pressure cylinder. The last ships in the U.S. Navy that used reciprocating engines carried between 60 and 80 degrees of superheat and showed a marked decrease in fuel consumption over ships using saturated steam.

An interesting example of the performance of a small engine with and without superheat is the tug *S.S. Lehigh*, the data for which are given in Table VII. This boat was fitted with Scotch boilers and fire-tube superheaters. The ihp was increased 27 per cent and the fuel consumption reduced 17 per cent by the use of 200 degrees of superheat. John Neill[1] gives average engine efficiencies for modern triple-expansion engines using steam at pressures around 220 lb as follows:

Degrees of Superheat	Engine Efficiency
0	0.60
100	0.64
200	0.68
300	0.715

In some cases where saturated steam is used, steam jackets have been fitted to some or all of the cylinder heads and cylinder walls to assist in reducing the cylinder condensation.

Recently some low-powered British ships fitted with Scotch boilers and triple-expansion engines, operating with steam pressures around 225 lb, have used steam temperatures as high as 630°F (240 degrees superheat) at the throttle. These ships have shown a steam consumption in the neighborhood of 20 per cent lower than engines operating on saturated steam. These engines use a moderate amount of lubricating oil in the cylinders, but this is not a serious consideration where Scotch boilers are used.

At the present time (1942) at least seven British ships have been fitted with triple-expansion engines of around 2,000 ihp that have reheating receivers[2] between the H.P. and I.P. cylinders in which the steam is reheated to a high degree by means of superheated steam at a temperature of around 750 degrees. These ships have Scotch boilers equipped with combustion-chamber superheaters that produce steam at pressures around 225 lb and 760°F at the superheater outlet. The superheated steam on the way to the main throttle is sent through a coil reheater in the intermediate pressure receiver.[3] The heating

[1] "The Modern Trend in Marine Steam Practice," *Trans. Inst. Mar. Eng.*, vol. 45, p. 47.

[2] Harry Hunter, "The Reheated Reciprocating Marine Steam Engine," *Trans. Inst. Naval Arch.*, 1938. J. B. O. Sneeden, "Steam Propelling Units for Cargo Ships," *Trans. Inst. Eng. and Shipbuilders in Scotland*, vol. 83. See also *The Shipbuilder*, vol. 44, pp. 299 and 619, and vol. 45, p. 137.

[3] A receiver is the steam pipe connecting two cylinders, such as the H.P. and I.P., which is made large enough to hold the steam exhausted from one

surface in the reheater is around 300 sq ft. This arrangement has the advantage that a high degree of superheat can be used and dry steam is ensured in all the cylinders and valves without the necessity of having an extremely high temperature at the throttle with its accompanying difficulties. As shown in the trial data below, the throttle temperature was reduced to 632°F (240 degrees superheat).

In the ships built by the Northeastern Marine Engine Company the steam leaves the L.P. cylinder dry or slightly superheated. The engines are fitted with two cam-operated poppet valves of the Lentz type (Art. 64) on the H.P. and I.P. cylinders—one for admission and one for exhaust. This gives one-directional flow of steam in the valves, which helps reduce heat losses. The L.P. cylinder has a balanced double-ported slide valve. The engines show an engine efficiency of around 72 per cent, compared to a theoretical engine operating with initial steam conditions taken at entrance to reheater and a final pressure equal to the condenser vacuum. Mr. Hunter, in the paper referred to above, gives the internal cylinder oil consumption for an 1,800 ihp engine as 0.8 imperial pint per day.

The results of a 4-hr trial of the engine of the *S.S. Conakrin* (1937), which is of the type described above, were as follows:

Ihp = 2,210
Shp = 2,096
Rpm = 84.5
Mrp = 34.7 lb per sq in.
Boiler pressure = 240 lb abs
Pressure at throttle = 225 lb abs
Vacuum = 26 in.
Temperatures of steam, °F:
- Superheater outlet = 762
- Reheater inlet = 747
- Engine stop valve = 632
- H.P. exhaust = 466
- I.P. chest = 597
- L.P. chest = 334

Boiler efficiency, per cent = 80.75
Steam consumption of main engine, including bleeding, = 9.37 lb per ihp per hr

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cylinder until the valve of the second cylinder opens to admit the steam. Because of the arrangement of cranks, steam exhausted from one cylinder cannot pass directly into the next cylinder of a multiple-expansion engine.

Oil per ihp per hr = 0.80 lb (all purposes)
Mechanical efficiency = 94.8 per cent
Engine efficiency (indicated), including bleed steam = 75 per cent

61. Vacuum.—From the expression for theoretical steam consumption

$$w_t = \frac{2{,}545}{h_1 - h_2}$$

it is evident that a high vacuum (a low back pressure) will reduce the value of h_2 and result in a lower value of w_t. A vacuum cannot be produced, however, without the expenditure of power, and if the vacuum is carried too high the cost of producing it may offset the gain in economy.

A study of the pressures and volumes of steam at low pressures will show that the volume increases rapidly as the pressure decreases. Thus the volume of 1 lb of dry steam at 28-in. vacuum is practically double what it is at 26 in., and the volume at 29 in. is nearly double that at 28 in. These large increases in steam volume can be easily taken care of in a turbine; but with a reciprocating engine they call for a large increase in the size of the L.P. cylinder and valves. This not only increases the size and first cost of the engine but also increases the area of the cylinder walls and offers greater opportunity for condensation. Further, the increased size of the cylinder, piston, and valves greatly increases the friction of the mechanism. The vacuum can be carried so high that the increase in friction in the L.P. cylinder and the cost of producing the vacuum offset the gain in economy due to the higher vacuum.

The foregoing facts, together with the increased power and size required for the air and circulating pumps and greater danger of air leakage, show at once that a high vacuum is not desirable with a reciprocating engine.

High vacua are often advocated for reciprocating engines, and trials are quoted to show the reduction in steam consumption. First costs and cost of producing the higher vacuum are seldom mentioned, however. Probably 27 in. or possibly 27½ in. as a maximum can be carried to advantage in the condenser. This will give 23 to 25 in. in the L.P. cylinder.

62. Influence of Cutoff on Engine Performance.—When the length of the cutoff in the H.P. cylinder is reduced, the ratio of expansion is increased, the pressure at release in the L.P. cylinder is reduced, and each pound of steam does a greater amount of work. Thus in an ideal engine, the water rate would decrease as the cutoff was shortened. However, in an actual engine operating on saturated steam, initial condensation takes place in the steam passages and clearance volume; hence the shorter the cutoff the greater is the ratio of clearance volume to the volume of steam admitted to the cylinder per stroke, and consequently the greater is the percentage of condensation per pound of steam used. This tends to increase the water rate. Thus there are two conflicting tendencies as the cutoff is shortened, which result in an optimum cutoff to obtain the lowest steam consumption per ihp. This optimum cutoff for minimum water rate varies between 20 and 30 per cent for single-cylinder engines operating on saturated steam and is probably somewhat higher for multiple-expansion engines. The optimum percentage is lower when superheated steam is used.

When the cutoff in the H.P. cylinder is shortened, the ratio of expansion is increased and the mrp is reduced, and consequently the size of the L.P. cylinder is increased for a given ihp. This results in a larger, heavier, and more expensive engine for a given horsepower, and hence there is an economical limit beyond which it is not feasible to reduce the cutoff, despite the gain in economy.

The triple-expansion engines formerly used on battleships had long cutoffs on the H.P. cylinder to reduce the size and weight of the engine for a given horsepower. In the *U.S.S. Delaware* the high pressure cutoff was 86 per cent of the stroke at full power, and the ratio of expansion was 8.71. Merchant-ship engines have cutoffs varying between 50 and 70 per cent and expansions of around 9 to 12 for triple-expansion engines and 12 to 14 for quadruple-expansion engines. Hence for the same steam conditions, merchant-ship engines would be expected to have lower water rates and higher engine efficiencies than naval vessels.

Two methods are available for reducing the power of an existing reciprocating engine for low-speed operation: (1) reducing the steam pressure in the high pressure steam chest by throttling (*i.e.*, partly closing the main throttle valve) and (2) shortening the high pressure cutoff. The former method reduces the availability of the enthalpy in the steam (the opposite effect obtained by increasing pressure) and results in an increased steam consumption per ihp at reduced power. The latter method increases the ratio of expansion and decreases the water rate somewhat for powers down to 50 per cent of full power.

Triple-expansion engines used for ship propulsion are fitted with two eccentrics at each cylinder, one for going ahead and the other for going astern (Figs. 33 and 34). These eccentrics operate the valve stems through a link, and the engine runs ahead or astern, depending on which set of eccentrics is in position under the valve stems (Fig. 33). In large engines the eccentrics are shifted by power by one of two methods: (1) a steam cylinder (Fig. 33); (2) a worm-and-wheel ("all-round") gear. When the worm-and-wheel gear is used, the linkage can be held at intermediate points so that the valve is actuated by the combined motions of the two eccentrics. This allows the cutoffs to be controlled (within limits) at the operating station. Although some ships are fitted with a steam reversing cylinder that allows the gear to be stopped at intermediate points, many of these reversing cylinders allow the gear to be placed only in the extreme positions, *i.e.*, full ahead and astern positions, and hence no cutoff control is possible by means of the reversing engine. However, this method of shifting the cutoff, whether by worm-and-wheel gear or steam cylinder, changes the cutoffs on all the cylinders and not on the H.P. cylinder alone, as is desired. Practically all marine engines have an adjusting-screw device connected with the link gear whereby the cutoff of each individual cylinder can be varied independently of the others. However, this screw is not conveniently located and is usually incapable of rapid operation when it is desired to change the cutoff for a short period of time. The result is that most ships fitted with reciprocating engines reduce the power by throttling.

If a ship is to operate for comparatively long periods at reduced speeds, good economy at low powers will be desirable, and hence such ships should have a means of shortening the cutoff for low-speed operation. The cutoff at full power should be sufficiently large to allow for this. Some vessels, on the other hand, such as tugs, wish to obtain full power when the rpm falls off through increased resistance caused by tows or bad weather. The loss in power $\left(\text{ihp} = \frac{PLAN}{33{,}000}\right)$ due to the falling off in rpm of a given engine must be made up by an increase in mep. An increase in mep (P in the preceding formula) can be brought about by increasing the area of the indicator card by lengthening the cutoff. Hence in vessels where an increase in power is desired, as discussed above, the cutoff at full power at normal rpm should be somewhat lower than might be desirable from the point of view of first cost.

Too short a cutoff cannot be used with triple-expansion engines, for this would lead to difficulty in maneuvering, since the ports would be open for only a small angle of crank movement. Unless the engine was stopped with the high pressure crank within this angle, the engine would refuse to start, although not on "dead center." However, this condition would not occur in the two types of engines described in the next two articles, which have in one case two high pressure cylinders at 90 deg and in the other case a number of single-expansion cylinders operating in parallel.

A number of makes of reciprocating engines of low and medium power are now available that are fitted with cutoff controls that can be rapidly adjusted at the control platform as occasion demands. Two of these engines are discussed in Arts. 63 and 64.

63. The Unaflow Engine.—The unaflow engine has been widely used for stationary work in both Europe and the United States, and the Skinner Engine Company of Erie, Pa., has developed an engine of this type for marine propulsion. In the unaflow engine the steam makes a complete expansion in a single cylinder and because of the unaflow principle, which greatly reduces initial condensation, an engine of this type is generally considered to show the same economy as a triple-expansion engine working between the same limits. The steam enters the cylinders through valves in the end and leaves at the end of the stroke through a ring of ports cut into the cylinder wall (similar to the exhaust ports of a two-cycle Diesel engine), which are uncovered by the piston. The steam thus has a one-directional flow, and the relatively cool exhaust steam does not flow over the cylinder walls and through the valve passages, thus cooling them and causing loss of heat by the incoming high pressure steam, as in a triple-expansion engine. The marine engines are double-acting and condensing, and the exhaust ports that serve both the head end and crank end of the cylinder are cut in the center of the cylinder wall. Figure 36[1] shows a diagram of a double-acting condensing unaflow engine. It will be noticed that the compression begins as soon as the exhaust ports are covered, yet for a condensing engine where the back pressure is low the clearance volume is very small. The Skinner engine uses cam-operated poppet valves (employing the same principle as in the Lentz engine, Art. 64) for the admission of steam to the cylinder. This type of valve allows the use of high superheat. Unaflow engines are constructed with a number of cylinders of the same size (usually from three to six) and obtain their power by parallel flow of steam through the cylinders instead of serial flow used in multiple-expansion engines. The Skinner engine uses a very small cutoff at full power, around 10 per cent of the stroke, and reduces the power by shortening the cutoff. As would be expected, the steam consumption per ihp per hour falls off slightly as the power is reduced. A cut of the Skinner engine is shown in Fig. 37.

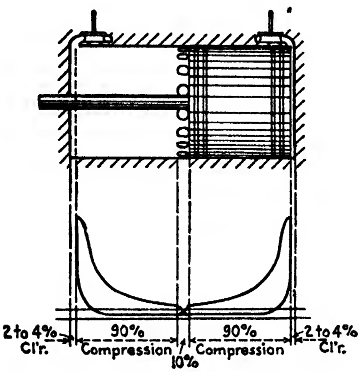


FIG. 36.—Diagram of cylinder of unaflow engine.

The latest marine installation (1941) of this make of engine is on the Great Lakes car ferry *S.S. City of Midland* (Père Marquette Railroad). The data for this ship are given below:

Engines: two—5-cylinder; 25-in. bore × 30-in. stroke
Normal power = 3,000 shp (each)
Throttle pressure = 325 lb
Throttle temperature = 640°F (210° superheat)
Vacuum = 26 in.
Rpm = 120
Mep = 73
Mechanical efficiency = 94 per cent
w_a full power = 10.2 lb per ihp per hr (approximate)
w_a half power = 9.6 lb per ihp per hr (approximate)
Mep at half power = 45 (rpm = 95)

[1] Taken from a paper by H. G. Mueller, chief engineer of the Skinner Engine Company, "The Development of the Marine Unaflow Steam Engine," *Marine Engineering & Shipping Age*, June, 1934.

Cutoff at full power = 8–10 per cent
Engine efficiency (on ihp) = 64 per cent (approximate)

This ship is fitted with Foster Wheeler water-tube boilers, which are equipped with Hoffman stokers for burning coal.

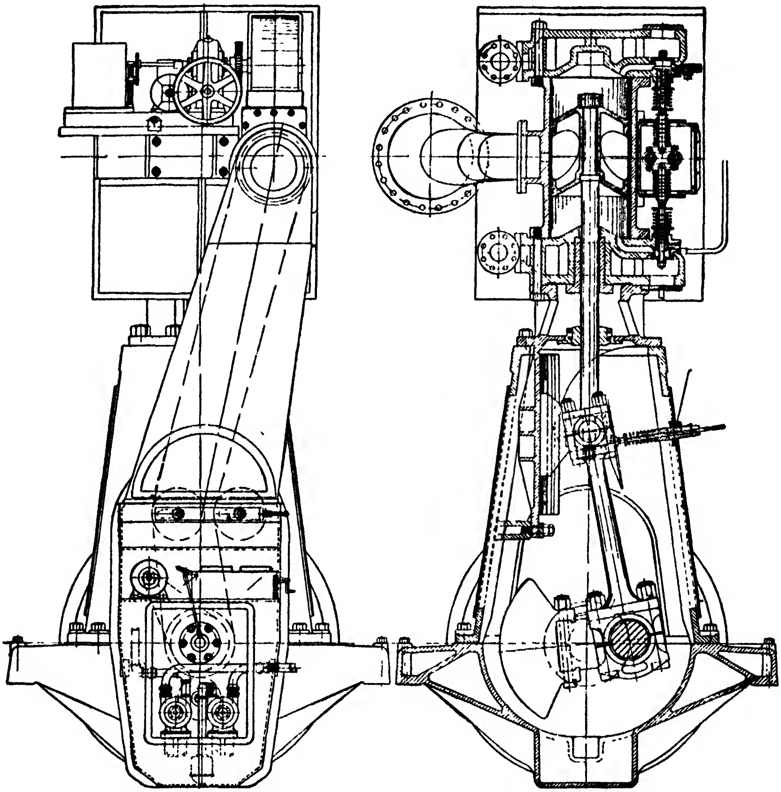


FIG. 37.—End view and cross section of Skinner unaflow engine.

Since these engines use superheated steam, lubricating oil must be used in the engine cylinders. Special oil separators are fitted, and no difficulties are anticipated from oil getting into the boilers.

As pointed out, unaflow engines are usually not compounded, but a few compound unaflow engines have been built. However, the engine mentioned has a ratio of expansion around 9.0, which is remarkably high for a single-expansion engine.

64. The Lentz Marine Engine.—Several engine builders in Europe, especially in Germany, have brought out reciprocating engines of small and moderate powers that have special features incorporated into their design and that operate under steam conditions that have reduced water rates appreciably over those of triple-expansion engines of 20 years ago. The Lentz Standard Marine engine is an engine of this class.

The Lentz engine[1] operates with steam pressures around 200 lb, steam temperatures of 500 to 650°F, and vacua around 27 in. It is a four-cylinder double-compound engine and works on the Woolf principle. The cranks of each pair of cylinders (H.P. and L.P.) are at 180 deg, and the cranks of the two H.P. cylinders making up the double-compound engine are at 90 deg to each other. Since the low pressure crank is at 180 deg to the high pressure crank, the steam exhausted from the head end of the H.P. cylinder passes directly into the head end of the L.P. cylinder (Woolf principle). No receiver is needed between the cylinders, as required in a triple-expansion engine, where the cranks are at 120 deg. By this arrangement there is no cutoff in the L.P. cylinder.

The Lentz engine is fitted with double-beat poppet valves (Fig. 38),[2] which are cam-operated. This type of valve is especially suitable for use with high steam temperatures, and because of the low mechanical friction a light valve gear can be used. The valves are arranged so that the steam always flows through the valves in one direction, and thus heat losses that occur with ordinary valve arrangements are avoided. As shown by the plan view of the top of the engine in Fig. 39,[2] one valve is used for admitting steam to the H.P. cylinder, a second valve serves both as an exhaust valve for the H.P. cylinder and as an admission valve to L.P. cylinder, and a third valve exhausts the steam from the L.P. cylinder to the condenser.

At full power, the Lentz engine employs a cutoff on the H.P. cylinder varying from 30 to 42 per cent. The ratio of expansions (neglecting clearances) varies between 11 and 16.5, which is as high as used with ordinary marine triple- and quadruple-expansion engines, where long cutoffs are the rule. The cutoff is

[1] E. C. Poultney, "Poppet Valves for Marine Steam Reciprocating Engines," *The Marine Engineer & Motorship Builder*, vol. 48, pp. 19–22. "The Lentz Marine Steam Engine," *The Shipbuilder & Marine Engine Builder*, vol. 36, p. 808.

[2] From *The Marine Engineer & Motorship Builder*, January, 1925.

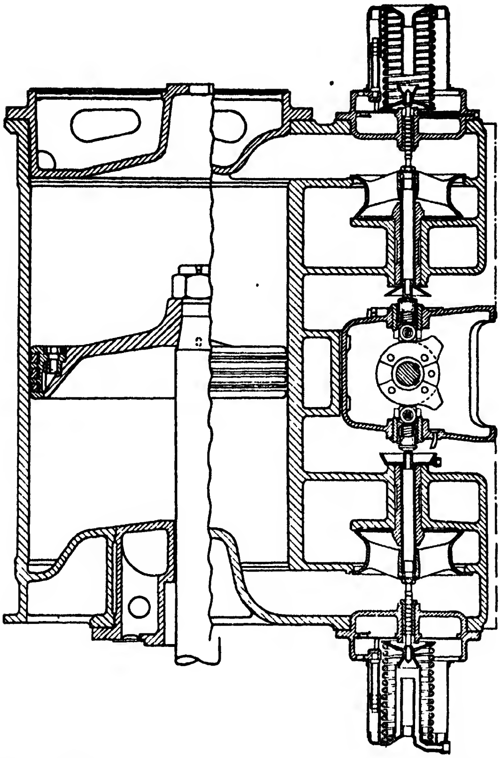


FIG. 38.—Section through L.P. exhaust valve of Lentz engine.

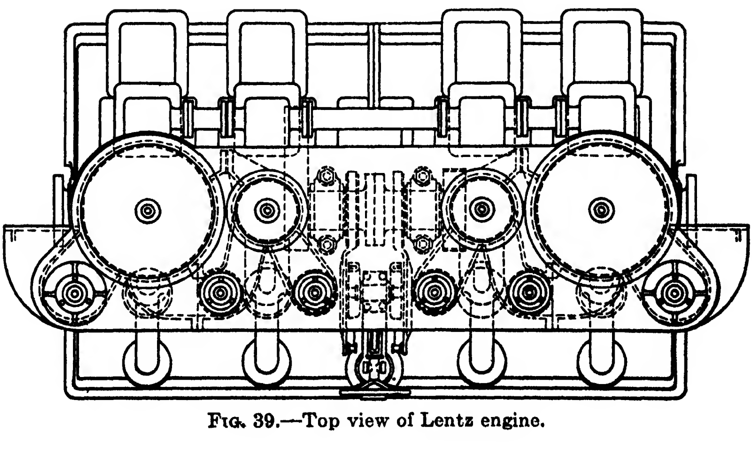


FIG. 39.—Top view of Lentz engine.

adjustable by hand gear at the operating station, and thus good water rates are obtained at reduced powers. When operated with steam pressures of 220 lb, 220°F of superheat, and 27 in. of vacuum, these engines show engine efficiencies of around 65 per cent. The low water rates obtained with these compound engines are due to (1) the moderately high superheat employed; (2) the elimination of receivers; (3) the large ratio of expansions due to the short cutoff; and (4) unaflow principle in the valves.

An interesting paper[1] by W. J. Muller, chief engineer of the Koninklyke Paketvaart Maatschappy (K.P.M.), gives trial data on one of these engines and notes on their use in K.P.M. steamers. An engine of 4,300 ihp $\left(2 \times \frac{560 \text{ mm} \times 1{,}200 \text{ mm}}{1{,}200 \text{ mm}} \text{ at } 93.6 \text{ rpm}\right)$ operating with 215 lb of steam pressure, 509°F steam temperature, and 27.2-in. vacuum showed a water rate of 11.03 lb per ihp-hr. This gives an engine efficiency of 0.665. The cutoff was 34.6 per cent. This engine was supplied with steam from a water-tube boiler. An oil extractor was fitted in the L.P. cylinder exhaust, which, Mr. Muller states, cut down the oil in the boiler feed to 5 mg per kg. The back pressure in the L.P. cylinder was 3.13 lb per sq in., with 1.47 lb per sq in. in the condenser.

In 1928, 24 of these engines were in operation in the small steamers of the K.P.M. East Indian fleet. The boilers of the ships were of the water-tube type and burned coal with mechanical stokers.

The engines of a number of existing ships have been partly converted to the Lentz type by replacing the H.P. cylinder with a new cylinder of slightly larger diameter, fitted with Lentz poppet valves. Usually superheated steam has been employed in the new engines.[2]

65. Combination Machinery.—During the period around 1909–1914, just before the geared turbine was first used for ship propulsion, a number of ships were built that were fitted with a type of machinery known as "combination machinery." This

[1] "Notes on the Lentz Standard Marine Engine as Fitted to Ships of the K.P.M.," *Trans. Inst. Mar. Eng.*, vol. 41, p. 357.

[2] See the article in *Mar. Eng. & Shipping Rev.*, 1937, p. 266, on the "Lentzification of Two Lake Bulk Carriers."

consisted of a three-screw arrangement with two triple-expansion engines on the wing shafts exhausting into a low-speed low pressure turbine directly connected to the center shaft. The steam left the reciprocating engines at a pressure around 10 lb abs, and about one-third of the power was developed on each shaft. It was an excellent arrangement for improving the economy, for it allowed the vacuum to be increased from around 26 in., that used with reciprocating engines, to 28.5 in. The engine efficiency for H.P. and I.P. cylinders is high when a moderate amount of superheat is employed. The L.P. cylinder of a triple-expansion engine, on the other hand, shows a low efficiency. Turbines do not show such high efficiency in the high pressure stages as they do in the low pressure stages, and a turbine can be easily bladed for a vacuum of 28.5 in. Thus the most efficient part of the reciprocating-engine cycle was combined with an efficient low pressure turbine. It must be remembered that this type of machinery was introduced as a competitor to the reciprocating engine and the direct-connected turbine, the geared turbine not having been introduced at this time.[1]

The first installation of combination machinery was on the *S.S. Otaki* in 1908. Among other ships that had combination machinery were the *S.S. Laurentic*, the *S.S. Olympic*, the *S.S. Minnekahda*, and the *S.S. Rochambeau*. The *S.S. Otaki* and the *S.S. Laurentic* each had sister ships operating on the same trade routes that were fitted with straight reciprocating engines on twin screws. Comparisons of the service performance of these sister ships showed a saving in fuel per ihp-hour of the order of 15 per cent. This type of machinery would probably have been widely adopted except for the fact that the geared turbine was introduced a few years after the first installations of combination machinery.

One of the disadvantages of this type of machinery was that the efficiency of the center screw was considerably lower than that of the wing screws. In order to obtain a turbine of reasonably small size, the direct-connected turbine on the center shaft was run at higher rpm than the wing shafts. On the *Otaki* the wing shafts turned at around 100 rpm and the center shaft at around 200 rpm. The result of this was a propeller of smaller diameter on the center shaft and hence a high wake fraction and

[1] See the footnote on p. 110.

a low speed of advance. This low speed of advance and the high rpm resulted in a reduced propeller efficiency.

66. Exhaust Turbines.—During the period 1925–1930 several closely related new types of machinery for ship propulsion made their appearance, in which the principle first introduced with combination machinery was employed. These new types all employed an exhaust turbine to receive the steam exhausted

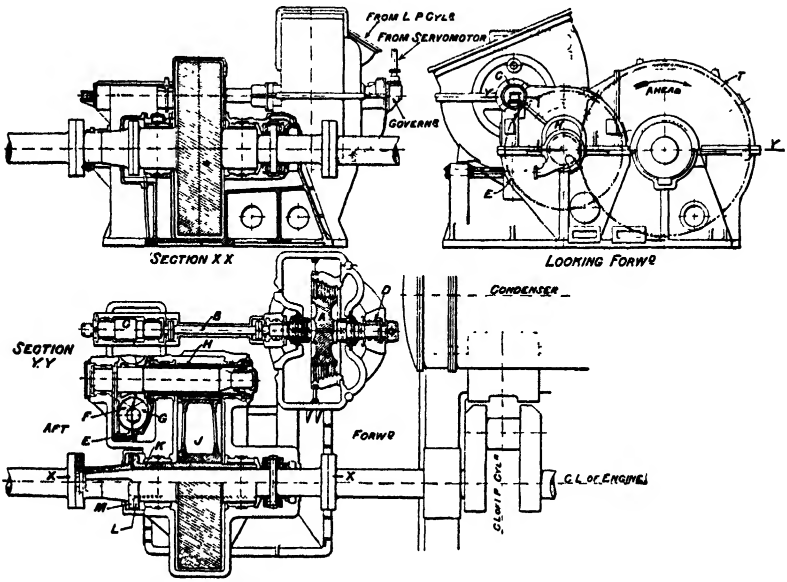


Fig. 40.—Arrangement of Bauer-Wach exhaust turbine installation on *S.S. Britannia.*

from the L.P. cylinder of a triple-expansion engine, but instead of using the three-shaft arrangement of the former type of machinery, all the new types were single-screw installations. These several related types will now be described briefly.

1. *The Bauer-Wach System.*[1]—The Bauer-Wach system consists of a single triple-expansion engine exhausting to a small high-speed exhaust turbine, which is geared on to the line shaft

[1] Gustav Bauer, "Recent Developments of the Exhaust Steam Turbine," *Trans. Inst. Eng. and Shipbuilders in Scotland*, vol. 71. R. J. Butler, "A Note on the Development and Operating Results of Bauer-Wach Installations," *Trans. Inst. Eng. and Shipbuilders in Scotland*, vol. 76.

of the reciprocating engine. A Vulcan hydraulic clutch is fitted between the exhaust turbine and the gearing, which, because of the small slip in this clutch, prohibits the unequal torque of the reciprocating engine from being transmitted to the turbine shaft. The turbine runs at about 3,000 rpm and is geared down through a set of double reduction gears to the slow speed (75 to 90 rpm) of the main engine shaft.

The arrangement of the Bauer-Wach exhaust turbine, coupling, and gearing, as fitted on the *S.S. Britannia,* is shown in Fig. 40.[1]

In Fig. 40, A is the exhaust turbine, C the high-speed pinion, E the first gear wheel, F the driving part of the Vulcan clutch, and G the driven part, H the low-speed pinion, J the main gear wheel, and X the line shafting.

The space taken up by the exhaust turbine and gearing is not large, and exhaust turbines can be added to existing vessels where an increase in speed or reduction in fuel consumption is desired.

The introduction of the Bauer-Wach exhaust turbine results in an increase in power of about 25 per cent on the same hourly fuel consumption or a reduction of about 20 per cent in the fuel consumption for the same power output. Naturally, a great many existing ships have had this exhaust turbine added to improve their performance, and a large number of new ships have employed this type of machinery. In 1938 it was reported that about 500 ships with a total ihp of around 1,500,000 were fitted with the Bauer-Wach system.

The exhaust turbine runs in only one direction and is not used in maneuvering or going astern. By letting the oil out of the Vulcan clutch the turbine can be readily disconnected from line shafting. When the main engine is reversed, a change-over valve sends the steam from the L.P. cylinder to the condenser instead of to the exhaust turbine.

An interesting example of an existing cargo ship that had the Bauer-Wach exhaust turbine added is the *Britannia,* which was built in 1926 and converted in 1928, when only two years old. The *Britannia* has a quadruple-expansion engine (see Table VII).

Data before and after the conversion are given in the following table.[1]

[1] From *Shipbuilding and Shipping Record,* Oct. 11, 1928.

	Before conversion	After conversion
Boiler pressure, lb	207	211
Superheat, °F	41 5	35
Vacuum, in. of mercury	26 83	28.04
Ihp	5026	3707 + 1315*
Rpm	85.5	85.1
Cutoffs	0.68-0.60-60-0.60	0.59-0.60-0.56-0.56
Referred mep	36.2	26.9
Oil per ihp per hr	1.04	0.832

* Shp of turbine changed to equivalent ihp.

The reduction in fuel on the *Britannia* with practically the same ihp output was exactly 20 per cent.

Two sister ships, the *S.S. Basil*, with triple-expansion engine, and the *S.S. Boniface*, with Bauer-Wach, showed fuel consumptions in service of 1.40 and 1.04 lb of coal per ihp per hr, respectively.[1]

Although the increase in power by adding a Bauer-Wach exhaust turbine is of the order of 20 per cent, the maximum torque on the line shafting is less than for the reciprocating engine alone because of the constant torque of the turbine transmitted through the double reduction gears. This is clearly brought out in Fig. 41,[2] which shows the torque diagrams taken on the *S.S. Cap Norte* before and after conversion. The *Cap Norte* had a triple-expansion engine 26.8 × 43.3 × 70.9 in. with stroke of 51.2 in. and ran at 80 rpm.

2. *Götaverken System.*[3]—The Götaverken system employs an exhaust turbine receiving steam from the reciprocating engine, as in the Bauer-Wach system. The turbine is used to drive a steam compressor, which takes the steam exhausted from the H.P. cylinder, compresses it, and delivers it at a higher pressure and temperature to the I.P. cylinder. The steam leaving the H.P. cylinder is sent through a separator to remove the moisture before it enters the compressor. The efficiency of the compressor is about 98 per cent because of the fact that the internal friction is not lost but raises the temperature of the steam, sending it to

[1] *Trans. Inst. Naval Arch.*, 1929, p. 129.

[2] From a paper by Dr. Gustav Bauer, *Trans. Inst. Eng. and Shipbuilders in Scotland*, vol. 71.

[3] *The Shipbuilder and Marine Engine Builder*, vol. 39, p. 364. *Shipbuilding and Shipping Record*, vol. 47, p. 163.

the I.P. cylinder in a superheated condition. About 65 per cent of the turbine output is used in compressing the steam and the remainder in raising its temperature. In the trials of the *S.S. Harpsa*,[1] a new ship fitted with the Götaverken system, the steam conditions at H.P. cylinder exhaust were 52 lb abs and 27.5

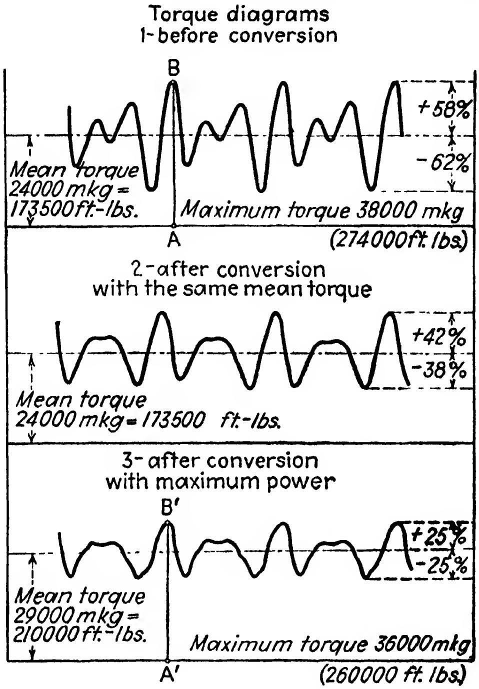


FIG. 41.—Torque diagrams taken on *S.S. Cap Norte* before and after fitting Bauer-Wach turbine.

degrees superheat; at the entrance to the I.P. cylinder the pressure was 78 lb abs and the superheat 49°F.

A diagrammatic arrangement of the Götaverken system is shown in Fig. 42.[2]

The exhaust turbine in this system is mechanically independent of the main engine. The exhaust-turbine compressor unit is of small size; it is usually mounted on top of the condenser in new

[1] *Shipbuilding and Shipping Record*, vol. 47, p. 165. See also *Shipbuilding and Shipping Record*, vol. 44, p. 235, and *The Shipbuilder and Marine Engine Builder*, vol. 40, p. 423.

[2] From a paper by J. B. O. Sneeden, *Trans. Inst. Mech. Eng.*, 1936; reproduced from *Shipbuilding and Shipping Record*, vol. 47, p. 163.

installations and conversion jobs, and the turbine exhausts directly into the condenser.

The gain in economy with the Götaverken system appears to be slightly less than with the Bauer-Wach system, but the weight and cost of the installation are less, and conversions are more readily carried out. If in a conversion job the advantage of this

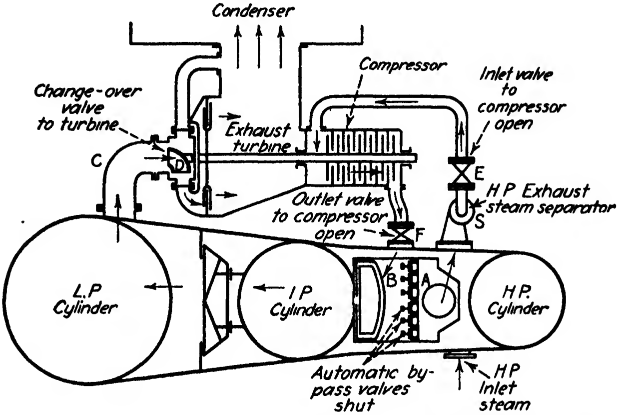


FIG. 42.—Diagrammatic arrangement of Rowan-Götaverken system.

system is taken as increased power, a new line shaft and crankshaft may be necessary as the increased power out of the I.P. and L.P. cylinders increases the torque in the shafting. However, if the ihp of the engine is kept the same as before conversion, the reduction in hourly fuel consumption will be around 17 to 18 per cent.

3. *Exhaust Turbo-electric Drive.*—In this system the exhaust turbine drives a dc-generator, and the electric power produced is used to drive a motor placed on the main shaft, aft of the triple-expansion engine. An installation of this system on the *S.S. City of Hongkong*[1] showed the following results on trial:

	Engine alone	Engine and motor
Vacuum, in.	27	28.7
Rpm.	75	84.5
Equivalent ihp	3190	4880
Water rate, lb per ihp-hr	12.1	9.1

[1] *Shipbuilding and Shipping Record*, Nov. 14, 1929. See also *The Shipbuilder and Marine Engine Builder*, vol. 37, p. 506, for a description of the installation in the *S.S. City of Barcelona.*

CHAPTER VI

THE GEARED TURBINE

67. General.—The elementary principle of the steam turbine consists in converting the potential heat energy of the steam into kinetic energy by expanding it through nozzles and directing the steam that leaves the nozzles at high velocity against a row of blades fastened to the turbine rotor. This blading is shaped to develop thrust from the jet action of the steam and thereby turn the rotor and produce power. The steam leaves the nozzles at high velocity, and to obtain good efficiency the turbine rotor must operate at a fairly high peripheral speed and hence at high revolutions.

In the last chapter we saw that the reciprocating steam engine was operated by steam pressure, and no attempt was made to take advantage of the kinetic energy of moving steam; in the steam turbine, on the other hand, the rotor revolves because of the impulse of the high-velocity steam impinging on the blading.

When high initial pressures, high superheats, and high vacua are used with turbines in order to obtain a good thermal efficiency, the velocity of the steam would be extremely high if it were expanded from the initial pressure to the back pressure in one set of nozzles. If a single row of moving blades were used under these conditions, the necessary peripheral speed of the blades and hence the rpm of the rotor would be prohibitive. On the other hand, if a low peripheral speed of the blading were used in conjunction with an extremely high steam velocity, the exit velocity of the steam leaving the blading would be high, and a large part of the kinetic energy of the steam would be lost in the exhaust. In order to reduce the turbine peripheral speed and still obtain high efficiency, various methods of compounding are resorted to, such as passing the steam through a series of moving, fixed, and moving blade rows or dropping the pressure in a series of steps by passing the steam successively through nozzles and moving blading until the pressure has been reduced to that of the condenser. Marine compound turbines usually have two or more separate casings in series, the steam being passed from one turbine to the other as it is in the cylinders of multi-cylinder compound reciprocating steam engines. The various methods of compounding turbines are discussed in the following articles.

When the turbine was first used for ship propulsion, it was connected directly to the propeller shaft, and the turbine and propeller shaft turned at the same rpm. This was not an economical arrangement, because a propeller usually requires low revolutions for good efficiency, and, as just pointed out, the turbine needs high blade velocity for good economy. In such installations, turbine rotors of large diameter were made to permit low revolutions, and many stages were employed. Such turbines were large[1] and heavy, and the efficiency was not high. The revolutions used were usually a compromise and were too low for good turbine performance and somewhat too high for good propeller efficiency. The turbines installed on the first *S.S. Mauretania* (1907), the *S.S. Aquitania*[2] (1913), the *S.S. Leviathan,* and many of the early turbine-powered naval vessels were of this type.

In order to allow the turbines to run at high rpm and thus show good efficiency with small, light, and short rotors and in order to have the propeller turn at the low rpm necessary for good efficiency, the geared turbine and turbo-electric drive were introduced for ship propulsion around 1911.[3] In a geared-turbine installation, either single or double reduction gears that have reduction ratios of from 1:6 up to as high as 1:85 are interposed between the turbines and the propeller shaft. With geared-turbine installations, the turbine or turbines driving each propeller shaft comprise a complete propulsion unit. One, two, three, or even four turbines make up such a unit, with the steam flowing through the turbines in series.

[1] The low pressure drum of the first *Mauretania* (1907) was 140 in. in diameter, and the blading of the last stage was 22 in. high. The shafts turned at 190 rpm, and the throttle pressure was 180 lb per sq in.

[2] In the *Aquitania*, the turbines were arranged for triple expansion, the H.P. casing being on one wing shaft and the I.P. casing on the other wing shaft; L.P. turbines were fitted on the two center shafts.

[3] See the footnote to Art. 56.

68. Steam Nozzles.—The purpose of turbine steam nozzles is to form efficient steam jets at the correct angle to the turbine blading. Steam is expanded in the nozzles from the initial pressure to the stage pressure, and the maximum velocity should be attained from the available energy in the steam with minimum frictional and eddy-making losses. For this reason the turbine nozzle (Fig. 43*b*) is provided with a nicely rounded form from the entrance or bowl at point 1 to the throat section at point 2,

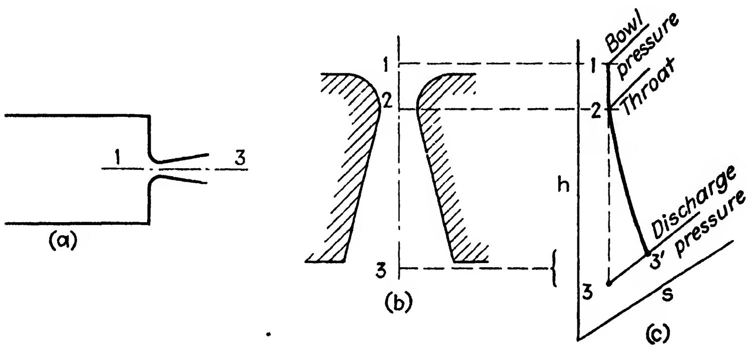


FIG. 43.

and the part of the nozzle from point 2 to the mouth (point 3) may be of a diverging form, as shown in Fig. 43*b* for reasons explained below.

If steam is flowing from a pipe of large diameter through the small nozzle, as shown in Fig. 43*a*, the velocity of approach need not be taken into consideration. If friction is neglected, the enthalpy of the steam at point 1 will be equal to the sum of the enthalpy and the kinetic energy at point 3. The energy equation between points 1 and 3 is

$$\frac{V_3^2}{2g} + 778h_3 = 778h_1$$

and

$$V_3 = 223.7\sqrt{h_1 - h_3}$$

Let us consider that a nozzle of the shape shown in (*a*) and (*b*), Fig. 43, is discharging saturated steam into a vessel or duct that has an absolute pressure of P_3 and that this pressure can be varied. If we start with a relatively high discharge pressure P_3 and gradually lower it, the throat pressure P_2 will decrease until it reaches a critical value of $0.58P_1$. Any further lowering of the pressure P_3 will not decrease P_2 or increase the nozzle discharge rate, *i.e.*, the velocity at point 2. The value $0.58\,P_1$ is for saturated steam; with the steam initially superheated, $P_2 = 0.55P_1$.

Thus, if the nozzle exit pressure or stage pressure P_3 is < 0.58 or $0.55P_1$, the throat pressure P_2 will equal 0.58 or $0.55P_1$. Hence, under these conditions, the throat velocity is

$$V_2 = 223.7\sqrt{h_1 - h_2}$$

where h_2 is the enthalpy after an isentropic expansion from P_1 to 0.58 (or $0.55)P_1$. The values of P given above are absolute pressures.

If the steam velocity beyond the throat of a nozzle becomes greater than about 1,500 ft per sec, a pressure wave started at the mouth of the nozzle cannot reach the nozzle throat. Thus the throat pressure remains at the critical value, even though the pressure at the nozzle mouth is much lower. As Osburne Reynolds aptly explained it, news of further lowering of the pressure cannot be telegraphed back to the throat.

In actual steam nozzles a friction loss occurs because of the steam flow along the walls of the nozzle, and hence the change between point 1 and point 3 is not an isentropic one, and the enthalpy at the mouth is h_3', as shown in Fig. 43c.

$$h_3' = h_1 - [(h_1 - h_3)e_n]$$
$$e_n = \frac{h_1 - h_3'}{h_1 - h_3}$$

where e_n = nozzle efficiency.
h_3 = enthalpy at point 3 after an isentropic expansion from point 1.
h_3' = enthalpy at point 3 in an actual nozzle.

The actual nozzle exit velocity is

$$V_3 = 223.7\sqrt{h_1 - h_3'} = 223.7\sqrt{(h_1 - h_3)e_n}$$

The efficiencies of nozzles reported from various tests show some variation; tests made by Warren and Keenan indicate a nozzle efficiency of about 97 per cent when the steam is initially superheated and the steam velocity is below that of sound.

For preliminary design of turbines, nozzle efficiencies of 94 to 95 per cent are often used for superheated steam and about 90 per cent for saturated steam. The nozzle efficiency decreases as the jet velocity is increased.

In calculating the throat velocity, a nozzle efficiency of 100 per cent can be used when the length between the entrance and the throat is comparatively short. The areas of the cross section of the nozzle at the throat and exit are made to conform to the changes in the volume and velocity of the steam. Convergent-divergent nozzles of circular cross section are usually reamed and have a straight taper from throat to exit for the sake of simplicity in construction. The proper cross-sectional areas for a nozzle can be obtained from the following expression:

$$A = \frac{Q}{V} = \frac{144wxv}{V} \text{ sq in.}$$

where Q = quantity of steam flowing, in cubic feet per second.
w = flow, in pounds per second.
v = specific volume of the steam, in cubic feet per pound.
x = quality.
V = steam velocity, in feet per second.

If the point h_3' in Fig. 43c is known, the quality or degree of superheat at exit can be found from the Mollier diagram. The specific volume of saturated or superheated steam for any pressure can be obtained from the steam tables. The length of a nozzle is determined by using a total taper from throat to exit of 1:5 to 1:10.

When the discharge pressure P_3 is less than the critical pressure P_2, the Grashoff formula

$$w = 0.0165A_2P_1^{0.97}$$

can be used to obtain or check the throat area A_2 of the nozzle.

Nozzles of simple impulse turbines or nozzles of the first few stages of the various types of compound turbines described in the following articles seldom extend completely around the periphery of a turbine wheel. When the nozzles extend only partway around the periphery, the turbine is said to have "partial" admission, and when they extend completely around, it is said to have "full" admission.

Example.—A feed pump is driven by a single-stage steam turbine having three nozzles. At the nozzle bowl the steam pressure is 325 lb per sq in.

gage, and the steam is dry and saturated. If the stage pressure P_3 is 15 lb per sq in. gage, calculate the velocities and required areas at the nozzle throat and mouth.

Total steam consumption = 3,600 lb per hr

V_3, nozzle exit velocity $= 223.7\sqrt{(h_1 - h_3)e_n}$

h_1 (from Mollier diagram) at 340 lb abs = 1204 Btu

$s = 1.499$

h_3, after isentropic expansion to 30 lb abs = 1022 Btu

$h_1 - h_3 = 182$ Btu

$V_3 = 223.7\sqrt{(182) \times 0.90} = 2{,}860$ ft per sec

h'_3, enthalpy after expansion, when friction is allowed for $= h_1 - \{(h_1 - h_3)e_n\} = 1{,}204 - \{(182)(0.90)\} = 1040$ Btu

x_3 = actual quality from the Mollier chart for a pressure of 30 lb abs and an enthalpy of 1040 Btu = 0.868

v_3, specific volume = 13.75 cu ft

$x_3v_3 = 11.95$ cu ft

Mouth area $= A_3$

$$A_3 = \frac{w(xv)}{V} = \frac{\left(\frac{3{,}600}{3{,}600}\right)\left(\frac{1}{3}\right) 11.95}{2{,}860} = 0.001395 \text{ sq ft or } 0.20 \text{ sq in.}$$

Diameter = 0.505 in.

P_2, pressure at nozzle throat = 0.58(340) = 197 lb abs

h_2, enthalpy at nozzle throat from Mollier chart = 1159 Btu

V_2, throat velocity $= 223.7\sqrt{h_1 - h_2} = 223.7\sqrt{1{,}204 - 1{,}159} = 1{,}500$ ft per sec

Quality at throat corresponding to 197 lb abs and entropy = 1.499 = 0.954. Under this condition the specific volume of the steam = 0.954 × 2.32 = 2.22 cu ft per lb

Throat area $= A_2$

$$A_2 = \frac{w(xv)}{V} = \frac{3{,}600}{3{,}600} \times \frac{1}{3} \times \frac{2.22}{1{,}500} \times 144 = 0.071 \text{ sq in.}$$

The corresponding nozzle diameter at the throat is .300 in.

When the stage pressure or exit pressure P_3 is greater than the critical pressure P_2, the nozzle should be converging; when the stage pressure is less than the critical pressure, the nozzle should have the convergent-divergent shape shown in Fig. 43(*b*).

Frequently turbines are operated at steam pressures less than that for which they are designed. Such a case would exist when the power is reduced by partially closing the throttle valve. This will decrease the initial pressure and also the stage pressures throughout the turbine. If the nozzle-mouth area is too great for this reduced-power condition, the steam will tend to leave the walls of the nozzle. This condition is termed "overexpansion." A relatively greater loss exists for such conditions than for underexpansion, where the nozzle-exit area is too small and the expansion is completed beyond the mouth of the nozzle.

69. Blade-velocity Diagram.—The turbine in its simplest form consists of a wheel with a single row of impulse blades fitted to the rim. Steam expands in one or more nozzles from the nozzle-entrance pressure to the turbine-casing pressure, thereby developing a high steam velocity at the nozzle exit.

After leaving the nozzle, the steam jet passes through the turbine blading, the blade-entrance shape being selected so the jet

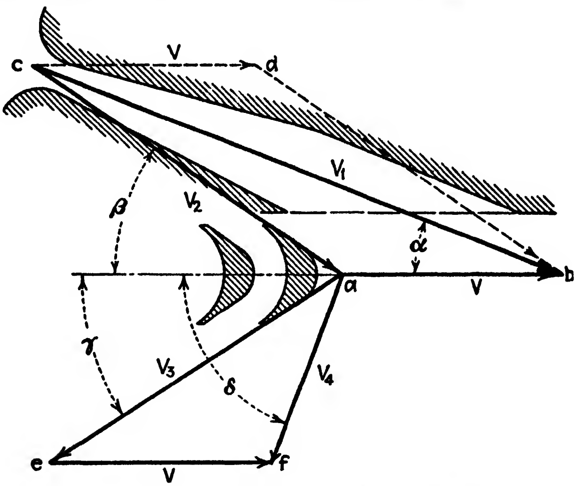


Fig. 44.—Blade velocity diagram for impulse turbine.

enters the blade passage with the minimum of shock. Thus, in the upper part of Fig. 44, which represents a velocity or vector diagram for single-row turbine blading, the blade-entrance angle is made tangent to V_2, the relative jet velocity at blade entrance. In Fig. 44, V_1 is the absolute jet velocity of steam at nozzle exit, and V is the peripheral velocity of the wheel. V_1 is laid out at the selected nozzle angle.

The purpose of the turbine blading is to convert an appreciable portion of the kinetic energy of the jet into torque. Hence the exit portion of the impulse blade is curved to reverse the direction of the steam jet, the blade passage being shaped to give a smooth channel for steam flow.

Blade-exit angle γ of an impulse turbine may at times be taken equal to the entrance angle β, as is done in Fig. 44. A slight increase in efficiency can be obtained by making γ slightly less than β.

The blade-exit-velocity diagram is drawn by laying out vector V_3, the jet velocity relative to the blade, at angle γ.

In ordinary impulse blading no pressure drop occurs in the moving blades, and hence no energy is available to increase the relative exit velocity V_3. The relative exit velocity will therefore equal V_2 if blade friction is neglected. Normally, blade friction is allowed for, and V_3 is obtained by multiplying V_2 by a friction factor, approximate values for which are given in Table VIII.

The absolute exit velocity V_4 is obtained by laying off the blade velocity V at the lower end of the vector V_3, as shown in Fig. 44.

Much higher efficiency could be obtained if the steam entered and left the blades in the same line of direction with which the blades are moving and with an entering velocity equal to twice that of the blade. Under these conditions V_4 would be equal to zero, and the maximum possible efficiency would be attained. Such an arrangement could not be used in practice because of the interference of the jet and blades; hence the steam must enter and leave at an angle to the blades, as shown in Fig. 44.

TABLE VIII

Velocity V_2 relative to blade, ft per sec	500	1,000	1,500	2,000
V_3/V_2, blade-friction factor, approximate	0.92	0.90	0 88	0.86

The blade-velocity diagram not only is useful in determining the entrance and exit angles of the moving blades but also can be used for calculating the work developed by the jet action and the blade-diagram efficiency.

70. Blade Efficiency.—In Fig. 44, let us assume that blade friction can be neglected so that $V_3 = V_2$.

The momentum of the steam $= \dfrac{w}{g} [V_1 \cos \alpha - (-V_4 \cos \delta)]$

The torque or moment of momentum

$$= \frac{w}{g} (V_1 \cos \alpha + V_4 \cos \delta)R$$

The work done $= \frac{w}{g} (V_1 \cos \alpha + V_4 \cos \delta)R\omega$ or

$\frac{w}{g} \{[V_1 \cos \alpha + (V_1 \cos \alpha - 2V)]\} V$, as expressed in terms of one variable V

where V_1, V_2, V_3, etc., are in feet per second.

w = weight of steam leaving nozzle, in pounds per second.

R = wheel radius to center of blades, in feet.

ω = angular velocity, in radians per second.

From the foregoing expression, it is seen that the work output will vary with the wheel speed V. By differentiating the expression for work with respect to V and placing this equal to zero, we can obtain the blade velocity V that will give maximum work

$$V = \frac{V_1 \cos \alpha}{2}$$

Thus the wheel velocity should be one-half the value of $V_1 \cos \alpha$ for maximum work. The expression $V_1 \cos \alpha$ is often termed the "initial velocity of whirl."

The efficiency of the blading is usually taken as blade work divided by the initial kinetic energy of the jet. The blade efficiency is

$$e_b = \frac{\frac{w}{g} (2V_1 \cos \alpha - 2V)V}{\frac{wV_1^2}{2g}}$$

and if the value of V is taken to give maximum work,

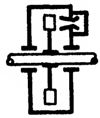
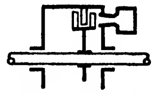
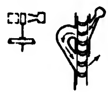
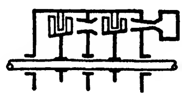
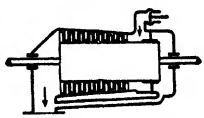
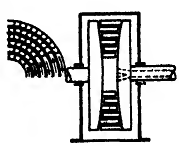
$$e_b = \cos^2 \alpha$$

This clearly shows the advantage of using small nozzle angles when they are practical.

Blade efficiencies are discussed further in Art. 79.

71. Classification of Turbines.—Turbines can be classified primarily as either impulse or reaction types. In the impulse type the pressure drop occurs only in the nozzles, the pressure in the turbine casing between nozzles being constant. Equal pressures on both sides of moving blades are ensured by allowing ample tip clearance of the blades and by drilling equalizer holes in the turbine disks. Impulse turbines can be further classified by the method of compounding employed.

TABLE IX.—TYPES OF STEAM

	Type		
Impulse Steam expands in nozzles to the stage pressure Partial peripheral admission in H.P. stages Blading designed only to reverse the steam jet, not as a nozzle. Flow area about constant Slight axial thrust	Single-stage; one row (DeLaval original)		
	Velocity compound single pressure stage Nozzles expand steam from chest to stage pressure Fairly high rpm	Multi-blade row—(Curtis type) (axial flow)	
		Pelton wheel Stumpf type tangential flow (helical)	
		Repeated flow type	
	Pressure-compound multistage (Rateau type)		
	Pressure-velocity compound a. Curtis b. Curtis—Rateau (Curtis stages followed by Rateau)		
Reaction Steam pressure drop in moving rows as well as in fixed rows; steam velocity greater at blade exit than at entrance Full peripheral admission	Axial flow (Parsons)		
		Curtis-Parsons	
	Radial flow (Ljungstrom)		

* Maximum shp per casing in marine practice.

TURBINES AND THEIR APPLICATIONS

Features	Rpm	Shp	Service	Notes
Separate nozzles expand to exhaust pressure; original type very high speed	8,000–30,000	5–300	Auxiliary generator and pump drive through gearing	Bad blade erosion
2 or 3 rows of moving blades, stationary guides between rows and opposite nozzles	1,200–10,000	5–1,500 1,500+	Small auxiliary generators; pumps, blowers; astern turbines	Often use reduction gearing for auxiliary drive. Built by all turbine manufacturers
Blading milled in rim of forged rotor. After jet impinges on blade the stationary guide redirects it on the wheel. Jet path is roughly helical	500–5,000	5–300	Small auxiliary generators; pumps, blowers	Very rugged. Efficiency only fair. Popular for noncondensing service
		5–300	Small auxiliary generators, pumps, blowers	
Equivalent to 2–30 single-stage turbines mounted on same shaft; steam flow through all in series	1,500–10,000	100–20,000*	Auxiliary generator, main propulsion turbines	High efficiency. Expensive construction. Common impulse propulsion turbine
Steam flow first through Curtis stage; Type *a*. Other Curtis stages follow first Type *b*. Rateau stages fitted after first	1,200–6,500	20–20,000*	Auxiliary generator; main propulsion turbines; astern turbines	Type *b*. Curtis wheel shortens rotor; lowers casing pressure
Blading set on a drum or conical rotor. Pressure drop in moving blade rows, and drum construction produces axial thrust, hence dummy pistons fitted; close blade clearances	1,200–5,500	1,500–20,000*	Main propulsion turbines	Especially efficient as L. P. units. H.P. low-powered reaction units have lower efficiency
Curtis H.P. stage followed by reaction stages			Marine propulsion	Results in a shorter turbine
Blading on two oppositely rotating disks so that one set of blades acts as guides for the other set, *i.e.*, no stationary guide blades		1,000–	Turbo-generators in stationary plants. A few early turbo-electric ships	High efficiency; intricate construction; costly. Not built in U.S.A. Compact

In the reaction turbine there is a drop in pressure accompanied by an increase in velocity in the moving blades as well as in the fixed guide blades. The fixed blading takes the place of nozzles of the impulse turbine. The reaction turbine consists of alternate rows of stationary and moving blades, the fixed blades being fastened to the turbine casing.

The fundamental types of turbines that are described in the following pages are:

1. Impulse type.
   a. Single stage.
   b. Velocity compound.
   c. Pressure compound.
   d. Pressure-velocity compound.
2. Reaction type.
3. Combined impulse and reaction type.

Turbines can be further classified by the direction of the steam flow relative to the rotor. Parallel flow is the common type, with the steam flow passing through the blading approximately parallel to the shaft line. In radial flow, as the name signifies, the steam flow is at right angles to the shaft. The steam may flow toward or away from the center. Turbines used for marine propulsion have the center line horizontal or approximately so. However, low-powered turbines are built with vertical shafts, and these are at times used in marine work for driving auxiliary pumps and blowers.

The number of stages of a turbine, the number of turbines or turbine casings, the steam-flow arrangement through them, whether in series or combined series and parallel, provide still other means of classifying turbines. Table IX (pages 148 and 149) gives in outline form the important features of the various types of turbines, including some not discussed in the text.

72. Single-stage Impulse Turbine.—This is the simplest type of turbine and consists of one set of nozzles and one revolving wheel. It was invented by DeLaval, a Swedish engineer, in 1889, and was one of the earliest commercially successful turbines. It is still referred to as the DeLaval turbine. The steam is expanded in convergent-divergent nozzles from the initial pressure to the exhaust pressure and consequently leaves the nozzles at high velocity with condensing units. The nozzle-exit velocity is usually between 3,000 and 4,000 ft per sec. Figure 45 shows in diagrammatic form the general construction of this turbine, the blade-velocity diagram, and the changes in enthalpy, steam pressure, and velocity throughout the turbine. Only a portion of the kinetic energy of the jet is absorbed by the turbine blading, and the steam leaves with an exit velocity of V_4, which is approximately one-third of the initial velocity. In order to obtain good blade efficiency extremely high turbine revolutions were employed, ranging from 8,000 to 30,000 rpm. The high steam velocities caused excessive blade erosion, and the high peripheral velocity of the wheel required expensive construction.

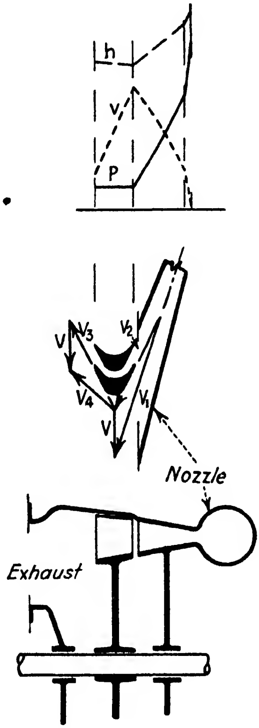


FIG. 45.—Diagram of simple impulse turbine.

73. Velocity-compound Turbine.—In this type of turbine the steam is expanded through a single set of nozzles from initial pressure to the stage or exhaust pressure, just as in the turbine described in the preceding article. Instead of attempting to absorb all the energy of the high-velocity steam in one wheel, the Curtis-type velocity-compound turbine uses two or more moving blade rows, with fixed blading attached to the turbine casing for reversing the flow of steam. The steam is thus passed through alternate rows of moving and fixed blading until its velocity has been reduced to a relatively low value. Obviously, with this type of turbine, good blade efficiency can be obtained with much lower revolutions than employed in the original DeLaval single-blade-row type of turbine. As now constructed, the Curtis velocity-compound turbine usually consists of two rows of moving blades and one set of fixed blades. It is employed in marine work to drive auxiliaries, for astern turbines, and as the first stage in multistage propulsion turbines, as described in Arts. 76 and 77. Figure 46 shows in diagrammatic form the general construction of a Curtis velocity-compound turbine employing two rows of moving blades. This figure also shows the blade-velocity diagram and the changes in steam pressure, velocity, and enthalpy throughout the turbine.

The steam leaves the nozzles at high velocity and acts on the first row of moving blades, where a considerable part of the kinetic energy of the jet is absorbed. As shown in Fig. 46, the direction of the steam is reversed in the fixed or guide blades so that it can effectively act on the second row of moving blades. These guide blades are located approximately opposite the exits of the nozzles. Thus, in this particular turbine, both the nozzles and the guide blades extend only partway around the periphery. The steam leaves the second row of moving blades with an absolute velocity of approximately one-quarter of the original jet velocity.

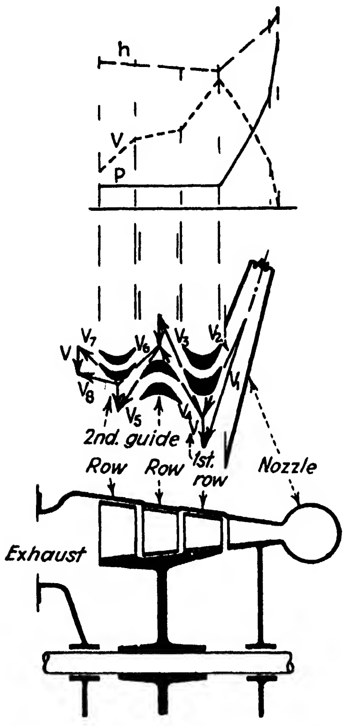


Fig. 46.—Diagram of velocity-compound turbine.

As shown in Fig. 46, there is a large drop in the velocity of the steam and hence in its kinetic energy in the first row of moving blades, compared with the second moving row. The work developed in the second row is only about one-quarter of that developed in the first row. There is some decrease in velocity in the guide blades through friction. The friction in the moving and fixed blades results in an increase in enthalpy of the steam.

There is no change in steam pressure in velocity-compound blading, hence the only change in specific volume and the quality of the steam as it passes through the stage is that due to the effect of friction. However, since the area for steam flow is proportional to quantity divided by the velocity, the height of the blades must be increased as the velocity decreases (see Fig. 46).

A double-blade-row single-stage-velocity compound impulse turbine such as is used on shipboard for driving pumps and

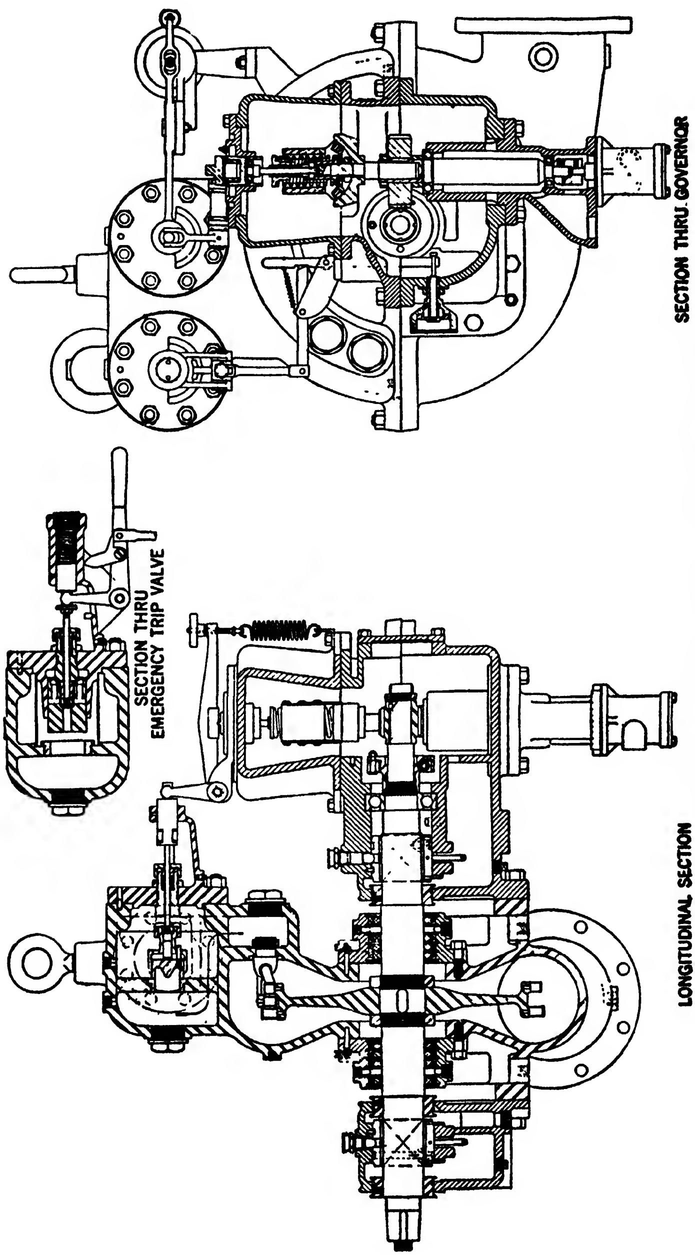


FIG. 47.—De Laval velocity-compound auxiliary turbine. (*The Nautical Gazette, November*, 1939.)

blowers is shown in Fig. 47. In this turbine the steam admitted through the turbine stop valve, passes through an emergency-trip valve, a steam strainer, and, finally, through the turbine-operating governor valve. The latter valve controls the flow of steam to the turbine nozzles and by this means holds the turbine revolutions within a fixed range. The emergency-trip valve is shown at the left-hand side of the end elevation in Fig. 47, and the governor valve is shown in cross section at the top of the longitudinal elevation. The emergency-trip valve is fitted to prevent the turbine from overspeeding in case of the failure of the operating governor. The valve is tripped mechanically by an excessive rotor speed.

74. Pressure-compound Turbine.—In the pressure-compound or Rateau turbine, the steam passes successively through a series of stages, each made up of a row of nozzles and a row of moving blades. Figure 48 shows in diagrammatic form the general construction of the pressure-compound turbine and the changes in pressure, velocity, and enthalpy through three stages.

The nozzles are designed so that the pressure drop and hence the enthalpy drop in the nozzles of each stage will provide a steam-jet velocity that will give a good turbine efficiency. The energy of the steam leaving the first set of nozzles is partly kinetic in the form of velocity and partly potential in the form of enthalpy. Much of the velocity acquired in the first set of nozzles is converted into mechanical work in the first row of moving blades. The enthalpy remaining in the steam is employed in creating velocity in remaining nozzles. This type of turbine virtually consists of a number of the early DeLaval single-blade-row turbines mounted on the same shaft and arranged so that the steam flows in series through each of the units.

The blades are mounted on disks attached to the shaft, and the nozzles are placed in diaphragms that separate each wheel from the adjacent ones. Since the pressure in each stage is different from that in the adjacent ones, the diaphragms must be made steamtight in order to avoid serious leakage between stages. This is accomplished by extending the diaphragms to the shaft, as shown in Figs. 48, 51, 52, and 53, so that the area for this steam leakage will be small.

The first-stage nozzles extend only partially around the periphery of the wheel (partial admission) but in the low pressure stages they extend completely around the periphery to take care of the large increase in the specific volume of steam as the pressure is reduced. The nozzle areas and the blade height are also increased at the low pressure end for the same reason. The general construction of this turbine and the greater number of

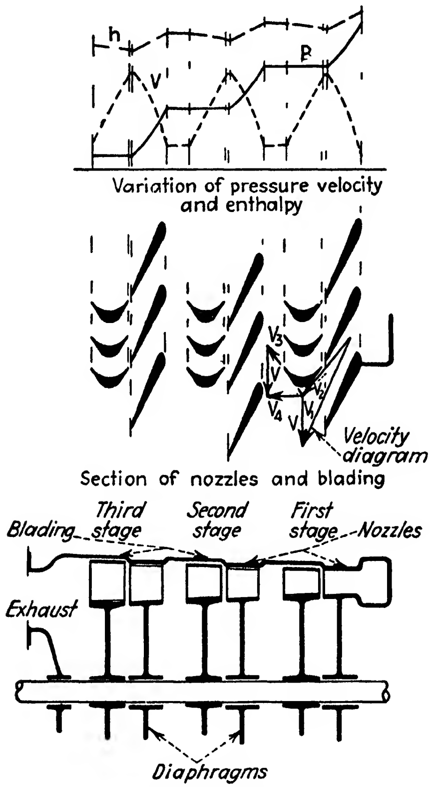


FIG. 48.—Diagram of pressure-compound turbine.

stages used make a turbine of this type larger and more expensive than the previous types, but the low steam velocities make it more efficient. It is suitable for moderate and large powers and is used for the largest steam turbines employed in marine and stationary service. High and low pressure turbines of this type are shown in Figs. 52 and 53 and will be described in more detail in Art. 76.

75. The Parsons Reaction Turbine.—The first commercially successful reaction turbine was developed by Charles A. Parsons, and this type of turbine still carries his name. The multistage reaction turbine operates both by impulse of the steam entering the moving blades and by the reaction of the steam leaving the blades. The turbine rotor is fitted with blading suitable for absorbing the kinetic energy of the steam leaving the fixed blading, and the moving blades are shaped so as to form steam jets that give a reaction effect as the steam leaves. Thus both the guide blades on the casing and the moving blades on the rotor

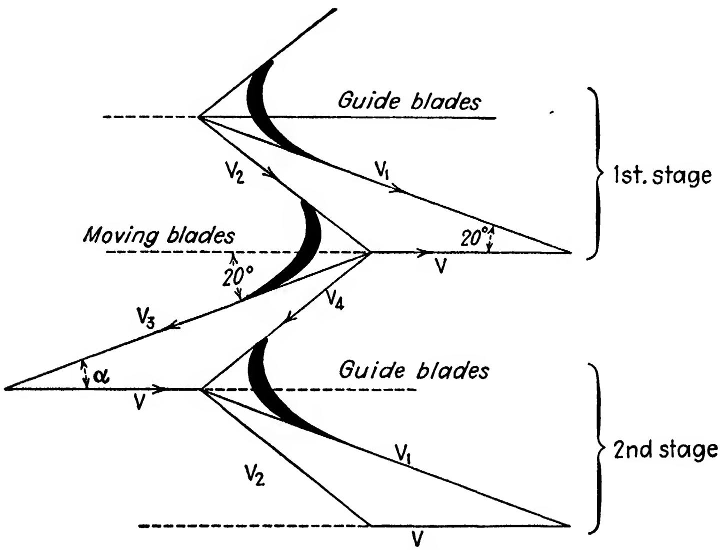


FIG. 49.—Velocity diagram of Parsons turbine.

serve as nozzles, and hence a pressure and enthalpy drop must occur across each row of blades. Because of the pressure drop across the blading, full peripheral admission of the steam must be used with reaction blading, even in the high pressure stages.

A reaction turbine consists of a large number of annular rings of fixed and moving blades. A row of fixed and moving blades constitutes a stage. Since there is a pressure drop and an increase in velocity across each row of blades, the steam pressure decreases gradually along the rotor from the high pressure end to the exhaust end of the casing.

A blade-velocity diagram for a Parsons turbine is shown in Fig. 49. The steam leaves the fixed guide blades with an absolute velocity of V_1. The peripheral velocity of the wheel is V; hence the velocity of the steam relative to the moving blades is V_2. Because of the drop in pressure in the moving blades, the steam leaves the moving blades with a relative velocity V_3 greater than that with which it entered. The absolute velocity of the steam entering the second set of guide blades is V_4. Because of the drop in pressure in the guide blades, the steam leaves with an absolute velocity V_1 greater than V_4. Reaction blades of normal type have exit angles of 18 to 20 degrees. If friction losses are neglected, the enthalpy drop required for a reaction stage, consisting of one guide and one moving row, is

$$h = 2\left[\left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2\right]$$

The drum type of construction is usually employed for the rotors of reaction turbines, the rings of moving blades being mounted on the periphery of the drum and the rings of guide blades on the inside of the casing (see Fig. 50). Because of the difference in pressure that exists between each row of blades in this type of turbine, the tip clearances of both the fixed and moving blades must be small to reduce leakage around the ends of the blades. The drum construction provides a stiff rotor that ensures minimum deflection and hence allows small blade clearances to be used.

Since there is no change in pressure in the blading of an *impulse*-type turbine, the radial clearance of the blading can be made as large as desired. The change in pressure in impulse turbines takes place in the nozzles between the stages.

The diameter of the rotor of a reaction turbine must be large enough to ensure that its deflection is small, so as to maintain blade clearances and also to avoid critical speeds. Blade heights of less than ½ in. are very inefficient because of the large percentage of tip leakage. Because of the foregoing limiting conditions and the fact that full peripheral admission is used, the area through the blades for passage of steam cannot be made very small, and hence turbines of this type are not suitable for the passage of small volumes of steam associated with turbines of low horsepowers. The minimum economical power of a reaction turbine is about 2,000 shp for one propulsion unit.

In order to obtain a small difference in pressure between each side of the blade rows and hence reduce leakage and secure a high efficiency, a relatively greater number of stages are employed with a reaction turbine. To avoid the necessity of small tip clearances of the blading in a reaction turbine, "end-tightened" blading, or some modification of it, is now employed for the H.P. turbine and at times for the L.P. turbine to reduce leakage. These types of blading are discussed in Art. 83.

The drum-type rotor of a reaction turbine is subject to axial thrust because of the drop in pressure in the moving blades and the unequal steam pressures acting on the two ends of the drum. To overcome this, balancing devices or dummy pistons are fitted. In a drum-rotor having no dummy, the steam force at the high pressure end is equal to the initial pressure times the cross-sectional area of the drum minus the shaft area at the gland packing. The axial thrust is usually partly equalized by extending the high pressure end of the rotor cylinder to form a dummy cylinder (Fig. 54). The space between the dummy and the shaft-end packing is connected to the exhaust end of the turbine casing by an equilibrium pipe or the equivalent so that the axial thrust due to the difference in steam pressures at the ends is small. Dummy labyrinth packing (Art. 86) is fitted around the cylindrical surface of the dummy piston to prevent steam leakage between the annular equalizing space and the first high pressure stage of the turbine. The dummy cylinder and dummy packing are clearly shown in Figs. 54, 55, and 58.

Thrust bearings are fitted to all turbines to locate definitely the longitudinal position of the rotor (see Figs. 52, 54, 55, and 58), and any end thrust of a reaction turbine that is not taken up by the dummy piston is carried by this thrust bearing.

Provision for enough area for steam flow through the moving blades and fixed guide rows at the low pressure end of a turbine is a problem in all types of turbines. Besides increasing the height of the blades, as shown in Fig. 57, the blade-exit angle at the exhaust end of the Parsons turbine is increased to give a greater steam-flow area. Semiwing- or wing-blade sections shown in Fig. 69 are fitted in the last low pressure stages. Large blade-exit angles are also used for the last stages of an impulse turbine.

Figure 50 shows the H.P. cylinder of the three-casing pure Parsons turbine fitted on the "Beaver" class of ships (1928) of the Canadian Pacific Steamship Company. This turbine operates with a pressure of 250 lb and a temperature of 650°F. The rotor is a solid forging.

When reaction turbines are used for ship propulsion they are usually modified as discussed in Art. 77.

76. The Impulse Turbine for Ship Propulsion.—Turbines used for ship propulsion may be contained in a single casing or divided up into two, three, or four units. With turbo-electric drive, the complete expansion takes place in one casing, although the total power of the ship may be developed by two or more turbines. With geared turbines of power less than around 2,000 shp per screw, single casings are often used. For powers of over 2,000 shp per screw, the usual American practice is to use two casings, a high pressure and a low pressure turbine, which are connected in series. Each turbine drives the reduction gearing through a separate pinion.

When the impulse turbine is used for ship propulsion, the usual practice for many years has been to employ a combined pressure velocity-compound turbine for the high pressure unit and a pressure-

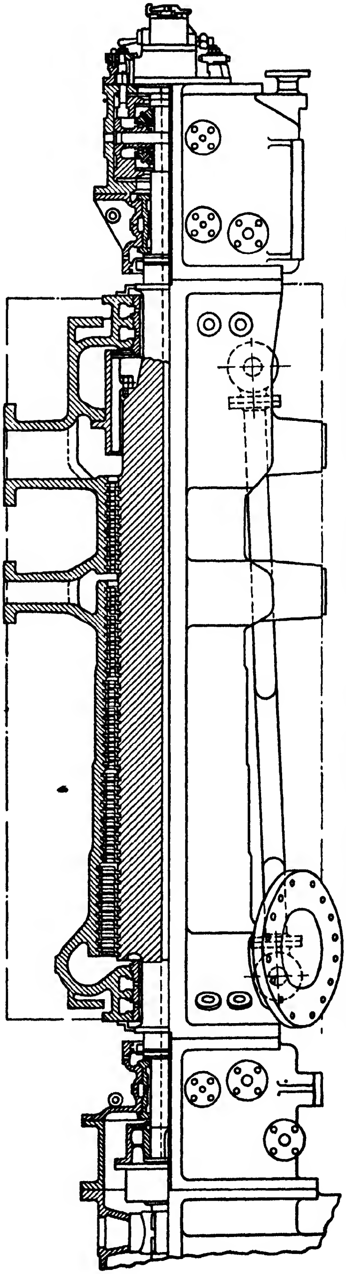


FIG. 50.—High-pressure Parsons turbine. *S.S. Beaverdale.* (*The Shipbuilder, July, 1928.*)

compounded or Rateau turbine for the low pressure turbine. In such turbines the high pressure unit is made up of a velocity-compound stage consisting of two rows of moving blades

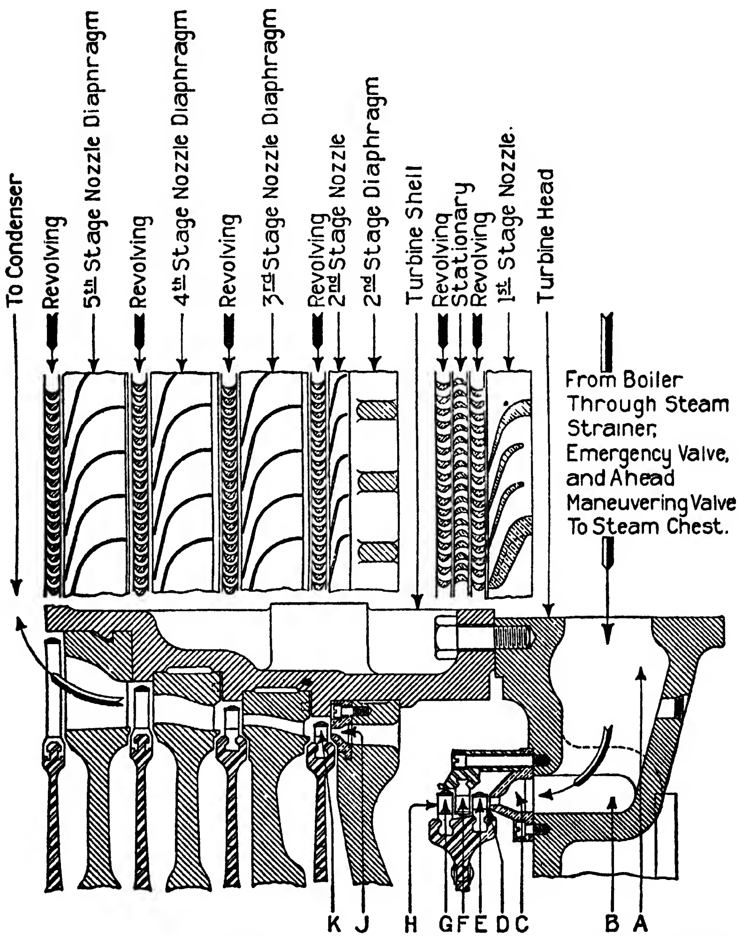


Fig. 51.—Diagram of pressure-velocity compound turbine. (*General Electric Company.* 1920.)

and one row of fixed blading, followed by a series of pressure-compound stages. Turbines of this type are known as "pressure-velocity-compound" or "Curtis-Rateau" turbines. By using a Curtis velocity-compound wheel in the first stage, a large initial pressure drop can be obtained that reduces the pressure drop required in the remaining stages and decreases the number of stages and lowers the maximum pressure in the casing.

A single-casing turbine of this type is shown in Fig. 51; the steam enters through the chest, A, passes through the nozzles, C, and thence through the moving blades, E, the fixed blades, F, and the second row of moving blades, G. From here the steam passes through the nozzles at J and thence through the blading on the wheel of the first pressure-compound stage.

At the present time, when impulse turbines are used, the tendency is toward using pure pressure-compound turbines. Figure 52 shows a high pressure turbine built by the De Laval Steam Turbine Company. This turbine is designed to operate with a low pressure unit in a cross-compound-geared turbine installation. It is of the pressure-compound or Rateau type, *i.e.*, all the stages are pressure-compound. Steam enters the turbine through the steam chest shown at the right of the figure. The turbine power output is controlled by the nozzle valves shown at the right. Each hand wheel regulates the flow of steam to a group of first-stage nozzles.

As will be observed in Fig. 52, the first-stage wheel is larger in diameter than the wheels in the remainder of the stages. Since this wheel is larger in diameter, it will have a higher peripheral speed than the other wheels, and the first-stage nozzle-exit velocity can be greater. This will allow a larger pressure drop assigned to the first stage, with a consequent lower steam pressure in the turbine casing and less steam-leakage losses by the diaphragm packings of the other stages. In this turbine the rotor and wheels are machined out of a solid forging. The diaphragms are split on a plane through the shaft center line and are fitted with labyrinth packing.

As shown in Fig. 52, a transverse I-beam type of support is fitted at the forward end of the turbine. This allows the turbine to expand forward with an increase in temperature but maintains the athwartship position of the forward turbine bearings.

Figure 83 shows a pure pressure-compound turbine as fitted on the turbo-electric ship *Viceroy of India*. In this case, as is usual with turbo-electric installations, the complete expansion takes place in one casing instead of in two or more, as with geared-turbine installations. The turbine shown in Fig. 83 operates with a throttle pressure of 375 lb and a temperature of 700°F.

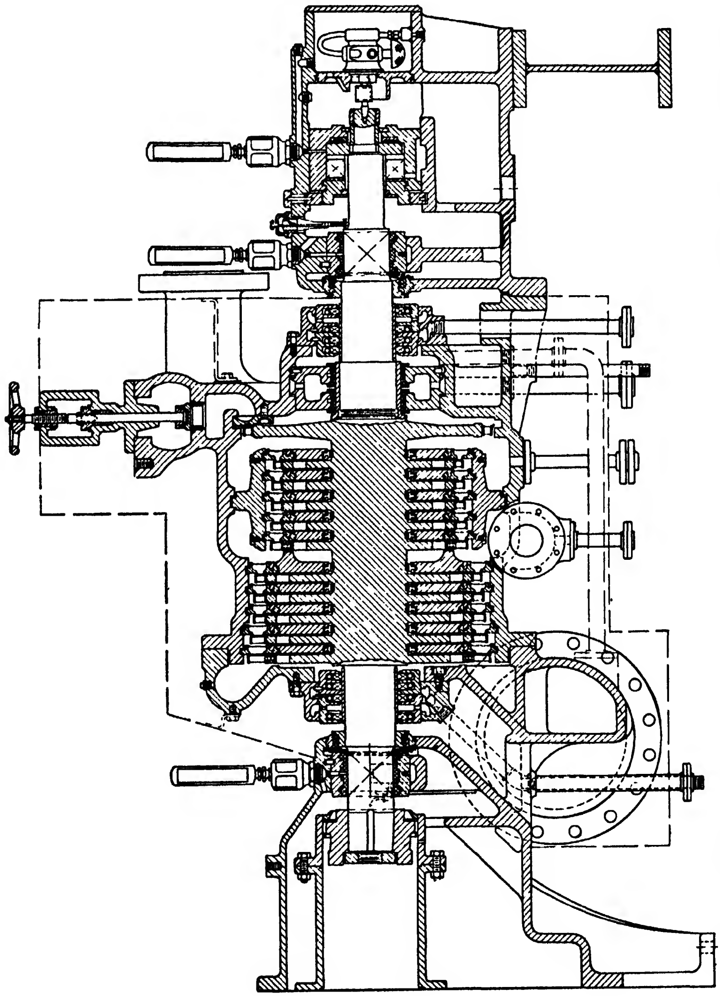


Fig. 52.—High-pressure impulse turbine (pressure-compound). (*De Laval Steam Turbine Company.*)

Figure 53 shows the sectional elevation of a General Electric pressure-compound low pressure turbine used for marine propulsion. The steam exhausted from the low pressure end of the high pressure casing enters the left-hand or aft end of the low pressure casing and flows forward through seven pressure-compound impulse stages. Each stage has full peripheral admission. By having the steam enter the aft end of the low pressure turbine, a more direct steam connection is made to the high pressure

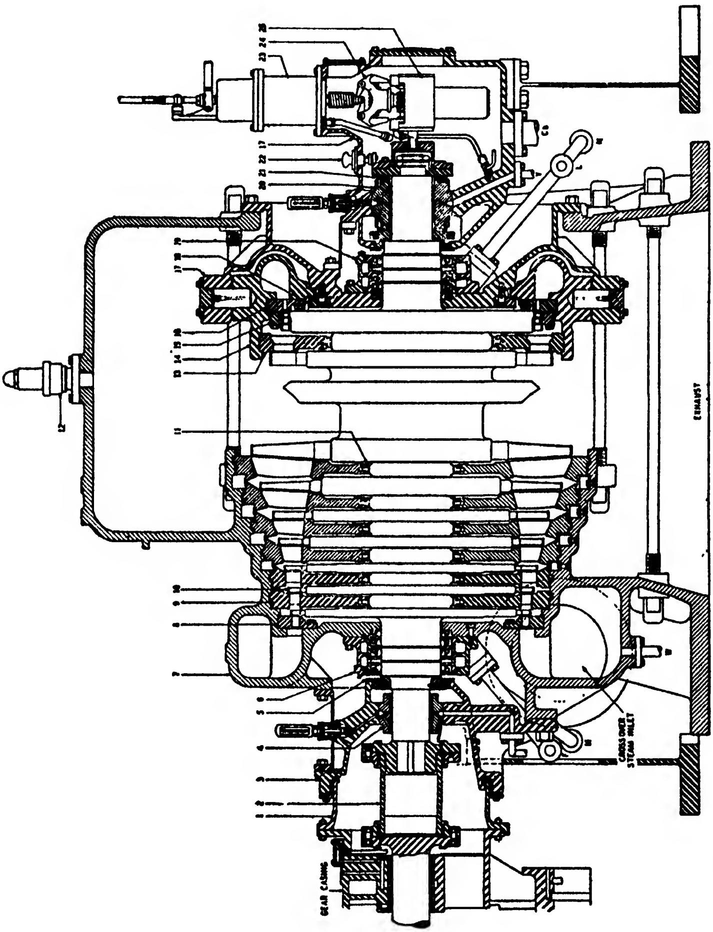


FIG. 53.—Low-pressure impulse turbine (pressure-compound). (*General Electric Company.*) (*From The Nautical Gazette, June,* 1940.)

casing, and more space is allowed between the condenser and the reduction gears.

The astern turbine is located in the forward end of the casing, adjacent to the exhaust trunk, so there is a common exhaust to the condenser. This arrangement also allows the astern turbine wheels to rotate in a vacuum during ahead operation, thus reducing windage losses. As shown in Fig. 53, the astern unit consists of a Curtis velocity-compound stage with two rows of moving blades, followed by a Rateau stage. A deflector disk is fitted on the rotor between the ahead and astern turbine, as shown in Fig. 53, so that the exhaust steam from the astern turbine will clear the ahead blading when going astern. The

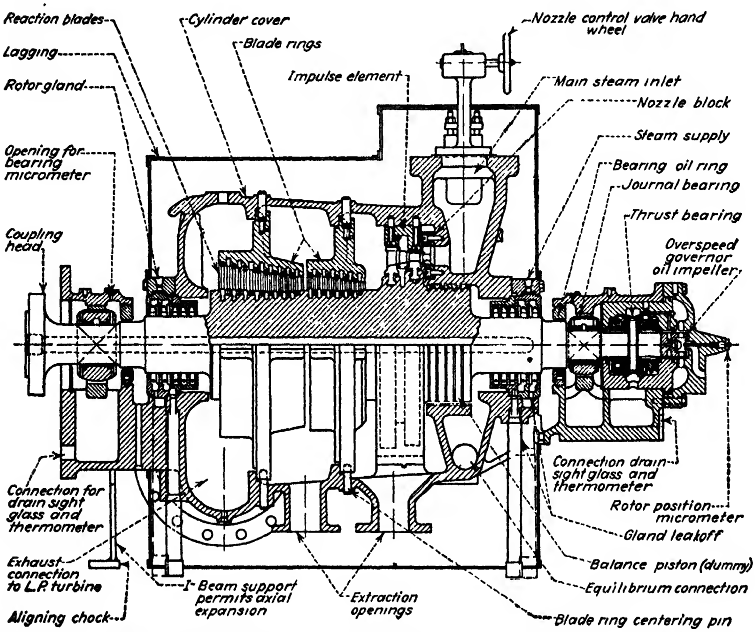


FIG. 54.—High-pressure combined impulse and reaction turbine. (*Westinghouse Electric & Manufacturing Company.*)

exhaust steam leaves the turbine casing at the bottom, and the condenser is placed underneath the turbine.

77. The Combined Impulse-and-reaction Turbine.—As already pointed out, the leakage between successive rows of blading in the Parsons turbine takes place over the end of the blades, and the clearance at the tip of the blades is made small to reduce this leakage. At the high pressure end of the Parsons turbine the blades are extremely short because of the small specific volume of the stream. Since the clearance at the end of the blades is of about the same magnitude for all blade lengths, the percentage of clearance must be very much larger with the short high pressure blades than with the longer blades. In consequence, the percentage of leakage by the rows of blades or stages at the high pressure end is high, resulting in a low turbine efficiency at this section.

Because impulse turbines are more efficient in the high pressure ranges and the reaction turbine is more efficient at low pressures, some turbine builders employ an impulse turbine for the high pressure unit of a two casing installation and a reaction turbine for the low pressure unit.

When reaction turbines are employed, the usual practice is to use a single Curtis velocity-compound impulse stage (Fig. 54) at the high pressure end of a reaction turbine. The remaining blading of the high pressure turbine and all the blading of the low pressure turbine are of the reaction type. Figure 54 shows a high pressure Westinghouse turbine that is of this type. The low pressure unit of this turbine is shown in Fig. 55.

In Fig. 54 the steam from the boiler enters at the right-hand end, passes through two rows of moving blading and one row of fixed blading of the velocity-compound impulse stage, and then enters the reaction blading marked "blade rings" and passes successively through rings of fixed and moving blades. The steam leaves the H.P. turbines through the exhaust connection to the L.P. turbine, shown at the lower left-hand end of Fig. 54.

The steam exhausted from the H.P. turbine enters the left-hand or aft end of the L.P. turbine (Fig. 55) and passes to the right through the reaction blading. After leaving the last row of reaction blading, the steam enters the condenser through the exhaust connection shown at the bottom of the L.P. turbine.

The astern turbine shown at the right end of Fig. 55 consists of two velocity-compound impulse stages arranged in series.

The use of the Curtis stage in the high pressure turbine does away with a number of rows of the inefficient, short high pressure blading and materially shortens the turbine, giving a shorter and stiffer rotor.

Figure 56 shows a photograph of the Westinghouse H.P. turbine illustrated in Fig. 54, with the cylinder cover removed. The Curtis velocity-compound wheel shown at the right in Fig. 54 is at the left-hand end of Fig. 56. Figure 57 shows a photograph of the Westinghouse L.P. turbine with the cylinder cover removed.

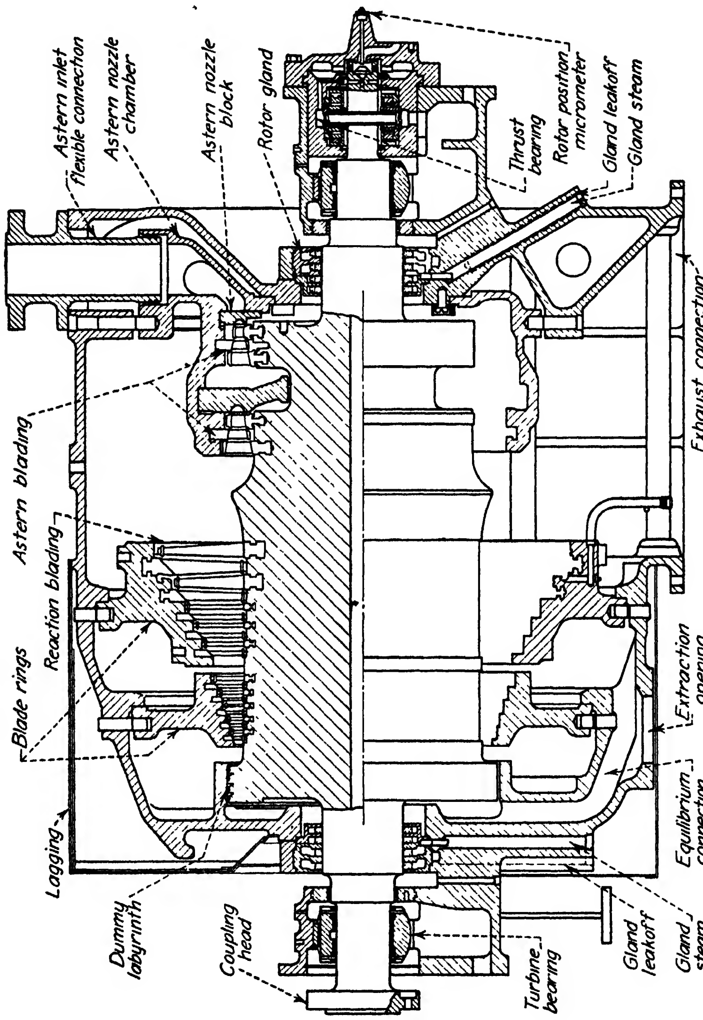


Fig. 55.—Low pressure reaction turbine. (*Westinghouse Electric and Manufacturing Company.*)

Figure 58 is a Bethlehem low pressure reaction turbine. The steam from the H.P. combined impulse and reaction unit enters the left-hand end of Fig. 58. It will be noted that the solid-forged rotor is tapered in order to give it an increased diameter at the low pressure end. The equilibrium connection is made by means of an axial hole in the rotor. The astern turbine consists of two velocity-compound Curtis stages, one with three moving blade rows and the other with two moving blade rows.

78. Astern Turbines.—Since the blading in a steam turbine will allow it to operate only in one direction, special astern stages

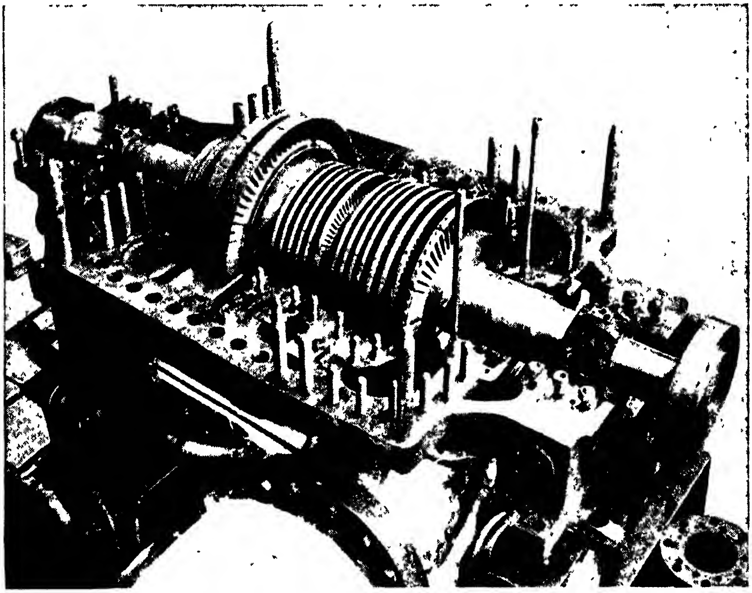


FIG. 56.—Westinghouse H.P. turbine (combined impulse and reaction type). Cylinder cover removed.

are fitted to geared turbines. The usual American practice with cargo ships is to fit the astern blading at the exhaust end of of the low pressure rotor (Figs. 53, 55, and 58) and to fit no astern blading in the high pressure casing. With turbo-electric drive, the turbogenerator rotates always in one direction, the propelling motor being reversed to go astern; hence no astern turbines are required with this type of machinery (see Fig. 83). In the case of geared turbines for merchant-ship service, the astern turbine is designed for about 80 per cent ahead torque at one-half the ahead revolutions; so approximately 40 per cent of the ahead full power is developed in going astern. In high-powered naval vessels, the astern turbine develops about 25 per cent of the ahead power. In merchant ships the foregoing powers can usually be obtained by fitting a pressure-velocity-compound astern turbine, consisting of a first-stage two-blade-row Curtis wheel, followed by one Rateau stage (see Fig. 53). When low blade speeds are employed, the Rateau second astern stage wheel is replaced by another two-row Curtis stage, as in Fig. 58. Astern reaction stages have been used in Great Britain but are not favored in the United States.

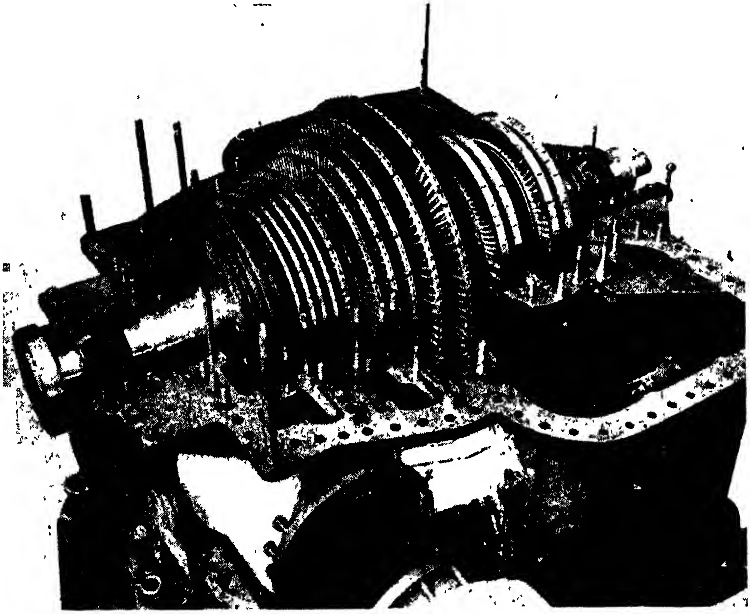


FIG. 57.—Westinghouse L. P. reaction turbine. Cylinder cover removed.

The astern turbine should be located in the low pressure end of the low pressure turbine for reasons given below. A deflector projection on the rotor or a series of guide vanes is fitted so that the exhaust steam from the astern turbine will clear the ahead low pressure blading when going astern. This location of the astern turbine provides a simple arrangement for the exhaust steam and allows the astern turbine to rotate in a vacuum, with consequent low windage loss during ahead operation.

Lloyd's rules require that two of the turbine rotors be fitted with astern blading; hence many passenger and cargo ships

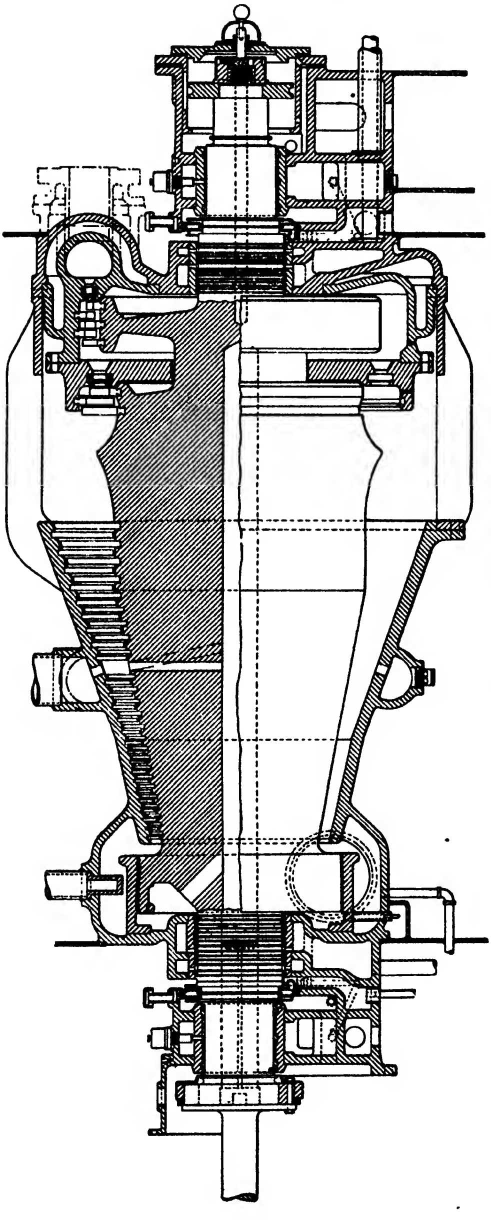


FIG. 58.—Low pressure reaction turbine. (*Bethlehem Steel Company, Shipbuilding Division.*)

(especially European) have astern blading in the intermediate casing as well as in the low pressure casing. The two groups of astern blading operate in series. When the high pressure

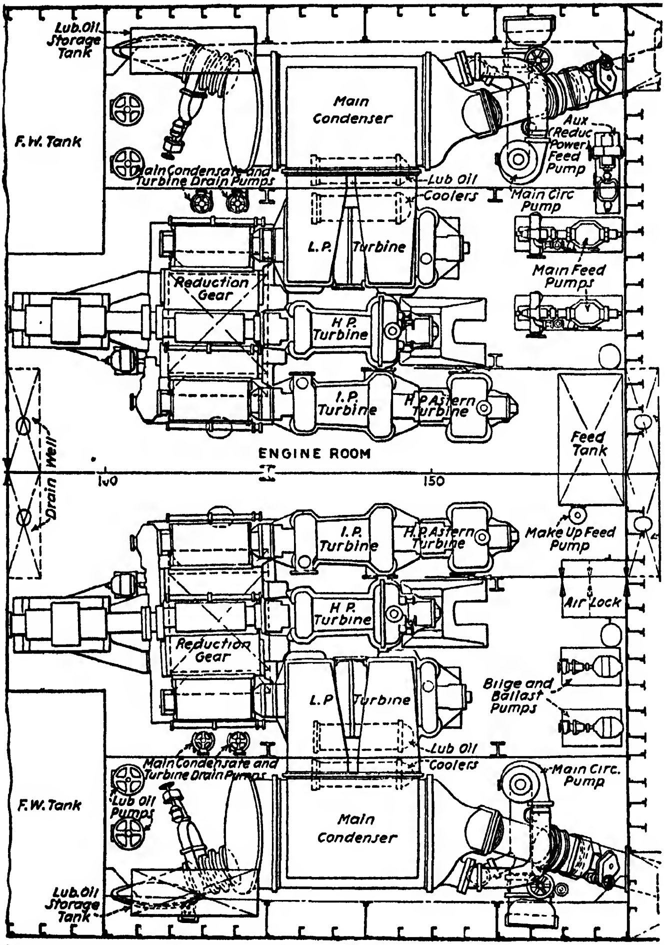


Fig. 59.—Turbine arrangement of *S.S. Manhattan.* (*Marine Engineering and Shipping Age.*)

astern turbine blading is in the same casing as the intermediate ahead blading, in order to reduce windage, the astern blading is separated from the ahead blading by a diaphragm and operates in a vacuum when the turbine is turning ahead. In the *S.S. Manhattan* (Fig. 59) a compound astern turbine is used with the high pressure astern unit in a separate casing attached to the shaft of the intermediate ahead turbine.

During ahead operations, the astern turbine blading, because of the direction of rotation, creates considerable windage, but the loss of power on account of windage is only about 1 per cent of the ahead turbine output when the astern blading is all in the L.P. casing. When astern blading is fitted in the intermediate casing, the engine efficiency of the installation is reduced somewhat through the windage of the additional astern blading.

Astern turbines are very inefficient because of the small number of stages employed, and consequently the exhaust steam entering the condenser is often superheated. In going astern, some of the gland steam will pass through the ahead blading and carry away part of the heat caused by windage. The temperature of the ahead blading may increase to a high value during astern operation, and hence care should be used to limit full-powered astern operation to a reasonable period of time.

79. Thermodynamics of the Steam Turbine.—The performances of steam turbines are usually compared by means of their engine efficiencies. The engine efficiency was defined in Art. 12 as follows:

$$\text{Engine efficiency} = \frac{2{,}545}{w_a(h_1 - h_2)}$$

where w_a = steam rate of the actual turbine, in pounds per shp-hour.

h_1 = enthalpy of steam at throttle conditions.

h_2 = enthalpy of steam after a reversible adiabatic expansion from the initial conditions to the condenser pressure.

In a geared-turbine unit used for marine propulsion, the shp is measured by a torsion meter located on one of the lengths of line shafting. Since the steam rate w_a is determined by using this horsepower, the engine efficiency thus includes the efficiency of the bearings, reduction gears, and the thrust bearing.

Engine efficiency cannot be calculated by the expression given above when steam is extracted from the turbine for feed heating, because the amounts of steam entering the throttle and condenser are not equal. Acceptance tests to determine the steam rate of a turbine are usually conducted with no steam being bled from the turbine.

The losses that take place in a turbine that result in an engine efficiency of less than 100 per cent are due to nozzle friction, blade losses, windage, leakage between stages, throttling, leaving losses, radiation, and the losses due to mechanical friction. Except for radiation, which is very small, leakage at the shaft-end glands, and mechanical friction, the losses must result in each pound of steam leaving the turbine with an enthalpy h_2' (Fig. 60) greater than that after an isentropic expansion h_2.

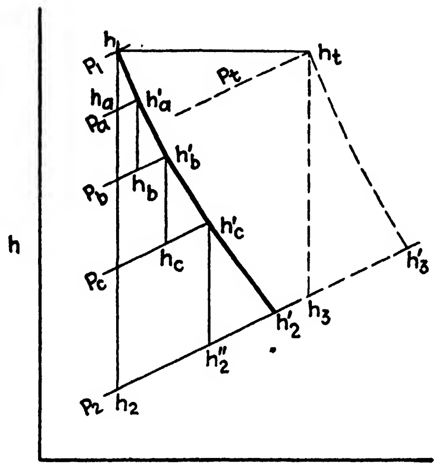


FIG. 60.

In the Mollier diagram for a multistage turbine shown in Fig. 60, let h_1 represent the initial condition of the steam at the throttle pressure P_1 and h_2, the final condition after an isentropic expansion to the condenser pressure P_2.

If P_a is the pressure at the entrance to the second stage, the available adiabatic enthalpy drop for the first stage is $h_1 - h_a$, and the steam condition at the entrance to the second stage of an ideal turbine would be h_a. The friction between the steam and the surfaces of the nozzles and blades creates a certain amount of heat, and since there is no way for this heat to leave the turbine except by radiation and in the exhaust steam, these friction losses appear as an increase in the enthalpy of the steam. This results in an increase in the quality of the steam at a given pressure and hence an entropy value greater than for an isentropic expansion. The result is that the steam at the end of the first stage is represented by the condition h'_a, and the useful work done in the stage is $h_1 - h'_a$, which is less than $h_1 - h_a$. The stage efficiency for the first stage is

$$\frac{h_1 - h'_a}{h_1 - h_a}.$$

In like manner, the available enthalpy drop in the second stage is $h'_a - h_b$, but because of the heating of steam by friction, its actual condition at the end of the second stage and entrance to the third stage is h'_b.

The steam leaves the turbine with the condition h'_2, and the actual work done per pound of steam is $h_1 - h'_2$. Hence the engine efficiency (excluding the efficiency of the bearings and gears) is

$$\frac{h_1 - h'_2}{h_1 - h_2}.$$

A curve drawn through the points on the Mollier diagram that represent the condition of the steam at the beginning of each stage and entrance to the condenser is known as the "condition curve." The condition curve in Fig. 60 is $h_1 - h'_a - h'_b - h'_c - h'_2$. Such a curve shows the quality of the steam at each stage entrance and exit throughout the turbine and hence is very useful in turbine design in determining the actual steam-flow areas through the various nozzles and blades.

If the engine efficiency is known, the end point h'_2 of the condition curve is readily calculated as

$$h'_2 = h_1 - \left[(h_1 - h_2) \frac{\text{engine efficiency}}{e_m} \right]$$

The sum of the enthalpy drops available in each stage divided by the total adiabatic drop in enthalpy, $h_1 - h_2$, is known as the "reheat factor." Thus the reheat factor RF for the turbine in Fig. 60 is

$$\frac{(h_1 - h_a) + (h'_a - h_b) + (h'_b - h_c) + (h'_c - h''_2)}{h_1 - h_2}$$

An inspection of Fig. 60 shows that the sum of the stage isentropic enthalpy drops for a multistage turbine is greater than the available isentropic enthalpy drop in a single expansion from P_1 to P_2, and consequently the reheat factor is greater than unity.

In studying the action of the steam in a turbine, it is helpful for illustrative purposes to consider the engine efficiency made up of the product of several component factors. If we neglect the losses due to radiation, throttling, and exit velocity, these factors would be

$$e_n \times e_b \times WF \times LF \times RF \times e_m$$

where e_n = nozzle efficiency.
e_b = blade efficiency.
WF = windage factor.
RF = Reheat factor.
LF = a factor to cover leakage between stages and shaft-end leakage.
e_m = mechanical efficiency.

Although such an equation can be used for analyzing the component efficiencies of a single-stage turbine, it is obvious that it does not lend itself readily for application to cases of multistage turbines operating in both saturated and superheated regions.

The windage factor *WF* covers the rotational losses that are made up mostly of disk friction and the pumping action of idle blading in stages having partial peripheral admission of the steam and in the astern blading. Windage losses are considered to be inversely proportional to the specific volume of the steam, directly proportional to the cube of the peripheral velocity and the square of the wheel diameter for disk friction and the first power of the wheel diameter for blade windage. The heights and arrangement of the blading and the side clearance of the rotating disks are also factors. Rotational losses increase with the percentage of moisture and decrease with superheat. Windage losses calculated by the various formulas in use vary considerably.

The mechanical efficiency appears in the foregoing expression because of the fact already mentioned that engine efficiency includes the mechanical friction losses in the gears, thrust block, and shaft bearings up to the torsion meter.

Factors influencing nozzle and blade efficiencies were discussed in Arts. 68 and 70. Figure 61[1] shows the approximate blade

[1] The curves in Fig. 61 for two-row Curtis turbine and the Rateau turbine have been taken from Sterling's "Marine Engineers' Handbook" and the curve for the Parsons turbine from a paper by S.S. Cook, "Modern Marine Steam Turbine Design," *Trans. Inst. Mar. Eng.*, vol. 50.

efficiencies for several types of turbines plotted against V/V_1, *i.e.*, the peripheral speed of the blades divided by the theoretical steam-jet velocity. The curves in this figure show that maximum blade efficiency of a two-row velocity-compound turbine is appreciably less than for a single Rateau stage. This is because high steam velocities are employed in velocity-compound turbines, and considerable friction is developed because of this high velocity and the longer path of the steam through the two moving

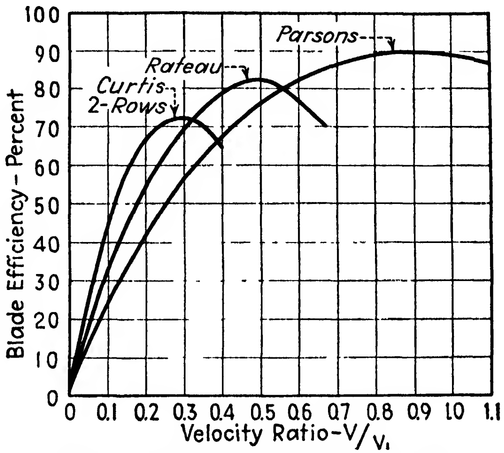


FIG. 61.—Approximate blade efficiencies.

rows of blades and the one fixed row of blades. However, the velocity-compound turbine develops its maximum blade efficiency at a V/V_1 ratio about one-half that of the Rateau turbine, which gives it a distinct advantage when low revolutions and a small number of stages are desirable. This makes it especially suitable for auxiliary and astern turbines.

The Parsons turbine shows greater efficiencies than the other types but requires a high V/V_1 ratio for good efficiency.

80. Influence of Steam Conditions on Turbine Performance.—The steam rate of a turbine depends upon (1) the difference in the enthalpy of the steam at the initial conditions and after an isentropic expansion to the back pressure and (2) the engine efficiency

$$w_a = \frac{2{,}545}{\text{engine efficiency} \times (h_1 - h_2)}$$

In Art. 15 we saw that the value of $(h_1 - h_2)$ could be increased by raising the initial pressure, increasing the amount of superheat, and lowering the temperature of the steam leaving the turbine, *i.e.*, by decreasing the exhaust pressure P_2.

Although the steam rate is a measure of the turbines' performance it is not a true criterion of over-all plant economy unless the turbines in question operate between the same initial and final conditions. As was demonstrated in Art. 15, the real measure of plant efficiency is the amount of fuel burned per shp-hour. Conditions that will give a low turbine steam rate do not necessarily give the maximum cycle efficiency or lowest fuel consumption; hence the turbine steam rate should not be considered independent of cycle efficiency.

In the previous article we saw that steam friction raised the enthalpy of the steam passing through a turbine by increasing its quality and was one of the causes for the steam expansion in the first stage to be represented by the line $h_1 - h'_a$ instead of by the isentropic line $h_1 - h_a$ in Fig. 60. The wetter the steam the greater is the friction and the more the condition line slopes off to the right. It is generally considered that stage efficiencies decrease about 1 to 1.3 per cent for each 1 per cent increase in the average percentage of moisture (average between the inlet and outlet of the stage). When the steam is superheated, steam friction due to the passage of the steam through the blading and nozzle is reduced, and the windage resistance is less.

If the entire expansion of the steam in the turbine or a large portion of it took place above the saturation line on the Mollier diagram, the engine efficiency would be higher and the condition line would lie closer to the isentropic line in Fig. 60 than for a turbine where all or a large part of the expansion took place below the saturation line. The efficiency of an individual stage is not increased by the degree of superheat; so long as the steam is above the saturation point the steam friction is reduced. However, a high degree of superheat at the throttle ensures that the steam will be above the saturated condition for a greater part of its passage through the turbine.

The purpose of the reheating cycle is to be able to carry superheat further through the turbine without the necessity of using a high initial temperature when high boiler pressures are adopted. The steam is withdrawn from the turbine when its expansion has brought it down to the saturated condition. It is sent back to the boiler to be resuperheated and then returned to the turbine to continue its expansion through the remainder of the blading to the condenser.

In Art. 15 we saw that the use of superheated steam did not increase the cycle efficiency of the ideal turbine as much as an increase in the steam pressure did. From the foregoing discussion, it is evident that superheat results in a practical gain by increasing the engine efficiency and hence bringing the actual cycle closer to the ideal cycle.

By reference to the Mollier diagram, it will be seen that superheating the steam increases the entropy and moves the expansion line of the ideal turbine further to the right, resulting in a higher quality of the steam at exhaust and a greater amount of enthalpy rejected to the condenser. Increasing the pressure, on the other hand, decreases the entropy and results in less enthalpy in the steam rejected to the condenser. Superheat increases the engine efficiency but does not increase greatly the cycle efficiency of the ideal cycle; increasing the pressure decreases the engine efficiency (by increasing the wetness of the steam in the lower pressure stages) but increases the cycle efficiency of the ideal turbine. Hence the true criterion must be the thermal efficiency which is the product of the cycle efficiency and the engine efficiency.

In practice, both steam pressure and superheat are increased to give the highest values of cycle efficiency and engine efficiency within the limits set by practical considerations of maximum temperature, blade erosion, and other economic factors.

In addition to increasing steam friction, wet steam causes blade erosion, and practical considerations of maintenance generally set the lower limit for the quality of the steam at the entrance to the condenser from 87 to 89 per cent. Blade erosion not only increases maintenance expense but also causes a loss of engine efficiency in service.

A common procedure today is to set a limit on the percentage of moisture at exhaust of around 11 to 13 per cent and then to determine the maximum initial temperature that metallurgical conditions will allow or some temperature below this that it seems desirable to use. If these two conditions are set and the engine efficiency can be closely estimated, the steam pressure and degree of superheat are fixed.

Thus suppose that for a 28.5-in. vacuum we fix upon the limits of moisture at exhaust as 12 per cent, the upper desirable limit of temperature as 700°F and assume the engine efficiency (excluding gears and bearings) to be 0.80. (Including gears and bearings, the engine efficiency would be $0.80 \times 0.97 = 0.776$.) By a series of calculations it will be found that the desired conditions are fulfilled with an initial pressure of 525 lb abs and a superheat at the throttle of 228 degrees.

For initial pressure of 525 lb abs and a temperature of 700°F, $h_1 = 1{,}356$.

After an isentropic expansion to $1\frac{1}{2}$ in. of mercury, $h_2 = 882$.

$$h_1 - h_2 = 474$$
$$h_2' = h_1 - [(h_1 - h_2) \times 0.80] = 977 \text{ Btu}$$

On the Mollier diagram, the quality at $1\frac{1}{2}$ in. pressure and an enthalpy value of 977 is 12 per cent.

$$w_a = \frac{2{,}545}{0.776 \times 474} = 6.93 \text{ lb per shp-hr}$$

If the upper limit of moisture had been fixed at 11 per cent, a lower boiler pressure would have resulted.

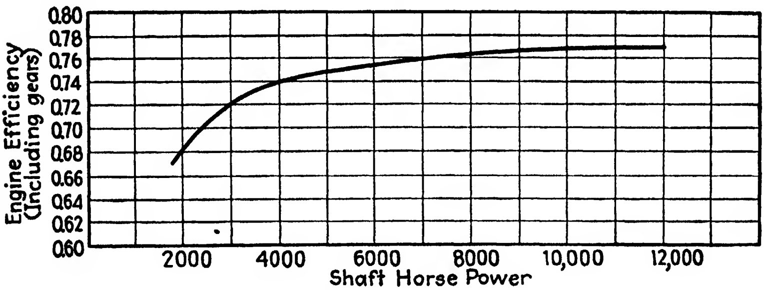


FIG. 62.—Engine efficiencies.

Other factors besides superheat that influence the engine efficiency of a turbine at full power are the shp of the turbine, the number of stages, the rpm, and the design and operating conditions effecting leakage between stages, shaft-end leakage, etc. The engine efficiency decreases with horsepower, especially in turbines of small horsepower, because such losses as windage, interstage leakage, and gland and dummy leakage are greater in percentage for small turbines than for turbines of large horsepower.

Figure 62 shows approximate engine efficiencies plotted against shp for marine turbines operating with throttle pressures of around 400 lb, steam temperatures of around 700°F, and vacua of 28.5 in. The steam temperatures and pressures for the smaller horsepower units were in some cases somewhat lower than those of the larger units (see also Fig. 140).

81. Number of Stages.—In order to obtain good turbine efficiency with moderate turbine revolutions and blade velocities, many stages will be required. As pointed out in Art. 70, in order to obtain high blade efficiency with pressure-compound turbines, the jet velocity at each stage should be such that the ratio of V/V_1 is around 0.4 to 0.5. The jet velocity, in turn, fixes the enthalpy drop required per stage. If the wheels in all the stages of a pressure-compound turbine have the same peripheral velocity and Δh is the average enthalpy drop required per stage to obtain the jet velocity, the number of stages is:

$$n = \frac{(h_1 - h_2)RF}{\Delta h}$$

where RF = reheat factor and h_1 and h_2 have the previously defined meanings.

The numerator of the preceding equation represents the total energy available in a multistage turbine. Since superheat, moisture, and windage influence stage efficiency, equal enthalpy drops are not always allotted to each stage, as suggested by the preceding equation.

The engine efficiency of a reaction turbine is sometimes plotted against the expression

$$\frac{\Sigma n\, D^2 \times \overline{rpm}^2}{(h_1 - h_2)10^6}$$

where n = number of stages.

D = pitch diameter of the moving blades, in inches.

$\Sigma n\, D^2$ = sum of squares of pitch diameter of all the moving blades.

h_1 and h_2 = enthalpies for initial conditions and final conditions after an isentropic expansion.

Engine efficiencies increase as the foregoing expression increases, and they will be about the same for equal values of this expression.

This expression is useful in that it shows the flexibility of turbine design. A turbine can have a small number of stages of moderate diameter and a high rpm or a large number of stages and a low rpm and still have the same engine efficiency.

82. Nozzles and Diaphragms.—The cross sections of turbine nozzles can be either circular or rectangular. Circular nozzles are easily reamed, and a smooth surface is thus obtained. Nozzles of rectangular cross section can be placed closer together than those of circular cross section, and since they give a more continuous band of steam entering the blading, they give higher turbine efficiencies than circular nozzles, provided that the nozzle walls are smooth surfaces. In order to obtain smooth surfaces, rectangular nozzles are often built up of machined surfaces welded together or riveted. Rectangular nozzles are shown

FIG. 63.—Nozzle plate for astern turbine. (*Allis-Chalmers Manufacturing Company.*)

in Figs. 51 and 64. A nozzle block made up of circular nozzles is shown in Fig. 63.

Steam enters an H.P. marine-propulsion turbine through several passages, all but one of which are controlled by nozzle valves. Each nozzle valve regulates the flow of steam through a group of nozzles varying from 1 to about 20. These nozzles are located opposite the passage from the valve. The flow of steam to the first stage and hence the power output of the turbine are controlled by these valves. The nozzles are usually constructed in blocks of several nozzles, each of which is bolted to the turbine casing or steam chest. The hand wheels controlling the nozzle valves are shown at the right-hand side of Fig. 52.

The exit height H, in inches, of rectangular nozzles can be determined by the following formula, when there is full peripheral admission:

$$H = \frac{144 \times w \times \text{shp} \times xv}{3{,}600 \times V_1 \sin \alpha \times \pi D_p \times m}$$

where w = steam rate, in pounds per shp-hour.

v = specific volume of the steam, in cubic feet per pound.

D_p = pitch diameter of the blades, in feet, at nozzle exit.

α = nozzle angle (Fig. 44).

V_1 = absolute velocity of steam leaving the nozzle.

m = a factor to allow for the thickness of the nozzle-division walls.

Diaphragms having rectangular cross-section nozzles are often constructed by casting wrought-steel nozzle-division plates in

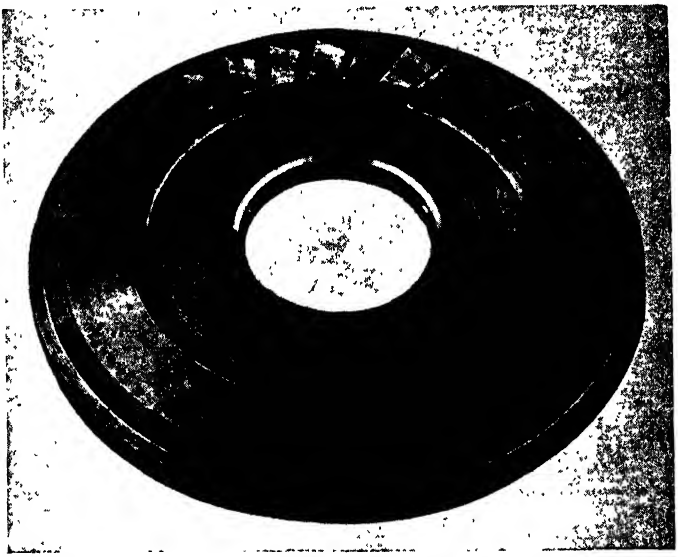


Fig. 64.—Interstage diaphragm with partial admission; for astern turbine. (*Allis-Chalmers Manufacturing Company.*)

the diaphragms. Where temperatures and stresses are low, cast iron can be used for diaphragms. For more severe conditions, cast steel is employed. Diaphragms may be constructed for full or partial peripheral admission. A diaphragm for partial admission is shown in Fig. 64. Diaphragms are usually made in two pieces, with the joint at the center line of the shaft corresponding to the horizontal joint in the turbine casing. The outer circumferences of the diaphragms are secured to the turbine casing (see Fig. 51). At the shaft the diaphragms are provided with labyrinth type packing to reduce steam leakage between stages.

83. Turbine Blading.—Steam turbine blades are subject to centrifugal forces, impulse forces, and to stresses due to blade vibration. Differences in expansion of rotor, casing, and the blading may cause actual rubbing, which may reduce the effectiveness of the moving blades or root fastenings. Uneven and severe temperature changes and the strains set up during starting or during astern operation have resulted in blade failure. In selecting material for turbine blades, consideration must be given to such factors as strength at operating temperature,

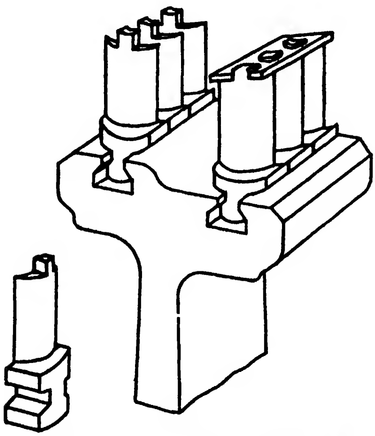


FIG. 65.—Curtis-type blading.

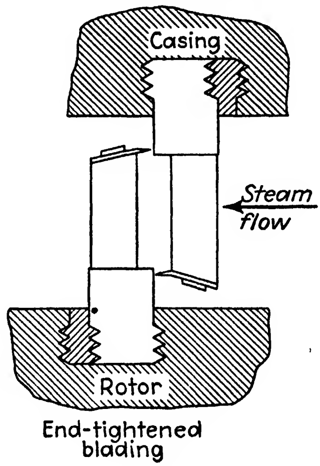


FIG. 66.

resistance to erosion, ease of fabrication, etc. Stainless steel (or iron) is favored where temperatures are high and the service is severe. Bronze blading is suitable for moderate temperature conditions and moderate velocities. Various methods are used for fastening the blades to the turbine casing and the disks or rotor. Several methods are shown in Figs. 65 to 68.

The common practice with impulse turbines is to use a flat type of shroud bands having blade tips riveted to the bands, as shown in Fig. 65. The bands are not continuous but are fitted in short sections, 4 to 12 in. long, to allow for changes in temperature. In some recently constructed turbines, the shroud ring has been welded to the blade tips.

The blading of the Parsons type of reaction turbine requires small radial clearance because of the pressure drop through the fixed and moving blades. In order to avoid small radial clearances and the possibility of the ends of the blades rubbing, Parsons developed the end-tightened blade (Fig. 66). The end tightening consists in fitting shroud rings to the fixed and moving blades, which have overhanging side extensions. These extensions are tapered to a fine edge and nearly touch the foundation ring on the adjacent blade. The axial clearance between the shroud bands is carefully adjusted so that only a small area for leakage exists. Thus the end-tightened blades do not require a small radial clearance, since the area for leakage is now an axial one instead of radial.

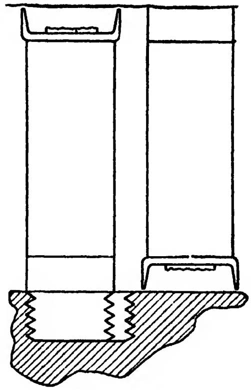


Fig. 67.

The Bethlehem and Allis-Chalmers reaction turbines use a channel shroud band with very thin edges in place of the end-tightened blades (Fig. 67). The Westinghouse turbine employs two thin sealing strips fitted into the casing opposite each row of moving blades and

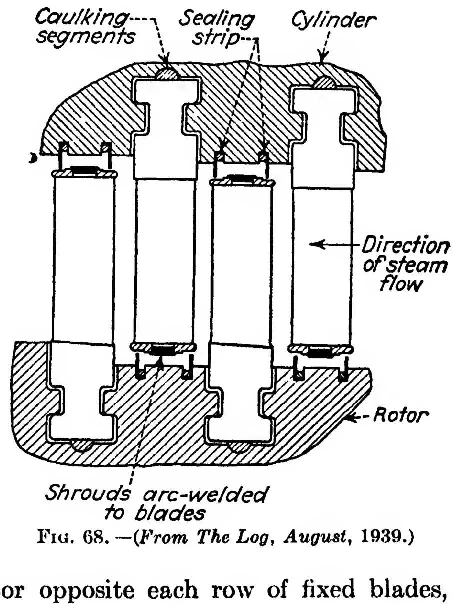


Fig. 68.—(*From The Log, August, 1939.*)

into the rotor opposite each row of fixed blades, as shown in Fig. 68.

When peripheral velocities of blades are high and the blades are long, a tapered blade section is desirable, in which case the root stresses can be reduced. Blades of this type are often fitted in the last rows of the L.P. turbine (see Figs. 53 and 55).

84. Turbine Rotors and Casings.—As noted previously, the disk-type rotor is the normal construction for the impulse turbine and the drum form for the reaction turbine. Stresses set up in a turbine rotor depend on the shape of the rotor, the peripheral

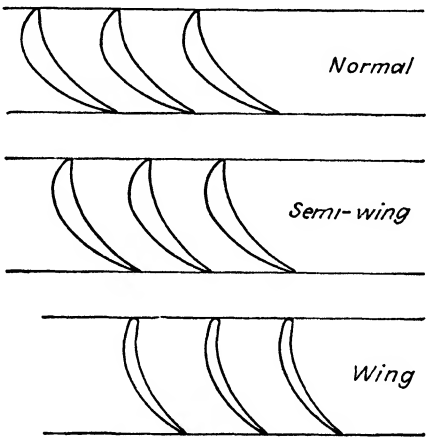


FIG. 69.—Types of Parsons blading.

velocity, and the force produced by the blading. In recent years there has been a distinct trend towards higher revolutions and peripheral velocities, and as a result the rotors of impulse-type turbines, which were formerly made up of disks keyed to the turbine shaft, have been replaced by rotors with disks and shaft machined out of a solid forging (see Fig. 53). This type of construction is lighter, permits a better diaphram packing design, and reduces the danger of slight shifting of the disks on the shaft, which would disturb the initial balance of the rotor.

Reaction-turbine rotors are often solid forgings (see Figs. 50, 54, 58), although built-up hollow rotors with the shaft ends attached to the hollow drum have been used (Fig. 84). If the hollow rotor of a reaction turbine is fairly long, of relatively large diameter, and is operated at high velocity it would be subject to high stresses from centrifugal force. For such conditions disk and shaft construction or a modified disk construction can be used with the outer rotor surface smooth as in the case of the drum-type design.

The casing of a turbine carries the stationary blading in velocity-compound and reaction turbines and supports the diaphragms in pressure-compound turbines. In all large turbines, the casing, shaft-bearing housings, and gland-packing casings are rigidly bolted together to ensure accurate alignment. In order to reduce distortion through changes in temperature, the casing is given as symmetrical a shape as possible and supported at the bearings rather than at the bottom of the casing. In turbines where high steam temperatures are employed, the first-stage nozzles and the stationary blading of the first stages are often constructed so that they can expand independently of the main turbine.

Cast steel is used for casings subjected to temperatures above 450°F. For temperatures less than 450°F, cast iron is usually employed, because it is less expensive and is easier to cast and machine. Thus the high pressure casing and the astern section of the low pressure casing are normally constructed of cast steel. The portion of the turbine casing forming the outer exhaust connection to the condenser is frequently made up of wrought-steel plates welded to the casing.

For convenience in assembly and inspection, the casings of multistage turbines are split or divided along a plane through the shaft axis. Often the steam connections to the turbine are fitted to the bottom half of the casing so that the upper half can be lifted for inspection without breaking pipe connections.

85. Governors and Methods of Controlling the Power Output of Turbines.—Two types of governors may be used with propulsion turbines: a speed-regulating governor and an emergency governor.

Since the propeller torque increases more rapidly than the turbine torque with an increase in rpm, a speed-regulating governor is not required for normal operation. Some marine-propulsion turbines are fitted with speed-regulating governors to prevent the turbine from overspeeding when the propeller is lifted partially out of the water during pitching. Since the turbine rotors and gears in a geared-turbine installation have an inertia or fly wheel effect and the time for acceleration during normal pitching is short, many marine engineers in the past have been of the opinion that speed-regulating governors are unnecessary on propulsion turbines. They are, however, fitted on turbines driving auxiliary generating sets, blowers, and centrifugal pumps (see Fig. 47).

The American Bureau of Shipping now requires an emergency governor to be fitted on propulsion turbines, although many installations are in operation without them.

Figure 70 shows a view of a speed-regulating governor for a General Electric marine-propulsion turbine that functions also as an emergency governor. Under normal operating conditions, oil pressure is maintained under the spring-loaded piston in the "operating cylinder" (shown in the upper center of Fig. 70). This oil pressure holds the double-seated "maneuvering and throttle valve" open. An excessive overspeeding of the turbine causes the revolving weights on the speed governor to move outward and open a valve connecting with the lower part of the "operating cylinder" and allows the oil to drain out. The spring now pushes down the piston of the operating cylinder, and the throttle valve is closed.

In Fig. 70 the throttle is also under the control of a "low oil pressure trip valve" (shown at upper right-hand side of the diagram). If for any reason the level of the oil in the lubricating-oil gravity tank, which supplies the main reduction gears, should become low, this valve will operate to drain the oil from the "operating cylinder" and hence close the main throttle and thus prevent damage to the gears.

When speed-regulating governors are fitted to auxiliary generating sets, two types are available: throttle governing and nozzle cutout governing. In throttle governing, the governor mechanism regulates the flow area through a single throttle valve, which controls all the steam entering the turbine. Thus an increase in turbine speed is followed by a partial closure of the throttle valve, which reduces the quantity of steam flowing through the valve and hence lowers the revolutions.

When nozzle cutout governing is employed, the first-stage nozzles are arranged in groups with steam flow to each group controlled by separate valves. The governing mechanism is constructed so that it will close valves to these nozzle groups one at a time. Thus, at fractional powers, steam to some of the nozzles is entirely shut off, and steam at practically full pressure

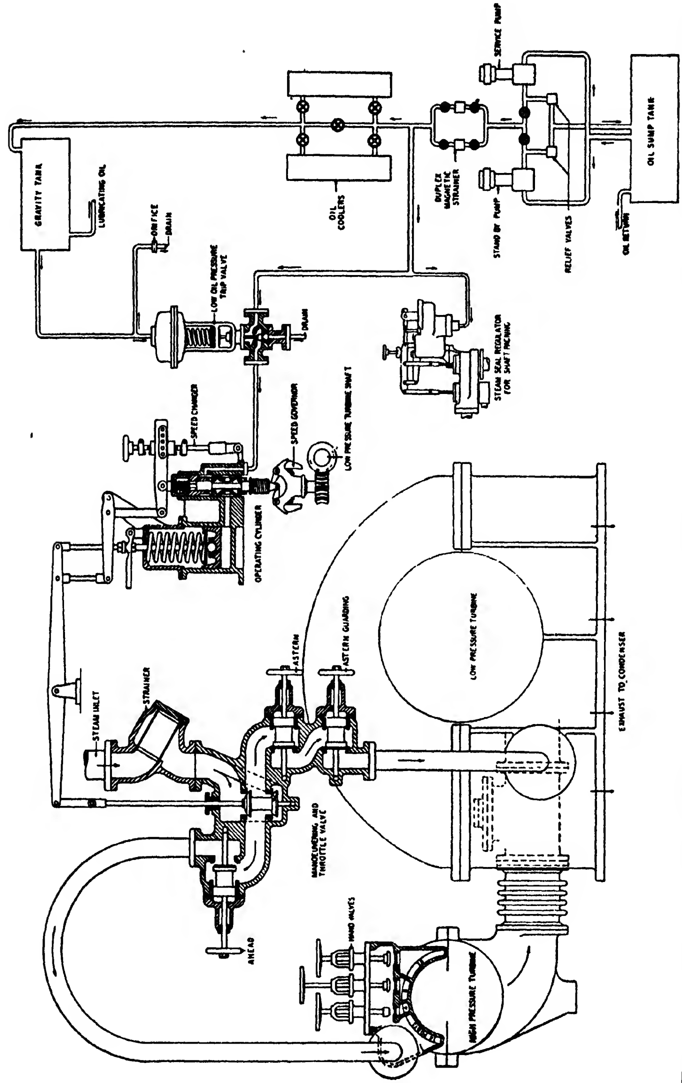


FIG. 70.—Diagram of turbine-governing mechanism. General Electric Company turbine. (*The Nautical Gazette, June*, 1940.)

acts on the other nozzles except for one group, which may be partially throttled. Thus most of the first-stage nozzles in use operate at the steam pressure for which they were designed. As shown below, nozzle cutout governing is a more economical method than throttling.

When it is desired to reduce the speed of the ship and hence the power output of the turbine by manual control, the common methods employed are (1) partial closing of the throttle valve and (2) cutting out first-stage nozzles.

Throttling, which is a reduction in pressure at constant enthalpy, lowers the pressure at the entrance to all the first-stage nozzles or guide blades, and this results in a reduced flow of steam into the turbine. The influence of throttling is shown in Fig. 60 in a somewhat exaggerated form. If the steam is reduced in pressure from P_1 to P_t by throttling, the condition after throttling is represented by point h_t. An isentropic expansion to the back pressure P_2 will result in condition h_3. It is clear from Fig. 60 that isentropic enthalpy drop after throttling $(h_t - h_3)$ is less than that without throttling $(h_1 - h_2)$. The condition line for the actual turbine after throttling would be represented by a line such as $h_t - h_3'$. This reduction in the isentropic enthalpy drop will naturally result in less enthalpy per pound being converted into work and hence an increase in the steam rate of the turbine where

$$w_a = \frac{2{,}545}{\text{engine efficiency } (h_t - h_3)}$$

As pointed out, when the power is reduced by cutting out nozzles, the remaining nozzles in use will operate under the same steam pressure for which they were designed, and no loss in isentropic enthalpy drop takes place, as is the case with throttling. However, the reduced steam flow at fractional powers will reduce somewhat the efficiency of stages beyond the first one, since their flow areas will not be correct for this condition. Thus nozzle-cutout governing is most efficient for turbines having few stages. It cannot be used with pure reaction turbines because full peripheral admission must be used with this type of turbine.

In marine-propulsion turbines that have a first-stage impulse wheel, hand-control valves are usually fitted so that the turbine output can be regulated by cutting out first-stage nozzles. Thus in Fig. 70, 4, 8, and 16 nozzles are under the control of the left-hand, center, and right-hand valves, respectively. All but one of these valves should be wide open or closed, the close adjustment for the desired power being obtained by regulating only one of them. The nozzle-control-valve combination should be used that will give nozzle-bowl pressures most nearly approaching the steam pressures at the throttle.

86. Glands and Drains.—Packing must be fitted where the turbine shaft extends through the casings in order to reduce

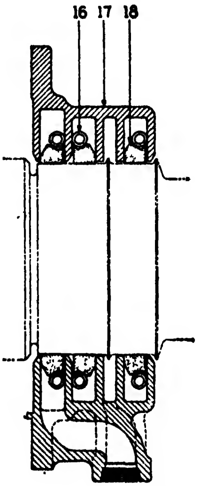


FIG. 71*a*. —Carbon-type shaft gland packing. (*General Electric Company.*)

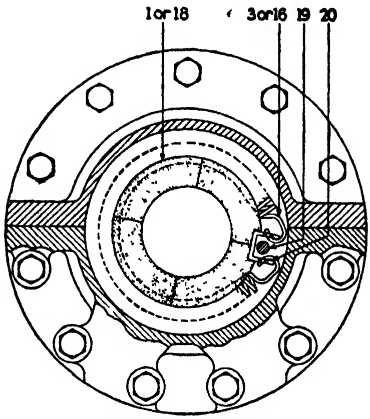


FIG. 71*b*.—Section through carbon packing. 18 indicates carbon rings; 16 indicates spring. (*General Electric Company.*)

steam leakage out or air leakage in, just as piston-rod packing must be provided for in double-acting steam engines.

In the ordinary two-casing marine steam turbine, which operates with inlet pressures of over 200 lb, the steam pressure at the exhaust end of the high pressure turbine will be above atmospheric. Hence steam will tend to leak out around the shaft at both ends of the high pressure casing and also from the high pressure end of the low pressure casing, which is at the same pressure as the exhaust end of the high pressure casing. The astern turbine is placed at the exhaust end of the low pressure turbine where shaft leaves the casing (see Figs. 55 and 58). Since the astern turbine is under a vacuum during ahead operation, there will be a tendency for the air under atmospheric pressure to leak into the casing around the shaft.

In order to reduce shaft-end steam leakage from the turbine to a minimum and thus improve the efficiency of the turbine and also to reduce air leakage into the turbine, gland packing (Fig. 71) is fitted around the shafts at the ends of the turbine casings. These glands are made tight either by packing made up of from three to five rings of carbon (Fig. 71) fitted to the shaft with very small radial clearance or by labyrinth packing. The

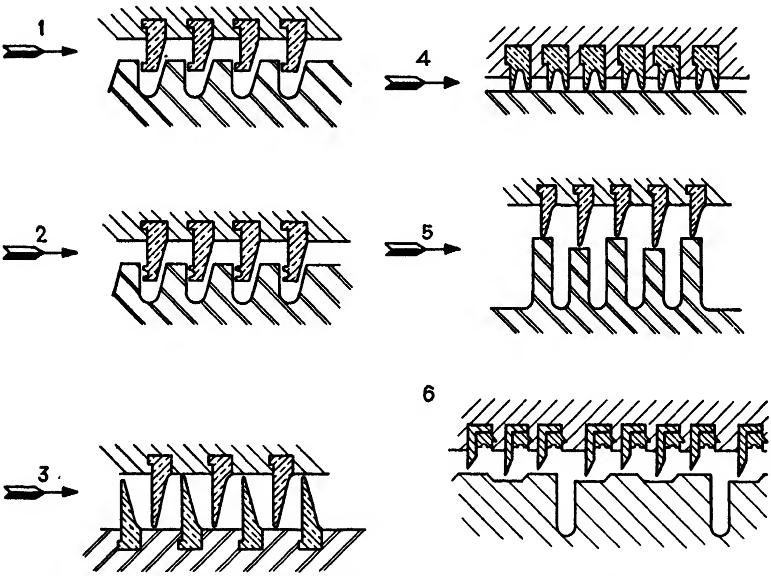


FIG. 72.—Types of dummy sealing strips developed by Westinghouse Electric & Manufacturing Company. (*From The Log, August*, 1939.)

latter type of packing consists of numerous metallic strips with very close clearances (Fig. 72(3)). The practice in the use of packing varies among turbine builders. Figures 54 and 55 show carbon packing used at both ends of the high and low pressure turbines. Figure 53 shows carbon packing at both ends of a low pressure turbine, and the low pressure turbine shown in Fig. 58 uses labyrinth packing at both ends. The greatest tendency of shaft-end leakage is, of course, at the high pressure end of the high pressure turbine. In reaction turbines where dummy pistons are fitted, labyrinth packing is used on the dummy, and this assists in preventing leakage. In impulse turbines, where no dummy pistons are used, both labyrinth and carbon packing may be fitted at the high pressure end of the H.P. turbine (see Fig. 52).

Dummy pistons are fitted with various types of sealing strips, a form of labyrinth packing. Types of sealing strips developed by Westinghouse are shown in Fig. 72, which has been taken from a paper by J. B. Walbridge.[1]

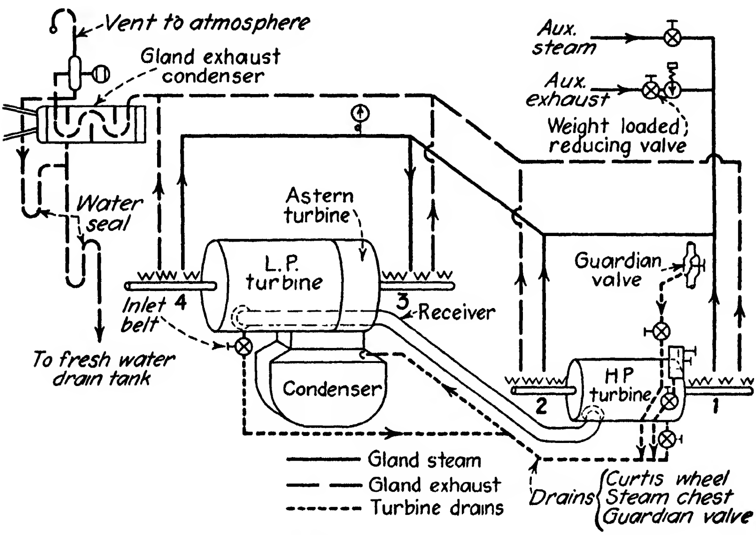


FIG. 73.—Diagram of gland steam piping and turbine drains.

In discussing these sealing strips, Mr. Walbridge points out that types 1 and 2 with axial clearance were the earlier types used, and these were followed by types 3, 4, and 5. Type 6 is the latest type of sealing used by Westinghouse. The steps in the rotor in type 6 are to break up any direct axial flow and increase turbulence and thus cause a greater pressure drop between the strips. Type 3 is used to a considerable extent for dummies and by some builders for shaft-end glands. Labyrinth packing is provided with greater clearances than carbon packing and depends on the greater number of strips to make it effective.

A detail of the carbon rings used by the General Electric Company for shaft-end packing is shown in Fig. 71.

[1] *The Log*, August, 1939.

When a turbine is operating ahead at full power, sealing steam under low pressure is introduced into a pocket within the gland at the astern end of the L.P. turbine. This steam is wire-drawn and is reduced in pressure as it leaks by the carbon rings or the labyrinth packing strips. The slow leakage of this steam into the turbine and toward the outer end of the shaft prevents air leakage into the turbine. Steam leaking from the high pressure glands can be used for sealing the gland at the vacuum end of

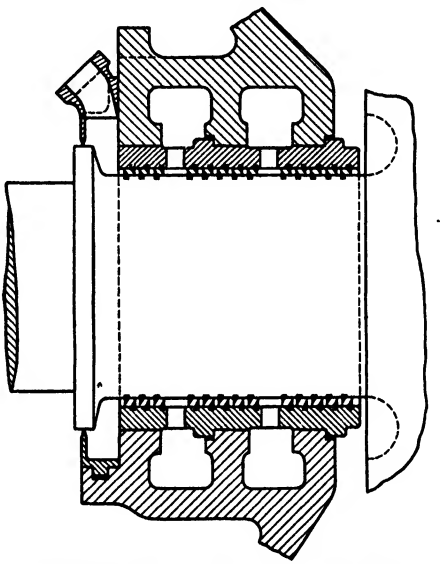


FIG. 74.—Shaft-end gland showing gland pockets and labyrinth packing.

the low pressure turbine. Such an arrangement is known as "self-sealing" (see Fig. 74).

A diagram of a gland system is shown in Fig. 73. The full lines in Fig. 73 show the direction of flow of the gland steam when operating ahead. When operating astern, all the shaft glands but the one at the astern stages (gland 3) will be under a vacuum. Under these conditions, steam from the auxiliary steam line or auxiliary exhaust will have to be supplied to glands 1, 2, and 4, since usually there will not be sufficient steam leaking from gland 3 to supply the other three glands.

Steam pressure in the gland system is maintained at about ½ to 2 lb per sq in. gage, this being sufficient to seal against air leakage. Naturally, gland steam will tend to leak out along the shaft. With some turbines it is allowed to escape into the engine room. This practice, however, increases the engine-room humidity and also causes a loss of fresh water from the feed-condensate system. To avoid these disadvantages, the glands of most main propulsion turbines are made in three sections, with two pockets between the sections (see Fig. 74).[1] The inner pocket is connected to the gland-sealing system mentioned above. The outer pockets are intended to collect steam leaking along the shaft from the inner pockets and are connected to a gland leak-off condenser. (See Fig. 73.)

The pressure in the gland exhaust system is maintained about $\frac{1}{3}$ lb below atmospheric pressure by connecting this system to a small gland exhaust (leak-off) condenser. Air is removed from this condenser, and the vacuum is maintained by use of an exhaust fan or steam jet. Any gland steam that is condensed is returned to the feed system. The gland leak-off system is indicated by dotted lines in Fig. 73. The main feed on its way to boiler is usually used as the circulating water for this condenser (see Fig. 130).

With impulse turbines it is necessary to reduce steam leakage by the annular space between the diaphragms and the turbine rotor. Since space is limited and such glands can be given attention only during a major overhaul period, a rugged type of labyrinth packing is used.

Reaction turbines are generally fitted with dummy cylinders (Art. 75) to reduce the axial thrust developed by reason of the drum type of rotor and the pressure drop in way of each moving row of blades. Steam leaking by the dummy cylinder does no work and must be reduced to a minimum. Various types of labyrinth packing are used for this purpose.

Pockets in steam lines and parts of the turbine casings that will not drain automatically are fitted with individual drain lines. The common arrangement of condenser under the L.P. turbine makes the astern turbine and last stages of the L.P. turbine self-draining.

The drains usually fitted to a combined impulse and reaction turbine are shown in Fig. 73. These consist of drain from steam chest, Curtis wheel, guardian valve (*i.e.*, special stop valve

[1] Taken from a paper by S. S. Cook, "Modern Marine Steam Turbine Design," *Trans. Inst. Mar. Eng.*, vol. 50.

in steam line to astern turbine), space between dummy and shaft-end packing, gland pockets, and receiver pipe.

87. Turbine and Propeller Revolutions.—As pointed out in the foregoing articles, turbines require comparatively high revolutions or a large number of stages for good efficiency when high steam pressures are employed. High steam pressures should be accompanied by high or moderately high steam temperatures (Art. 80). Since a large number of stages results in a long rotor and casing with possible temperature expansion difficulties, the present American trend is to use a moderate number of stages in two turbine casings.

In the last stages of a low pressure turbine, sufficient steam-flow area must be provided to pass a large volume of steam without employing high steam velocities with the accompanying losses. Such steam-flow area among other factors is proportional to blade height and pitch diameter. Thus a high-powered turbine, because of the large volume of steam passing through the low pressure blading, requires a low pressure rotor of large diameter. Since blade speeds are chiefly limited by stresses arising from centrifugal force, the peripheral velocity of the blading is limited, and for blades of the same construction the maximum blade speed is approximately the same for turbines of all powers. Hence high-powered turbines of large diameter operate at lower revolutions than turbines of smaller power and smaller diameter.

The amount of steam flowing through a high pressure turbine is a function of the horsepower, and hence for full peripheral admission the lower the power the smaller will be the annular-flow area of the blading necessary to pass the steam and the smaller will be rotor diameter. It is obvious that for a given peripheral speed (with other conditions remaining the same) the smaller the horsepower the higher will be the rpm of the high pressure turbine. The diameter of the high pressure rotor and hence its rpm will, of course, depend on whether full or partial admission of the steam is employed and also on the number of stages and hence the steam velocity used.

Because of better materials and extended experience, the revolutions of high pressure turbines have steadily increased in recent years, and at the present writing high pressure turbines for the merchant service for powers ranging between 2,000 and

8,000 shp run at rpm from 7,000 to 4,500. With turbines of larger powers, the revolutions are lower (see Table X).

To obtain a high propeller efficiency with a slow-speed cargo ship, low propeller rpm is required. The optimum rpm can seldom be obtained with this class of ship because of the impossibility of fitting a propeller of sufficiently large diameter. The rpm used with low-speed single-screw cargo ships ranges between 75 and 90. The faster the ship the higher are the optimum propeller revolutions.

When the turbine was first introduced, it was connected directly to the propeller shaft (Art. 67). The propeller was run at revolutions above the optimum and the turbine at revolutions lower than necessary for maximum efficiency; hence both propeller and turbine efficiency were sacrificed. This type of drive was fitted to fast passenger ships and naval vessels, but the low propeller revolutions required with slow-speed cargo ships made the direct-connected turbine unsuitable for this type of ship, and practically no direct-connected turbines were fitted in slow-speed merchant ships.

The foregoing factors led to the introduction of the mechanical reduction gearing for use with marine turbines. Both single and double reduction gears are used, the choice depending on several factors, as discussed in Art. 90. Table X gives data on turbine and propeller revolutions and data on reduction gearing for a number of typical ships.

88. Types of Reduction Gearing.—Mechanical reduction gears can be classified in the three following divisions:

1. Single or double helix.
2. Single or double reduction.
3. Articulated or nested.

With single helical gears (Fig. 40), there is an appreciable axial or end thrust. This can be reduced by using a small helical angle, but the lower efficiency of the single helical type offsets its simplicity and cheaper construction. Hence this is normally used only for auxiliary drives.

With double helical gears (Fig. 77), end thrust is avoided, and the driving pinion can move fore and aft slightly to equalize the tooth pressure. This type of gearing shows an efficiency of around 98.5 per cent for single reduction gears and 97 per cent

TABLE X

Ship	Date	No. of screws	Shp, each screw	Prop. rpm	Turbine data: Turbine press. temp.	Units	Type	Rpm	No. of stages	Gear data: Gear type	Reduction ratio	High-speed pin. Dia., in.	High-speed pin. Length, in.	Low-speed pin. Dia., in	Low-speed pin. Length, in.	Gear-wheel dia., in.	References†
America	1940	2	17,000	128	400 lb	H.P.	Impulse	3,300	8	D R.	25 78	13	30	14 8	51		*S.N.A.M.E.*,
					715°F	I.P.	Reaction	1,500	46	S R.	11 72	..		14 8	51	174	1940
						L P.	Reaction	1,500	20	S R.	11 72	...		14 8	51		
Associated	1938	1	3,600	85	375 lb	H.P.	{ Impulse	6,285	8	D R	73 95	..					*M.E.S.A.*,
					725°F	L.P.	{ Pres.-comp.	4,919	8	D R	57 87						vol. 43
Boston	1923	2	3,800	140	200 lb	H P.	Reaction	1,990		S R	14 21			8 25	34	117 25	S N.A.M.E.,
					389°F	L P.	Reaction	1,990		S.R							1930,
																	Plate 63
Bremen	1930	4	23,000	180	330 lb	H P.	Reaction	1,800		S R.				5 13			*S B.*, vol. 37,
					700°F	I.P.	Reaction	1,800		S R.	10	..		5 13		54	p 787
						L P.	Reaction	1,800		S R.				10 26			
City of Roubaix	1928	1	4,500	85	300 lb	H P.	Impulse-reaction	1,837		S R							*S.B.*, vol. 37,
					570°F	I.P.	Reaction	1,837		S R	21 61						p. 332
						L P.	Reaction	1,837		S.R.							
Delbrasil	1940	1	7,800	105	450 lb	H P.	{ Impulse	6,487	9	D R	61 79						*M.E.S.R.*,
					750°F	L.P.	{ Pres -comp	3,809	7	D R	36 28						July, 1940
Dixie	1928	1	8,000	90	350 lb	H.P	{ Impulse	3,240		D.R.							S N A M.E.,
					633°F	L P.	{ Pres.-vel comp	3,240		D R	36	12 8	28	18 5	60	128	1930, Plate
																	63
Empress of Britain	1931	4	20,800*	150*	375 lb	H P.	Impulse-reaction	1,365*		S R		19 75*					*S.B.*, Special
					700°F	I.P.	Reaction	1,365		S R	8 8	19 75	56*			174*	No., 1931
						L P	Reaction	1,365		S R.		19 75					

Examiner	1941	1	8,000	96	1,200 lb	H P	Impulse	8,010	6	D.R.	83 48	7 1	14 5				S.N.A.M.E.,
					740°F	I.P.	Impulse-reaction	5,035	20	D.R.	52 34	11 3	14 5	18 75	41 75	154	1941
						L.P.	Reaction	4,201	15	D.R.	43 75	10 3	17 0				
Exporter	1940	1	8,000	96	425 lb	H.P.	Impulse-reaction	4,450	23	D R.	46 33	10 2	21 5	20	34	149	S N.A.M.E.,
					740°F	L P.	Reaction	2,310	25	D.R.	24 06	19 6					1941
Gneisenau	1936	2	13,300	129	720 lb	H P.	Impulse	7,300	6	D.R.	56 75						W.R.H.,
					840°F	I.P.	Impulse	7,300	6	D.R	56 75						Mar 15,
						L P.	Impulse	2,180	12	S R	16 9						1936
King George V	1926	2	1,750	570	500 lb	H P	Reaction	6,000	17	S R.	10 52			5 13			*M.E.S.A*,
					750°F	2-I P.	Reaction	6,000	15–24	S R.	10 52	x	x	5 13		54	vol 31, p
						L P.	Reaction	3,000	13	S R.	5 26			10 26			561
Mormacport	1939	1	8,500	85	440 lb	H.P.	Impulse	5,012	7	D.R.	53	9 8	18 625	17 92	38 5	153 752	
					740°F	L P.	Impulse	3,459	11	D.R	40 7	14 2	18 625	17 92	38 5		
Orontes	1930	2	10,000	96	215 lb	H.P.	Impulse-reaction	1,388	51	S R.		*					*S.B*, vol 37,
					543°F	I P.	Reaction	1,388	35	S R	14 5	x	x	10 79	50	156	p 49
						L.P.	Reaction	1,388	12	S R							•
Sea-train New York	1932	1	8,000	105	375 lb	H.P.	Impulse	4,685		D R	44 62	10 66	25	16 9	56	124 7	*M E.S A*,
					771°F	L.P.	Impulse	3,445		D R	32 83	14.5	25	16 9	56		vol. 37, p.
																	416
Tannenburg	1936	2	6,250	250	880 lb	H.P.	Impulse	18,000	3	D R.	71						*S.B.*, vol. 43
					860°F	I.P.	Impulse	15,700	5	D R.	63	. .	.				p 56
						L.P.	Impulse	6,500		D.R.	26						
Ohio	1940	1	9,000	90	435 lb	H.P.	Impulse-reaction	5,978	15	D R	65 42	8 953	20	22 12	40	144 94	
					730°F	L P.	Reaction	4,484	10	D R	49 65	11 938					

* Inboard screw only.

† S.N A.M.E is the Society of Naval Architects and Marine Engineers. *M.E.S.A* is the *Marine Engineering and Shipping Age*, now *Marine Engineering and Shipping Review* (*M.E.S.R.*). *S B.* is *The Shipbuilder and Marine Engine Builder* (British). W.R.H. is Werft, Reederei, Hafen (German)

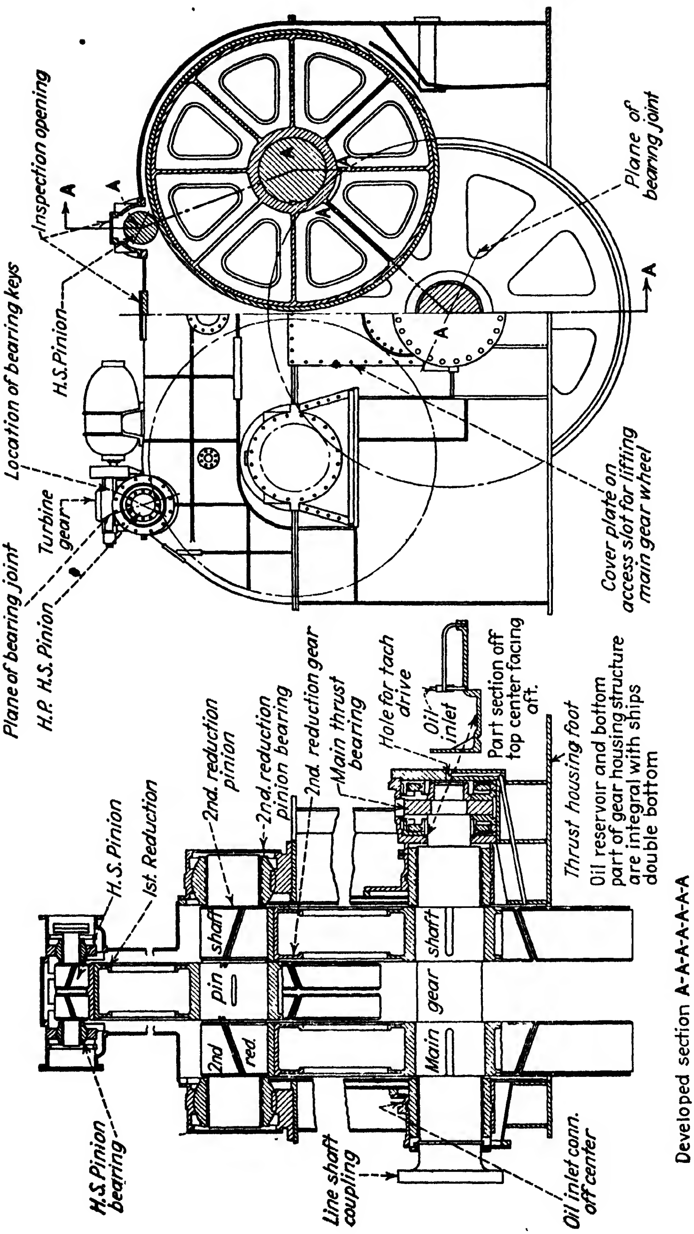


FIG. 75.—Double reduction gear assembly (nested type). (Westinghouse Electric & Manufacturing Company.)

for double reduction gears. Bearing losses are greater than those caused by friction between the gear teeth. The helical angle ranges from 20 to 45 deg.

The upper limit of speed-reduction ratios with single reduction gearing is usually around 20 with American practice, although the British have employed ratios as high as 24 (see Table X).

Single reduction gears are simpler, have fewer bearings, and are usually cheaper than double reduction gears. However, as a rule, the single reduction gears will require the use of a larger gear wheel (see Table X). Double reduction gearing naturally allows greater speed reduction ratios between the turbine and propeller shaft. Up to the present time, ratios as high as 84 have been used. In the United States the tendency is toward double reduction gears and high-speed turbines. Double reduction gears naturally allow greater latitude in the choice of turbine and propeller revolutions than single reduction gearing. As pointed out later, some installations use double reduction for the high pressure turbine and single reduction for the low pressure turbine.

When double reduction gears are employed, the gears may be arranged in several ways. Figure 75 shows a Westinghouse reduction gear of the nested type. The high-speed (first reduction) gear wheels driven by the H.P. and L.P. turbines are fitted between the two low-speed gear wheels that are keyed to the main gear shaft. The high-speed pinion of the high pressure turbine drives the left-hand first-reduction gear wheel, and the low pressure pinion drives the right-hand wheel. As will be noted in Fig. 75, the low-speed pinion is divided into two parts, one-half being forward of the high-speed wheels and the other half aft of the high-speed wheels.

A Falk articulated-gear arrangement fitted in connection with Bethlehem propulsion turbines is shown in Fig. 76. Here the high-speed elements are placed ahead of the low-speed gearing, and the high-speed gear wheels are separate from the low-speed gear wheel. Flexible couplings are placed between the high-speed and low-speed elements.

Fewer journal bearings are required with the nested gear (Fig. 75), and no flexible coupling is placed between the high-speed gear wheel and the low-speed pinion. Flexible couplings can be troublesome in operation, and their omission in the nested type decreases the possibilities of critical torsional vibrations. Also, with this arrangement, a very compact casing can be used. On the other hand, with nested gears, renewal of the low-speed pinion requires a new high-speed gear wheel, and alignment is probably more difficult to secure. Flexible couplings have

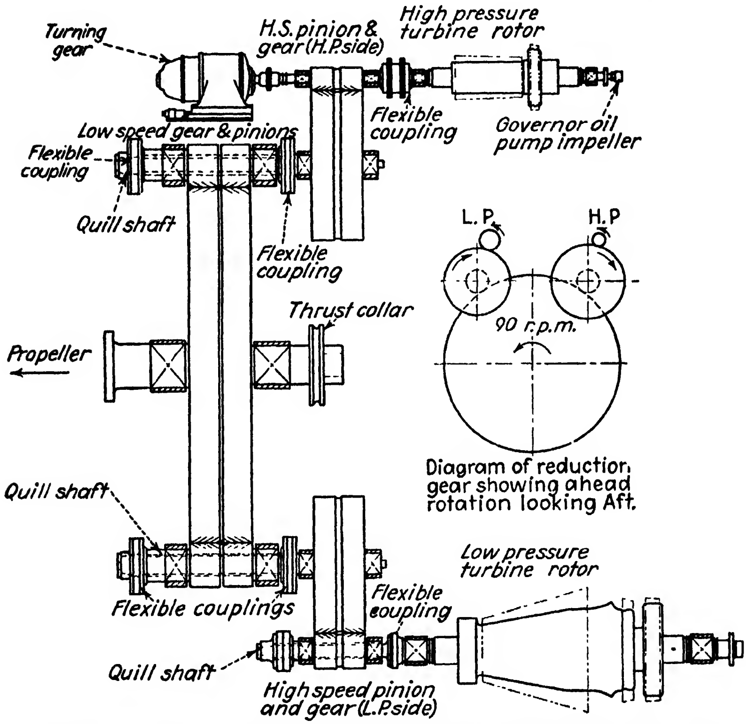


FIG. 76.—Falk double reduction gear assembly (articulated type) with Bethlehem Steel Company turbine installation. (Diagrammatic.)

advantages in that they allow a small fore-and-aft displacement of the low-speed pinion independent of the high-speed gear and permit some slight transverse adjustment so that the high-speed gear and low-speed pinion assume positions tending to give uniform tooth pressure. Other arrangements of gearing are shown in Fig. 78.[1]

Flexible couplings are always fitted between the turbines and the turbine pinions of the reduction gears. In articulated gears,

[1] From a paper by Robert Warriner, "Reduction Gears for Ship Propulsion," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1921.

as already pointed out, additional flexible couplings are placed between the first and second reductions. The flexible couplings between the high-speed and low-speed elements of an articulated reduction gear are now sometimes placed at the aft end of the low-speed pinions. With this arrangement the low-speed pinion is made hollow, and a quill shaft from the high-speed wheel extends through the hole in the pinion and is attached to the low-speed

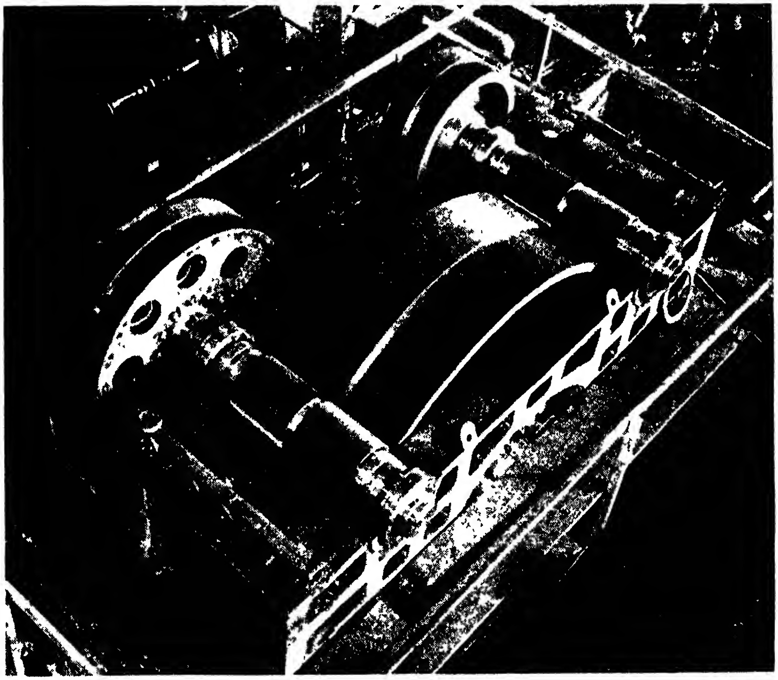


FIG. 77.—Reduction gears with high-speed pinions removed (see Fig. 76). (*The Falk Corporation.*)

pinion at its aft end by means of a flexible coupling. In the arrangement shown in Fig. 76 the flexible coupling between the first and second reductions is in each case divided into two parts, one part being forward and one aft of the low-speed pinion.

The teeth normally are of involute form; this type of tooth permits slight changes in distance between gear centers without affecting the angular velocity. Great care is used in cutting gears to assure high accuracy; tolerance in tooth thickness will run about 0.0005 in. The backlash, *i.e.*, total clearance between adjacent teeth, will range from 0.015 to 0.03 in. The circular pitch is about $\frac{5}{8}$ to $1\frac{1}{4}$ in. Some tooth pitting or formation of small cavities may occur in service, but unless it is abnormal in amount it is not considered serious. Since the pinion teeth are in contact more frequently than the gear teeth, the pinion should be of harder material than the gear teeth.

Gear wheels are either of cast steel or are composed of forged wrought-steel rims shrunk or welded to a cast or built up center. Rigid construction of the center or spider is obtained by using flat or conical disks, reinforced with diaphragms or by supporting the rims with especially reinforced arms. The reduction-gear casing may be a casting or be made up of welded steel plates with cast-steel bearing housings welded in. Since the reduction-gear casing must provide a rigid support for the bearings of the gears, it is well ribbed.

Most American reduction gears have the main thrust block of the Kingsbury type at the forward end of the gear casing as shown in Fig. 75. It is located just in front of the forward journal bearing of the bull gear. An independent thrust bearing and foundation aft of the reduction gearing is not required with this design.

89. Turbine and Gear Arrangements.—In most marine geared-turbine installations, the turbines are forward of the gears. This reduces the length of the propeller shafting and leaves more space for access around the turbines and gears. If the boilers are in their usual location at the forward end of the machinery space, this arrangement makes the steam lines shorter.

When the geared turbine was introduced, it was first applied to low-speed cargo ships, and single reduction gearing was used with two turbines, a high pressure and a low pressure unit. This was a logical step, since multicasing units had already been employed in many of the direct-connected turbines used on passenger and naval vessels. The first geared-turbine ship was the *Vespasian* (1909), a British cargo ship. In this ship the one-plane gear arrangement was employed, the center lines of the pinions and gears being in the same horizontal plane as shown in Fig. 78*a*.

In later single-reduction-gear installations, the pinions were placed above the horizontal plane, which passed through the center line of the propeller shaft (Fig. 78*b*). This arrangement produced a more compact foundation and in a few cases permitted the condenser to be placed under the low pressure turbine.

With the use of higher steam pressures and temperatures and hence with greater enthalpy drops, more stages were required

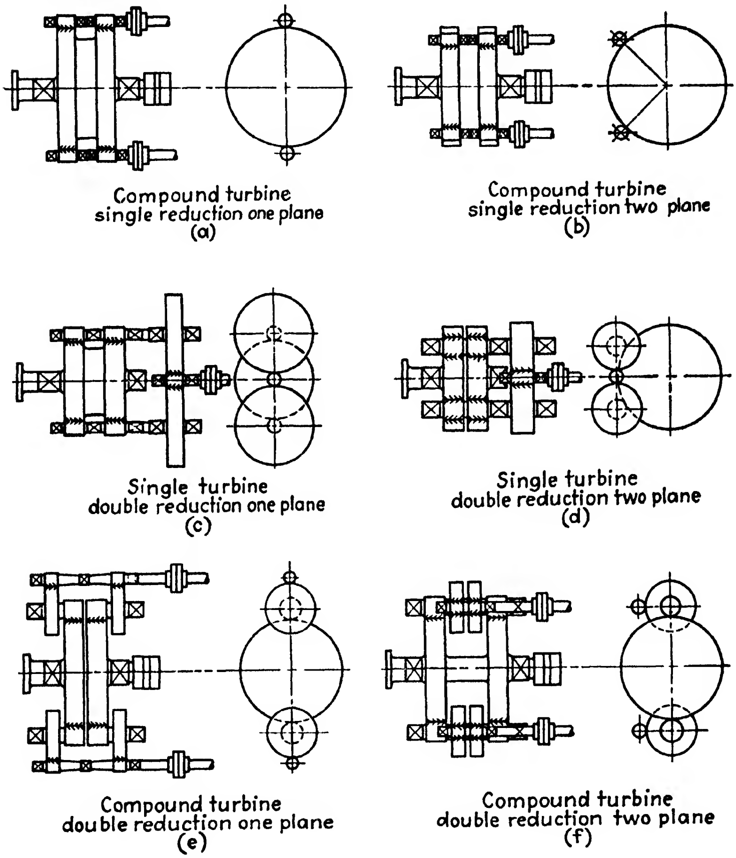


FIG. 78.—Gear arrangements.

when moderate turbine revolutions were used. In order to avoid the use of long rotors with their relatively low whipping or critical speeds, three and at times four turbines were employed, with the steam flowing in series through the several casings.

A very common arrangement for single reduction units was developed with three turbines and three pinions. The high pressure pinion was placed at the top of the gear wheel, and the intermediate and the low pressure pinions were placed close to the horizontal plane, as shown in Fig. 59. The *S.S. Mariposa*, the *S.S. Manhattan* (Fig. 59), and the *S.S. Excalibur* class of ships and also many British ships have propulsion units of this type. Since the main gear wheel was unusually large in diameter, the condenser in some installations was placed athwartships under the low pressure turbine. Figure 137 shows a three-casing arrangement with the condenser underneath the turbine.

An example of installations with four casings is the *S.S. Queen Mary*,[1] in which the high pressure and first intermediate casings are aft of the gears and the second intermediate and the low pressure casings are forward of the gears. The British destroyer *H.M.S. Acheron*[2] had a four-casing arrangement, three casings being forward of the gears and one aft.

Double reduction gearing was first introduced about 1913 and is today usually arranged with two casings and two high-speed pinions. Many American merchant ships constructed between 1917 and 1921 with powers under 3,000 shp were fitted with double reduction impulse turbines. These installations employed mostly single-casing turbines. Although the rpm of the turbines ranged from 3,000 to 3,500 and moderately high blade speeds were used, the engine efficiencies were rather low, because only from five to eight stages were fitted. The engines efficiencies obtained were somewhat better than those obtained with triple-expansion engines. A split-train type of single-plane reduction gearing similar to that in Fig. 78*c* was commonly used. For powers under 2,000 shp, double reduction gearing with single-casing turbines is still favored. A more compact gear arrangement than that shown in Fig. 78*c*, however, is adopted.

At the present time, the typical American steam-driven merchant ship of moderate powers per propelling unit is fitted with the two-turbine-casing arrangement of double-reduction-geared turbine. High blade speeds and high turbine rpm are employed that have reduced the number of stages and resulted in short rotors. The high pressure turbine in these installation

[1] *The Shipbuilder and Marine Engineer Builder*, June, 1936. (Special number).

[2] *The Marine Engineer & Motorship Builder*, 1931, p. 247.

runs at higher rpm than the low pressure turbine. For turbines of around 6,000 shp, the rpm of the high pressure rotor are in the neighborhood of from 5,000 to 7,000, and the low pressure rotor, in the neighborhood of 4,000. The gearing is arranged as in Figs. 75 and 76. The planes through the centers of the main gear and the low-speed pinions are at about 45 deg with the horizontal. The high-speed pinions are nearly over the high-speed gear wheels. The condenser can be fitted athwartship under both the low pressure and the high pressure turbines, thus assuring minimum pressure drop between turbine and condenser as well as good turbine drainage.

The British practice has tended more toward the three-turbine arrangement and single reduction gearing, with lower turbine rpm and peripheral speeds. Turbines of this type have been employed even with comparatively low powers.

Some low-powered installations using three turbine casings have been arranged with the H.P. and I.P. turbines connected in tandem and driving one of the two pinions.[1]

A number of geared-turbine installations have been made in which double reduction gearing has been employed for the H.P. turbine and single reduction for the I.P. and L.P. turbines. The *S.S. America* (Fig. 137) is of this type and has a reduction ratio for the H.P. turbine of 25.78 and for the L.P. turbine of 11.72. The German passenger ship *S.S. Gneisenau* has a three-casing arrangement with double reduction gearing for the H.P. and I.P. turbines and single reduction for the L.P. turbine. The data for this ship are given in Table X.

Cruising turbines are at times fitted in naval vessels to improve the economy during operation at low speeds.[2] Naval vessels, when cruising at low speeds, may develop powers of only 2 to 4 per cent of those required at full speed and with rpm between one-quarter and one-third of those developed at full power. By fitting cruising turbines, an appreciable reduction in the fuel rate at low cruising speeds is possible. When cruising turbines are employed, the steam flows first through the blading of the cruising turbine and then through the blading of the main turbine. The use of cruising turbines is the equivalent of increasing the number of stages in the turbine, which, of course,

[1] See *The Marine Engineer & Motorship Builder*, vol. 50, p. 102.

[2] See Art. 126.

is desirable at the reduced rpm. The cruising turbine is placed in a separate casing and is connected through reduction gearing and a clutch to the shaft of the high pressure turbine. In place of or in conjunction with a separate cruising turbine, additional blading is sometimes placed at the inlet end of the high pressure rotor, which is by-passed at full power.

90. Factors Influencing Reduction-gear Design.—The choice of single or double reduction gearing depends on the required size of low-speed gear wheel, the machinery space available, and the speed-reduction ratio between turbine rotors and propeller.

In merchant-ship installations, the tangential force at the pitch line of the reduction-gear pinions is allowed to reach 45 to 65 lb per in. of pinion diameter per in. of axial length of tooth face. Usually the pinions are carried in two bearings, one at each end; but at times three bearings are employed. The third bearing is fitted at the center of the pinion between the two helices. The pinions are subjected to transverse deflection because of tooth loading and torsional deflection from the torque transmitted. To prevent excessive deflection, the effective length of the pinions is limited to around 2.5 times the pinion-pitch diameter when two bearings are fitted and to around 4.5 times the pinion-pitch diameter when three bearings are used. The maximum peripheral speed used for the gear teeth is 250 ft per sec.

Since the allowable tooth load and the pinion length can be expressed as functions of pinion diameter, a formula can be developed that will give pinion diameter in terms of pinion shp, rpm, etc. The pinion diameter thus computed should preferably be 6 in. or over, since it should have more than 32 teeth in each helix.

After the pitch diameter of the pinion has been calculated, the size of the main gear wheel can be determined as soon as the reduction ratio is fixed. If the gear-wheel diameter for a single reduction is too great for gear-cutting equipment or for the desired machinery-space arrangement, double reduction gearing is fitted.

Many American engineers favor high turbine revolutions, and since double reduction gears have proved to be very reliable, they have been specified for most of the recent American geared-turbine installations. The use of this type of gearing allows a more flexible choice of turbine and propeller revolutions than when single reduction gearing or turbo-electric drive are employed.

The trend to high initial pressures and temperatures, however, has favored the installation of small high-speed turbines, which, in many cases, operate at 4,500 to 7,000 rpm. This requires the use of double reduction gearing for cargo ships and tankers.

91. Lubricating-oil System.—Two types of lubricating-oil systems are used on shipboard for lubricating the reduction gears, turbine bearings, and thrust bearings: the gravity and the pressure systems. In the gravity system, the lubricating-oil pump delivers the oil to an overhead gravity tank, and the oil flows from the tanks to the gears and bearings by gravity. In the pressure system, the oil is delivered direct to the gears and bearings by the lubricating-oil pump. The former type is used on merchant ships, where there is sufficient headroom, and the latter type on naval vessels, where the headroom is limited. The gravity system is used whenever possible, since it has the advantage that if the lubricating-oil pump stops for any reason, the oil supply in the gravity tanks is sufficient to last several minutes, allowing time to start the stand-by pump or to stop the turbine before the oil supply fails.

A gravity lubricating-oil system is shown in Fig. 79. This is typical of most gravity systems. The lubricating-oil pump withdraws the oil from the sump tank shown at the bottom of Fig. 79 and discharges it through the lubricating-oil cooler to the overhead gravity tanks located in the upper engine-room casing. The lubricating-oil for the turbines and gears is withdrawn from the gravity tanks at a point somewhat above the bottom. This allows the water and sediment to settle to the bottom of the tanks, where it can be drawn off and sent to the oil purifier. The oil supply to the gears and turbine bearings passes down the gravity pipe and through a strainer to headers alongside the turbines and gears. Branch lines lead from these headers to all the main journal bearings, thrust bearings, the gear-spray nozzles, and the flexible couplings. The oil drains from the bearings and reduction-gear casings to the lubricating-oil sump tank.

An overflow pipe is connected to the top of the gravity tanks as shown in Fig. 79, and the speed of the lubricating-oil pump is regulated so that there is a continuous discharge of oil down the overflow pipe. This pipe is fitted with sight glass, and as long as a supply of oil is passing down this pipe the engineer knows that the gravity tanks are full. If for any reason the oil level in the active gravity tank drops below three-quarters level, an alarm bell rings. A further drop in the oil level automatically closes the turbine throttle as pointed out in Art. 85.

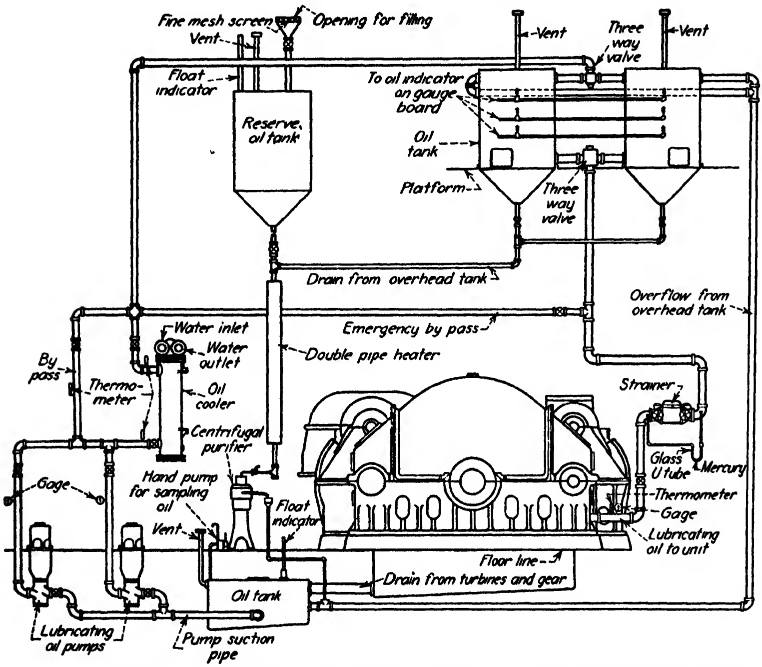


Fig. 79.—Diagram of gravity lubricating-oil system. (*De Laval Steam Turbine Company.*)

Centrifugal purifiers are installed in all lubricating-oil systems for removing water and sediment from the oil. The oil is withdrawn from the bottom of the gravity tanks as shown in Fig. 79. At sea, part of the oil supply can be continuously put through the separators if desired. In port, all the supply can be purified in a short time. A small heater is fitted in the line between the gravity tank and the centrifugal separator to allow heating of the oil when necessary before purifying it.

The lubricating-oil pressure should be at about 7 to 10 lb pressure at the highest bearings and at the gear nozzles. When friction in the piping valves, strainers, and cooler is allowed for, the overflow of the gravity tanks must be around 30 to 35 ft above the shaft.

The lubricating oil, besides lubricating the gears and bearings, carries away the heat generated by friction. Some heating of the oil also takes place in the gear case through heat transmitted along the shafts from the turbine casings. The efficiency of a single reduction gear is about 98.5 per cent and that of a double reduction gear, about 97 per cent. It is customary, however, to allow for a friction loss of about 5 per cent of the shp developed in fixing the size of the lubricating-oil cooler and pumps. This allows sufficient margin for thrust bearings, heating from the turbine as mentioned above, etc. The requirements for lubricating the gears call for a higher viscosity and hence a cooler oil than is required for bearing lubrication. At times provisions have been made for supplying cooler oil to the gears than to the bearings, but because of the additional complications the general practice is to supply all the oil at the same temperature, usually around 120°F. The rise in oil temperature in passing through the gears and bearings is usually between 10 to 20°. An oil cooler is fitted to bring the oil back to the desired temperature. Sea water is circulated through the cooler, usually from a branch connection from the main-condenser circulating water system. Calculations for the size of the lubricating-oil pump and lubricating-oil-cooler pump are carried out in Art. 162.

It is customary to make the capacity of the gravity tank sufficient to hold a 4- or 5- min supply. If we allowed a rise of 15° in the lubricating oil temperature passing through the gears, this would require a tank capacity of between 160 and 200 gal for an installation of 1,000 shp. The sump tank is usually made equal to the capacity of the two gravity tanks. An overhead reserve lubricating-oil tank is fitted that has about the capacity of one of the gravity tanks.

CHAPTER VII

THE TURBO-ELECTRIC DRIVE

92. General.—The turbo-electric drive is a type of machinery used for ship propulsion that accomplishes the same purposes electrically that the geared turbine does mechanically. This drive consists of one or more high-speed steam turbines driving alternating current generators that deliver power to one or more low-speed a-c motors on the propeller shafts. This form of ship propulsion was the first used by the U.S. Navy in an experimental installation on the *U.S.S. Jupiter* (1912) and on the battleship *U.S.S. New Mexico* (1918). Since then it has been used on a number of United States battleships and two airplane carriers, on U.S. Coast Guard cutters, and on a number of American passenger ships, several foreign passenger ships, and a large number of American tankers. When turbo-electric drive was first introduced, a few cargo ships were fitted with this type of machinery, but as far as is known, no pure cargo ships have used it since 1928.

Two single-screw bulk carries on the American Great Lakes, the *S.S. T.W. Robinson* (constructed in 1924) and the *S.S. Carl D. Bradley* (constructed in 1928), are fitted with turbo-electric drive (see Table XII).

The turbo-electric drive was introduced as a means of reducing the rpm between the turbine and propeller in competition with the geared turbine. The advantages originally claimed for this drive over the geared turbine were its ease of maneuvering, high reversing torque, the fact that no reversing turbine is required, quietness of operation, freedom from vibration, and, in some installations, high economy at reduced speeds.

93. Essential Features.—In a simple turbo-electric installation consisting of one turbogenerator and one propelling motor, the current is generated by a three-phase two-pole high-speed generator and is transmitted to a low-speed multipole motor on the propeller shaft. The generator is driven by a direct-connected steam turbine running at a speed of between 2,000 and 3,600 rpm. The motor usually runs at 90 to 130 rpm. In order to allow the use of a light high-speed generator, alternating current with voltages of around 2,000 to 4,000 has been used in all turbo-electric installations.

The frequency is fixed by the speed of the generator and the number of poles used. Since a turbo-electric drive is a unit in itself, standard commercial frequencies do not have to be used. The reduction in rpm between the generator and the motor depends on the ratio of poles in the two machines.

$$\text{Rpm of generator} = \frac{120 \times \text{frequency}}{\text{number of generator poles}}$$

$$\text{Rpm of motor} = \text{generator rpm} \times \frac{\text{generator poles}}{\text{motor poles}}$$

Thus a 60-pole motor connected to a 2-pole generator running at 3,600 rpm will turn at 120 rpm. The number of poles that can be fitted in a motor is limited to around 80; hence there is an upper limit to the reduction in rpm between generator and motor.

94. Types of Propelling Motors and Operation at Reduced Speeds.—Several types of motors have been used for ship propulsion as follows:

1. Induction motor with wound rotors and external resistance.
2. Double squirrel-cage motor.
3. Synchronous motor.

Diagrams of these three motor installations are shown in Figs. 80, 81, and 82.[1] With the exception of the ships of the U.S. Navy, almost all the turbo-electric-drive installations have used synchronous motors.

In turbo-electric installations, the generators and motors are tied together electrically, and the speed of the main motor and hence the speed of the ship are increased or reduced by increasing or reducing the speed of the generator and hence changing the frequency. This is accomplished by varying the speed of the turbine, which is exactly the way in which the speed of the ship is controlled with a geared-turbine installation. When the speed of the ship is controlled by changing the speed of the main

[1] These figures have been taken from a paper by W. E. Thau, before the Society of Naval Architects and Marine Engineers, 1921.

turbine, the economy falls off as the turbine rpm is reduced, and hence the ordinary turbo-electric drive has no better economy at low speeds than the geared turbine.

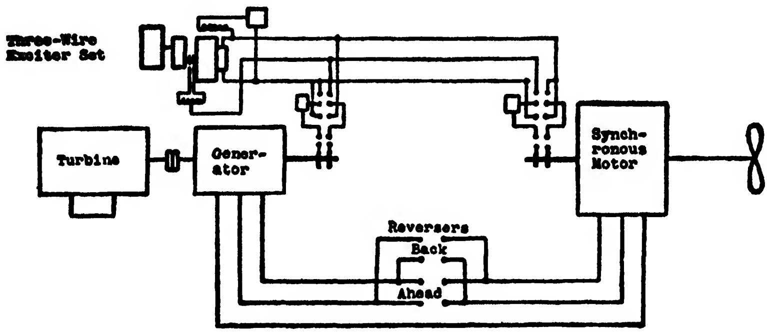


FIG. 80.—Diagram of synchronous motor installation.

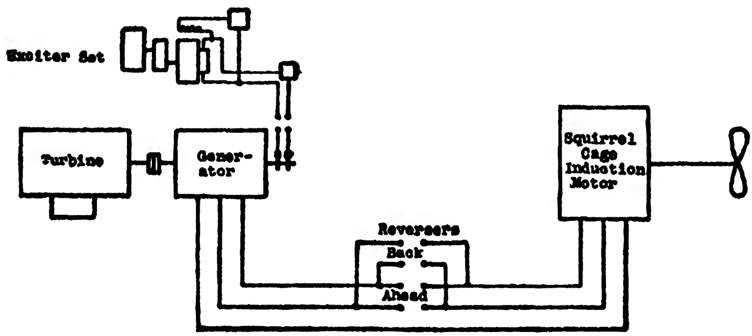


FIG. 81.—Diagram of squirrel-cage motor installation.

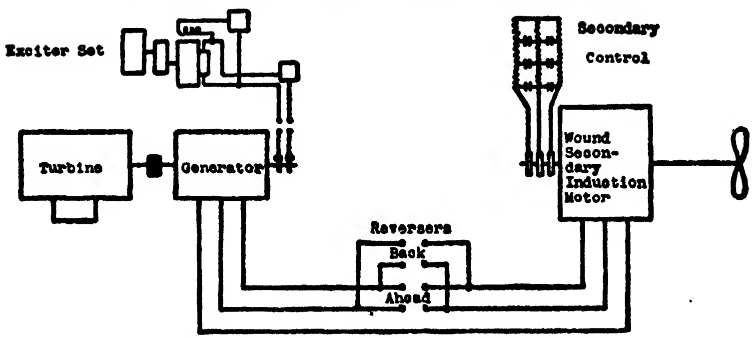


FIG. 82.—Diagram of induction motor installation.

The induction motors used on the United States battleships are fitted with two sets of pole windings so that it is possible to obtain two different motor speeds for one generator speed, as explained later.

The *U.S.S. New Mexico*, the first United States battleship to be equipped with turbo-electric propulsion, was fitted with motors having double squirrel-cage rotors. The high-resistance windings are effective during maneuvering and the low resistance, for normal speeds. The "California" class of battleships and the airplane carriers *U.S.S. Lexington* and *U.S.S. Saratoga* have squirrel-cage motors with secondary windings. The high-resistance squirrel cage is used for maneuvering and the secondary windings, for ordinary operating conditions. The *U.S.S. Tennessee*[1] has induction motors with wound rotors and external resistance. The external resistances with variable controls and liquid rheostats are used to give high torques during maneuvering.

The first two types of motors listed above have the advantage of high starting and reversing torques, which are essential in naval vessels. They also can be fitted with two sets of pole windings, which allow higher turbine revolutions and hence improved turbine steam rates at reduced speeds. Thus on the *Saratoga* (Table XI), at full speed, the turbine rpm is 1,668

TABLE XI.—PERFORMANCE DATA OF U.S.S. SARATOGA*

	1	2	3	4	5	6
Per cent full power for ship. .	Full power	68 5	38 6	14 5	10.4	2.5
Per cent generator power.. .	100	68.5	77 2	58.0	42.4	10.0
Turbine rpm	1,668	1,483	1,238	1,752	1,562	980
Per cent full-speed rpm (turbine)...................	100	89	74	105	94	59
Motor rpm.................	309	273	226	162	144	90
Per cent full-speed rpm (motor)	100	88.5	73	52.5	46.5	29
No. of generators in operation.	4	4	2	1	1	1
No. of motors in operation....	8	8	4	4	4	4
No. of motor poles used.......	22	22	22	44	44	44
Fuel, lb per shp-hr...........	0.950	0.965	1.13	1.14	1.19	2.38
Oil, lb per sq ft H.S. per hr....	0.97	0 68	0.64	0.45	0.40	0.26
No. of boilers in operation .	16	16	11	6	5	4

* Data from paper by J. A. Alexander, "Trial Trip Data of the *U.S.S. Saratoga*," *Trans. Society of Naval Architects and Marine Engineers*, 1929.

and the motor rpm is 309. The generators are 4-pole machines and the motors have 2-pole combinations, namely, 22 and 44.

[1] An installation similar to this has been used on the *S.S. Carl D. Bradley*, a bulk carrier on the Great Lakes.

At full speed the 22-pole combination is used on the motor. When the generator and propeller rpm have been reduced to about 50 per cent of full-speed rpm as the ship is slowed down, the motors are shifted to the 44-pole combination, and the generators can now be speeded up to the full-speed rpm while the motors still turn at half speed. This arrangement allows the turbine to operate at its full rated rpm when the ship is running at half speed. This results in a lower turbine water rate.

Another method of improving the economy at reduced speed of the ship is to shut down generators when more than one is installed. This allows the generator in operation to operate at nearer full load, with a consequent improvement in turbine water rate.

Table XI shows that when the percentage of power for the ship has been reduced to 38.6 (column 3), half the generators are shut down and only half of them are used. The load on each generator is thus increased from 38.6 to 77.2 per cent (line 2). When the ship's power has reached 14.5 per cent (column 4), which gives approximately half speed for the ship, the motors are shifted to the 44-pole combination, and all but one of the generators is shut down. The remaining generator is thus running at 58 per cent full power (instead of 14.5 per cent, as would be the case if all 4 were kept in operation), and the generator rpm is 1,752—slightly higher than the full-speed rpm. Turbine water rates are not given, but the fuel consumptions listed in the third line from the bottom indicate how the economy has been improved by cutting out generators and changing motor-pole combinations. Thus the fuel-consumption curve plotted against percentage of full power has a saw-tooth appearance, as shown in Fig. 104. The sharp drops in the curve indicate changes in generator and pole combinations. Fuel consumptions were not reported before making the changes indicated in columns 3 and 4, but the author has indicated by dotted lines in Fig. 104 the general trend the fuel-consumption curve would probably take. Actual trial data are shown by the solid lines. Figure 104 also shows the improvement in fuel economy on the *S.S. President Hoover* and the *Viceroy of India* achieved by shutting down one of the two generators. Neither of these ships had pole-changing motors.

When an induction motor with wound rotor and external resistance is used, the speed of the ship can be controlled by use

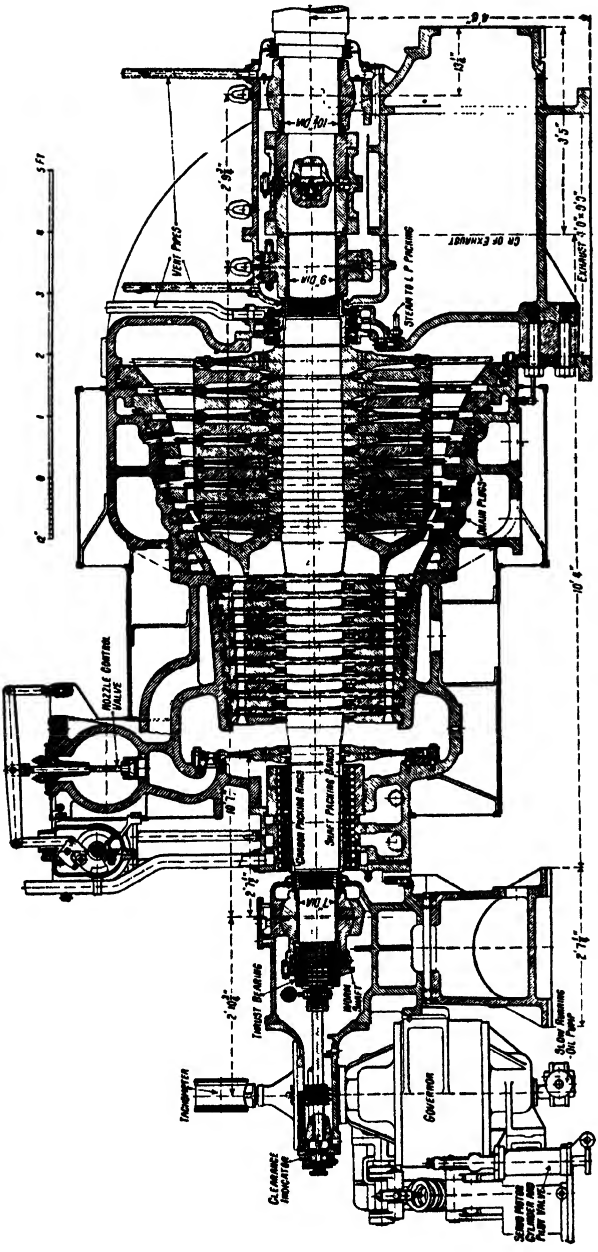


FIG. 83.—Longitudinal section of British Thomson-Houston turbine installed on S.S. *Viceroy of India.* The expansion is completed in one casing. (*The Marine Engineer & Motorship Builder.*)

of the external resistance, the generator and hence turbine speed remaining constant. This would give a higher thermodynamic efficiency for the turbine, compared to the usual method of speed control, *i.e.*, slowing down the turbine, but the electrical loss due to the resistance would more than offset the thermodynamic gain.

All merchant ships fitted with turbo-electric drive, and the U.S. Coast Guard cutters, are equipped with synchronous motors. Since the synchronous motor has no inherent starting torque, squirrel-cage bars are fitted to these motors, and they are started as induction motors. The advantages of the synchronous motor are:

1. Unity power factor and hence a cheaper, smaller, and lighter motor and generator.
2. Larger air gap.

The synchronous motor, however, does not have the starting and reversing torque that an induction motor has and hence is not so well suited for a ship that has to do a great deal of maneuvering. A pole-changing synchronous motor, although theoretically possible, is not practicable and none has been used.

A ship fitted with a synchronous motor is started in the following manner:

1. The turbine is started, and the governor control lever is set at idling speed with the generator field-control lever set at "stop" position.

2. The reversing switch (see Fig. 80) that controls the leads between generator and motor is set in the "ahead" position.

3. The field lever is set to give double excitation on the generator field (240 volts). The main motor field is not excited. The motor now starts as an induction motor.

4. When the motor has attained constant speed as an induction motor, the field-excitation control is set to put excitation on the motor field.

5. The generator is now speeded up, and the motor speeds up as a synchronous motor in step with the generator.

6. Before reaching full speed, the field-excitation lever is set to the "run" position, which reduces the generator field excitation to its normal value (120 volts).

Reversing is accomplished by slowing down the turbine to idling speed, opening the field switch, reversing two of the

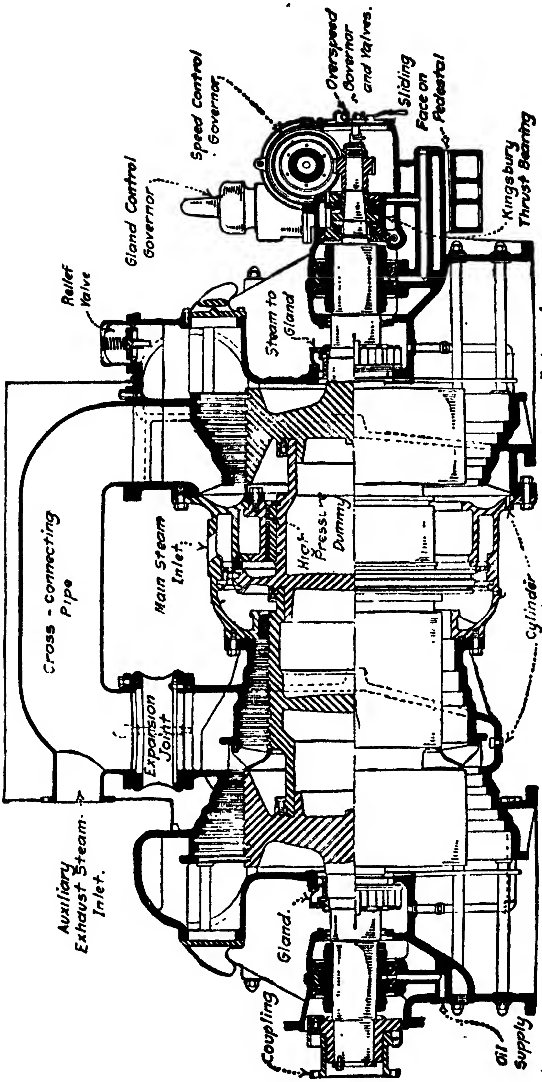


FIG. 84.—Impulse-reaction turbine, *U.S.S. Tennessee* (1920), turbo-electric drive. The expansion is complete in one casing; low pressure blading arranged for double flow.

connections between generator and motor, and repeating the operations outlined above. The turbine is not reversed as in a geared turbine installation, and hence no reversing turbine is required.

95. Special Features.—So far all the turbo-electric installations have used complete expansion of the steam in one turbine casing and have not used two and three casings, as has been done with geared-turbine installations (see Fig. 83). This has the disadvantage that all blading must turn at a constant rpm, which will be fixed by the lower rpm requirements of the low-pressure blading. As pointed out in Chap. VI, the H.P. and L.P. turbines of a geared-turbine ship run at different rpm.

As mentioned above, no reversing blading is required. This allows a turbo-electic drive to deliver as much power to the propeller when going astern as when going ahead. However, because of the lower propeller efficiency in reversing, the same propeller thrust cannot be attained in going astern as in going ahead.

Since d-c current is necessary for excitation, nearly all the ships fitted with turbo-electric drive have had the auxiliaries driven by d-c motors, the current being supplied by separate turbine-driven d-c generators.

96. Installation on U.S.S. Pontchartrain.—Each of the U.S. Coast Guard cutters of the *Pontchartrain*[1] class is fitted with a single-screw turbo-electric drive with a synchronous motor and one generator. Most of the auxiliaries are driven by a-c motors, the current being supplied through a transformer from the main generator. This arrangement allows the auxiliaries to operate with a fuel economy of that of the main generator instead of receiving their power from a small less efficient auxiliary generating set. All the propulsion auxiliaries, such as blower, circulating pump, etc., being driven by synchronous motors in step with the main alternator will slow up and speed up with the main unit.

An auxiliary set is installed with a turbine, a d-c machine, and an a-c machine on the same shaft. In normal full-speed operation, the a-c machine acts as a motor driving the d-c machine, as a generator for supplying d-c current for lighting, etc., and also drives the turbine, which is kept under a vacuum.

[1] Q. B. Newman, "The Central Power Station Goes to Sea," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1928.

TABLE XII.—TYPICAL TURBO-ELECTRIC DRIVE INSTALLATIONS

Ship	Type of ship	Total motor shp, normal	Type of motor	No. of generators	No. of motors	Generator rpm	Motor rpm	No. of generator poles	No. of motor poles	Voltage	Frequency
U.S.S. New Mexico	Battleship	28,000	Induction	2	4	2,100	167	2	24–36	2,300	33.2
*U.S.S. Saratoga**	Carrier	180,000	Induction	4	8	1 755	317	4	22–44	5,000	58.5
S.S. Argentina	Passenger	13,500	Synchronous	2	2	2,640	110	2	48	3,700	44
S.S. President Coolidge	Passenger	26,500	Synchronous	2	2	2,660	133	2	40	4,000	44.3
S.S. Viceroy of India	Passenger	17,000	Synchronous	2	2	3,110	110	2	.....	3,200	51.8
S.S. Scharnhorst†	Passenger	26,500	Synchronous	2	2	3,120	130	2	48	3,100	52
S.S. Normandie‡	Passenger	130,000	Synchronous	4	4	2,250	225	4	40	5,500	75
S.S. J. W. Van Dyke	Tanker§	5,000	Synchronous	1	1	3,600	90	2	80	2,300	60
S.S. Carl D. Bradley	Bulk carrier	4,800	Induction	1	1	3,600	104.5	2	...	2,300	60
U.S.S. Pontchartrain	U.S. Coast Guard	3,000	Synchronous	1	1	3,600	163.5	2	44	2,300	60

* *Jour. Am. Soc. Naval Eng.*, 1928, p. 438.
† *The Shipbuilder and Marine Engine Builder*, 1935, p. 457.
‡ *The Shipbuilder and Marine Engine Builder*, June, 1935.
§ Five tankers owned by the Atlantic Refining Company in this class all have turbo-electric installations.

As the ship slows down, when the frequency reaches 40 cycles, the throttle of the auxiliary turbine automatically opens, the a-c motor on its shaft is disconnected from the main generator, and the turbine now drives the a-c machine as a generator that supplies current for all the auxiliary motors. This method of driving the auxiliaries is used for all speeds below two-thirds (40 cycles). As the speed of the ship is increased, when a frequency of 40 cycles is reached the a-c auxiliary machine is brought into synchronism with the main generator, and the auxiliary turbine throttle is closed by hand. On trial, this ship was run at full speed, with the current for the auxiliaries supplied by the main alternator, as designed, and during a second trial with the auxiliary turbo-generating set supplying the current. The fuel consumptions were 0.823 and 0.89 lb per shp per hr, respectively, an advantage of 7.5 per cent in favor of the designed condition. Although the gain in economy may not be worth the added complications for a lower powered installation, it holds considerable merit for a high-powered installation.

An interesting modern installation of turbo-electric drive is that of the tanker *S.S. J. W. Van Dyke* and her four sister ships, belonging to the Atlantic Refining Company.[1] This is a single-screw installation with 5,000 shp on the main motor. The boilers operate under a pressure of 600 lb, with a steam temperature of 825°F. The two boilers are of the type shown in Fig. 11, and each has a heating surface of 3,222 sq ft, a superheating surface of 1,094 sq ft, an air-heating surface of 2,412 sq ft, and a furnace volume of 426 cu ft. Most of the auxiliaries are driven by a-c motors, with current supplied by the main generator and the auxiliary set operating as described above for the *Pontchartrain.* The fuel consumption on trial was 0.594 lb per shp per hr for main unit and auxiliaries and 0.502 lb for propulsion only. Data for the generator and motor are given in Table XII, which gives the particulars of a few representative turbo-electric installations.

[1] See the paper by Lester M. Goldsmith, A.S.M.E., 1938, reproduced in *Mar. Eng. & Shipping Rev.*, 1938, pp. 548–558.

CHAPTER VIII

THE DIESEL ENGINE

97. General.—The Diesel engine is an internal combustion oil engine and has the following distinguishing characteristics:

1. It operates on a heavy petroleum fuel oil.
2. It uses a high compression pressure (300 to 450 lb per sq in.).
3. The fuel is injected under pressure at or near the end of the compression stroke.
4. The fuel is ignited by the heat of compression.
5. It has low fuel consumption (0.38 lb per bhp-hr).

The Diesel engine is named after its inventor, Dr. Rudolph Diesel, but, as pointed out in a later article, modern oil engines using airless (solid) injection of the fuel do not operate on the true theoretical Diesel cycle. However, the name "Diesel" is still almost universally applied to the modern oil engine that uses compression ignition and that has been developed from the air-injection engine that did operate on the Diesel cycle. These engines are also frequently referred to as "compression-ignition engines."

98. Oil-engine Cycles.—A pressure-volume (PV) diagram of the theoretical Diesel cycle is shown in Fig. 85*a*. The heat is added with an increase in temperature along the constant-pressure line *ab*; adiabatic expansion takes place along *bc*; the heat is rejected along the constant-volume line *cd*; and an adiabatic compression from *d* to *a* completes the cycle.

The heat received along *ab* is

$$Q_1 = c_p(T_b - T_a)w$$

The heat rejected along *cd* is

$$Q_2 = c_v(T_c - T_d)w$$

The efficiency of the cycle becomes

$$\frac{Q_1 - Q_2}{Q_1} = \frac{c_p(T_b - T_a) - c_v(T_c - T_d)}{c_p(T_b - T_a)} = 1 - \left[\frac{1}{n}\frac{(T_c - T_d)}{(T_b - T_a)}\right]$$

where $c_p/c_v = n$.

This can be reduced to

$$\text{Efficiency} = 1 - \left[\frac{1}{n}\left(\frac{V_a}{V_d}\right)^{n-1} \times \frac{\left(\frac{V_b}{V_a}\right)^n - 1}{\left(\frac{V_b}{V_a}\right) - 1}\right]$$

The ratio V_d/V_a is known as the "compression ratio."

From the preceding expression for efficiency it will be seen that the greater the compression ratio the higher will be the

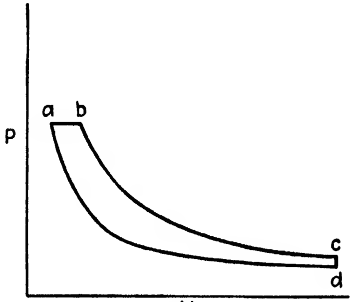


FIG. 85a.

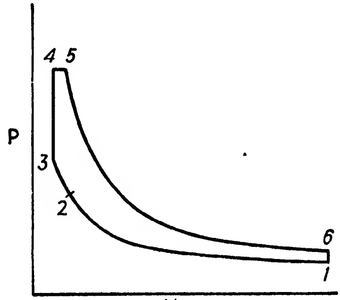


FIG. 85b.

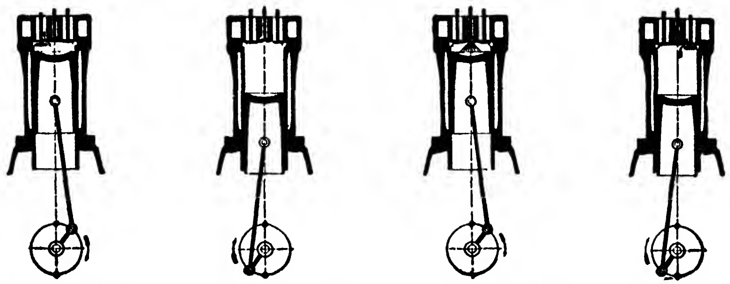
efficiency. Also, the smaller the ratio V_b/V_a the higher will be the efficiency. Diesel engines show a higher efficiency than the Otto engine because of the higher compressions used.

In an actual air-injection engine operating on the Diesel cycle, fuel injection begins at point *a* and continues to point *b*. The fuel begins to burn as soon as it enters the cylinder because of the heat of compression. The greater the length of the line *ab* the larger will be the area of the card and the mep. The power of an engine is reduced by injecting less fuel per stroke, *i.e.*, shortening the line *ab*; hence, from the foregoing expression, theoretically, an engine operating on the Diesel cycle should show a higher efficiency and lower fuel consumption per ihp per hour at reduced loads than at full load. Most engines show a slight reduction in fuel per ihp-hour as the power is reduced (see Fig. 98).

As pointed out in Art. 97, modern oil engines using solid injection do not operate on the true Diesel cycle shown in Fig. 85a. A theoretical PV diagram for a solid injection-oil engine is shown in Fig. 85b. Fuel injection begins at the point

2 during the adiabatic compression 1 to 3 and continues up to point 5. Point 2 is at a crank angle around 10 to 15 deg before upper dead center. Because of the low compression when fuel injection begins, the fuel does not start to burn instantly, as it did for the engine operating on the cycle shown in Fig. 85*a*. From point 2 to point 3 there is a delay period, and just before the end of the compression is reached the fuel starts to burn because of the heat of compression. Since fuel has been entering the cylinder from point 2 to point 3 and has not been burning, when it does start to burn at point 3 a sharp rise in pressure 3 to 4 results. From 4 to 5, fuel continues to enter and burn at constant pressure, as in the engine illustrated by the PV diagram in Fig. 85*a*. Cards from actual engines, of course, do not have sharp corners, as shown in the theoretical PV diagrams in Figs. 85*a* and 85*b*. A card from an actual engine using solid injection is shown in Fig. 100. A fuller discussion of the PV diagram for the solid-injection engine is given in Art. 109.

99. Operation of Four-cycle Diesel Engine.—In the four-stroke cycle, or, as it is commonly called, the four-cycle engine, four strokes of the piston, or two revolutions, are required to complete the cycle. As the piston moves down on its suction stroke *a* (Fig. 86) a charge of pure air containing the necessary oxygen for combustion is drawn in through the inlet port in the cylinder head. The inlet valve was opened mechanically by a cam at the end of the previous upward stroke. After the piston has completed the suction stroke *a*, the inlet valve closes, and the piston starts on an upward stroke *b* (Fig. 86), compressing the charge of pure air into the clearance volume to a pressure of between 350 and 450 lb per sq in. About the time that the piston reaches the end of its upward stroke, the fuel-injection valve opens for a short period, and finely atomized oil is sprayed in the combustion space under high pressure. Because of the heat of compression, the oil is ignited and burns generating heat. The piston has now started downward on its third stroke, *c*, and the hot gases generated by the combustion of the oil expand and force the piston downward on its working or power stroke. Near the end of the working stroke the exhaust valve in the cylinder head opens, part of the gases escape, and the pressure in the cylinder drops to approximately that of the atmosphere. The piston now starts up on its exhaust, or scavenging stroke *d*, and all the products of combustion except those contained in the clearance volume are forced out through the open exhaust port. The exhaust valve now closes, and the cycle is repeated. It will be observed that four strokes of the piston and two revolutions of the shaft are necessary to complete the cycle. Since the engine shown in Fig. 86 is a single-acting engine, there is only one working stroke during the cycle. Four-cycle double-acting Diesel engines have been built for ship propulsion; then there is a cycle taking place at the bottom of the piston as well



(a) First stroke (admission stroke). Piston travels down; admission valve open; cylinder is being filled with pure air.

(b) Second stroke (compression stroke). Piston travels up; all valves closed; air in cylinder is being compressed.

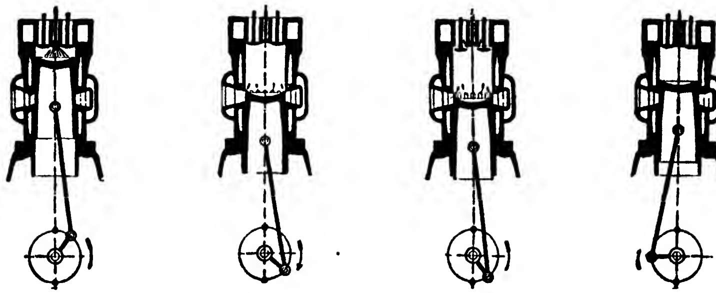
(c) Third stroke (power stroke). Piston travels down; fuel valve open near top dead-center, but closed at fraction of stroke; gases expand.

(d) Fourth stroke (exhaust stroke). Piston travels up; exhaust valve open; burnt gases are being expelled from cylinder.

FIG. 86.—Four-cycle Diesel engine. (*Courtesy of Busch-Sulzer Bros.-Diesel Engine Co.*)

as on top, and there will be a working stroke every revolution instead of every two revolutions as in the single-acting four-cycle engine. The engine described, which has only one cylinder, would have to be provided with a large flywheel so that enough energy could be stored up during the one working stroke to carry the piston through the three idle strokes. In practice, all marine four-cycle single-acting engines used for propulsion have six or more cylinders connected to the crankshaft. For example, if an eight-cylinder single-acting engine were used, there would be four working strokes each revolution of the crankshaft, and no flywheel would be necessary.

100. Operation of the Two-cycle Engine.—In the two-stroke cycle, or two-cycle engine, the cycle is completed in two strokes of the piston or one revolution. If we refer to Fig. 86, it will be seen that the two idle strokes *d* and *a*, in which the burned gases are pushed out of the cylinder and the new change of air is drawn in, will have to be dispensed with if the cycle is to be completed in two strokes. In other words, the working stroke will have to be followed by a compression stroke in which a new charge of air for combustion is compressed into the clearance volume. At the end of the working stroke *c*, the cylinder is filled with burned gases, and hence in a two-cycle engine some means must be provided for clearing out these burned gases at the end of the working stroke and filling the cylinder with a



(a) Piston traveling down; all valves and ports closed; fuel injected and burnt during first part of stroke; gases expand.

(b) Piston traveling down; piston uncovers exhaust ports; burnt gases escape through exhaust ports, reducing their pressure to atmospheric.

(c) Piston traveling down; exhaust ports still uncovered; scavenging valves open, air under light pressure enters through scavenging valves, and blows gases out of cylinder.

(d) Piston traveling up; piston covers exhaust ports; scavenging valve closed; air in cylinder is being compressed.

FIG. 87.—Two-cycle Diesel engine. (*Courtesy of Busch-Sulzer Bros.-Diesel Engine Co.*)

new charge of pure air before the piston starts its compression stroke.

This clearing of the cylinder volume of the products of combustion and filling the cylinder with a new charge of air is termed "scavenging." Various means are employed to scavenge two-cycle Diesel engines; a few of these are described in the following articles.

101. Scavenging of Two-cycle Engines.—All two-cycle engines employ one or two rings of ports in the cylinder wall, which are uncovered by the piston as it moves downward. These openings in the cylinder walls may be used for allowing the exhaust gases to escape, for admitting the scavenging air, or for both purposes. The principle of the two-cycle engine is shown in Fig. 87. Here the ring of openings around the cylinder wall function as exhaust ports. As shown at *b*, Fig. 87, the piston in its downward movement on the working stroke uncovers this ring of ports in the cylinder wall shortly before the piston reaches its lowest position.

The burned gases under pressure in the cylinder escape through these ports, and the pressure in the cylinder quickly drops to that of the atmosphere. Shortly after the exhaust ports start to uncover and the pressure in the cylinder has been relieved, the scavenging air is admitted. In Fig. 87, the scavenging air is admitted by the opening of mechanically operated valves in cylinder head. The scavenging air is blown into the cylinder under a pressure of about 2 lb per sq in. This scavenging air blows the products of combustion out of the exhaust ports and leaves the cylinder filled with a new supply of pure or nearly pure air. Thus this so-called "scavenging" process is in reality a scavenging and cylinder-charging process combined. A two-cycle engine must be provided with a scavenging compressor for supplying this scavenging air.

If the piston starts to uncover the exhaust ports when the crank is at an angle of 50 deg from the bottom dead center, the piston will have traveled approximately 81 per cent of its downward stroke. This early uncovering of the exhaust ports is, of course, necessary to allow sufficient time at the end of the working stroke and the beginning of the compression stroke to clear the cylinder of the products of combustion and charge the cylinder with a new supply of air. The scavenging begins at about 12 deg of crank angle after the exhaust ports start to uncover, which, for the example used above, will be at about 38 deg before bottom dead center. The scavenging will continue while the piston is moving upward and end at approximately 38 deg of crank angle after dead center. Thus the scavenging process will have about 76 deg of crank angle or approximately one-fifth of a revolution. The crank angles for the uncovering of the exhaust ports and the beginning of scavenging vary with different makes of engines, and when an engine runs at high rpm the exhaust ports must uncover earlier to allow sufficient time for scavenging. For an engine turning at 120 rpm or 1 revolution in ½ sec, the time of scavenging for the crank angles given above would be approximately ⅒ sec.

Many of the earlier makes of two-cycle Diesel engines had scavenging valves in the cylinder head, as illustrated in Fig. 87. This method of scavenging has now been abandoned in favor of other methods, and it is believed that no make of marine engine now employs valves in the head for scavenging.

The commonest and simplest method of supplying scavenging air to the cylinders is the port-scavenging method. Various forms of port scavenging are in use, and a few of them are described in the following pages. The earliest and simplest form of port scavenging is illustrated in the diagram in Fig. 88. This engine employs ports in the cylinder wall for both scavenging air and exhaust gases. As the piston descends on its working stroke, it uncovers the exhaust ports shown on the right-hand side of the cylinder. This allows the gases in the cylinder to escape and lowers the pressure in the cylinder to approximately atmospheric. A short time after the exhaust ports are uncovered by the piston, another group of ports on the other side of the cylinder is uncovered, through which the scavenging air enters. The diagram shows the piston in its lowest position and the scavenging process taking place. The scavenging ports are inclined upward so that the air will take the path shown by the arrows. It will be noted that the exhaust ports cover half of the periphery of the cylinder wall and the scavenging ports the other half. The exhaust ports are located higher up in the cylinder wall than the scavenging ports so that they will be uncovered first to relieve the pressure in the cylinder before scavenging air is blown in. The diagram is for a single-acting engine, but a double-acting engine could also use this method of scavenging.

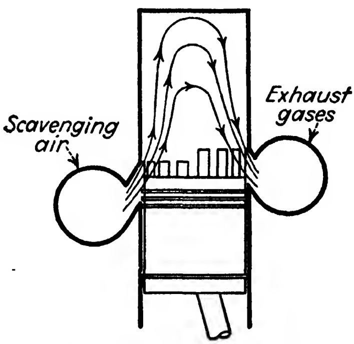


FIG. 88.—Diagram of engine using port scavenging and port exhaust.

The piston of two-cycle single-acting engines must be made long enough so that the ports in the cylinder walls are fully covered when the piston is in the upper part of the cylinder; otherwise scavenging air and exhaust gases would be blown into the crank case. This is accomplished by using a skirt on the piston, as shown in Figs. 90 and 91.

When port scavenging is used, the amount of scavenging air blown into the cylinder is considerably in excess of the piston displacement in order to do as thorough a job as possible of cleaning out the upper corners of the cylinder. Usually the excess air is in the neighborhood of 30 to 50 per cent. The scavenging is not as well done with port scavenging as with some of the methods of scavenging described below, but the construction of the engine is simple as the only valve necessary in the cylinder head is the fuel-injection valve, and no valve gear is necessary for operating scavenging or exhaust valves. The opening and closing of the exhaust and scavenging ports in a two-cycle engine employing port scavenging as shown in Fig. 88 are entirely automatic.

The method of scavenging shown in Fig. 88 is now used only on small engines for reasons that will be indicated later. However, modifications of this method are employed in the Sulzer, Richardsons-Westgarth, Nordberg, and M.A.N.[1] engines.

102. Types of Two-cycle Engines.—Figure 89 illustrates a third method of taking care of the exhaust gases and scavenging for a two-cycle engine. This method is used by the General Motors Corporation on their single-acting high-speed Diesel engines. The scavenging air is blown in through a double row of ports, which are uncovered by the piston, and the exhaust leaves through two exhaust ports in the cylinder head, which are opened and closed by mechanically operated valves. This method is just the reverse of that shown in Fig. 87. Burmeister and Wain also use this method of exhaust and scavenging on their single-acting two-cycle engine.

When exhaust valves are located in the cylinder head, the whole periphery of the cylinder can be used for scavenging air instead of half the periphery, as is the case when both scavenging and exhaust ports are fitted in the walls. By using the whole periphery for scavenging air, ports of less height can be used; or, if the height is not decreased, a greater amount of scavenging air can be blown in during a given period of time.

It will be noted that the path of the scavenging air in Fig. 89 is directly upward and does not follow a reverse path, as shown in Fig. 88. Obviously this uniflow method of scavenging is more effective than that shown in Fig. 88.

The Busch-Sulzer engine (Fig. 90) has long used the port-scavenging system, with two rows of scavenging ports one above the other. The upper ports are controlled by an automatic valve that does

[1] The common trade name for the German engine built by Maschinenfabrik Augsburg Nürnberg.

not allow the scavenging air to enter the cylinder until the pressure in the scavenging trunk is greater than that in the cylinder. This is illustrated in the Busch-Sulzer single-acting engine shown in Fig. 90. It will be observed in Fig. 90 that the upper row of scavenging ports on the left-hand side of the cylinder is above the exhaust ports and that they are uncovered by the piston on

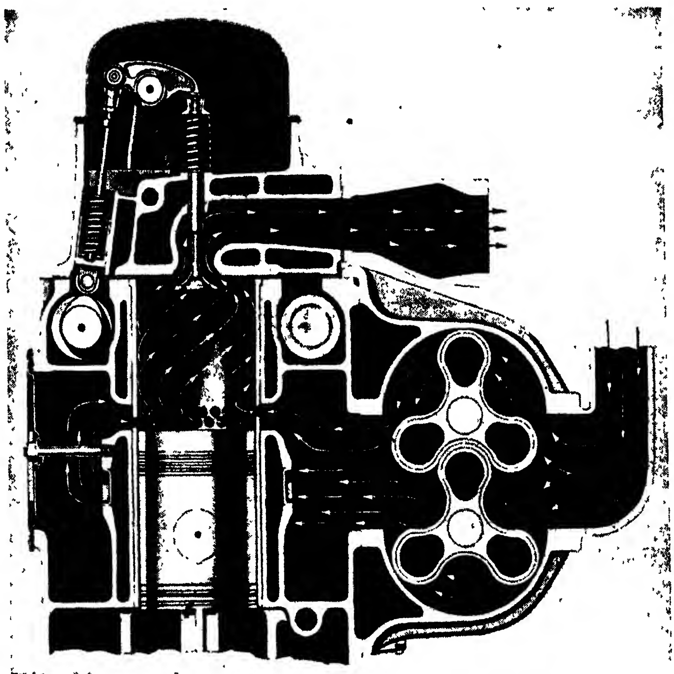


FIG. 89.—Section through cylinder of two-cycle single-acting engine, showing scavenging blower, scavenging air ports, and exhaust valve. (*General Motors Corporation.*)

its downward stroke before the exhaust ports are uncovered. The lower row of scavenging ports is uncovered after the exhaust ports, as in the arrangement for port scavenging shown in Fig. 88. However, since the upper row of ports are held closed by an automatic valve, scavenging air does not enter through these until the pressure in the cylinder has been relieved by the uncovering of the exhaust ports. The valve also prohibits the higher pressure gases in the cylinder from entering the scavenging air piping.

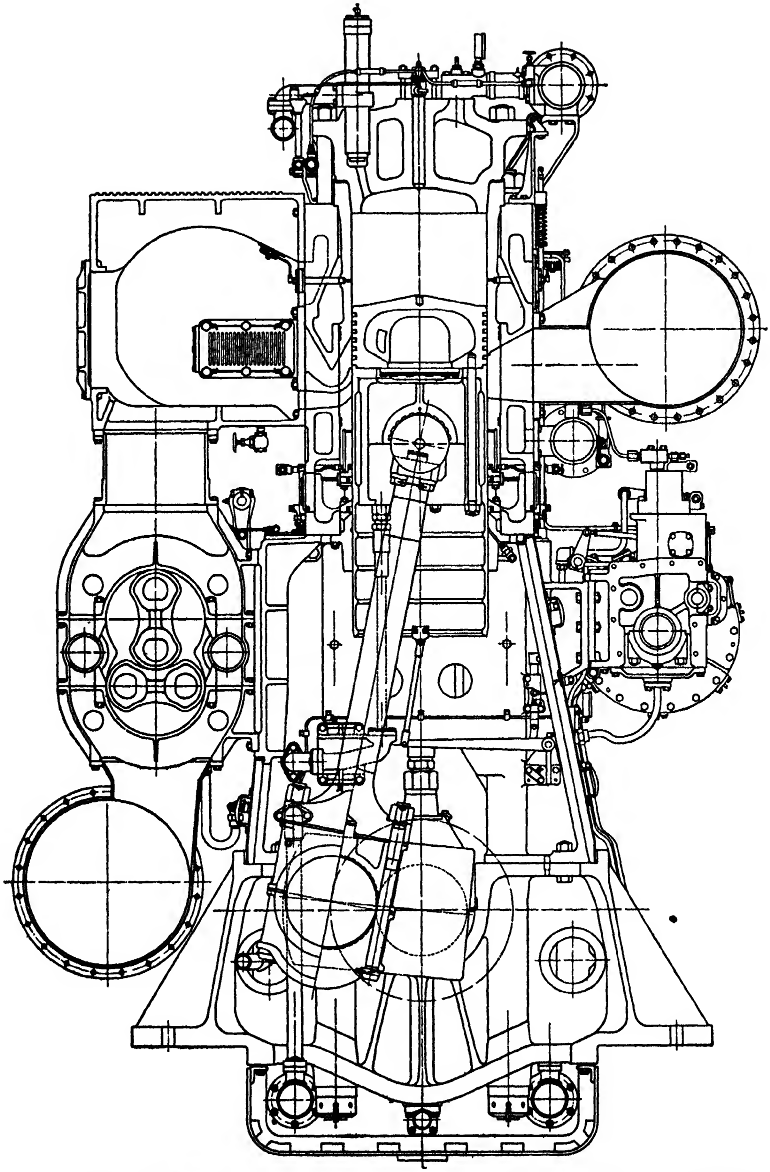


FIG. 90.—Cross-sectional elevation of Busch-Sulzer two-cycle, single-acting engine. This type of engine was used on the U.S. Maritime Commission's C-3 geared Diesel ships.

At about the same time that the scavenging air starts to enter the cylinder through the lower series of ports, the automatic valve allows the air to blow in through the upper row. The advantage of this system is that the area for the scavenging ports is increased, and scavenging air can be blown in through the upper ports on the compression stroke after the exhaust ports have been covered by the piston. This allows a certain amount of supercharging to take place, and the extra amount of air blown into the cylinder allows more fuel to be burned per stroke, with a resultant increase in mep.

In the engine shown in Fig. 88, the scavenging ports are covered before the exhaust ports, and the pressure in the cylinder will be practically atmospheric until the exhaust ports are covered. This reduces the effective compression stroke by the height of the exhaust ports.

Figure 91 shows a later development of the Sulzer engine, where only one row of scavenging ports is used. These are uncovered by the piston before the exhaust ports, but the automatic valve does not open until the pressure in the cylinder is less than that of the scavenging air. However, on the compression stroke, combustion air can be blown in after the exhaust ports are covered, as in Fig. 90.

Figure 92 shows a Burmeister and Wain-type double-acting two-cycle engine. In a double-acting engine, the piston rod occupies the center of the lower head, and no simple arrangement of exhaust valves is possible as in Fig. 89. In the engine shown in Fig. 92, the scavenging air enters through a ring of ports in the center of the cylinder and leaves through exhaust ports in the two ends of the cylinder. The exhaust ports are covered and uncovered by piston valves, which are operated from the auxiliary crankshaft shown in Fig. 93. The piston rod slides back and forth through the center of the lower piston valve. A study of Figs. 93 and 94 will make clear the construction and operation of this engine.

The arrangement of exhaust and scavenging ports for the M.A.N. engine is shown in Fig. 95. This cylinder diagram is for the engines of the German battleship *Deutschland*. The engines of this ship were fitted with mechanically operated valves in the exhaust, which allow the exhaust ports to be closed before the scavenging ports are covered. This allows a greater

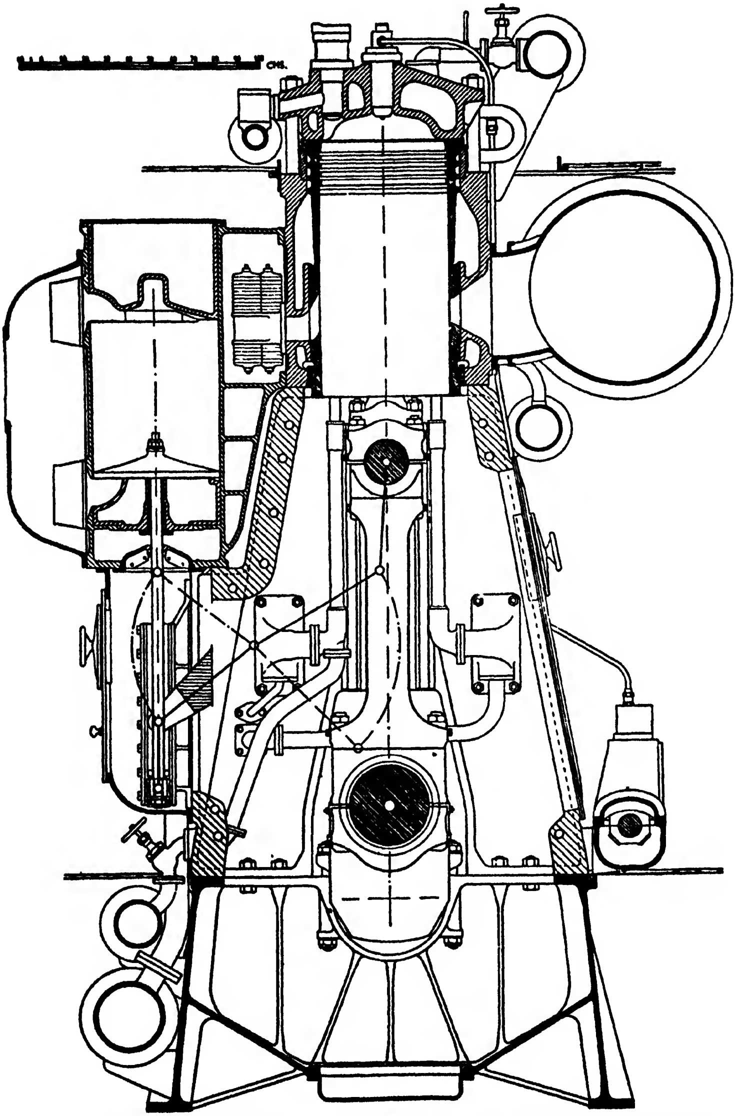


FIG. 91.—Cross-sectional elevation of Sulzer type two-cycle, single-acting engine installed on *M.S. Prins Albert*. (*British Motor Ship*.)

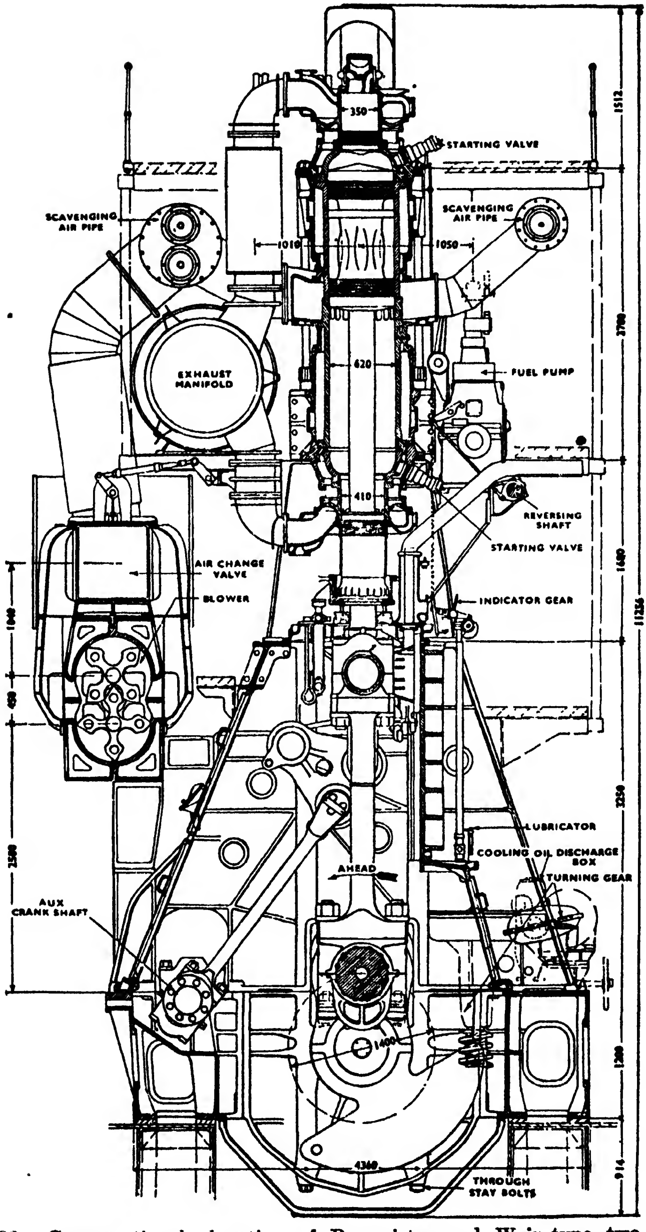


FIG. 92.—Cross-sectional elevation of Burmeister and Wain-type two-cycle double-acting engine. (*British Motor Ship.*)

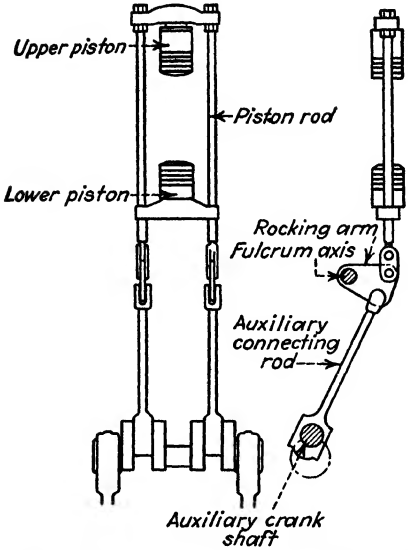


FIG. 93.—Exhaust-valve gear. Burmeister and Wain double-acting engine. (*British Motor Ship.*)

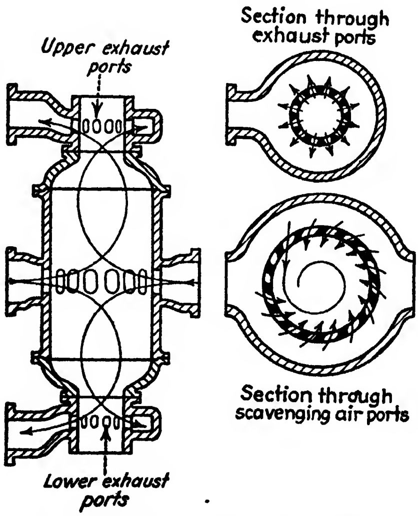


FIG. 94.—Method of scavenging Burmeister and Wain double-acting engine. (*British Motor Ship.*)

effective compression stroke for the engine than for that shown in Fig. 88.

Another type of engine that should be considered in connection with scavenging is the opposed piston engine. This type of engine is illustrated by the Sun Doxford engine, built by the Sun Shipbuilding & Dry Dock Company, shown in Fig. 96. This engine, which is of the two-cycle type, has two pistons working in each cylinder. The two pistons have a common combus-

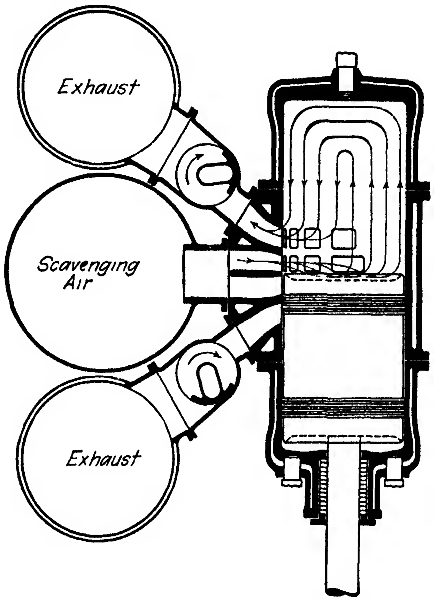


Fig. 95.—Diagram of cylinder of M.A.N. two-cycle, double-acting engine. (*British Motor Ship.*)

tion space at the center of the cylinder, one moving downward and the other upward on the working stroke after the injection of the fuel oil. The lower piston is connected to the crankshaft in the customary manner through a crosshead and connecting rod. The motion of the upper piston is transmitted to the crankshaft by means of two piston rods, two crossheads, and two connecting rods. The two connecting rods of the upper piston are attached to two cranks on the main shaft, which are located just forward and just aft of the crank to which the connecting rod of the lower piston is attached. There are thus three cranks for each cylinder, the two outer cranks being at 180 deg to the center crank. The

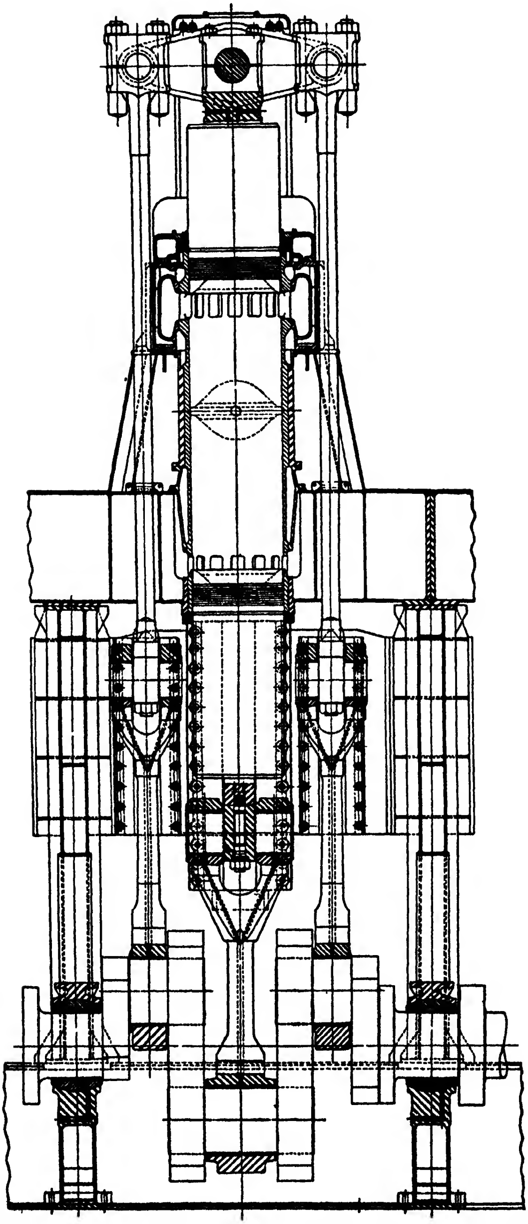


Fig. 96.—Longitudinal elevation through cylinder of Sun Doxford opposed-piston engine.

engine has no cylinder heads, since the upper piston takes the place of the cylinder head. Two fuel-injection valves are placed horizontally, one opposite the other, in the clearance space at the center of cylinder. As the gases expand after the injection of the fuel oil, one piston moves downward and one upward. As shown in Fig. 96, the upper piston upon nearing the end of its upward motion uncovers a ring of exhaust ports located in the cylinder wall and the lower piston shortly afterward uncovers a ring of scavenging air ports. The scavenging air rises and forces out the burned gases. This uniflow scavenging ensures a rapid and complete clearing of the cylinder of burned gases. It will be noted that both the exhaust and scavenging ports form a complete ring around the cylinder and not halfway around as in other engines using port scavenging and port exhaust.

The advantages claimed for this engine, many of which are inherent with the Junkers type, are as follows:

1. Piston loads balanced so that no stress comes on columns or cylinders.
2. Cylinder heads with all their complications eliminated.
3. Less fore-and-aft space required.
4. Good balance due to pistons moving in opposite directions. Allows excellent balance with a four-cylinder engine.
5. Rate of expansion of gases double that for other engines of same piston speed, allowing operation at low rpm.
6. Excellent scavenging.

It should be pointed out that whereas the opposed piston engine is short, it is also high. Some Doxford engines have been constructed with the stroke of the upper piston less than that of the lower one to improve the balance. The engine shown in Fig. 96, installed on the U.S. Maritime Commission's C-2 ships, has a 55-in. lower stroke and a 40-in. upper stroke, with a bore of 32 in. The Burmeister and Wain engine (Fig. 92) and the General Motors engine (Fig. 89) also have uniflow scavenging.[1]

103. The Scavenging Air Supply.—The scavenging air is produced by a number of different methods:

1. Reciprocating pump driven by the main engine.
    a. On the main crankshaft, in line with the working cylinders.
    b. On the side of the engine, driven by a lever from the main cross-head (Fig. 91).

[1] For other methods of scavenging, see the paper by W. S. Burn, "The Development of the Two-stroke Cycle Oil Engine," *Trans. Inst. Naval Arch.*, 1937, p. 278.

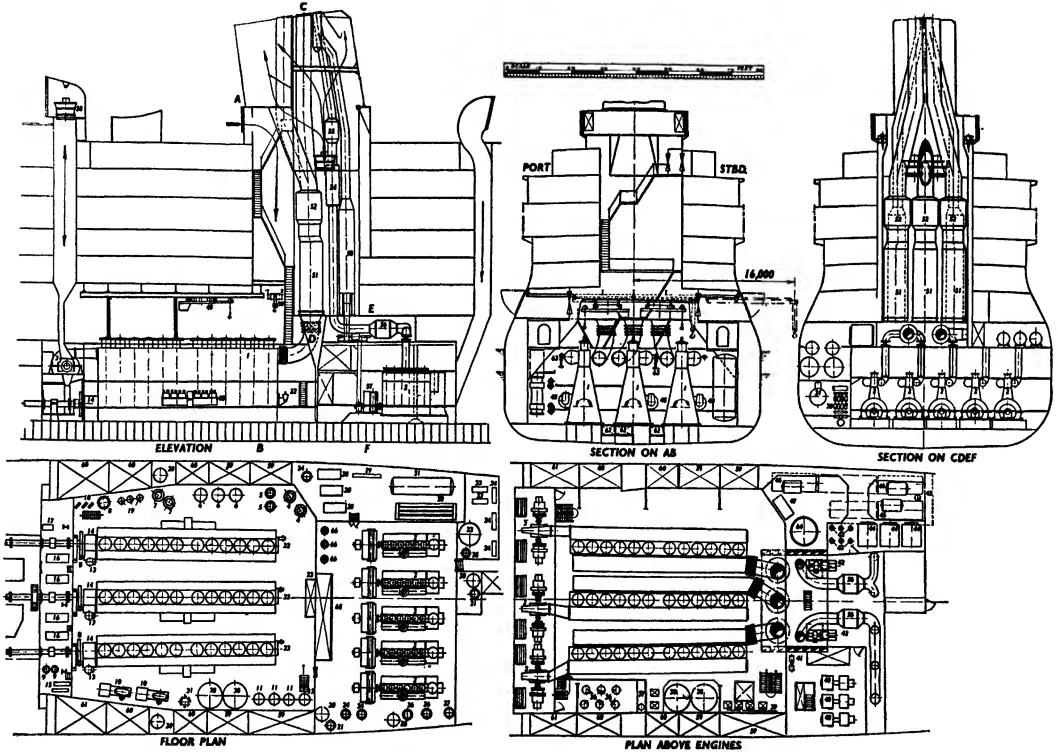


Fig. 97.—(*British Motor Ship.*)

ENGINE-ROOM PLANS OF THE M.S. "ORANJE"

(1) Main engines. (2) Auxiliary engines. (3) Scavenge air pumps. (4) Piston cooling pumps. (5) Sea-water cooling pumps (main engines). (6) Fresh-water coolers (main engines). (7) Fresh cooling water pumps (main engines). (8) Emergency bilge pump. (9) Fuel transfer pumps. (10) Air compressors. (11) Sanitary pumps. (12) Ballast pumps. (13) Turning motors. (14) Thrust block. (15) Daily fuel supply pumps. (16) Lubricating oil pumps. (17) Reserve lubricating oil pumps. (18) Lubricating oil filters. (19) Lubricating oil coolers. (20) Sewage tanks. (21) Sewage pumps. (22) Manœuvring platform. (23) Control instrument board. (24) Salt cooling water pumps (auxiliary engines). (25) Fresh-water coolers (auxiliary engines). (26) Fresh cooling water pumps (auxiliary engines). (27) Reserve lubricating oil pump (auxiliary engines). (28) Refrigerating compressors. (29) Control board for refrigerating installation. (30) Ammonia condenser. (31) Fresh-water cooler for air conditioning plant. (32) Cold water tank for air conditioning plant. (33) Circulating pumps for air conditioning plant. (34) Circulating pumps. (35) Emergency bilge pump. (36) Separators. (37) Vice bench. (38) Starting air reservoirs. (39) Switch boxes. (40) Transformers for power service. (41) Transformers for telegraphs. (42) Fuel oil and fan motor. (43) Norit filters. (44) Aeration tanks. (45) Ventilator. (46) Evaporator. (47) Warm water tank. (48) Fuel pumps. (49) Rotary crane. (50) Transverse crane. (51) Exhaust gas boilers. (52) Spark arresters for the main engines. (53) Oil-fired boilers. (54) Silencers (auxiliary engines). (55) Spark arresters (auxiliary engines). (56) Exhaust gas boilers (auxiliary engines). (57) Generators. (58) Ventilators. (59) Fuel oil tanks. (60) Fuel oil bunkers. (61) Fuel for boilers. (62) Lubricating oil drain tank. (63) Automatic starting gear. (64) Steam tank. (65) Main switchboard. (66) Boiler circulating and feed pumps.

2. Rotary blower attached to engine and driven by chain or gearing from the main crankshaft (Figs. 89, 90, and 92).
3. High-speed turbo blowers driven by separate electric motors (Fig. 97).

When the compressor is of types 1 or 2, it is usually customary to fit one or two to take care of all the working cylinders. However, in the Sulzer engines used on the *M.S. Prins Albert* (Fig. 91), a separate scavenging pump was used for each of the 12 cylinders, one being driven off each crosshead. When independent turbo blowers are used, several are usually installed so that the stopping of any one will not require the shutting down of the engines.

The attached blowers have the advantages that they reduced the engine-room length and require less power to drive them. The independent blowers have the advantage of flexibility and no loss in main-engine bhp due to the power necessary to drive the blowers.

104. Diesel Engine Auxiliary Systems.—Before proceeding to a more detailed study of Diesel engines, the student should become familiar with the various auxiliary systems required for a Diesel engine installation. These are briefly outlined below. In connection with the following discussion, the reader should study the machinery layout of the *M.S. Oranje* shown in Fig. 97.

1. *Fuel Oil System.*—The fuel oil system consists of the main bunker tanks, transfer pump, settling tanks, service tanks, centrifugal separators, booster pump, engine-driven fuel-injection pumps, fuel-injection valves in the cylinders, together with the necessary piping valves and strainers, etc. When heavy, viscous fuels are used, approaching those used by steamships, oil heaters will also be required.

2. *Starting-air System.*—Diesel engines are started and reversed by compressed air under a pressure of around 350 lb per sq in. The air for starting is supplied by a motor-driven compressor (frequently called the "maneuvering compressor"). Sufficient air for starting and maneuvering is kept in starting-air tanks in the engine room (Fig. 97, No. 38). The starting-air system consists of the maneuvering air compressor, with necessary air coolers, starting-air tanks, starting-air valves on the engine cylinders, together with the necessary piping.

3. *Circulating Water System.*—A cooling-water system must be supplied with an oil-engine installation for cooling *directly* or *indirectly* the following: cylinder jackets, cylinder heads,

crosshead guides, exhaust manifolds, exhaust piping, pistons (and piston rods of double-acting engines), exhaust valves, lubricating-oil, maneuvering compressor jackets and air coolers, and, at times, the barrels of fuel-injection valves. In small engines the piston and exhaust valves are not cooled.

The jackets, heads, and exhaust piping of the auxiliary engines are usually but not always cooled by one or more separate systems (see Fig. 97, No. 24). The pistons and piston rods are cooled by a separate systems; usually, lubricating oil is employed as the cooling medium. This oil must be cooled, like the main lubricating supply, by the main circulating water system.

The main cooling-water system therefore circulates water through the main cylinder jackets and heads, the exhaust piping, lubricating-oil cooler, piston-oil cooler, maneuvering compressor system and through the crosshead guides, exhaust valves, and fuel-injection valves when required.

Both salt water and fresh water are used for the main cooling system. When a separate fresh-water system is employed, an extra fresh-water cooler and pump for circulating sea water through the fresh-water cooler must be supplied. The advantages of the separate fresh-water system are that higher temperatures of the cooling water can be used, salt-water fittings are not necessary beyond the fresh-water cooler, muddy and other objectionable overboard water cannot get into the engine jackets, main circulating-water piping, and circulating-water pumps Two or more circulating-water pumps are usually supplied (see Fig. 97, Nos. 5, 6, and 7).

4. *Piston-cooling System.*—The piston-cooling system consists of the piston-cooling pump, piston-oil cooler, and necessary piping. Since the pistons are reciprocating, it is necessary to have telescoping pipe connections or elbow joints between the piston and the stationary part of the engine for the delivery and return piping. When the engine is double-acting, the cooling oil is passed up through the piston rod to the piston, and the warm oil returns through a passage in the center of the rod. The flexible connections are then made to the cross-heads. Oil is the usual cooling medium for the piston, although fresh water is sometimes used.

5. *Lubricating-oil System.*—The lubricating of an oil engine may be divided into three parts:

1. Journal bearings, crankpins, crossheads (or wrist pins), and camshaft bearings.
2. Pistons of working cylinders.
3. Rocker arms, cam rollers, etc.

In large engines, the journal bearings, crankpins, crossheads, and camshaft bearings are lubricated by the main lubricating-oil supply, the oil being circulated under pressure from the sump by a motor-driven lubricating-oil pump. A lubricating cooler must be in the circuit, as mentioned under (3), to remove the heat that the lubricating oil has absorbed in its passage through the lubricating-oil circuit. Lubricating-oil storage tanks, sump tank, centrifugal separator or purifier, strainers, piping, and fittings and valves complete the system.

The working pistons are lubricated by forced-feed mechanical lubricators, the oil being injected when the pistons are at the bottom of their travel. Rocker arms, etc., are usually oiled by hand.

6. *Scavenging-air System.*—Scavenging air was fully discussed in Art. 101. The complete system consists of inlet muffler, piping, scavenging-air pumps, and air manifold on the engines.

7. *Exhaust System.*—The exhaust system consists of the exhaust valves (if fitted), exhaust manifold on engine, exhaust piping, muffler, and any device that may be fitted for the utilization of waste heat.

8. *Auxiliary Generating Sets.*—The auxiliary generating sets are driven by auxiliary Diesel engines (Fig. 97, No. 2) and supply electric power for operating the fuel-oil-transfer pumps, starting-air compressor, circulating-water pumps, piston-cooling pumps, lubricating-oil pumps, scavenging-air blowers (when independent of the main engine), and power for ship's use, such as sanitary and fresh-water pumps, ballast and fire pumps, steering, lighting, refrigeration, ventilation, etc.

9. *Injection-air System.*—The older Diesel engine installations that use the air-injection system for injection of the fuel oil have an additional system for supplying the injection air. This system is made up of a three-stage compressor on the main crankshaft for compressing air to around 1,000 lb per sq in.; intercoolers and aftercoolers for cooling the compressed air; high pressure air-storage flasks; and high pressure air piping from compressor to fuel-injection valves. The system must be

supplied with the necessary cooling-water service for cooling compressor jackets, intercoolers, and aftercoolers.

105. Engine Dimensions and Mep's.—Two horsepowers are customarily used in discussing oil engines, namely, indicated horsepower (ihp) and brake horsepower (bhp). The former is obtained by taking indicator cards, as with steam engines, and is the power developed in the cylinders of the engine. The bhp is measured by means of a mechanical brake or imput to a generator and is the equivalent of the shp used with geared turbines. The bhp is the ihp minus the frictional horsepower. The relationship of the two is

$$e_m = \frac{\text{bhp}}{\text{ihp}}$$

where e_m = mechanical efficiency.

The ihp developed in the head end or crank end of a cylinder is obtained by the familiar formula

$$\text{Ihp} = \frac{P_iLAN}{33{,}000}$$

and the total ihp of the engine is obtained by adding up the ihp developed above and below the pistons of all the cylinders. If we assume for discussion that the mean effective pressure (mep) is the same for all cylinders and the area of the piston rods can be neglected

$$\text{ihp} = \frac{P_iLANn}{33{,}000}$$

where P_i = indicated mep (often referred to as the "mean indicated pressure," mip), which is obtained by dividing the area of the indicator card by its length and correcting for the spring used.

L = length of stroke, in feet.

A = effective area of the piston. In a double-acting engine the effective area of lower side of the piston is the piston area minus the area of the piston rod.

N = working strokes per minute. In a four-cycle single-acting engine, N is one-half the rpm; in a two-cycle double-acting engine it is 2 × the rpm.

n = number of cylinders.

Obviously the P_i is likely to vary somewhat from cylinder to cylinder, and the effective A is different for the top and bottom of the cylinder; hence in practice the expression given above for ihp cannot be used directly in calculating horsepower. The total power of an engine is obtained by adding up the individual powers developed in the top and bottom of each cylinder. The preceding expression does, however, have value in discussing the power of oil engines.

Bhp can be expressed in a similar manner by the expression

$$\text{Bhp} = \frac{P_b L A N n}{33{,}000}$$

where P_b = brake mean effective pressure.

The value of P_b is obtained by measuring the bhp during a shop trial and solving the preceding expression for P_b. Obviously, it cannot be obtained on shipboard in the simple manner in which P_i is obtained.

Costs, weights, fuel consumptions, and mep's of Diesel engines are expressed in terms of both ihp and bhp. However, since the bhp is the power that is required to drive the ship at a given speed, costs, weights, and fuel consumptions have more significance when expressed on a bhp basis.

The relationship of these factors is as follows:

$$\begin{aligned} \text{Bhp} &= e_m \times \text{ihp} \\ P_b &= e_m \times P_i \\ f_b &= f_i \div e_m \end{aligned}$$

where f_i = fuel per ihp per hour.
f_b = fuel per bhp per hour.

Brake mep's of marine engines vary between 60 and 90, being lower for two-cycle engines than for four-cycle engines because of cylinder temperature considerations (see Art. 111). Supercharged engines (Art. 112) have mep's somewhat higher than the upper limit given above.

The working strokes per minute N, used in the foregoing general expression depends upon the rpm, whether the engine is two- or four-cycle and whether it is single- or double-acting. For engines connected to the propeller shaft, the rpm is fixed within rather narrow limits for a given speed by considerations of propeller efficiency. By adopting the geared Diesel drive or the

Diesel-electric drive, the engines can be run at higher revolutions than those required for best propeller efficiency. Obvious inertia effects, which vary with the mass of the reciprocating parts times the square of the rpm, limit the upper rpm that can be used in engines of large power. Considerations of piston speed and time for scavenging in two-cycle engines also exert an influence on the revolutions.

The expression LAn (stroke, piston area, and number of cylinders) represents the factors in the bhp expression that fix the engine cost (material and labor), weight, and space occupied. An increase in the bhp by increasing any or all of the factors L, A, and n will increase the engine cost and weight.

Obviously, therefore, to obtain additional power for the least cost and weight increase, P_i and N should be increased, not bore, stroke, or number of cylinders. Hence there is always an incentive to make P_i as large as possible and to push up the rpm to keep engine costs and weights down.

The maximum bhp that can be developed for an engine with fixed values of L, A, N, n, and e_m depends directly on the value of P_i. Much confusion exists in regard to the horsepowers of Diesel engines because of the values of P_i used. If an engine is designed with a high value of P_i, it may not be able to develop its rated power except for short periods, cannot maintain its designed bhp in service at sea day in and day out, and may not have any reserve of power for an emergency. On the other hand, if an engine of the same rpm and bhp is designed with a conservatively low P_i, so that it can maintain its power satisfactorily in service and have an overload capacity for short periods, the combined value of $L \times A \times n$ will have to be increased, which will add to the size, weight, and cost of the engine. These facts must be kept clearly in mind in comparing costs and weights per horsepower. In order to have an engine that will perform satisfactorily in service with low maintenance expense, the mep must be reasonably low. On the other hand, in order to reduce Diesel engine costs and weights, the designers and research men in the Diesel field are bound to direct their energies to the possibilities of using higher mep's in Diesel engines. Values of service P_i's used in a number of Diesel installations are given in Table XVIII. The factors influencing the value of the indicated mep are discussed in Art. 111.

106. Mechanical Efficiency.—The mechanical efficiency of an engine is

$$e_m = \frac{\text{bhp}}{\text{ihp}} = \frac{\text{ihp} - \text{friction horsepower}}{\text{ihp}}$$

$$= 1 - \left(\frac{\text{friction horsepower}}{\text{ihp}}\right)$$

When the indicated horsepower of an engine is reduced the frictional horsepower decreases at approximately the same ratio as the rpm, but as the ihp of a marine engine decreases nearly as the cube of the rpm, it can be seen from the above expression that the mechanical efficiency falls off with a decrease in power.

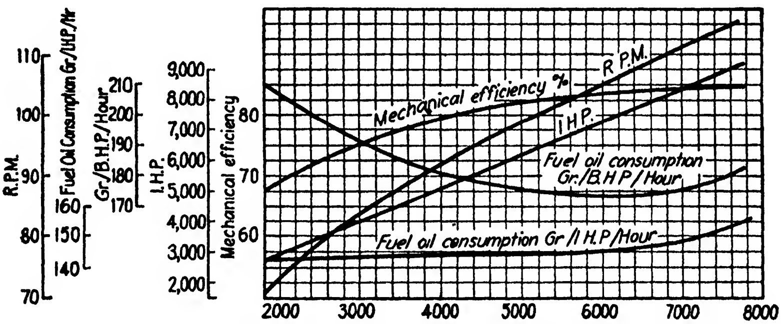


FIG. 98.—Curves showing results of shop test on a Burmeister and Wain-type two cycle, double-acting engine. (*British Motor Ship.*)

As the power is reduced, the fuel burned per ihp per hour remains fairly constant or decreases slightly. However, as e_m is decreasing and $f_b = f_i \div e_m$, the fuel per bhp per hour f_b increases as the load is reduced. These relationships are clearly shown in Fig. 98, which shows the trial performance of a Burmeister and Wain-type double-acting two-cycle engine at various bhp outputs.

As a rough average, the mechanical efficiency at full power of a large, slow-speed four-cycle engine using solid injection can be taken as 85 per cent, being somewhat lower for single-acting engines than for double-acting engines. Two-cycle engines with attached scavenging pump will have about the same value for mechanical efficiency. Two-cycle engines with independently driven scavenging blowers will have a mechanical efficiency about 5 points higher, or about 90 per cent. Engines using air injection lose about 10 points in mechanical efficiency over solid-injection engines because of power absorbed by the high pressure air compressor.

107. Fuels for Oil Engines.—Fuels generally used in Diesel engines are distillates and are not residual fuels such as are used in steamships. In some cases, special light residual fuels that have been found suitable are used, and at times Diesel engines are operated on blends of distillates and residual fuels. The viscosity of the fuels is usually such that no heating of the oil is required. In recent years motor ships have begun to use residual fuels to some extent. These are of such high viscosity that heating is required before the fuels are delivered to the engine fuel pumps. Outstanding among such motor ships that use heated residual fuels are the C-2 and C-3 types of motor ships built in 1939–1941 by the U. S. Maritime Commission.

1. *Viscosity.*—One of the most important characteristics of Diesel fuels is viscosity. A low viscosity is essential to ensure free flow through the piping and atomization at the injection valves of the engine. The Saybolt Universal viscosimeter is generally used for measuring the viscosity of the lighter Diesel fuels as well as lubricating oils; the Saybolt Furol viscosimeter, however, is used for the heavier oils now being used with Diesel engines. As noted in Table XIII, the viscosities for unheated oils range from 50 to 250 SSU at 100°F. The higher the rpm the lower is the viscosity required. As shown in Table XIV, A.S.T.M. No. 2 Diesel fuel classification sets a maximum viscosity of 45.5 SSU at 100°F for fuel for high-speed engines. In order to avoid seepage at the valve, a viscosity below 30 to 35 is generally considered undesirable. A low pour-point temperature is essential, especially for ships operating in cold waters, to ensure that oil carried in the double bottoms can be pumped without the necessity of heating. Fuels should also have sufficient lubricating qualities to lubricate the pump plungers.

2. *Ash.*—The solids remaining after a fuel oil has been subject to burning at a high temperature are termed the "ash." The ash is made up of the sediment in the fuel, the noncombustible chemicals in solution in the water contained in the fuel, and the noncombustible chemicals in solution in the fuel itself. The two former constituents can be removed by centrifuging the fuel, but the third cannot. An extremely low ash content is an

TABLE XIII.—CHARACTERISTICS OF DIESEL FUEL OILS FOR SLOW-SPEED ENGINES

	1	2	3	4	5	6	7	8 (bunker *C*)
Type of fuel		Distillate	Distillate	Residual		Residual		Residual
Viscosity SSU at 100°F		100	40	840	67	540–1,700*	50–250	8,500*
Viscosity SSF at 122°F						30–70		300†
Specific gravity, at 60°F		0.886	0.825–0.85			0.97–0.945		0.96
A.P.I. gravity at 60°F		28	40–35			14–18	14–38	16
Ash, per cent by weight	0.05	0.05	0.01	0.10	0.03		0–0.05	0.13
Carbon residue (Conradson), per cent by weight	3.5	2.5	Trace	8.0	3.0	5–9	0.2–10.0	14.0
Water and sediment, per cent by volume	0.5	0.5	1.0	1.0	0.5	1.0	0.01–2.0	2.0
Sulphur, per cent by weight	2.0		0.65		2.0	1.5	0.2–2.5	2.5
Flash point, °F	140		145	150	150	150–200	125–200	150
Pour point, °F	35		0		30	35		
Hard asphalt, per cent by weight				4.0			0.5–2.5	
Heat value, Btu per lb				18,250	18,700	18,714	17,700–19,150	

* Approximate values.

† Viscosity 300 SSF at 122°F = approximately 8,500 SSU at 100°F.

1. A.S.T.M. proposed (1940) Diesel-oil classification 4; values are all maxima allowable except flash point, which is a minimum.
2. A typical American Diesel fuel.
3. A Diesel fuel sold by one oil company.
4. Heavy Diesel fuel, British Standards Institution, 1937 (*British Motor Ship*, vol. 21, p 217).
5. "Marine" Diesel fuel, British Standards Institution, 1937, for engines of less than 800 rpm.
6. Fuels used in some of the U.S. Maritime Commission motor ships. Viscosity = 30–70 SSF at 122°F = approximately 540–1,700 SSU at 100°F. Centrifuged and heated to 180°F before injecting into engine.
7. Range of specifications of a large number of American engine manufacturers for slow-speed engines, below 500 rpm, (*British Motor Ship*, vol. 21, p. 86). All values except flash point are maxima.
8. A typical bunker *C* residual fuel oil of maximum allowable viscosity (not cracked).

essential requirement for Diesel engine fuels, for ash is one of the causes of excessive wear of cylinder liners and piston rings and hence of heavy maintenance expenses. It will be observed that in the various specifications and analyses given in Tables XIII and XIV, the ash content of the distillate fuels ranges between 0.01 and 0.05 per cent. The higher the engine rpm the lower is the permissible ash content. The ash content of bunker *C* fuels usually is much higher than this (see fuel No. 8, Table XIII, and Art. 21).

TABLE XIV.—DIESEL FUEL-OIL CLASSIFICATION
(American Society for Testing Materials, 1941)

Grade of Diesel oil	Flash, °F, min.	Water and sediment, per cent by vol., max.	Carbon residue, per cent by wt., max.	Ash, per cent by wt., max.	Viscosity, SSU, 100°F, max.	Sulphur, per cent by wt., max.	Pour point, °F	Cetane No., min.	Type of engine using
1	100	0.05	....	0.01	....	0.5	0	50	High speed
2	140	0.05	0.20	0.01	45.5	1.0	20	45	High speed
3	140	0.10	1.0	0.02	65	1.5	35	35	Medium speed
4	140	0.50	3.5	0 05	....	2.0	35	30	Low speed

3. *Carbon Residue.*—The hard carbon and tarry residues left in the engine after combustion are objectionable characteristics of Diesel fuels, because they necessitate constant cleaning of valves. Carbon residue is also considered the cause of some of the liner wear. A carbon residue by the Conradson test of over 2.5 to 3.5 per cent is generally considered undesirable for the distillate fuels, but, as shown in Table XIII, the residual fuels now being used have percentages of carbon residue ranging from 5 to 9.

4. *Sulphur.*—A low sulphur content is usually specified for Diesel fuels, because experience indicates that fuels high in sulphur are unsatisfactory for use in oil engines, and it now appears that excessive liner wear is associated with an oil that has a sulphur content of over 2 per cent. At low temperatures, the sulphur dioxide in the exhaust gases may form sulphuric acid, which will attack the exhaust piping.

5. *Liner Wear.*—The use of low-grade residual fuels in Diesel engines has usually resulted in excessive liner wear. This has necessitated more frequent liner replacements, which has

increased the maintenance expense of a Diesel engine using such fuels. This excessive liner wear has generally been considered to be due to the higher ash content of these heavier residual fuels. Although ash has been the largest contributing cause of this liner wear, other characteristics of the fuel and operating conditions also have an influence. Broeze and Gravesteyn, in a very instructive paper, "Fuel and Wear in Diesel Engines,"[1] have pointed out that the following factors influence liner wear:

1. Oil with sulphur content above 2 per cent (wear probably due to the formation of sulphur trioxide).
2. Ash, especially ash containing oxides of iron and silica.
3. Carbon residue. However, since a high sulphur content is usually associated with high carbon residue and high ash content, it is difficult to separate the influence of these three factors.
4. Increased loads on the engine. The authors suggest using a reasonably low mean indicated pressure (P_i) when engines are operated on low-grade fuels.
5. Incomplete combustion. Incomplete combustion may be caused by improper air and fuel mixture, too low a percentage of excess air, or overcooling of the engine at reduced loads.
6. Lubricating oil. The authors suggest using an oil of low viscosity. No doubt the method of applying lubricating oil to the cylinders has an influence on liner wear. Liner wear is greater in two-cycle engines than in four-cycle engines, no doubt because of the absence of the idle stroke of the four-cycle engine, which is made use of to lubricate the cylinder walls.

In recent years, the application of chromium plating to the liner walls has been very effective in reducing wear. Cases have been cited in which the proper application of chromium plating has decreased liner wear by a marked degree. The liners of the engines of the motor ships constructed by the U.S. Maritime Commission are chromium-plated.[2]

6. *Ignition Quality.*—One of the essential characteristics that should be possessed by a Diesel fuel is ability to ignite rapidly when injected into the engine cylinder. At the present time the ignition quality or delay period is rated in terms of a scale of cetane numbers. Cetane is one of the hydrocarbons existing in petroleum that has excellent ignition qualities. A fuel oil is tested[3] in a standard engine, and the engine is then operated on

[1] See *British Motor Ship*, vol. 19, p. 216.
[2] *Trans. Soc. Naval Arch. & Mar. Eng.*, 1940, p. 335.
[3] See *British Motor Ship*, vol. 21, p. 238, for methods of conducting these tests.

various mixtures of cetane and alpha methylnaphthalene. The percentage of cetane in the blended fuel that gives the same ignition quality as the tested fuel is termed the "cetane number" for the fuel in question. Since a 100 per cent cetane would have a cetane number of 100, it is obvious that the higher the cetane number the smaller will be the delay angle for a given compression ratio. The cetane number is an important characteristic of fuels for high-speed engines. The specified minimum values for cetane numbers average between 30 and 50 (see Table XIV).

7. *Evaporation Curve.*—All petroleum products are a complex mixture of hydrocarbons. Petroleum fuels do not evaporate at a constant temperature as does a homogeneous fuel such as alcohol, but parts are distilled off at low temperatures, and other parts require higher temperatures. A Diesel fuel might begin evaporation at around 400°F, and a temperature of 650–700° be required to evaporate the last of the fuel. The end-point temperatures of the evaporation curve, *i.e.*, the temperature range required to evaporate the last 10 per cent of the fuel, gives a good indication of burning properties of the fuels. The shape of the evaporation curve of a fuel is much more important for high-speed engines than for slow-speed engines.

108. Use of Residual Fuel Oils in Diesel Engines.—At the present writing (1942), the average fuel consumption of a cargo motor ship is around 0.38 lb per bhp per hr for all purposes; that of a modern steamship is around 0.60; and the new U.S. Maritime Commission's steamships have obtained values as low as 0.55 on trial. This marked decrease in the fuel consumption of steamships has greatly decreased the advantage that the motor ship had 20 years ago, when fuel consumptions were of the order of 0.42 for the motor ship and 1.00 for the steamship. If the cost of fuel per ton for motor ships were the same as for steamships (and if we could assume that the fixed charges and maintenance costs for the two types of machinery were equal), the motor ship would have an annual fuel cost of around 37 per cent less than that of the steamship for fuel consumptions of 0.38 and 0.60 lb per bhp hour.

The fuel costs per ton of Diesel fuel and bunker *C* are not the same, however. In the fall of 1940, the prices in New York were approximately $1.25 per bbl for bunker *C* and $1.70 per bbl for Diesel oil. In July, 1941, the New York prices were $1.35 and $2 per bbl; in 1930, the prices were around $1.10 and $2.05.

If we take the July, 1941, prices and correct them for the difference in specific gravities of the two fuels, the costs per ton are as follows:

Bunker *C*, \$1.35 × 6.54 bbl per ton (assuming 13° A.P.I.)
= \$8.83 per ton
Diesel oil, \$2.00 × 7.21 bbl per ton (assuming 28° A.P.I.)
= \$14.42 per ton

Thus, whereas the ratio of fuel consumption per bhp-hour is 0.60/0.38 = 1.58 and the ratio of fuel costs per ton is 14.42/8.83 = 1.64 the advantage that the motor ship has in fuel consumed per bhp-hour is more than lost in the cost of fuel per ton.

True, this cost ratio is only in one place, the Atlantic seaboard of the United States. In European, Asiatic, and South American ports, the ratio of fuel costs is appreciably lower, although the cost per ton of each fuel is much higher than in New York.

It will be shown later that many steamships have a fuel consumption appreciably higher than the value used above. This is especially the case in ships with machinery installations of small horsepower, where the refinements used to get this low consumption are not possible or feasible. In such cases Diesel machinery does not suffer from the comparison outlined above.

However, with prices as they are today, a motor ship burning Diesel fuel oil is at a serious disadvantage, compared with the modern steamship using 0.60 lb of oil per shp per hr. Hence there is a strong incentive for the motor ship to burn a lower grade and hence a cheaper fuel. If a Diesel engine could burn a fuel such as listed in column 6 of Table XIII, which costs, say, around \$10 a ton, the ratio of fuel costs would be 10.00/8.83 = 1.13, and the advantage in fuel costs would be with the motor ship.

Diesel engines burning heavy residual fuel have shown a higher unit fuel consumption, namely 0.40 to 0.42 lb per bhp per hr instead of 0.38. If we use a value of 0.41, the ratio of fuel costs becomes

$$\frac{\$8.83 \times 0.60}{\$10 \times 0.41} = 1.29$$

i.e., the steamship's fuel cost is 29 per cent higher than that of the motor ship.

Other factors besides fuel costs must, of course, be taken into consideration in any complete comparison. Such factors are the higher first cost of machinery and higher lubricating-oil consumption of the Diesel installation; annual repairs and maintenance; reduced weight and volume of fuel carried by the motor ship; and absence of make-up feed-water weight on the motor ship.

Burning these heavier fuels in Diesel engines has a number of disadvantages and objectionable features, and at the present writing the over-all relative advantage of these fuels is not fully known. The use of these oils in Diesel engines requires that they be heated from bunker to injection valve tip to reduce the viscosity so that the oil can be pumped and injected into the engine cylinder. In port, the oil and injection valves must be heated by warm-water circulation so that the oil will not become too viscous to prevent starting. To overcome this difficulty, some engines use a lighter Diesel oil for starting and before stopping so that the piping, injection valves, and fuel pumps will be filled with an oil that will not require heating in port. With heavy residual oil, there must be more routine clearing of valves and piston rings that may increase operating expenses through the employment of shore gangs or payment of overtime to the crew. The variation in the types of residual oil bunkered at various world ports will require engine adjustments to be made. The increased ash, carbon residue, sulphur, and poorer combustion will cause liners to be renewed more often, which will result in higher maintenance costs. Just how much this item will amount to can be determined only over a period of time. Although these fuels are now being successfully used for the main propelling engines, which run at low rpm, their use in the auxiliary engines, which run at 300 to 400 rpm, may not be so successful, and it may be advisable to use a lighter Diesel fuel oil in the auxiliary engines. The U. S. Maritime Commission ships, which use a residual fuel, are heating the oil to around 180°F and are maintaining a jacket-water temperature of around 170°F.

109. Fuel Injection.—All marine Diesel engines built today use airless injection, *i.e.*, the fuel is injected into the cylinder under high oil pressure without the assistance of injection air, which was used on all the early Diesel engines. This type of injection is also referred to as "solid injection" and "mechanical injection."

The first Diesel engines built used air injection and were equipped with three-stage air compressors that compressed the air used for fuel injection up to a pressure of around 1,000 lb per sq in. The air pressure was in communication at all times with the injection valve. A measured amount of fuel was pumped up to the barrel of the injection valve by the fuel oil pump while the valve was closed, and at the correct point in the cycle (about 5 deg before dead center), the valve was opened by cam action. When the valve opened, the injection air blew into the cylinder and carried the fuel along with it, atomizing it into a fine foglike spray. The injection air not only atomized the fuel but created a turbulence and distributed the fuel particles throughout the combustion space. The injection valve closed at about 30 deg after dead center.

With the solid-injection system, the fuel-injection valve is usually held closed by a spring, and at the proper point in the cycle a plunger of the fuel oil pump sends a measured amount of fuel to the injection valve under a pressure around 5,000 to 8,000 lb per sq in. The spring-loaded valve opens under this oil pressure, and the fuel is forced in the combustion space at high velocity through a number of very small openings. When the plunger of the fuel pump has completed its effective stroke, the oil pressure falls off, and the injection valve closes under the action of its spring.

The airless-injection engine was developed in order to have a simpler and less complicated engine by eliminating the air compressor with its air coolers and high pressure piping and also to improve the mechanical efficiency of the engine and hence lower its fuel consumption per bhp per hour. In eliminating the compressor, however, all the benefits of the high pressure injection air in assisting combustion were lost.

Difficulties were experienced in the development of the solid-injection engine in securing proper penetration of the fuel when the oil was sent in as a spray of very fine particles and in securing rapid combustion through the absence of turbulence in the combustion space. The final result of the development of this method of fuel injection is that compression pressures have been reduced over those used with air injection, and the fuel injection begins at a crank angle appreciably before dead center. High oil pressures and numerous jets of oil are used. In large engines two fuel-injection valves are frequently employed, for it was found that two valves in the cylinder head gave better distribution of the fuel than a single valve centrally located.

A theoretical indicator card for an engine using solid injection is shown in Fig. 99. To bring out clearly the events of the cycle,

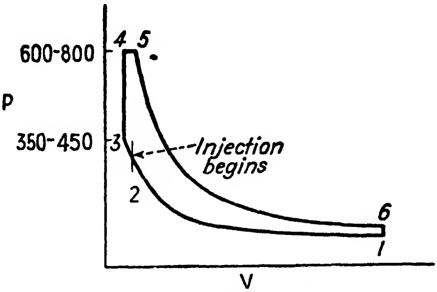


FIG. 99. – Theoretical PV diagram for an engine using solid injection.

this PV diagram has been drawn with sharp corners. In the actual indicator card the corners are rounded off, and the events of the cycle are not so easily located as in Fig. 99. An actual indicator card from a two-cycle engine is shown in Fig. 100.

In Fig. 99, during the compression from 1 to 3, the fuel injection begins at point 2, say 10 deg before upper dead center. Because of the low compression and absence of turbulence when the fuel is injected, the fuel does not start to burn instantly, as in the true Diesel engine (Fig. 85*a*). From 2 to 3, there is a delay period. At or just before point 3, burning of the fuel starts at a nuclear point somewhere in the fuel-air mass, and the flame spreads almost instantaneously through the combustion space, resulting in an explosion similar to that in a gasoline engine and a rapid increase in pressure from 3 to 4. The maximum pressure is thus greater than in engines operating on the true Diesel cycle. Fuel continues to be injected up to point 5, somewhat after upper dead center, and all the fuel that enters the combustion space after ignition has taken place burns at or approximately at

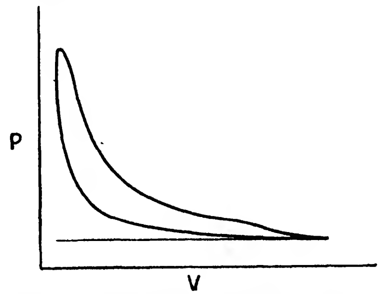


FIG. 100.—Indicator diagram from a two-cycle solid-injection engine.

constant pressure, as in the true Diesel engine. Thus, with solid injection, we have a dual cycle, part Otto and part Diesel.

The ignition delay period for a given fuel and degree of turbulence is more or less a fixed period of time, and the faster an engine runs the earlier must the fuel injection begin. In high-speed engines (800 rpm and upward) all the fuel is sometimes injected before dead center. In such a case, the constant pressure part of the PV diagram (4–5 in Fig. 99) does not exist. The card becomes a pure Otto card, and since more fuel is in the combustion space when ignition takes place the rise in pressure is greater, often reaching 800 to 900 lb per sq in. Obviously, engines running at high rpm must use a fuel that ignites more quickly than an engine running at low rpm.

Two methods of operating the fuel-injection valves are used with marine engines. The common method is to use spring-loaded valves in the cylinder, which are opened by oil pressure. The fuel pump has a separate plunger for each cylinder (or each end of each cylinder, in double-acting engines), which is driven by cam action so timed that the fuel is injected into the cylinder at the proper point in the cycle.

Another method is the so-called "rail" system, as used on the Doxford engine. Here a constant oil pressure is maintained in the fuel-delivery pipe or "rail" leading to the injection valves, and cam-operated injection valves are opened at the correct time to admit fuel to the cylinder; the fuel flows into the cylinder as long as the cam holds the injection valve open.

Usually the fuel pump, with all its plungers, is located on the side of the engine (see Figs. 90 and 92) and operated by a short camshaft. In the General Motors Diesel engine the valve and pump plunger are incorporated into one unit for each cylinder and located on the top of the cylinder. A camshaft extending along the engine operates plungers on the top of each cylinder by means of push rods and rocker arms (Fig. 101).

The power output of an oil engine is controlled by the effective stroke of the fuel pump. The actual stroke of the fuel-pump plungers is a fixed amount, but the length of time that fuel is actually delivered to the pipes leading to the injection valve is controlled by the governor and hand control. Various methods are used to control the effective stroke of the fuel-pump plunger. The most common method is that used in the Bosch fuel pump

Here a spiral groove is cut in the pump plunger and two ports in the bushing in which the plunger works. By rotating either the bushing or the plunger, the position of this spiral groove can be varied with respect to the ports in the sides of the bushing. In this way, as the plunger starts down on its delivery stroke, it is possible to turn the plunger or bushing so that all or part of

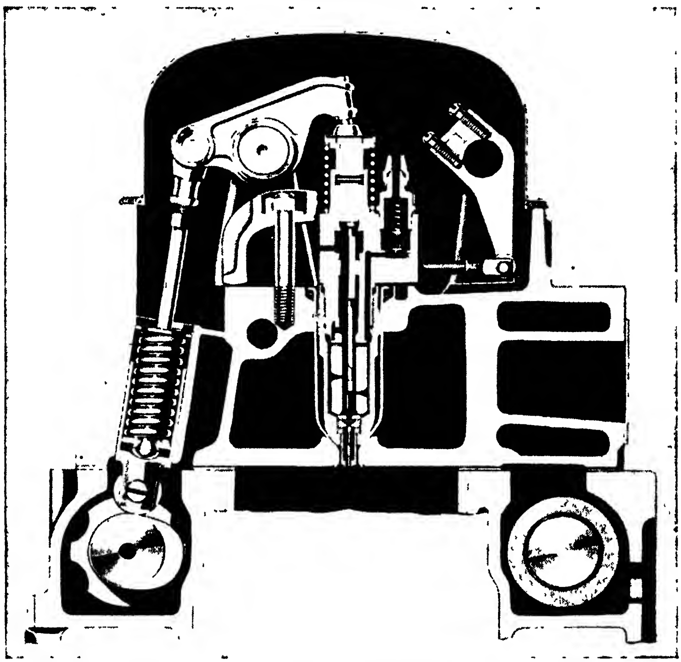


FIG. 101.—Fuel injection mounting. (*General Motors Corporation Engine.*)

the oil is discharged back through one of the ports into the oil suction pipe instead of being sent to the injection valve. When all the oil pumped by a plunger is sent to the injection valve, the engine operates at its maximum overload capacity; when all the oil is sent back into suction line, the engine stops. A full comprehension of the operation of the fuel pump can be obtained only by a study of an actual pump. In Fig. 101, the pump plunger, ports, and gear for rotating the plunger can be clearly seen.

Rapid combustion in oil engines is secured by one or more of the following methods:

1. High oil pressure, with one or more injection valves with numerous short oil jets.
2. Reduced compression in order to allow penetration of the oil jets.
3. Injection before maximum compression.
4. Two injection valves so situated that the jets impinge on one another.
5. A large amount of excess air for combustion.
6. Turbulence of the combustion air so that the air will tend to seek out the fuel as well as the fuel to seek out the air. This turbulence is commonly obtained by sending the scavenging air into the cylinder with a rotary motion so that the air will be in motion during fuel injection.
7. Precombustion chambers.

The precombustion-chamber engine was developed when solid injection began to be used, as a means of overcoming some of the difficulties experienced when air injection was given up. A great variety of these engines is built, but the horsepower seldom exceeds 500 or 600. In these engines the fuel oil is injected into a small precombustion chamber that is connected to the top of the main cylinder by one or more restricted openings. During compression, the air from the main cylinder is forced into the precombustion chamber through the restricted openings at high velocity, and turbulence is created in the precombustion chamber. Oil is injected into the small chamber under a comparatively low pressure. The more volatile parts of the oil ignite, causing an explosion that forces the whole contents of the precombustion chamber out through the restricted opening into the main cylinder. The violent turbulence created as the oil particles, air, and burning gases pass through the restricted area causes the oil to burn rapidly. Some engines using a precombustion chamber require the addition of a small amount of heat, such as a glowing punk, in starting from a cold condition. Details of construction and operation of this type of engine are fully described in the catalogues of the various concerns that build them. They are usually of the two-cycle single-acting type.

110. Volumetric Efficiency.—The volumetric efficiency, e_v, of a Diesel engine is

$$\frac{\text{Volume of pure air at atmospheric pressure and temperature in cylinder at beginning of compression}}{\text{Volume of piston displacement}}.$$

Figure 102 shows the negative loop of the indicator card of a four-cycle engine, taken during the exhaust and suction strokes, with the pressures greatly magnified. It will be observed that the pressure during the exhaust stroke is above atmospheric pressure (approximately 1 lb), and the suction pressure is about the same amount below atmospheric pressure. This area represents the negative ihp during the exhaust and suction strokes. The piston displacement is $V_1 - V_2$, and the clearance volume is V_2. The burned gases are pushed out of the cylinder from point 1 to point 2 during the exhaust stroke, but the clearance volume V_2 is not cleared. During the suction stroke the burned gases expand from point 2 to point 3, and fresh air is drawn in from point 3 to point 4. At point 4, when compression begins, the pressure in the cylinder is less than atmospheric. The effect of a volumetric efficiency of less than unity is to reduce

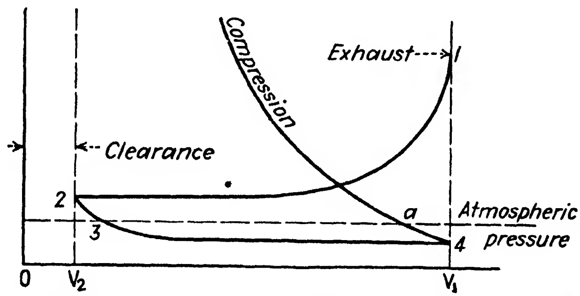


FIG. 102.—Negative loop of indicator diagram of four-cycle engine with pressures greatly magnified.

the weight of air in the cylinder at the beginning of compression and hence the amount of oxygen in the combustion space when the fuel is injected. Less fuel can consequently be injected, resulting in a reduced mep and a lower ihp.

The two-cycle engine does not have the two idle strokes of the four-cycle engine; hence the card has no negative loop. As the piston of a two-cycle engine starts upward on its compression stroke, scavenging is taking place until the scavenging ports are closed, and, if the engine has exhaust ports in the cylinder wall (Fig. 88), air is forced out of the exhaust ports until the piston covers them. Thus compression does not begin until the exhaust ports are covered. If we assume that the crank angle at which the exhaust ports are covered by the piston is 50 deg after bottom dead center, the effective compression stroke is approximately 84 per cent of the stroke, allowing a ratio of crank to connecting rod of 1:4.5. When scavenging is complete, the burned gases are completely cleared from the clearance volume of a two-cycle engine. Because of facts pointed out above and because of the fact that cylinders of two-cycle engines are seldom scavenged 100 per cent, the volumetric efficiency of a two-cycle engine is less than unity.

In order to increase the effective stroke, some two-cycle engines use exhaust valves that open 15 or more deg before the scavenging ports uncover and close at practically the same time as the scavenging ports; thus the effective compression stroke is reduced only by the height of the scavenging ports.

Some engines using scavenging ports and exhaust ports have had a mechanically operated valve fitted in the exhaust pipe. Such an installation is illustrated in Fig. 95, which shows a cylinder diagram of the M.A.N. engine as fitted on the battleship *Deutschland.* On the downward stroke, the exhaust valve is open, as shown in Fig. 95. On the upward stroke, the mechanically operated valve in the exhaust closes as soon as the scavenging ports cover, thus increasing the volumetric efficiency.

111. Factors Influencing Brake Mep.—Brake mep, as already explained, is

$$P_b = e_m \times P_i$$

Thus any factors influencing the mechanical efficiency will influence the brake mep. For a constant value of e_m, P_b will vary directly with P_i.

The value of P_i depends directly on the area of the indicator card, which, in turn, depends upon the amount of fuel burned per stroke. The amount of fuel that can be burned depends (1) upon the amount of oxygen that is present in the combustion space when the fuel is injected, the rapidity with which the fuel and air can be brought together and (2) upon the maximum temperature that the metal can stand.

Most marine Diesel engines in use today operate with an excess air ratio of around 100 per cent—*i. e.*, the air in the combustion space when the fuel is injected is about twice the theoretical amount required for combustion (Art. 41). This large excess air ratio used is due to two causes: (1) poor turbulence, which does not allow the injected fuel and necessary air for combustion to come together quickly, and (2) the necessity to keep down the maximum temperature developed in the cylinder.

Other factors influencing the mep are the volumetric efficiency and the degree of scavenging. Obviously, any factor that cuts down the quantity of air in the cylinder at the beginning of compression will reduce the weight of air in the combustion space and the quantity of fuel that can be burned with smokeless combustion.

The four-cycle engine suffers in comparison with two-cycle engines that are well scavenged because of the lower volumetric efficiency; the two-cycle engine, on the other hand, loses some card area through the early opening of the exhaust and the idle part of the working stroke during scavenging, when no pressure exists on the piston. This is shown clearly in the card in Fig. 100, where the "toe" of the card is cut off, and during the last 7 per cent of the stroke the card is reduced to a straight line. Many two-cycle engines also suffer in comparison with four-cycle engines because of poor scavenging.

The ratio of pure air at atmospheric pressure and 60°F temperature compressed in an engine per minute to that theoretically required per minute for combustion we shall term the "air ratio" and designate it by AR

$$AR = \frac{\dfrac{LANne_v}{144} \text{ cu ft per min}}{\dfrac{f_b \times \text{bhp}}{60} \times \dfrac{14.2}{0.0765} \text{ cu ft per min}}$$

where L = stroke, in feet.
A = area of one piston, in square inches.
N = working strokes per minute.
n = number of cylinders.
e_v = volumetric efficiency.
f_b = fuel consumption per bhp per hour.

Here we have assumed that 14.2 lb of air are theoretically required to burn 1 lb of fuel. This value will vary, of course, with the composition of the fuel (see Art. 41).

If we substitute for bhp its value $\dfrac{e_m P_i LANn}{33{,}000}$, this expression reduces to

$$AR = \frac{74e_v}{f_b e_m P_i}$$

and

$$P_i = \frac{74e_v}{AR \times f_b \times e_m} = \frac{74e_v}{AR \times f_i}$$

If we assume that the heating value of the fuel is 19,000 Btu per lb, the indicated thermal efficiency is

$$e_t = \frac{2{,}545}{19{,}000 \times f_i}$$

and

$$P_i = \frac{550e_ve_t}{AR}$$

We thus see that, neglecting any considerations of temperature limitations, the indicated mep developed by an engine depends directly on the volumetric efficiency, the indicated thermal efficiency, and the excess air where

$$AR = 1 + \frac{\text{per cent excess air}}{100}.$$

Example.—If an engine operates under the following conditions, estimate the value of P_i:

$$\begin{aligned} e_v &= 90 \text{ per cent} \\ f_b &= 0.38 \\ e_m &= 85 \text{ per cent} \\ \text{Excess air} &= 110 \text{ per cent} \\ P_i &= \frac{74 \times 0.90}{2.10 \times 0.38 \times 0.85} = 98 \end{aligned}$$

It is evident that if the excess air ratio could be reduced by better fuel atomization and turbulence in the combustion space the mep could be improved. It should be borne in mind, however, that any increase in P_i must be accompanied by more fuel burned by stroke and an increase in the temperature of combustion. When future research in metallurgy produce a metal that will stand higher temperatures than those used at present and when combustion research allows a reduction of the amount of excess air used, we can expect higher mep's.

112. Supercharging.—In order to increase the indicated mep of four-cycle engines and allow them to compete more successfully with two-cycle engines, supercharging has been introduced. Supercharging consists of blowing air under pressure into the cylinder of an engine during the suction stroke and thus having a pressure at the beginning of a compression greater than that of the atmosphere instead of below that of the atmosphere, which is the usual practice with four-cycle engines (Fig. 102). So far, supercharging has been used only commercially with four-cycle engines[1] and, as far as is known to the author, all these engines have been single-acting.

By increasing the pressure in the cylinder at the beginning of the compression stroke, a greater weight of air is present for combustion—*i.e.*, the volumetric efficiency is increased above unity, and more fuel can be burned per stroke and a higher mep obtained. The usual practice is to use a supercharging air pressure varying between 2 and 5 lb per sq in., which increases the bhp between 30 and 40 per cent. At times, increases of 70 per cent in the bhp have been attained. The value of supercharging is that it allows an increase in mep without any increase in the maximum temperature during combustion. If the weight of air present in the combustion space is increased 30 per cent, it is possible to inject and burn about 30 per cent more fuel per stroke without any greater rise in temperature during combustion than was the case before supercharging.[2]

When supercharging is employed with four-cycle engines, the timing of the inlet and exhaust valves is changed so that they are both open a short time at the beginning of the suction stroke. Thus, when the supercharging air is blown into the cylinder through the inlet valve, it completely scavenges the clearance space. The exhaust valve then closes, and the pressure in the cylinder is greater than atmospheric during the suction stroke. By scavenging the clearance volume and bringing the cylinder pressure up to only atmospheric, the volumetric efficiency of a four-cycle engine is increased from around 87 to around 108 per cent.

Since supercharging raises the ihp of an engine with little increase in the friction horsepower, the mechanical efficiency is

[1] The method of scavenging the Sulzer two-cycle engine described in Art. 101 allows a small amount of supercharging. Two ships, the *M.S. Felix Roussel* and the *M.S. Aramis*, fitted with Sulzer engines, had their power increased about 30 per cent when they were, respectively, four and six years old, by using the upper scavenging ports for supercharging, with an air pressure of about 8 lb per sq in. See *Mar. Eng. & Shipping Rev.*, September, 1936, p. 508.

[2] See *British Motor Ship*, May, 1934, p. 52.

increased, and hence the gain in bhp is greater than the increase in P_i.

When supercharging is employed without any change in clearance volume, the pressure at the end of compression will increase, as will the maximum pressure at the end of combustion. If the clearance volume is increased, it is possible to have the pressure at the end of compression with supercharging the same as without supercharging. This, however, decreases the com-

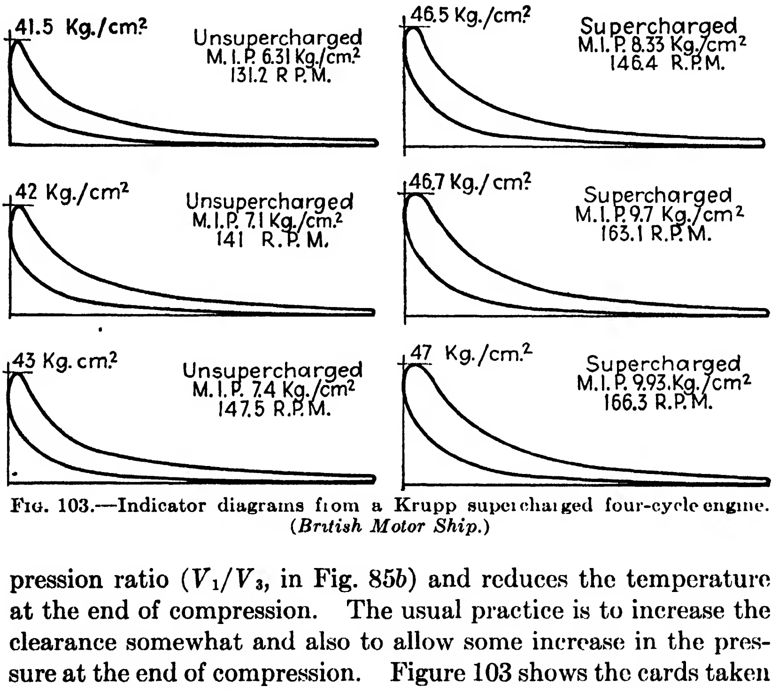


FIG. 103.—Indicator diagrams from a Krupp supercharged four-cycle engine. (*British Motor Ship.*)

pression ratio (V_1/V_3, in Fig. 85*b*) and reduces the temperature at the end of compression. The usual practice is to increase the clearance somewhat and also to allow some increase in the pressure at the end of compression. Figure 103 shows the cards taken from a Krupp engine with various degrees of supercharging.[1] Conversion of kilograms per square centimeter to pounds per square inch can be made by multiplying by 14.22.

Air for supercharging is obtained either by an engine-driven blower similar to those used for scavenging two-cycle engines or by a blower driven by an exhaust-gas turbine. The Büchi system uses a single-wheel turbine driven by the exhaust gases to drive a high-speed turbo blower for supercharging.

[1] From *British Motor Ship*, December, 1937, p. 326.

Examples of ships fitted with single-acting four-cycle supercharged engines are the *M.S. Venus,*[1] the *M.S. Reina del Pacifico,*[2] and the *M.S. Maron.*[3] In 1934, Shaw, Savill, and Albion constructed three ships that were fitted with two 12-cylinder Burmeister and Wain four-cycle single-acting engines of 6,000 bhp each that were supercharged 30 per cent by the Büchi system.[4]

Interesting trials were carried out with the supercharged engine on the *M.S. Maron.*[3] This engine used air injection. During all the trials, the supercharge blower was in operation. The results of trials 1 and 4 are given below:

	Trial 1	Trial 4
P_i	101	154
Ihp	2,145	3,230
e_m	69 2	81.1
Rpm	117 2	115 6
Blower pressure, lb per sq in. gage	1 76	4 92
f_i	0 29	0 33
f_b	0 412	0.407
Exhaust temperature, °F	608	757
Btu transfer per min to liners and cylinder heads	32,600 (16.6%)	56,200 (16.4%)
Compression pressure (air injection), lb per sq in.	496	595
Exhaust-turbine rpm	2,530	4,000

113. Reversing of Oil Engines.—All marine engines used for ship propulsion, except when used with Diesel-electric drive, must be able to run astern. Diesel engines are started ahead and astern by means of compressed air under a pressure of around 350 lb per sq in. The starting air is admitted to the cylinders through starting-air valves. In a four-cycle engine, the working stroke is the only one during which starting air can be admitted. Since the exhaust valves open around 35 to 40 deg before bottom dead center, only about 135 deg of crank angle are available in two revolutions, or 720 deg during which starting air will be

[1] *British Motor Ship*, vol. 12, pp. 86–93; vol. 15, pp. 116–117.
[2] *British Motor Ship*, vol. 12, pp. 8–18.
[3] *Marine Engineer, and Motorship Builder*, vol. 53, p. 101; pp. 143–144.
[4] *British Motor Ship*, vol. 15, pp. 42–45; p. 191.

effective. Hence, if the cranks of a four-cycle engine are arranged symmetrically, the number of cylinders required in order that a single-acting four-cycle engine may be able to start from any position in which it may happen to stop is $^{720}\!/_{135}$, or six cylinders. Two-cycle engines and four-cycle double-acting engines can start from any position with half this number, or three cylinders.

The usual method of designing a Diesel engine so that it can run astern is to provide two sets of cams for each valve, one group set for ahead operation and the second group for astern operation. When it is desired to reverse the engine, the gear is shifted so that the cam followers are moved from the ahead to astern cams. This is usually done by shifting the camshaft longitudinally the thickness of a cam. Thus a four-cycle double-acting engine using air injection might have a maximum of 16 cams per cylinder —8 for ahead operation and 8 for astern (4 inlet cams, 4 exhaust cams, 4 fuel cams, and 4 air-starting cams). A two-cycle double-acting airless injection engine using port scavenging and port exhaust would need only half this number of cams as a maximum. Each make of engine has its own special valve gear and method of reversing. A detailed description of valve gears is beyond the scope of this book.

114. Geared Diesel Engines.—In order to reduce the cost, space requirements, and weight of Diesel engine installations and in order to obtain large powers on a propeller shaft, the geared Diesel engine has been introduced. In this type of machinery two or four engines are connected through single reduction gearing to a single shaft. The rpm of the engines are considerably above the rpm used with direct-connected installations. The rpm of the propeller can be made as low as desired. This allows the speed of the engine to be increased, reducing its size and weight, and yet maintaining the proper propeller rpm for high propeller efficiency.

In order to keep the unequal turning moment of a Diesel engine from being transmitted to the gears, Vulcan hydraulic clutches or electromagnetic clutches are now usually interposed between each engine and the reduction gearing. There is no mechanical connection between engine and gearing with either type of clutch. The hydraulic clutch has a slip of about 3 per cent, and the electromagnetic clutch a slip of around 2 per cent at full load, and hence unequal turning moments and torsional vibrations of the engine cannot be transmitted to the gearing. The over-all efficiency of the hydraulic clutch is around 97 per cent, and that of the gearing around 98 per cent. The efficiency of the electromagnetic clutch is around 97 per cent.

The electromagnetic clutch consists of two concentric electromagnets. The primary multipole magnet is usually mounted on the pinion shaft, and the secondary magnet is usually attached to the engine shaft and surrounds the primary magnet. The two magnets are separated by an air gap varying between 0.2 and 0.4 in. The primary magnets are excited from an outside source through slip rings on the shaft; the secondary windings are of the squirrel-cage type and are short-circuited.[1]

The geared Diesel drive is lighter and probably somewhat cheaper than direct-connected Diesels. The engine rpm of installations on merchant ships are usually in the neighborhood of 200 to 300; hence the reduction in speed between engine and propeller is much smaller than with geared-turbine installations. Higher engine rpm will no doubt be used in the future. The increase in engine rpm makes the engines cheaper than the slower turning engines of direct-connected installations, but from this saving in cost must be subtracted the additional cost of the gearing and special clutches. It does not appear that geared Diesel drive is any shorter than a direct-drive installation, but the height of the machinery is reduced considerably, which, of course, is an important advantage in passenger ships and naval vessels. A table showing the characteristics of some of the important geared Diesel engine installations is given on page 266.

In a geared Diesel engine installation, the engines geared to the same shaft can be cut out as desired by means of either type of clutch. To reverse the ship, the engines are usually reversed. However, the U. S. Maritime Commission's C-3 type geared Diesel ships have electromagnetic clutches and frequently reverse

[1] The following papers and articles give full information on these two types of clutches: "Hydraulic Couplings for Internal Combustion Engines," by Alison, Olson, and Nelden, *Proc. Nat. Conf. Oil and Gas Power Div. A.S.M.E.*, 1940. "Electric Slip Couplings for use with Diesel Engines," by A. D. Andriola, *Proc. Nat. Conf. Oil and Gas Power Div. A.S.M.E.*, 1940. "Vulcan Gearing and Hydraulic Coupling," *Mar. Engineer & Motorship Builder*, vol. 53, p. 175. "Electro-magnetic Slip Couplings for Use with Geared Diesel Engines for Ship Propulsion," by G. L. E. Metz, *Trans. Inst. Mar. Eng.*, vol. 49, p. 237.

TABLE XV.—TYPICAL GEARED DIESEL ENGINE INSTALLATIONS

Ship	Type	Gross Tonnage	No. of screws	No. of engines per shaft	Bhp per engine	Engine rpm	Propeller rpm	Make of engine†	Type of clutch
M.S. Astri	Cargo	2,556	1	2	850	300	130	Atlas	Electromagnetic
M.S. St. Louis	Passenger	17,000	2	2	3,100	225	110	M.A.N., D.A.	Hydraulic
U.S. Maritime Commission C-3.	Cargo	7,900	1	4	2,225	240	85	Busch-Sulzer, S.A.	Electromagnetic
U.S. Maritime Commission C-2.	Cargo	6,200	1	2	3,155	225	92	Nordberg 2 cycle, S.A.	Hydraulic
M.S. Oslofjord	Passenger	18,700	2	2	3,950	212	90	M.A.N., D.A.	Hydraulic
Rotterdam Lloyd ship	Passenger	21,000	2	4	3,500	215	120	Sulzer, S.A.	Electromagnetic
Deutschland	Battleship		2	4	7,100	450	250	M.A.N., D.A.	Hydraulic
*M.S. Rio Hudson**	Cargo and passenger	12,500	1	2	4,500	180		Sun Doxford, opposed	Electromagnetic

* U.S. Maritime Commission design C-3 passenger ship

† D.A. = double acting. S.A. = single acting.

the ship in the following manner. During maneuvering, two of the four engines are set to run ahead and two astern, and only two engines are used at one time. To reverse when the ship is going ahead at half power (79 per cent speed) under two engines, all that is necessary is to de-energize the clutches of the two driving engines and energize the two clutches of the engines that are turning in the reverse direction. This is quickly carried out at the engine-control station. If full power is required during maneuvering, it will be necessary to stop two engines and start them in the opposite direction.

115. Diesel-electric Drive (Direct Current).—Another means of reducing the rpm from the engine to the propeller is the Diesel-electric drive. Like the geared Diesel drive, however, the reduction in rpm between engines and propeller shaft is not large, being of the order of 1:2 to 1:4 instead of the 1:30 to 1:40 used with turbo-electric drive and reductions up to 1:80 used with the geared turbine. The main advantage of the Diesel-electric drive is the excellent means of speed control it provides. This drive also allows full torque to be developed on the propeller shaft at reduced rpm.

From our formula for bhp

$$\text{Bhp} = \frac{P_b L A N}{33{,}000}$$

it is evident that with a direct-connected engine a reduction in revolutions due to bad weather or to a tug's picking up a tow, the bhp of the engine will be reduced as the rpm fall off. Unless the engine is run under normal operation conditions with an mep below maximum, so that the mep can be increased when the rpm fall off, there will be a reduction in bhp at a time when full power is usually desired. As we saw in Art. 105, the operation of a Diesel engine at full power with a large reserve in mep necessitates an increase in the expression (LAn) to obtain the designed bhp. This, of course, increases the cost and weight of the engine. As will be pointed out later, a falling off in rpm of a ship propelled by Diesel-electric drive does not necessarily mean a reduction in the power transmitted to the propeller.

A Diesel-electric drive consists of a number of medium-speed Diesel engines (we shall assume four for discussion), running at constant rpm in one direction and directly connected to d-c generators. The generators are connected in series with a low-speed motor on the propeller shaft. When a number of generators are connected together in series, usually a double-armature motor is used to reduce the voltage across each motor armature. The Ward-Leonard system is used, the motor rpm being controlled by the excitation of the generator field. The field on the main motor is held constant during maneuvering and ordinary operation. The engines driving the generators run at constant speed from full load to zero load, the fuel supply being regulated by the governor. The exciters are usually located on an extension of the generator shafts and supply current for the auxiliary motors as well as for excitation of generators and main motor. The voltage output of the generator depends on the strength of its field current and the revolutions of the motor depend on the voltage impressed upon its armature. The power output of the motor is that necessary to maintain the rpm caused by the voltage impressed on its armature. Thus the speed of the propeller and its direction of rotation are under direct control by use of a generator field rheostat. This rheostat control can be located on the bridge or in the wheel house, and by a simple control handle (and an indicator similar to an engine telegraph) the navigating officer has direct control of the speed and direction of the main motor.

A second system of Diesel-electric drive is now in use in which the engine rpm are varied as well as the generator field. In this system the engines are run at a constant reduced rpm for low ship speeds. For high speed operation the engine rpm are increased and the speed of the ship can be varied by changing the engine rpm. This method of control results in a reduced fuel consumption and reduced wear and tear on the engines when operating at low ship speeds.

In both systems the motor field is fitted with a rheostat for controlling the rpm under special conditions as noted below.

A tug powered with a Diesel-electric drive can develop the full power of the main engines on the propeller shaft at reduced revolutions by adjusting the motor field until the propeller rpm is such that the main motor is taking the full power of the engines. In the meantime, the engines are running at full rpm, and since the power developed is the same as before, when the tug was running free at full-rated rpm, the mep of the engines has not increased.

TABLE XVI.—TYPICAL DIESEL-ELECTRIC-DRIVE INSTALLATIONS

Ship	Type	Current	Total motor shp	No. of screws	No. of engines	Engine rpm	Motor rpm	Bhp per engine	Reference§
M.S. Courageous	Cargo	Direct	4,000	1	4	250	60	1,200	*U.S.S.B.*, 1928 conversion
M.S. Brunswick	Tanker	Direct	2,800	1	4	225	95	750	*B.M.S.*, October, 1928
M.S. Port Houston	Fireboat	Direct	720	2	2	425	265	550	
M.S. New York Central No. 34	Tug	Direct	650	1	2	265	145	400	
M.S. Navajo†	U.S. Navy tug	Direct	3,060	1	4	750	.	950	*M.E.S.R.*, November, 1940
M.S. E. J. Moran‡	Tug	Direct	1,350	1	2	750	200	950	
M.S. Lock Fyne	Small passenger	Direct	....	2	2	330	430	1,000 ihp	*B.M.S.*, June, 1931
M.S. Wuppertal	Cargo	Alternating	..	1	3	250	125	2,600	*B.M.S.*, vol. 18, pp. 42–50
*M.S. Patria**	Passenger	Alternating	15,000	2	6	250	110	5 × 2,140 kw 1 × 1,600 kw	*B.M.S.*, vol. 19, p. 371

* Two synchronous motors, 3,500 volts.

† A geared Diesel-electric drive. Propeller rpm not published. Four motors of 765 bhp each, turning at 1,350 rpm, geared to propeller shaft. General Motors engines.

‡ A geared Diesel-electric drive. Two motors of 675 bhp each, turning at 875 rpm, geared to propeller shaft.

§ *B.M.S.* is the *British Motor Ship*, London. *M.E.S.R.* is the *Mar. Eng. & Shipping Rev.*, New York. *U.S.S.B.* is United States Shipping Board.

In order to improve the fuel economy in operating at long periods at reduced power, part of the generators can be shut down. Thus, suppose that the ship had an installation of four generators. If it was desired to operate at half power (79 per cent speed) two generators could be cut out and the ship operated under two generators, each of which would be running at full power and full rpm and hence at minimum fuel consumption per bhp per hour. In shutting down two generators, however, the voltage on the main motor would be cut in half. This would cut the motor rpm down to one-half the full-speed rpm. According to our general laws of ship propulsion, power varies approximately as V^3, and V varies as rpm. Half rpm would, therefore, result in half speed and one-eighth power instead of half power and 79 per cent speed (79 per cent rpm), as desired. Hence, when part of the generators are cut out, the motor field must be weakened sufficiently to bring the motor up to the desired rpm so that it will absorb the full power of the generators that are in operation.

The electrical loss in a Diesel-electric drive is of the order of 15 per cent; hence the fuel consumption per propeller shp per hour for a Diesel-electric drive is $0.38/0.85 = 0.45$ lb per shp per hr instead of 0.38 obtained with direct-connected Diesels. The geared Diesel drive has a loss of about 2 per cent in the gears and about 3 per cent in the clutch, or a total over-all efficiency of about 95 per cent.

The Diesel-electric drive makes an ideal installation for craft that do a great deal of maneuvering because of the easy manner in which the speed of the ship can be controlled. No reversing gear is necessary, and a considerable saving is made because large quantities of starting air do not have to be provided. It also has an advantage in ships where full propelling power can be used for other purposes, as on a hydraulic dredge, fireboat, salvage tug, and tanker.

For cargo ships that do little maneuvering, the Diesel-electric drive offers no special advantages and, as a rule, costs and weighs as much as the direct-drive installation. On passenger ships it may offer advantages if the ship is to operate for long periods at low powers.

Since a Diesel-electric drive is usually composed of a number of small units, repairs can readily be made at sea, and the installation is more reliable than a single-screw ship, since the ship can operate at reduced speed if one or more of the engines break down.

The thermal efficiency of a small Diesel engine is practically the same as that of an engine of large power, whereas the efficiency of small steam turbines is much lower than that of large turbines. Hence a Diesel-electric installation can be subdivided into a group of small units far better than a turbo-electric drive.

116. Diesel-electric Drive (Alternating Current).—During 1937 and 1938, a number of German merchant ships were fitted with Diesel-electric installations in which a-c current was used. The advantage of a-c current over d-c current is that the transmission efficiency is around 90 per cent against 85 per cent for d-c current and the cost of the electrical machinery is about 20 per cent cheaper. However, when a-c current is used, all the advantage of bridge control is lost. The a-c Diesel-electric drive is thus similar in nature to the geared Diesel but uses electric reduction instead of mechanical. Its adoption by the Germans, who originated the geared Diesel drive, may be due to the fact that it is lower in first cost than a geared installation. The engines are slowed down to reduce the speed of the ship, and the motors are reversed to make the ship go astern, the engines always turning in one direction. Engines can be cut out at reduced speeds, as with the geared drive and the d-c Diesel-electric drive. The particulars of a few Diesel-electric drives are given in Table XVI.

117. Utilization of Waste Heat of Diesel Engines.—The heat balance for a Diesel engine, given in Art. 5, shows that about 30 per cent of the heat supplied in the fuel leaves in the exhaust. The temperature of the gases as they leave the engine is in the neighborhood of 650 to 800°F. Unlike the steam plant, where the large amount of heat discharged into the condenser is at too low a temperature to be used, the heat in the exhaust of a Diesel engine is at a high temperature, more in line with the dry stack gases leaving a boiler than the waste heat of a steam turbine.

Various means have been employed to utilize this waste heat as follows:

1. Generation of steam by means of an exhaust-gas boiler for (*a*) heating the ship; (*b*) operation of auxiliary machinery.
2. Exhaust-gas turbines for driving supercharging blower.
3. The Still engine, a combination Diesel- and steam-engine cycle, which uses both the heat in the cooling water and the exhaust gases.

4. Hot-water heating, using heat of exhaust and cooling water.
5. Combined steam-turbine and Diesel installation, in which the Diesel waste heat is used for heating the boiler feed water.

The last two of the preceding methods of using the waste heat have been employed on shore but never on shipboard.

Exhaust-gas boilers, when fitted with economizers for heating the feed water, are able to produce between 0.70 and 0.90 lb of steam (at 60 to 100 lb per sq in.) per bhp per hr. This is sufficient steam to heat a passenger ship or to drive the auxiliary machinery of a cargo motorship.

A number of Diesel engine cargo ships and tankers[1] have been built in which the auxiliary machinery, including the steering gear, is driven by steam supplied by a waste-heat boiler. Most cargo motor ships today are fitted with waste-heat boilers that generate steam for heating the ship, heating water, and heating the fuel oil when heavy fuel oil is used. These boilers are usually fitted with a separate furnace that allows the boiler to be fired with oil in port.

An interesting example of a ship fitted with an exhaust-gas boiler in which the steam produced is used to drive the auxiliaries is the *M.S. Sutherland*,[2] a British tramp. This ship is powered with a three-cylinder Doxford engine of 1,800 bhp. No Diesel auxiliary generating sets are fitted. The exhaust boiler receives the gases from engine at 650°F, and the gases leave the boiler at 380°F. The circulating water pump and lubricating oil pump are driven off the main-line shafting. The steam produced by the boiler at 120-lb. gage drives a 12.5-kw steam-turbine-driven generator, the steering gear, the ballast pump, general service pump, bilge pump, and air compressor and heats the ship. The fuel consumption in service is 0.35 lb per bhp per hr, and the daily fuel consumption is 6.2 tons of oil per day. The bunker capacity is 780 tons, giving the ship a radius of 28,000 miles. The outstanding merit of this installation is not its low unit fuel consumption but its low first cost, since the cost of the Diesel auxiliary generating sets and auxiliary electric motors is eliminated. The only disadvantage is that steam winches with their high fuel consumption must be used in port instead of the Diesel-electric winch drive usually employed on motor ships.

[1] See the paper by Robert L. Hague, "Performance of Standard Oil Tankers," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1932.

[2] *British Motor Ship*, vol. 15, p. 335; vol. 16, p. 362.

TABLE XVII.—HULL AND MACHINERY CHARACTERISTICS FOR TYPICAL DIESEL-DRIVEN CARGO SHIPS AND TANKERS (DIRECT DRIVE)

	U.S. M.C. C-2	*M.S. Waiwera*	*M.S. Sakito Maru*	*M.S. British Columbia Express*	*M.S. Sutherland*	*M.S. Esso Augusta*
Type	Cargo	Refrig.; cargo	Cargo	Refrig.; fruit	Tramp	Tanker
Date	1939	1934	1939	1937	1935	1940
L.O.A, ft.	459	535.5	506	380.5	423	547
Dead weight	8,600	....	9,500	4,000	9,215	17,950
Speed, knots	15.5	16.0	17.0	16.5	10.5	15.5
Service bhp	6,000	12,000	9,600	6,100	1,800	7,500
No. of screws	1	2	2	2	1	1
Type of engine	Opposed	4 cycle, single acting, super-charged	2 cycle, single acting	2 cycle, single acting	2-cycle, opposed	2-cycle, opposed
Make of engine	Sun Doxford	B & W	Mitsubishi	Gotaverken, B & W	Doxford	Sun Doxford
Rpm	92	112	128	160	115	92
No. of cylinders per engine	4	10	7	9	3	5
Bore and stroke, mm	32 in. × 95 in.	740 × 1,500	720 × 1,250	500 × 900	520 × 2,080	32 in. × 95 in.
Nationality	U.S.	British	Japanese	Norwegian	British	U.S.
Reference*	*M.E.S.R.*, vol. 44, p. 370	*B.M.S.*, vol. 15, pp. 42, 191	*B.M.S.*, vol. 20, pp. 66, 436	*B.M.S.*, vol. 17, p. 380	*B.M.S.*, vol. 16, p. 362	

* *M.E.S.R.* is the *Mar. Eng. & Shipping Rev.* *B.M.S.* is the *British Motor Ship.*

TABLE XVIII.—HULL CHARACTERISTICS AND MACHINERY DATA FOR TYPICAL MOTOR PASSENGER SHIPS

	M.S. Oranje	*M.S. Dominion Monarch*	*M.S. Capetown Castle*	*M.S. Prins Albert*	*M.S. Venus*	*M.S. Vega*	*M.S. Patria*		*M.S. Oslofjord*
Date	1939	1939	1938	1937	1931	1938	1938	1939	1938
Nationality	Dutch	British	British	Belgian	Norwegian	Norwegian	German	Dutch	Norwegian
L.B.P., ft.	605	682†	685	360	395	415	560	610	545
Gross tonnage	20,000	26,500	27,000	3,300	5,600	7,500	16,600	21,000	18,700
Service speed	21.0	19.5	20.0	23.5	19.5	20.5	17.0	21.0	19.0
Total service bhp.	37,500	32,000	24,000	15,000	10,250	10.600	15,000	27,000	15,800
No. of screws	3	4	2	2	2	2	2	2	2
Bhp per engine.	12,500	8.000	12,000	7.500	5,125	5,300	See below	3,500	3,950
Type of machinery	Direct connected	Direct connected	Direct connected	Direct connected	Direct connected supercharged	Direct connected	A-c Diesel electric	Geared	Geared
Rpm.	145	133	102	257	160	133	250–110	215–120	212–92
Cycle	2 cycle, single acting	2 cycle, opposed	2 cycle, double acting	2 cycle single acting	4 cycle, single acting	2 cycle single-acting	2 cycle single-acting	2 cycle single-acting	2 cycle double acting
Make of engine.	Sulzer	Doxford	B & W	Sulzer	B & W	C R.A. Sulzer	M.A.N.	Sulzer	M.A.N.
No. of cylinders per engine	12	5	10	12	10	10	8	8	7
Bore and stroke, mm.	760 × 1,250	725 × 2,250	660 × 1.500	580 × 840	630 × 1,150	650 × 1,200	520 × 700	580 × 840	530 × 760
P_i (service)	81*	..	96		135	..	....	..	63.5
P_b (service)	....	84.1	79.8	70.8		65‡	...	59.3‡	51.3‡
Scavenging blower drive.	Motor	Recip. attached	Rotary attached	Recip. attached	Supercharged, chain driven	Recip. attached	Rotary attached	Recip. attached electromag. clutch	Recip. attached
Remarks	....	....	....	English Channel boat	Bergen-Newcastle service	Bergen-Newcastle service	6 generators 5 × 2,140 kva 1 × 1,600 kva 3,500 volts	4 engines per shaft	Vulcan clutches, 2 engines per shaft
Reference§	*B.M.S.*, vol. 19, p. 238; *B M.S.*, vol. 20, p. 158	*B.M.S.*, vol. 19, p. 87, 198	*B.M.S.*, vol. 19, p. 43	*B.M.S.*, vol. 18, p. 230	*B.M.S.*, vol. 12, p. 86	*B.M.S.*, vol. 19, p. 82	*B.M S.*, vol. 19, p. 311	*B.M.S.*, vol. 20, p. 286	*B.M.S.*, vol. 19, p. 98

* Maximum pressure = 680 lb per sq in.
† L.O.A.
‡ Calculated from published data.
§ *B.M.S.* is the *British Motor Ship*.

CHAPTER IX

COMPARISON OF TYPES OF PROPELLING MACHINERY

118. General.—A large number of types of propelling machinery are available today, and the selection of the proper machinery for a given ship is not always a simple matter.

The various types of machinery in use today on identical trade routes show that wide differences of opinion still exist regarding propelling machinery and that the various shipowners are giving different relative weights to the factors influencing choice of machinery. In some cases, it is apparent that not enough study is given to the selection of machinery for a new design.

The installation that gives the most economical performance in service is not necessarily the one that shows the lowest fuel consumption, for hand in hand with a study of fuel and steam consumption must go a study of fixed charges, cost of maintenance, weight, space occupied, revolutions and propeller efficiency, engineering risk involved, and the general feasibility and reliability of the installation as a whole. As an example, a case is on record where a quadruple expansion (land installation) showing excellent fuel consumption was replaced by another engine because of the high over-all maintenance cost. Here is a case where the low fuel consumption was outweighed in actual practice by other considerations.

Another point that should not be overlooked is the interrelation between efficiency of the power plant and the operating efficiency of the ship as a whole. Obviously, if the repairs or overhauling of a particular unit of a ship's power plant prolonged the ship's stay in port over that necessary for discharging and loading the cargo, it would be poor judgment to install such a unit, no matter what gain in fuel economy resulted. For a slow- or medium-speed cargo ship, the fuel cost is not a large proportion of the total operating expenses, and hence a reduction in fuel consumption should not be accomplished at the expense of some more important factor in the operating disbursements or revenue.

The power plant and fuel consumption have other relations to the ship in that they affect the cubic feet of cargo space, the tonnage measurement, and the cargo dead weight. We should not lose sight of the fact that a reduction in steam consumption is usually accompanied by a reduction in the size (and cost) of the boilers, piping, auxiliaries, and bunkers, which, in turn, increases the carrying capacity of the ship. The problem of machinery selection is a complex one, and many times detailed calculations for annual operating expenses and profits for ships with two or more types of machinery operating on the trade route in question must be carried through before an intelligent selection of the machinery for a proposed ship can be made.

The tendency in the past has often been too conservative in regard to innovations and improvements in a ship's power plant. There has been some justification for this, for if troubles arise with new types of machinery at sea the engineer cannot call on the manufacturer for help and assistance, as with a land installation, but must make his own repairs and solve his own problems. Further, when a ship is at sea the superintending engineer does not have access to the plant, as with a land installation, and the chief engineer is thrown absolutely on his own resources. A breakdown of a ship's machinery at sea is always a more serious affair than in a power plant on shore; at times it may seriously endanger the ship.

119. Types of Machinery.—In the selection of machinery for a merchant ship, we have the following types from which to choose:

1. Triple- and quadruple-expansion reciprocating engines.
2. Lentz compound and unaflow engines.
3. Reciprocating engines with exhaust turbines.
4. Geared turbines.
   a. Single reduction.
   b. Double reduction.
5. Turbo-electric drive.
6. Direct-connected Diesel engines.
   a. Two- and four-cycle.
   b. Single- and double-acting.
   c. Supercharged engines.
7. Geared Diesel engines.
8. Diesel-electric drive (d-c or a-c current).
9. Geared Diesel-electric drive.

With the first five types, we can use either Scotch or water-tube boilers and coal or oil. With types 6 and 7, we can use either regular Diesel fuel oils or heavy and cheaper fuel oils that require heating.

TABLE XIX

	S.S. Red Jacket	*S.S. Exporter*	*S.S. President Jackson*
Type	C-2	Cargo	C-3 passenger
Gross tonnage	6,085	6,736	9,300
Speed	15.5	16.0	16.0
Shp	6,000	8,000	8,500
No. of screws	1	1	1
Pressure superheater outlet, gage	465	450	450
Temperature of superheater outlet, °F	750	750	750
Pressure turbine throttle, gage	440	425	440
Temperature of turbine throttle, °F	740	740	740
Vacuum, in. of mercury	28.5	28.5	28.5
Boilers, no. and type	2 Foster Wheeler	2 B & W	2 B & W
Turbine rpm	H.P. 6,100 L.P. 4,050	H.P. 4,450 L.P. 2,310	H.P. 4,500 L.P. 2,290
Propeller rpm	91.4	96	85
Turbine builder	General Electric	Bethlehem	Newport News
Boiler generating surface	5,948	11,250	
Air heating surface	4,960	6,634	
Economizer heating surface	3,360	None	None
Condenser cooling surface	5,500	9,533	7,800
Type of condenser	2 pass	1 pass	2 pass
Boiler feed pump	Motor-driven plunger	Steam centrifugal	Steam centrifugal
Feed heaters	2 closed + 1 D.C.	2 closed	2 closed + 1 D.C.
Auxiliary generators	2–250 kw	3–150 kw	3–300 kw

If steam machinery is selected, we must fix on the boiler pressure and superheat to be used, the type of auxiliary machinery, whether steam- or motor-driven. If the auxiliaries are to be motor-driven, several methods are available for driving the auxiliary generating set (Art. 171).

Figure 104 gives data for the fuel consumption of various types of propelling machinery at full power and at reduced powers. The explanation for the dotted lines in the performance curve of the *U.S.S. Saratoga* is given in Art. 94.

TABLE XX

	S.S. J. W. Van Dyke	*S.S. Examiner*
Type of machinery.....	Turbo-electric	Geared-turbine, reheat cycle
Shp............. ..	5,000	8,000
Pressure, superheater outlet, lb abs....	640	1,200
Temperature, superheater outlet, °F.. ...	835	740
Vacuum, in. of mercury	28 5	28 5
Feed heaters......	2 closed + 1 D.C.	3 closed + 1 D.C.
Feed pump....... . . .	Steam centrifugal	Aldrich-Groff variable-stroke-motor-driven
Auxiliaries............ ...	A-c current from main generator	Motor-driven d-c
Fuel consumption, lb per shp-hr†...............	0.610	0.513*

* Designed value.

† Fuel for all purposes. Corrected to 18,500 Btu.

The American steam cargo ships that were built during the period 1939–1941 have been fitted with double-reduction geared turbines and water-tube boilers, with most of the auxiliaries driven by d-c motors. In some cases, the boiler feed pump is motor-driven, and in other cases a steam-turbine-driven centrifugal pump is employed. The feed water is heated by two or three heaters in series, which are supplied by steam extracted from the main turbine. High-speed geared-turbine generating sets working under full boiler pressure and temperature, and at a 28½-in. vacuum, are used to supply the electric power.

Data for two modern American cargo ships and one combined passenger and cargo ship are given in Table XIX.

The fuel consumption of the *Red Jacket* on official trials was 0.545 lb of oil per shp-hr (18,500 Btu). A heat balance and diagram of the feed system for the *S.S. Challenge*, a sister ship of the *Red Jacket*, is given in Fig. 130.

Higher pressures and temperatures have been used on two other American merchant ships, the tanker *S.S. J. W. Van Dyke*[1] and the cargo ship *S.S. Examiner*.[2] The reheat cycle was used on the latter ship.

120. Factors Influencing Selection of Machinery.—The important factors to be considered in the selection of the machinery are outlined below, but not necessarily in their order of importance.

1. Reliability and engineering risk.
2. Type of ship; passenger, cargo liner, tramp, collier, tanker, trawler, etc.
3. Trade route.
   a. Length of nonstop run (influences bunker capacity).
   b. Bunkering ports; kinds and cost of fuels available.
   c. Types of crew available.
   d. Repair facilities available.
   e. Percentage of voyage spent in port.
   f. Cargo available—measurement or dead weight.
   g. Average sea temperature, as influencing vacuum.
4. Amount of ship to be installed (size and speed of ship).
5. First cost of machinery.
6. Repair and maintenance expenses and time required for repairs.
7. Fuel costs.
8. Weight of machinery.
9. Weight of fuel for a given length of voyage.
10. Space occupied by machinery (influence on hold capacity and tonnage measurements).
11. Space occupied by fuel.
12. Requirements for economy at reduced speed.
13. Rpm and number of screws (propeller efficiency).
14. Maneuvering ability.
15. Noise and vibrations.
16. Auxiliaries (drive to be used; use in port, etc.).

Of the foregoing factors, reliability is, of course, the first and most important consideration, for breakdowns not only increase operating costs but also influence the ship's time for a round voyage and the reputation of the ship and steamship line.

The type of ship often influences the selection of the machinery, since naturally modern types of the more complicated machinery that would be suitable for passenger or cargo liners would not

[1] Several ships of this class were built. The second ship, the *S.S. E. J. Henry*, had the same boiler pressure but a temperature of 910° at the superheater outlet.

[2] See the paper, "A 1,200-pound Reheat Marine Installation," by B. Fox and R. H. Tingey, *Trans. Soc. Naval Arch. & Mar. Eng.*, 1941.

be suitable for tramps and colliers, for example. Also, the power requirements in port for winch or pump operation would exert an influence on the selection of the auxiliaries and auxiliary generator sets and might have some influence on the selection of the propelling machinery. The percentage of the year spent in port varies with the type of ship, and this factor sometimes has an influence in determining what type of main engine should be used.

Two of the main factors in fixing the type of machinery for a ship, as well as the characteristics of the ship itself, are the trade route and the trade in which the ship is to be engaged. The various items that will have an influence on this factor have been listed above and show clearly how the trade route will exert an influence on machinery and bunker spaces, fuel costs, and hence on the type of machinery and fuel to be used. Machinery suitable for one trade route may not be suitable for another trade route.

The amount of horsepower to be installed, which, of course, will vary with the size and speed of the ship, is one of the outstanding factors to be given consideration in machinery selection. When the shp is small, the fuel per shp-hour often becomes a minor consideration (especially on short runs), and a small horsepower does not warrant such refinements as high boiler pressures, superheat, air heaters, extracted steam for feed heating, etc. Because steam machinery of small powers has a higher fuel consumption per shp than steam machinery of large powers, Diesel machinery must be given serious consideration for such installations.

With a large horsepower installation, the fuel per shp-hour becomes important, and in a naval vessel where a large shp is accompanied by a high speed of the ship, the necessity of keeping the ship's displacement low to reduce resistance makes the saving in machinery weight an important item.

The first cost of the machinery influences the annual charges for depreciation and insurance and also influences the percentage of profit received for a given annual revenue from cargo.

The annual fixed charges and upkeep cost of the various parts of the machinery equipment must be weighed against the gain in economy each contributes to the fuel consumption. For example, it is possible that some refinement such as a superheater carrying a high degree of superheat with a reciprocating-engine installation might add more to the annual fixed charges and maintenance expenses than the saving it brought about in the annual fuel bill.

The variation in fuel costs at the various ports of the world naturally will have an influence not only on the fuel selected, such as coal or oil, but also on the machinery, whether steam or Diesel are used. Another factor that should be given consideration is the probable relative change in the prices of coal, bunker *C*, and Diesel fuel oil that is likely to take place during the life of the ship. The physical life of a ship is usually fixed at 20 years. What is likely to be the relative price of the three fuels 10 or 15 years hence is a question that every shipowner must face when ordering a new ship.

The weight of machinery and fuel carried influences the dead weight carrying capacity of a cargo ship of given displacement and influence the required displacement of high-speed passenger and naval ships. For low-speed cargo ships carrying measurement cargo, these factors are not of serious concern; for high-speed naval vessels they are of paramount importance.

The space occupied by the machinery and the fuel economy at reduced speed are considered at length in later articles.

The auxiliaries seldom receive the attention in machinery selection that they deserve. Here such factors as method of driving generators, percentage of year spent in port, "hotel load" on passenger ships, and type of cargo winches to be used should be given consideration at the time that the type of propelling machinery is selected.

Sea temperatures on the trade route of a given ship will exert an influence on the vacuum that can be carried economically and may make a reciprocating-engine or a Diesel engine installation more suitable than a geared turbine. A round-the-world voyage starting from Boston in May showed sea temperatures of 50°F at Boston, 84° in the Caribbean, 68° on the west coast of the United States, 78° at Honolulu, 86° at Singapore, 92° in the Red Sea, 78° in the Mediterranean, and 70° at Boston upon the completion of the voyage in August.

121. The Reciprocating Engine.—With the exception of the slow-speed four-cycle Diesel engine, the reciprocating engine, as represented by the three- and four-cylinder triple-expansion and the quadruple-expansion types, is the heaviest and bulkiest prime mover used for ship propulsion. The marine reciprocating engine shows a higher steam consumption than the geared turbine, partly because of the lower steam pressure, lower superheat, and lower vacuum used with the reciprocating engine and partly because of the lower engine efficiency that is inherent with the reciprocating engine. It was pointed out in Art. 61 that it was inadvisable to carry more than 27 in. of vacuum with a reciprocating engine, whereas with a turbine, 28½ and at times even 29 in. is possible. Reciprocating engines can, however, carry fairly high steam pressures and superheat, but generally when this type of machinery is used the same conservatism that leads to the adoption of the reciprocating engine results in the use of lower steam pressures and generally little or no superheat. We thus find the turbine working between more extreme limits and hence with a larger cycle efficiency.

For moderate and large powers, the engine efficiencies of the geared turbine are higher than those of ordinary triple-expansion engines. However, as pointed out in Art. 80, the engine efficiency of geared turbines falls off rapidly as the power is reduced below 1,000 or 1,500 shp. On the other hand, some of the modern types of reciprocating steam engines, such as those described in Arts. 63 and 64, show exceedingly good engine efficiencies, and hence the unaflow and Lentz types of reciprocating steam engines should be given consideration in any comparison of propelling machinery for powers around 1,500 to 3,000 shp.

Some of the latest types of reciprocating engines built in Great Britain (Art. 60) that use steam temperatures of around 700°F have shown marked improvements in economy over the ordinary reciprocating-engine installations using low steam pressures and temperatures.

Also, reciprocating engines fitted with one of the types of exhaust turbines have shown good economy and have much to recommend them for certain low-powered cargo ships. Around 1930, some shipowners adopted the reciprocating engine and exhaust turbine because it allowed improved economy without the use of superheated steam.

If steam and fuel consumption and weight and space considerations were the only criteria in selecting machinery, the ordinary triple- and quadruple-expansion engine would receive little consideration today for powers above 1,500 or 2,000 shp. The triple-expansion engine, however, is an installation of proved reliability; repairs can generally be made in any part of the world, the first cost is usually lower than that of a geared-turbine installation, and marine engineers are familiar with it.[1] For these reasons, it continued to be used to a considerable extent in many foreign cargo liners, tramps, and colliers up to 1942. It is doubtful, however, whether ordinary triple-expansion engines (and especially the quadruple-expansion engines) such as were commonly installed in ships 20 years ago will be used to any extent in the future. The reciprocating steam engine, with the improvements discussed in Arts. 60, 63, 64, and 66, probably will continue to be used in many tramps and low-powered cargo ships. In these fields, however, the Diesel engine is a strong competitor.

122. The Geared Turbine and Turbo-electric Drive.—If steam machinery is to be used for ships of moderate or large powers, it appears from the foregoing article that our choice of machinery narrows down to the geared turbine and turbo-electric drive. These two types of machinery do not differ greatly from one another in weight and space occupied. The geared turbine, however, shows a somewhat lower steam rate and is appreciably lower in first cost. The efficiency of the gearing of a geared-turbine drive is about 97 per cent; the transmission efficiency of the turbo-electric drive is about 90 per cent (97 per cent generator efficiency $\times$ 93 per cent motor efficiency). The geared turbine also has a loss of about 1 per cent due to windage in the astern turbine. The use of two turbine casings with a geared-turbine installation allows the H.P. turbine to turn at high rpm, and a somewhat lower steam rate is obtained than is possible when only one turbine casing is used, as with the electric drive. The double-reduction geared turbine is able to secure a greater reduction in rpm between turbine and propeller than is possible with electric drive.

A good comparison between the two types of machinery is offered by the trials of two modern ships, the *S.S. J. W. Van*

[1] Some of the low-speed cargo ships built in the United States under the emergency shipbuilding program, which started in 1940, have been fitted with triple-expansion steam engines. Thus at the present writing the triple-expansion engine continues to be an important type of machinery for the propulsion of ships of the American Merchant Marine.

Dyke, with turbo-electric machinery, and the *S.S. Red Jacket*, with geared turbine machinery as shown below:

	J. W. Van Dyke, 1938	*Red Jacket*, 1939
Shp installed	5,000	6,000
Boiler pressure, lb gage*	630	463
Initial steam temperature, °F	825	766
f, lb per shp-hr†	0 61	0.55

* At superheater outlet.
† Fuel for all purposes; corrected to 18,500 Btu.

When economy at low power is desired, as on naval vessels, the geared turbine will today show as good economy and often better at reduced speeds than the turbo-electric drive. The turbo-electric drive offers certain advantages in maneuverability and reversing power, but these characteristics of the geared turbine are usually considered entirely satisfactory. The geared turbine has been selected for most cargo-ship installations in place of turbo-electric drive because of the appreciably lower cost of the former, the slightly better steam rate, and the fact that the turbo-electric drive offers no special advantages not possessed by the geared turbine. The electric drive offers certain advantages for tankers because of the possibility of using the main generators for supplying power for the cargo pumps. Upkeep costs are probably in favor of the electric drive. In some of the earlier installations, turbo-electric drive was selected because of some doubt as to the reliability and upkeep costs of reduction gears. However, with the successful records of the geared-turbine drive, this is not an important consideration today.

The claim has been put forward for the electric drive that it eliminates the shaft alley, but this is accomplished only by a reduction in cargo space caused by the motor compartment aft. It is also questionable practice to place a piece of machinery in a compartment that has no communication with the engine room except by way of the deck.

For passenger ships, the turbo-electric drive offers certain advantages, and it is in this type of ship that most of the turbo-electric installations have been made. The advantages are quietness of operation (elimination of the noise of the gears) and the good economy at about three-quarters speed, when one of the two generators can be shut down.[1]

123. Space Occupied by Machinery.—The adoption of an improved type of propelling machinery that results in a reduction of the machinery space may lead to an increased operating expense due to higher port dues that are charged against net tonnage. The various national rules for net tonnage measurement allow a deduction of 32 per cent of the gross tonnage for the machinery-space allowance, provided that the actual tonnage measurement of this space is above 13 and under 20 per cent of the gross tonnage. If the space does not lie within these limits, the deduction for the machinery space is 1.75 times the tonnage of the actual space. The 32 per cent deduction does not apply to Panama and Suez Canal tonnage measurements, and hence this factor is not so important today as it was before 1938, when the 32 per cent rule was used at Panama for net tonnage calculations.

A. T. Wall[2] cites the following example:

A ship 580 × 70 × 45 ft, with a gross tonnage of 20,000, was fitted with reciprocating engines and Scotch boilers. The space occupied by engines and boilers works out to 14 per cent, allowing a reduction from gross tonnage of 32 per cent, or 6,400 tons. With an allowance of 1,350 tons for crew space, the total reduction becomes 7,750 tons. This gives a net tonnage of 12,250 tons. If the same ship is fitted with turbine machinery and water-tube boilers, the actual space is 10 per cent, which allows a reduction of 10 × 1.75 or 17.5 per cent instead of 32 per cent, as before. This results in a net tonnage of 15,150—an increase in net tonnage of 24 per cent, with an actual gain of only 5 per cent in cargo space. Thus the owner would be required to pay port tonnage dues on 2,900 additional tonnage by the adoption of more efficient machinery unless care was taken in the design to increase the machinery space by additions to the light and air spaces so that it measured slightly more than 13 per cent of the gross tonnage.[3]

[1] A paper by W. E. Thau, "Modern Turbine Propulsion," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1929, gives an excellent comparison of the merits of the geared turbine and turbo-electric drive.

[2] "The Tonnage of Modern Steam Ships," *Trans. Inst. Naval Arch.*, 1919.

[3] The student should consult, Chap. 2, vol. I, of Rossell and Chapman's "Principles of Naval Architecture," published by the Society of Naval

A geared-turbine installation occupies considerably less space than the reciprocating-engine installation and about the same space as the turbo-electric drive. The geared-turbine installation usually occupies slightly less space than straight Diesel drive and about the same space as or a little more space than the geared Diesel drive, depending on the rpm used with the latter type of machinery.

Some machinery installations that occupy exceedingly small fore-and-aft space and yet secure the full 32 per cent machinery allowance are the three types of single-screw ships designed (1939) by the U.S. Maritime Commission. These ships have the boilers and main engines in the same compartment (see Figs. 132 and 133). The length of these installations are given below:

Type	Machinery	Shp	Length of machinery space, ft
C-1	Geared turbine	4,000	45
C-2	Geared turbine	6,000	47 5
C-3	Geared turbine	8,500	50
C-2	Straight Diesel	6,000	47 5

124. The Diesel Engine.—The outstanding feature of the Diesel engine is its low fuel consumption. The fuel consumption of a Diesel installation averages around 0.38 lb per shp per hr, compared to around 0.57 to 0.60 for the modern oil-fired geared-turbine ship of 4,000 shp and upward and around 0.80 to 1.00 lb for oil-fired reciprocating-engine installations.

Today the first cost of a Diesel engine installation is only slightly higher than that of a high pressure geared-turbine installation. Data published[1] in 1940 showed that the U.S. Maritime Commission's motor ship cost approximately 8 to 10 per cent more than the geared-turbine ships. Some of the published bids show prices for motor ships less than 1 per cent higher than those for steamships. The ratio of European costs is usually smaller than in the United States.[2]

Architects & Marine Engineers, New York for a full discussion of this important subject.

[1] *Trans. Soc. Naval Arch. & Mar. Eng.*, 1940, p. 343.

[2] See *British Motor Ship*, vol. 15, p. 262.

The cost of Diesel machinery naturally depends on the rpm of the engine, whether the engine is two- or four-cycle, single- or double-acting. An increase in revolutions naturally lowers the first cost, weight, and size of the engine, but for a direct-connected installation an increase in revolutions usually lowers the propeller efficiency; so the rpm cannot be increased to too great an extent in order to reduce the first cost.

The upkeep cost over a period of years for Diesel machinery produced by experienced builders is probably no greater than that of a modern geared-turbine installation. When heavy fuel oil is used in Diesel engines, upkeep costs may prove to be higher than for steamships.

The cost of the engine-room crew for a Diesel ship is about the same as that of a steamship, and today little difficulty is experienced in obtaining trained engineers for motorships.

Diesel fuel oil costs considerably more than boiler fuel oil (bunker *C*), and it is this factor that is retarding the adoption of Diesel machinery for powers of 3,000 shp and upward. Fifteen and twenty years ago, when the fuel consumption of the steamship was around 1.00 lb of oil per shp per hr and that of the motor ship was around 0.42 lb, this difference in price of the fuels was not serious. Today, however, with the fuel consumption of the modern geared-turbine ship of moderate and high powers in the neighborhood of 0.60 lb per shp-hr, the motor ship has an advantage of only 37 per cent in unit fuel consumption over the steamship, and frequently the cost of Diesel fuel is about this much higher than bunker *C* fuel. As pointed out in Art. 107, some recent motor ships are attempting to overcome this handicap of fuel price by utilizing a lower grade and cheaper fuel.

The lubricating-oil consumption of the Diesel engine is appreciably higher than for steam machinery and is a factor adverse to the Diesel engine.

One serious disadvantage of Diesel machinery is that it does not have the overload capacity that the geared turbine has and hence cannot be forced so readily when the ship is behind schedule or when an extra amount of speed or power is desired for short periods. If Diesel machinery is to have an overload capacity, a reserve in mep must be allowed. This results in a lower service mep being employed than might otherwise be the case. A reduction in service mep for a fixed rpm means an increase in the bore or stroke (or number of cylinders), with its attendant increase in first cost.

A geared-turbine installation, on the other hand, can be built with an appreciable overload capacity with little increase in the first cost. An increase in power can be obtained by simply cutting in a few more nozzles and increasing the rate of combustion of the boilers.

The fuel consumption of small Diesel engines is practically the same as that of large engines unless high rpm is employed in the small engines. The unit fuel consumption of the geared-turbine steamship, on the other hand, increases as the shp of the installation is reduced. This is due partly to the lower engine efficiency of the small turbine and partly to the fact that the numerous refinements used with large shp steam installations to reduce fuel consumption are not warranted in small shp installations. The result of this is that the Diesel engine has a definite advantage over the geared-turbine installations of small horsepowers. No exact value can be given for the horsepower at which the motor ship is more economical than the steamship, since many factors are involved, such as first cost of machinery, assumptions for annual repairs, fuel costs, gain in carrying capacity of the motor ship, etc. As already pointed out, the improved types of reciprocating steam engines often have lower steam consumptions than the geared turbine at small horsepowers and hence are more competitive with the Diesel engine in powers around 1,000 to 2,000.

One advantage of the motor ship over the steamship is the reduced amount of fuel carried for a voyage of given length. If we assume the fuel consumption of the motor ship as 0.38 and that of the steamship as 0.63, the motor ship will carry 40 per cent less fuel than the steamship for a given length of voyage. For two ships of the same displacement, the greater the distance between bunkering ports the greater will be the cargo deadweight of the motor ship. Today, where much of the world's cargo is carried on a measurement basis, this advantage may not be so great as when dead weight cargo is carried. However, even if cubic cargo is carried, since the motor ship carries less fuel for a given voyage, it can purchase more fuel at ports where prices are favorable, and for the same bunker capacity as the steamship the motor ship can often bunker for a round voyage whereas a steamship cannot.

The marine Diesel engine was developed in Europe before it was developed in the United States, and the Europeans have had far more experience with this type of machinery than we have had in the United States. The result is that many European steamship owners have large fleets of successful motor ships, whereas there are comparatively few large motor ships under the American flag. The Japanese have followed the European lead, and practically all the merchant ships built in Japan in recent years have been propelled by Diesel engines. In the United States, on the other hand, the development of marine steam machinery has been carried further than in any other country, especially the machinery for cargo ships, as represented by the U.S. Maritime Commission's C-1, C-2, and C-3 type ships. The result is that the ratio between the unit fuel consumption of steam and Diesel machinery is smaller in the United States than in Europe and Japan.

The geared Diesel engine and the a-c Diesel-electric drive have been developed to bring the cost, weight, and space requirements of Diesel machinery down so that these types of drive are better able to compete with steam machinery.

A machinery comparison should be for a definite ship on a specified trade route. General comparisons are of little value. Often a comparison is made by designing a steamship and then fitting Diesel machinery to the same hull instead of laying out a special design to obtain all the advantages of Diesel machinery (see also Art. 108).

125. Comparison of the Two- and Four-cycle Diesel Engine.—Two-cycle and four-cycle engines working on both the single- and double-acting principle have been used for ship propulsion. The four-cycle single-acting engine was used in a great many of the earlier motor ships. Double-acting four-cycle engines were used on several of the large passenger liners, but today no engines of the latter type are being built. The predominating types today for ship propulsion are the two-cycle single-acting engine for small powers and the two-cycle double-acting engine for large powers. Four-cycle single-acting engines are used in the smaller powers for propulsion and for auxiliary drives. The reason for the wide use of the two-cycle engine is its greater bhp for a given set of engine dimensions and revolutions and consequently its lower cost per horsepower. A few years ago, the four-cycle single-acting trunk piston engine employing supercharging was an active competitor of the two-cycle engine, but at the present writing not so many engines of this type are being built for ship propulsion as formerly.

British Motor Ship, in its January issue each year, gives data on the engines built throughout the world during the past year. The following table is a summary from that publication.

TABLE XXI.—TYPES OF DIESEL ENGINES BY NUMBER OF SHIPS AND TOTAL IHP

	1928	1930	1935	1938
Two-cycle single-acting	73	102	47	111
	...	...	181,330	525,600
Two-cycle double-acting.	12	10	33	46
	...	...	213,150	489,600
Four-cycle single-acting	100	120	49	66
	. .	..	237,800	183,500
Four-cycle double-acting	4	8	0	0
			0	0

126. Fuel Consumption at Reduced Speed.—As the speed of a ship and hence the power output of the engine are reduced, the fuel consumption per shp-hour increases with practically all types of marine propelling machinery. The amount of fuel consumed per hour, of course, decreases, as the reduction in power is greater than the increase in the unit fuel consumption. The rate at which the unit fuel consumption increases varies with the type of machinery and the method employed to decrease the power of the machinery. A few typical curves are shown in Fig. 104.[1] It will be observed that the curve for the Diesel engine is fairly flat down to about 30 per cent full power. It will also be noted that modern steam machinery gives a curve as flat as that of Diesel machinery down to about 50 per cent of full power. The older types of steam machinery, as shown by the curves of installations dated 1920, had very high unit fuel consumptions at reduced speeds.

Cargo ships, as a rule, seldom operate for long periods at reduced speed; hence the fuel consumption at reduced powers is not an important consideration. For passenger ships it is usually important for powers from 100 to 40 per cent, and for naval

[1] The reason why part of the curve for the *U.S.S. Saratoga* is drawn dotted is explained in Art. 94.

vessels it is of extreme importance for all powers from 100 down to around 5 per cent. Since power for a ship varies approxi-

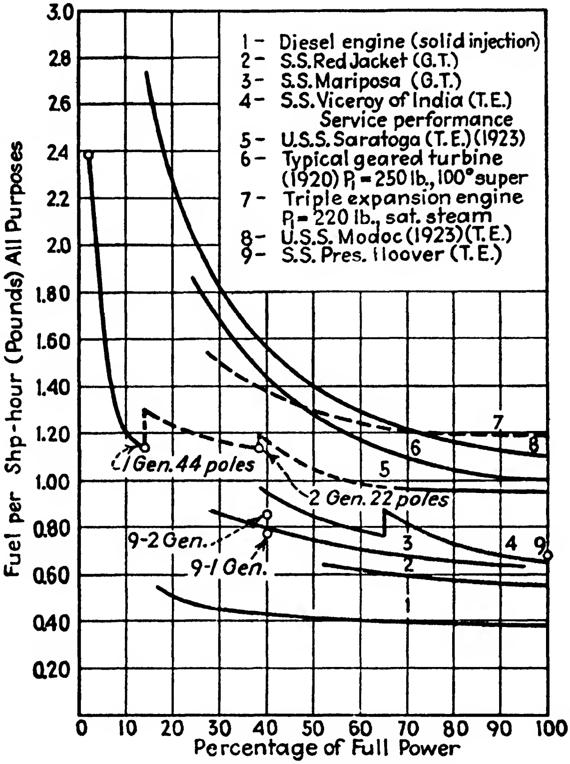


FIG. 104.—Fuel-consumption curves for typical ships.

The sources of data, shp, and initial steam conditions for ships in the above figure are as follows:

S.S. Red Jacket, Marine Engineering and Shipping Review, April 1940. 6,000 shp. Throttle pressure = 440 lb per sq in. gage, and 740°F.* (Based on 18,500 Btu.)

S.S. Mariposa, Trans. Soc. Naval Arch. & Mar. Eng., 1932, p. 349. 22,000 shp. Pressure in H.P. chest = 361.5 lb per sq in. gage; temperature at superheater outlet = 670°F. (Based on 19,000 Btu.)

S.S. Viceroy of India, Trans. Inst. Naval Arch., 1930, p. 54. 17,000 shp. Boiler pressure = 375 lb per sq in gage and 700°F.*

U.S.S. Saratoga, Trans. Soc. Naval Arch. & Mar. Eng., 1929, p. 201. 180,000 shp. Pressure H.P. chest = 265 lb per sq in. gage and 485°F. (Based on 19,500 Btu.)

U.S.S. Modoc, Trans. Soc. Naval Arch. & Mar. Eng., 1923, p. 143. 2,600 shp. Boiler pressure = 200 lb per sq in. gage and 90° superheat. (Based on 18,900 Btu.)

S.S. President Hoover, Trans. Soc. Naval Arch. & Mar. Eng., 1931, p. 202. 32,000 shp. Pressure in H.P. chest = 266 lb per sq in. gage and 166° superheat. (Based on 18,500 Btu.)

* Designed conditions.

mately as the cube of the speed, half speed requires only about one-eighth power, and 79 per cent speed requires half power.

When the power is reduced by cutting out nozzles, most modern geared-turbine installations show satisfactory fuel performance down to around three-quarters speed, and naval installations have fairly flat performance curves down to half speed (12.5 per cent power) when cruising blading is fitted.[1] However, when the speed is reduced below one-half (and the power decreased below 10 per cent), the unit fuel consumption per shp per hour for steamships increases very rapidly. Diesel-electric drive using d-c current, as pointed out in Art. 115, shows a good fuel performance at reduced speed when four or more engines are employed, since engines can be cut out at reduced powers and the remaining engines operated at full mep and full rpm. Engines can also be cut out at reduced speeds with a geared Diesel engine installation and a-c Diesel-electric drive; but here the revolutions of the engines have to be reduced directly with the speed of the ship, as in a direct-connected Diesel installation.

In order to improve the performance of steam naval vessels at low speeds, Diesel cruising engines have been installed on several German cruisers,[2] and one Norwegian naval vessel[3] employs steam turbines for full power and a Diesel-electric drive at low speeds.

In these installations the Diesel cruising engines are also used to assist the steam machinery at full power, and thus the weight of the machinery is not increased by engines that are not used at full speed. On the *S.S. Olaf Tryggvason*, the electric motors are fitted on the line shafting after the geared turbines. The motors can be used alone or can be used to assist the turbines. Full power of the Diesel engines can be obtained at all propeller revolutions by adjusting the motor field rheostat. The four cruising Diesel engines of the *Leipzig* are geared to the center shaft, and the geared turbines are on the two wing shafts. In order that the Diesel engines can turn up their full revolutions and hence develop full power at all speeds of the ship, an adjustable pitch propeller was fitted. In cruising, the turbines are shut down and disconnected from the propeller shaft. The propeller shafts are turned by electric motors so that the wing propellers will not cause a drag when the ship is being driven only by the Diesel engines on the center shaft.

[1] See Art. 89.

[2] *British Motor Ship*, vol. 12, p. 396. *The Shipbuilder and Marine Engine Builder*, vol. 39, pp. 285–287.

[3] The *Olaf Tryggvason*, *British Motor Ship*, vol. 15, pp. 270–272.

CHAPTER X

CONDENSERS AND CONDENSER AUXILIARIES

127. General.—Practically all condensers used on shipboard are of the surface type. A surface condenser consists of a shell into which the steam enters and a series of tubes through which the cooling water passes. The condensed steam and the cooling water do not come into direct contact with one another. Only this type condenser will be discussed in this chapter. As pointed out previously, ships operating in fresh water sometimes use jet condensers in which the cooling water and steam mix together and both the condensate and cooling water are discharged overboard.

The reasons for using surface condensers are: (1) To lower the back pressure on the prime mover and thus reduce the enthalpy of the steam h_2 and cause a greater amount of the energy existing in the steam at the throttle to be converted into useful work $(h_1 - h_2)$. This, of course, reduces the steam consumption of the prime mover and the fuel consumption of the ship. (2) To save the condensate so that it can be used over again as boiler feed water.

Theoretically, the higher the vacuum the greater will be the efficiency of the prime mover; but practical considerations impose limits to which the vacuum can be economically carried. With a reciprocating engine, 25 to 27 in. is about the limit with sea water of from 60 to 65°F. Vacua greater than this require an undue size for the L.P. cylinder in order to handle the increased steam volume. The increase in the capital charges due to the larger low-pressure cylinder and condenser, together with the additional cost of producing a higher vacuum and the losses brought about by increased cylinder condensation and friction of the engine, will more than offset the gain in economy and horsepower due to the higher vacuum.

With a turbine, the disadvantages mentioned above for reciprocating engines caused by the larger cylinder volumes do not exist, and the economical limit of vacuum is probably around

28.5 in., with a sea temperature of 70°F. With colder sea water, a higher vacuum can be carried economically. The capital charges on equipment and the cost of producing vacua higher than 29 in. will, as a rule, more than offset any reduction in steam consumption unless the sea water is at a low temperature. Ships that operate regularly in warm water are at a disadvantage when geared turbine machinery is installed, as they cannot carry a high vacuum. This condition might at times make another type of machinery, such as reciprocating steam engines or Diesel engines, more advantageous.

The following table shows the theoretical steam consumption for two turbines, both having the same initial conditions but one operating with a 27½-in. and the other with a 28½-in. vacuum.

	Case 1	Case 2
Pressure at throttle, lb abs	395	395
Temperature at throttle, °F	680	680
Vacuum in condenser	28 5	27.5
h_1	1,352	1,352
h_2	895 5	921
$h_1 - h_2$	456 5	431
Theoretical steam consumption w_t	5.58	5.90

The theoretical steam consumption for case 2 is 5.75 per cent higher than for case 1.

The main function of a condenser is, of course, to condense the steam coming from a reciprocating engine or turbine. However, a condenser performs a secondary and very important duty, namely, cooling the air contained in the entering steam in order to reduce its volume so that it can be readily removed from the condenser. Thus the condenser functions as an air compressor as the partial pressure of the air in the condenser is increased and its volume decreased by the cooling water passing through the condenser. Proper allowance must be made in condenser design for this cooling of the air.

A condenser for a high-vacuum installation has three important auxiliaries: (1) a circulating pump, which supplies the comparatively cold sea water to condense the steam and cool the entrained air and thus produce the vacuum; (2) a condensate pump for removing the condensate from the bottom of the condenser; and (3) an air ejector for removing the air from the condenser shell as fast as it enters, thus allowing the vacuum to be maintained in the condenser. In installations carrying a low vacuum, such as a reciprocating engine, the condensate pump and air ejector are combined into one unit known as an "air pump." This is discussed in a later article.

In the majority of merchant-ship condensers, the cooling water makes two passes through the condenser, first through the lower half of the tube bank and then through the upper half of the tube bank, the inlet and outlet water connections being on the same end of the condenser (Fig. 107). Such condensers are known as "two-pass condensers." One-pass condensers are also often used on shipboard (see Fig. 111). In this type, the cooling water enters at one end, passes through all the tubes, and is discharged overboard at the other end of the condenser. Three-pass condensers have been employed at times in order to reduce the length of the condenser, but they are extremely rare in marine practice.

128. Heat Transfer.—A great many factors influence the rate of heat transfer through condenser tubes, and exact results cannot be obtained by any simple formula. The application of the recent developments in the theoretical approach to heat transfer in condensers and other heat-transfer apparatus is beyond the scope of this book.

George H. Orrok has given the following equation for heat transfer between steam and water in condensers:

$$U = 350C_1C_2\left(\frac{P_s}{P_t}\right)^2\sqrt{V}$$

where U = Btu transfer per square foot per degree difference per hour.

C_1 = cleanliness coefficient (varies between 1.00 and .50).

C_2 = material coefficient (see the Table XXII).

V = velocity of cooling water in feet per second.

P_s = partial pressure of the steam in the condenser.

P_t = total pressure of steam and air.

The foregoing equation for U does not allow for all the factors influencing heat transfer. Other factors influencing the rate of heat transfer are the velocity of the steam passing over the tubes, the temperature of the injection water, the temperature of the entering steam, the arrangement of the tubes, and the presence of a film of condensed water on the tubes. The Heat Exchange Institute[1] gives curves that show small variations in U with tube diameter, inlet water temperature, and condensation per square foot. The preceding equation, however, gives values for marine condenser using ¾-in. tubes that check fairly well with these curves for full-powered operation and inlet water temperatures varying between 60 and 75°F.

The air-richness factor P_s/P_t will, of course, vary throughout the condenser, decreasing rapidly at the bottom. Orrok gives values of $P_s/P_t = 0.95$ to 0.97 for tight condensers. Table XXIII shows this ratio varying between 1.00 and 0.99 in the upper part or steam-condensing zone of the condenser and between 0.90 and 0.50 in the lower part of the condenser, where there is a reduction in the temperature of the steam-air mixture of 20°F. In two-pass condensers, about three-quarters of the heat transfer between the steam and cooling water takes place in the second (upper) pass of the circulating water; hence, in an efficiently operated condenser where the temperature drop of the steam-air mixture is around 7 to 10°, an air-richness ratio of around 0.95 would appear a reasonable value to use in condenser calculations.

In early marine installations where reciprocating steam engines were used, water velocities of between 3 and 5 ft per sec were the usual practice. With the introduction of the turbine, and hence higher vacua, the velocity of the cooling water was increased in order to keep down the size of the condenser. Velocities of 8 and 10 ft per sec were common, and even higher velocities were employed in some of the earlier turbine-driven high-speed ships where weight and space were important. Tube failures, however, were very frequent with these high velocities (Art. 132), and today velocities of over 8 ft per sec are seldom used.

TABLE XXII.—MATERIAL COEFFICIENT (C_2)

Copper	1.00	Tin	.79
Admiralty	.98	Zinc	.75
Aluminum lined	.97	Monel metal	.74
Admiralty (oxidized)	.92	Shelby steel	.63
Aluminum bronze	.87	Admiralty (badly corroded)	55
Cupro-nickel	.80	Glass	.25

[1] "Standards of Heat Exchange Institute," Part 1, New York, 1939.

With condensers for low vacua, where the rise in temperature of the cooling water between inlet and discharge is large because of the higher temperature existing in the condenser, the speed of the cooling water will have to be reduced below 6 ft per sec in order to keep the length of the condenser within reasonable limits.

The horsepower of the circulating water pump increases nearly as the cube of the water velocity; hence high velocities while reducing the size and cost of the condenser (by increasing U) will increase the cost of the circulating pump and the cost of producing the vacuum.

Condenser tubes are seldom perfectly clean; hence it is well to allow for slightly dirty tubes in designing condensers. It is suggested that a cleanliness coefficient C_1 of .90 be allowed for trial conditions. If lower values are allowed in fixing the condenser cooling surface, it would be possible to maintain the full vacuum in service with less cleaning of tubes. Values for the material coefficient C_2 are given in Table XXII.

Calculations for condenser are carried through in Arts. 156 and 157, and a study of these calculations will make clear the foregoing discussion.

129. Air Leakage.—The most important factor in maintaining a high vacuum is the presence of air in the condenser. The duty of the air ejector or air pump is to remove the air from the condenser at the same rate at which it enters. The capacity of the air ejector or air pump and the power required to operate it depend on the amount of air leakage and not necessarily on the vacuum carried or the size of the plant. The possibility of air leakage, of course, increases with the amount of vacuum, the size of the propelling unit, and the amount of feed water used.

As we have already seen, the presence of air in a condenser decreases the rate of heat transfer between the exhaust steam and the cooling water and thus lowers the efficiency of the condenser. Further, the presence of air in a condenser lowers the temperature in the condenser for a given vacuum, which often causes the condensate to leave the condenser at a lower temperature. This results in a lower temperature of the feed water entering the feed-water heater and reduces somewhat the over-all efficiency of the plant.

Not only must the condenser and exhaust trunk to the condenser be kept tight but glands must be designed so that there is practically no air leakage, and the boiler feed water should be as free of air as possible. Drains containing air and exhaust of auxiliary engines should not be led to the main condenser.

With reciprocating engines there may be considerable leakage around the low pressure valve and piston rod, since the pressure in the cylinder is less than that of the outside air. The steam seal fitted on the L.P. rods of the later ships of the U. S. Navy that employed triple-expansion engines reduced the possibility of air leakage and also prevented the entrance of lubricating oil on the L.P. rod into the cylinder. The methods employed to prevent air leakage around the shaft into the low pressure turbine are discussed in Chap. VI.

When a condenser is in operation, the weight of air entering and leaving in a given time must be the same. If the volume of this air when it reaches the air pump or air ejector suction is large and the air-removing apparatus is not of proper size to handle it, the vacuum falls until an equilibrium has been established between the air pump and the condenser.

Methods of cooling (and hence compressing) the air and removing it from the condenser and the hourly quantity of air that may be expected to enter a condenser are discussed in later articles.

130. Thermodynamics of the Condenser.—The exhaust entering a condenser is a steam-air mixture, *i.e.*, a mixture of two gases, and hence Dalton's law applies.

Dalton's law states:[1]

1. The pressure of a gaseous mixture (composed of two or more gases or a gas and a vapor) contained within a vessel is the sum of the pressures that the separate gases would exert if each occupied the vessel alone.

2. Each constituent behaves as if the others were not present, *i.e.*, each occupies the volume of the vessel, and each exerts its own pressure, corresponding to the temperature within the vessel.

A mixture of water vapor, steam, and air in a condenser follows this law closely. The temperature at any point within the condenser must be the same for both air and steam and must correspond to that given in the steam tables for the partial pressure of the steam. The vacuum or absolute pressure in the

[1] The student is referred to textbooks on thermodynamics for a complete statement and explanation of Dalton's law.

condenser is the sum of the partial pressure of the air and the partial pressure of the steam.

The following equations express Dalton's law and give the relations of volume, pressure, and temperature in a condenser:

If V_a = volume of air present.
V_s = volume of steam present.
t_a = temperature of air.
t_s = temperature of steam.
P_t = total pressure existing in the condenser.
P_a = partial pressure of air.
P_s = partial pressure of steam.
W_a = weight of air present in condenser.

We have

$$V_a = V_s = V_t$$
$$t_a = t_s = t_t$$
$$P_t = P_a + P_s$$
$$P_a = \frac{W_a R(t_a + 460)}{144 V_a}$$

The following example will make clear the application of Dalton's law to a condenser:

$$\text{Vacuum in condenser} = 29 \text{ in.}$$
$$P_t = 1.0 \text{ in. of mercury } (0.491 \text{ lb per sq in.})$$

Suppose that the temperature existing in the lower part of the condenser is 70°F. From the steam tables, the partial pressure of the steam P_s, corresponding to a temperature of 70°, is 0.739 in. of mercury (0.363 lb per sq in.).

We now have

$$P_t = 0.491 \text{ lb per sq in.}$$
$$t_s = t_a = 70°\text{F}$$
$$P_s = 0.363 \text{ lb per sq in.}$$

V_s (volume occupied by 1 lb of steam at 0.739 in. of mercury) = 867.8 cu ft

$$P_a = P_t - P_s = 0.491 - 0.363 = 0.128 \text{ lb per sq in.}$$
$$P_a = \frac{W_a R T_a}{144 V_a} = \frac{W_a \times 53.3 \times 530}{144 \times 867.8} = 0.128$$
$$W_a = 0.562 \text{ lb of air per lb of steam}$$
$$\frac{P_s}{P_t} = 0.74$$

The ratio of the weight of air to the weight of steam in a condenser varies through the condenser. At the inlet of the condenser, the ratio of air to steam is very small, and its effect is negligible. As the mixture passes through the condenser, the steam is condensed, and the weight of water vapor present is constantly getting smaller, whereas the weight of air remains the same. Thus the ratio of air to water vapor increases rapidly until at the air-pump suction it is high.

The partial pressure of the air increases as the steam is condensed, becoming a maximum in the bottom of the condenser. The increase in the partial pressure of the air causes a reduction in the partial pressure of the steam, the total pressure in the condenser remaining constant. This reduction in the partial pressure of the steam results in a lowering of the temperature of the mixture in the lower part of the condenser. In the types of condensers formerly in general use and still employed to some extent today, no attempt was made to cool the air in the condenser without cooling the condensate or to reheat the condensate in the bottom of the condenser. The result was that the condensate was cooled to a temperature lower than that corresponding to the vacuum, which reduced the temperature of the water entering the feed heater. This depression of the condensate temperature allows the water in the condenser to absorb more air, which may be carried to the boiler in solution in the feed water. Formerly, a large depression in the temperature of the condensate was often indicative of large air leakage, the circulating pump having been speeded up to cool the air so that it could be readily removed by the air pump. However, low condensate temperatures may be due to poor condenser design or to improper regulation of the cooling-water supply. With modern types of condensers and air ejectors, large depressions of the condensate temperature seldom occur.

The table on page 302, from a paper by Stuart and Senner[1] shows the action of a mixture of 1 lb of air and steam in a condenser.

In the table on page 302 the air leakage has been taken as 0.0002 lb of air per lb of steam. This corresponds to 1.2 cu

[1] The *Journal of the American Society of Naval Engineers*, 1920 (modified from a paper by Gibson and Bancel, in the *Transactions of the American Society of Mechanical Engineers*, 1915).

ft of free air per min, with a steam consumption of 27,000 lb per hr.

$$1.2 \text{ cu ft of free air per min} = 1.2 \times 0.076 \times 60 = 5.45 \text{ lb per hr}$$

$$\frac{5.45}{27{,}000} = 0.0002 \text{ lb of air per lb of steam entering condenser}$$

A study of this table will indicate the action taking place in the condenser. At the entrance of the condenser, the total pressure is 1 in. of mercury. The partial pressure of the air by Dalton's law is

$$P_a = \frac{W_a RT}{V_a} = \frac{0.0002 \times 53.3 \times 539}{657 \times 144} = \frac{1}{16{,}000} \text{ lb per sq in.}$$

$$= \frac{1}{8{,}000} \text{ in. of mercury}$$

where $V_a = V_s$, corresponding to the volume of steam at 79°, which is the temperature at the top of condenser.

W_a = pounds of air present per pound of steam.

As the steam is condensed by the cooling water, the weight of steam decreases rapidly from the top to the bottom of the condenser. The preceding equation will show that the partial pressure of the air P_a increases as the volume of vapor is decreased (line 3).

The increase in the partial pressure of the air causes a decrease in the partial pressure of the steam (line 2), since the total pressure is constant. This decrease in P_s causes the temperature in the condenser to decrease (line 1), since $t_s = t_a$ = temperature corresponding to P_s in the steam tables.

Both the increase in the partial pressure of the air and the decrease in the condenser temperature from top to bottom cause the volume occupied by the given weight of air to be decreased. (line 7).[1]

The values given in line 7 show clearly how the increase in the partial pressure of the air within the condenser reduces the volume of the air from a value of 657 cu ft per lb of steam-air mixture entering the condenser to 0.412 cu ft at a temperature of 72°F or 1/1,600 of its original volume and to 1/4,000 of its original volume at 59°, the temperature given for the air-pump suction.

[1] The values in Table XXIII, which were taken from steam tables, differ slightly from those given in the Keenan and Keyes tables.

TABLE XXIII

Table showing typical action in a condenser supplied with 1 lb of steam-air mixture, containing 0.0002 lb of air (1.2 cu ft per min, with 27,000 lb of steam per hr), illustrating the process of steam condensation and air compression at a constant vacuum of 29 in.

1. Temperature of mixture, °F	79	79	79	79	76	72	59
2. Partial pressure of steam, in. of mercury	1 0	1 0	1 0	0.99	0.90	0.80	0.50
3. Partial pressure of air, in. of mercury	1/8,000	1/4,000	1/1,000	1/100	1/10	1/5	1/2
4. Air present, lb	0 0002	0.0002	0.0002	0 0002	0 0002	0.0002	0 0002
5. Steam condensed, lb	000	0 5000	0.8748	0 9873	0 9987	0.9993	0 99967
6. Steam present, lb	0 9998	0 4998	0.1250	0.0125	0.0011	0.0005	0 00013
7. Volume of air present, cu ft	657	328	82	8 2	0 823	0.412	0 164
8. Volume of mixture, cu ft	657	328	82	8 2	0.823	0 412	0.162
9. Volume of mixture, cu ft per lb of air	3,285,000	1,640,000	410,000	41,000	4,115	2,060	820
10. Air per 1,000 cu ft mixture, lb	0 0003	0.0006	0.0024	0 024	0 243	0 485	1 22
	Condenser entrance						Air-pump suction
	Steam condensing in this zone				Air cooling and compressing in this zone		

As is pointed out later, the design of a condenser should be such that the air can be cooled to the desired temperature in the condenser (in this case, 59°) and still be able to withdraw the condensate at a temperature only a few degrees below that of the vacuum (in this case, about 79°F).

131. Types of Condensers.—Practically all condensers fitted today in marine installations where a high vacuum is carried are designed to cool the air without cooling the condensate an undue amount and to reheat the condensate to nearly the temperature of the steam entering the condenser. The separate air cooling is accomplished by setting aside a certain group of the condenser tubes for air cooling and placing a baffle plate over these tubes so that the incoming steam and the condensed steam dropping from the upper tubes will not strike the air-cooling tubes. The reheating of the condensate in the bottom of the condenser is brought about by leaving one or more open passages through or around the tube bank so that part of the steam can rush down and strike the water surface before it is condensed.

A Weir regenerative condenser is illustrated in Fig. 105. The tube arrangement in one-half of the condenser is shown, the left-hand half being similar to the right-hand half. This condenser is constructed with a space below the tube banks that allows water to accumulate, which has a large surface area. The air-cooling tubes are shown at the right-hand side of the section under the baffle *D*. The air-injector suction for the right-hand half of the condenser is at *E*. It will be observed that the tubes are omitted in the center of the condenser, allowing a free passage *C* for the steam down to the large water surface in the bottom (see Fig. 128). The steam enters through the top *J*, the entrance area having a width nearly equal to that of the condenser and a length nearly that of the tubes. Most of the steam is condensed in the upper banks of tubes *B* and drips down to the bottom. Part of the steam and the entrained air passes down the open center space *C*, sweeps over the water surface, and rises into the space under the sloping baffle plates *D*. Here the remainder of the steam is condensed all the air collects, and the air with a small amount of water vapor is drawn out by the air ejector at *E*. The descending drops of condensed water that have been partially cooled by the circulating water pass through this steam as it sweeps over the water surface and are reheated to nearly the temperature of the steam. The steam coming in contact with the water surface tends to heat the condensate standing in the bottom to the steam temperature. The condensate thus being heated to the steam temperature can absorb only a very small amount of air and hence leaves the condenser nearly air-free. As pointed out in Art. 147, this condenser is used in conjunction

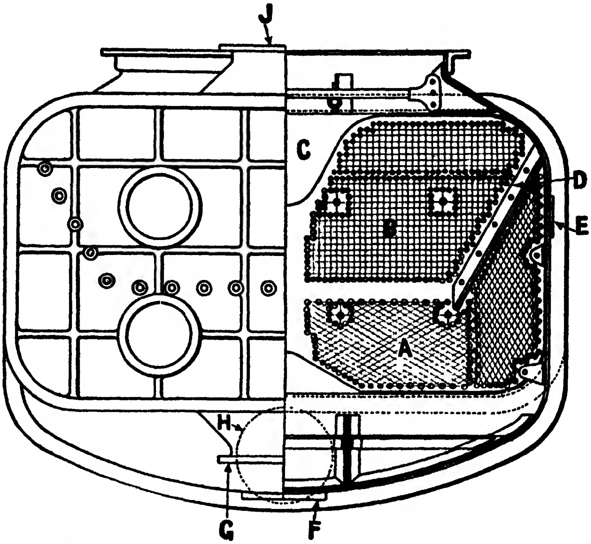


FIG. 105.—Part section of typical Weir regenerative condenser. (*W. & J. Weir, Glasgow.*)

A = first pass of cooling water
B = second pass of cooling water
C = clear steam lane to bottom
D = baffle over air-cooling tubes
E = air-ejector suction
F = condensate outlet
G = circulating water inlet
J = circulating water discharge

with the Weir closed-feed system. The feed tank (or surge tank) has been replaced by the water volume contained in the bottom of the condenser, the condenser also serving as a deaerator. Figure 106 shows a Weir regenerative condenser under construction. This condenser is of the same type as that shown in Fig. 105. The division plate in the water box separating the two passes of the cooling water is clearly shown. It will be noted that this division plate separates tube banks *B* in Fig. 105 from tube bank *A*. The connection for the circulating water inlet is shown at the bottom and the circulating water discharge at the top (covered over). The connection for air-ejector suction (*E* in Fig. 105) is shown on the right-hand side of the condenser.

In the Foster Wheeler condenser shown in Figs. 107 and 108, the air-cooling tubes are placed in the center—in the first (lower) pass of the cooling water—under an inverted V-shaped baffle plate. The air outlet to the air ejector is taken from the top

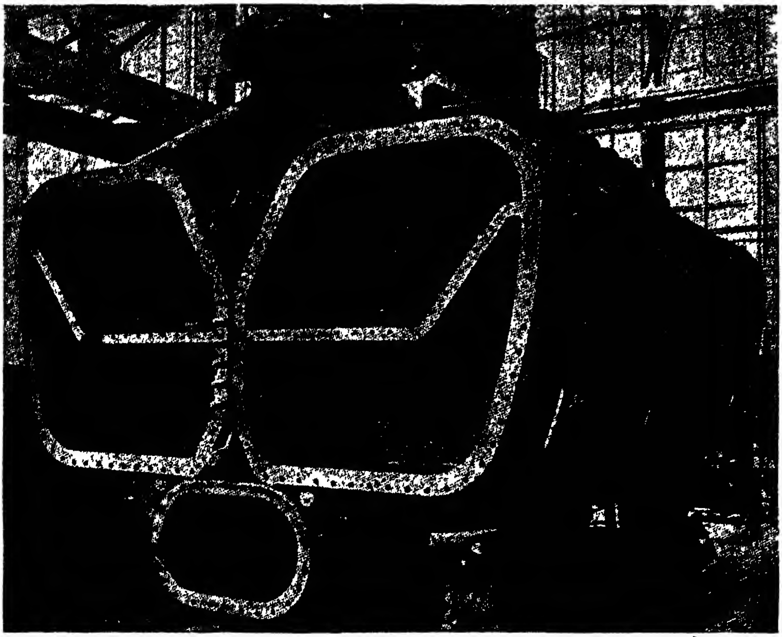


FIG. 106.—Weir regenerative condenser under construction.

of this space. The cylindrical hot well containing the condensate is attached to the bottom of the condenser. A by-pass area, which is shown on the right-hand side of the condenser (Fig. 108) and which extends over part of the length, allows some of the incoming steam to sweep down to the surface of the water in the hot well.

It will be noticed that the tubes are arranged on radiating lines with the steam space decreasing toward the bottom of the condenser. Part of the steam uncondensed in the upper banks and the entrained air passes through the lower tubes and up under the inverted V.

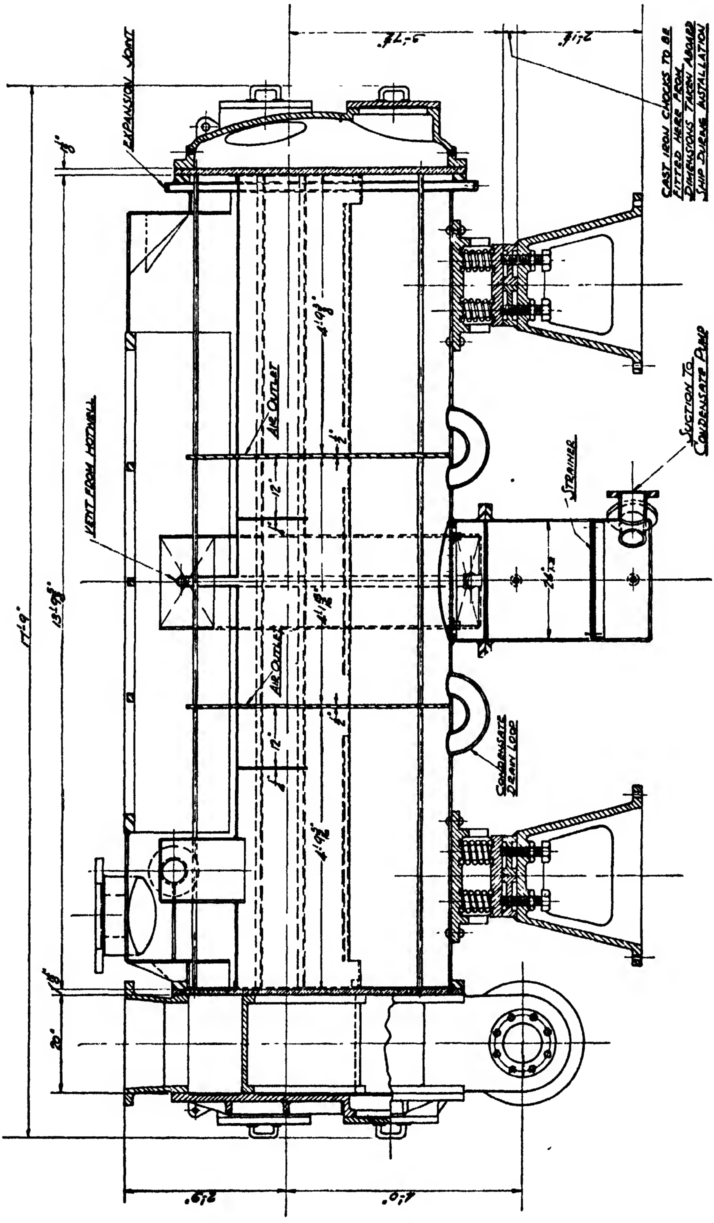


FIG. 107.—Longitudinal elevation of Foster Wheeler two-pass condenser of 4,800 sq ft of cooling surface.

This condenser is arranged for hanging under the L.P. turbine, and the spring supports relieve the turbine casing of part of the condenser weight and also do away with the need of an expansion joint between turbine and condenser.

Condensers are given various cross-sectional shapes today. Frequently the shape is made to suit the space limitations on shipboard. Many marine condensers are now placed directly

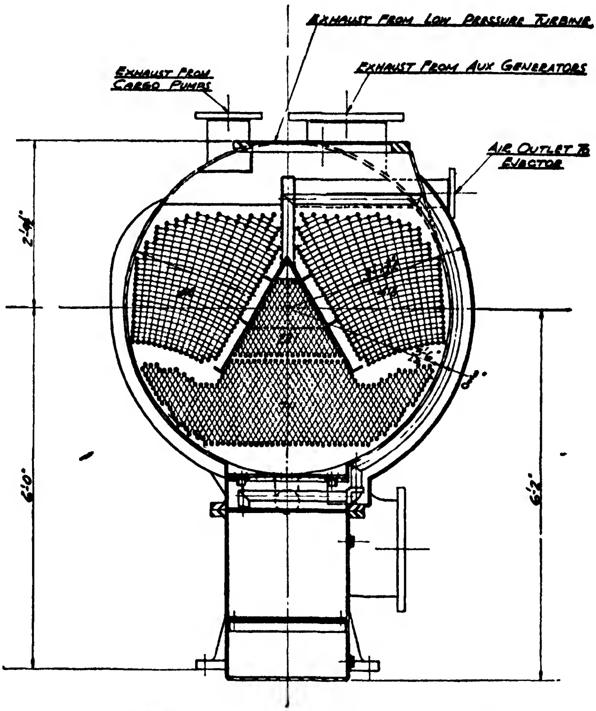


FIG. 108.—Cross section of Foster Wheeler two-pass condenser.

underneath the L.P. turbine, and a large top area for the entrance of the steam and a small depth are desirable; hence condensers frequently have a rectangular shape. A general rectangular cross section is used in the Weir regenerative condenser (Fig. 105) to allow a large water surface in the bottom, as discussed above. The circular section shown in Fig. 108 makes for a simple and inexpensive construction.

The condenser should be located as close to the prime mover as possible to reduce the possibility of air leakage and to reduce the pressure drop between the turbine and the condenser. The modern practice of placing condensers under the L.P. turbine makes an ideal arrangement. With reciprocating engines the exhaust pipe from the L.P. cylinder is usually of rather small cross section, and the condenser is placed further from the exhaust outlet from the engine (Fig. 113) than is the case with turbines. However lower vacua and smaller steam volumes are involved in reciprocating-engine installations. Most of the condensers fitted on merchant ships are of the two-pass type. The first pass of the circulating water is practically always through the lower part of the condenser. This gives a large temperature difference between the air-vapor mixture and the cooling water, thus reducing the volume of air to be handled by the air ejector. A two-pass condenser is shown in Figs. 107 and 108. The circulating water enters through the flanged pipe shown in the lower right-hand side of Fig. 108 (and the lower left-hand end of Fig. 107), passes up into the water box[1] on left-hand end of condenser and through the lower bank of tubes to the right hand. Here it makes a 180 deg turn in the space between the head and the tube sheet, passes back through the upper half of the tubes to the left-hand end, and leaves at the top through the upper flanged pipe shown at the left-hand end of Fig. 107. A horizontal division plate is placed in the water box at the left-hand end to separate the inlet and discharge water (see Fig. 106).

Single-pass condensers are universally used in cruisers and destroyers where scoops on the outside of the ship are used in place of circulating pumps. They have also been used on a number of American cargo and passenger ships (Figs. 59 and 135). In a one-pass condenser, the water enters at one end and leaves at the other. A single-pass condenser is illustrated in Fig. 109. The drawing in Fig. 109 is for the main condenser shell without the end water boxes attached. The two ends of this drawing represent the tube sheets that are shown in Fig. 110. The water boxes are separate castings that are bolted onto the flanges of the tube sheets. Covers are placed on the outside of the water-box castings. The shell of the condenser shown in Fig. 109 is built up of plates welded together, and no expansion

[1] The water boxes are the space at the ends of the condensers between the tube sheets and the heads. A water box with horizontal division plate is clearly shown in Fig. 106.

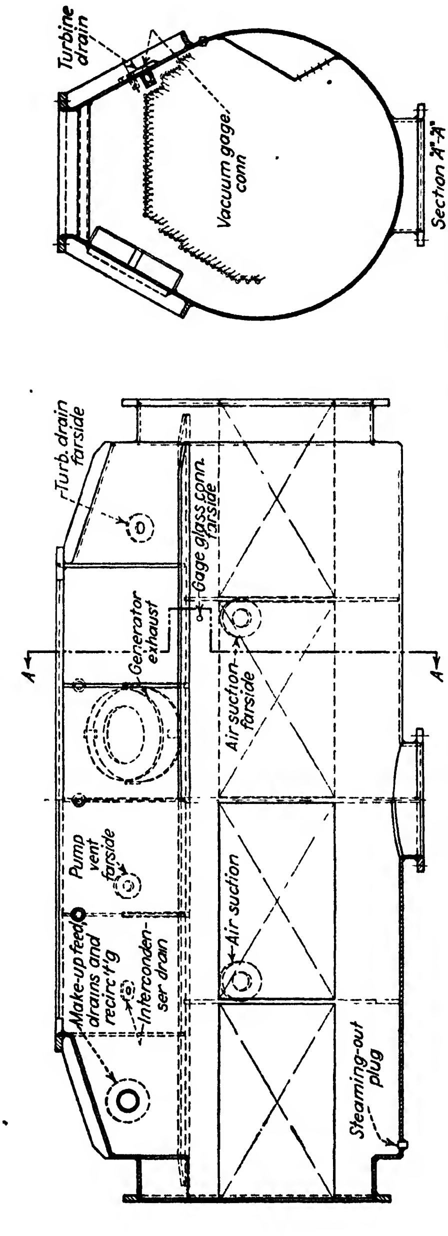


FIG. 109.—Single-pass condenser. (*Bethlehem Steel Company, Shipbuilding Division.*)

joint is used. The tubes for air cooling with baffle plate are shown at the right-hand side of the condenser. Single-pass condensers are longer than two-pass condensers, but slightly higher water velocities can be used because of the reduced turbulence. However, as shown in Art. 156, high water velocities require an increased tube length, and for this reason high velocities can seldom be used.

A number of American merchant ships have been fitted with condensers with the so-called "divided" water box. In these

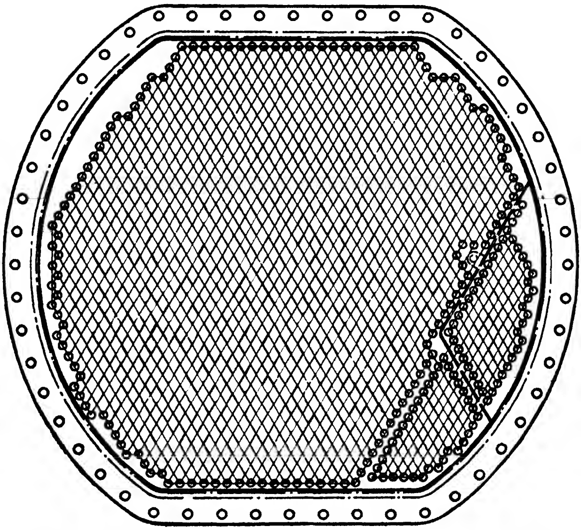


FIG. 110.—Tube sheet of condenser shown in Fig. 109.

condensers the two water boxes are divided into two parts by a vertical division plate. In a two-pass condenser one water box will also have a horizontal as well as the vertical division plate. Two inlet connections from the circulating pump and two overboard discharge connections are fitted to the condenser. By this arrangement half of the condenser can be cut out while the ship is under way, and thus half can be opened up for cleaning or plugging leaky tubes.

132. Condenser Tubes.—The condenser tubes are secured at each end of the condenser in vertical tube sheets. Formerly one end of the tube was securely fastened to one tube sheet by expanding the tube and left free to move in the other tube sheet because of changes in temperature. The free end was packed against salt-water leakage by screwed ferrules and cotton packing. Later, to secure a tighter joint, a metallic packing similar to that used for packing piston rods was substituted for the cotton packing. The tubes are supported by supporting plates at one or two points between the tube sheets.

In order to avoid the possibility of salt-water leakage into the condenser through tube packing and thence into the boiler, the present trend with turbine and water-tube boiler installations is to expand the tubes into both tube sheets and to provide for heat expansion by using a slightly curved tube or by placing expansion joints in the condenser sheet. Such an expansion joint is shown in Fig. 107. Some recent condensers with welded shells have been built with the shell diameter increased a short distance beyond the tube sheets and so designed that a small change in length can take place between tube sheets as the tubes change in length through changes in temperature (Fig. 109).

When the reciprocating steam engine was the universal type of machinery for propelling the steamship, condenser tubes were made of admiralty metal, consisting of 70 per cent copper, 29 per cent zinc, and 1 per cent tin. These tubes proved very satisfactory and had a long life. With the introduction of the turbine and high vacua, much higher cooling-water velocities were used, and condenser tubes began to fail after a very short life. In some cases it became necessary to retube the condenser within 2 years. The pitting that took place on these tubes, especially near the ends, caused shipowners to believe that these failures were due to erosion caused by turbulent water of high velocity that contained entrained air bubbles. However, after considerable study and research, it was proved that the final failure was due to corrosion by the breaking down of the very thin corrosion-resisting film that forms on the tubes. The film formed on the admiralty-metal tubes was able to withstand the low velocities formerly used but could not withstand the impinging action of the higher velocities. It has been found that a copper-nickel-alloy tube, made of 70 per cent copper and 30 per cent nickel, is able to withstand the higher water velocities and show a reasonably long life, because the thin protective film formed on such tubes is not broken down so easily as that on the tubes made of a copper-zinc alloy.[1]

[1] This subject is treated by Robert Worthington in an article in *Metal Progress* for July, 1933.

The cost of a copper-nickel tube is nearly twice that of an admiralty metal, and hence the life of a copper-nickel tube must be at least twice that of an admiralty tube to make its use economical. The coefficient of heat transfer for the former is also less than for the latter. Tubes made of an alloy of copper, zinc, and aluminum (2 per cent) and an alloy of copper, zinc, and small percentages of aluminum, and nickel show high resistance to corrosion, and since they are cheaper than the cupronickel tubes, they are also being used on shipboard. Admiralty metal is still used by some steamship companies, but the water velocity in the condensers having these tubes is kept low.

Condenser tubes used in merchant ships are usually ¾ in. outside diameter and No. 18 B.W.G. thickness. Naval vessels use a ⅝-in.-diameter tube of No. 16 or 18 B.W.G. The smaller tubes are used on naval vessels to make a lighter condenser.

133. Influence of Sea Temperature on Vacuum.—When the temperature of the sea water increases with a given vacuum, the mean temperature difference between the exhaust steam and the cooling water decreases. This reduces the rate of heat transfer and for a given cooling-water velocity calls for an increase in the cooling surface. Further, the temperature difference of the cooling water between injection and discharge is reduced, which requires a circulating pump of increased capacity and increased horsepower. These increases in first cost and operating expenses will usually result in a lowering of the designed vacuum for vessels that are to operate in warm water. An example will make this clear.

Example.—Suppose that a ship is designed to operate under the following conditions:

Vacuum = 28.5 in. (91.7°F)
w_a = 7.65 lb per shp-hr
h_2' at exhaust = 1,005 Btu per lb
Water velocity = 5 ft per sec
Sea temperature t_1 = 60°F
Overboard discharge 10° below vacuum temperature t_2 = 82°F
Rise in temperature of water through condensers $t_2 - t_1$ = 22°F
Condensate temperature = 88°F
U = 503 Btu per sq ft per deg difference per hr (see Art. 156)
Shp = 5,400

Under these conditions the mean temperature difference between water and steam is 18.8°F, the required condensing surface is 4,150 sq ft, and the required capacity of the circulating pump is 3,470 gpm.

Now consider the case where the sea temperature is 70°F.

$$
\begin{aligned}
t_1 &= 70°\text{F} \\
t_2 &= 92° - 10° = 82°\text{F} \\
t_2 - t_1 &= 12°\text{F} \\
t_m &= 14.85°\text{F (see Art. 156)} \\
\text{Cooling water required} &= 3{,}470 \text{ gpm} \times {}^{22}\!/_{12} = 6{,}360 \text{ gpm} \\
\text{Cooling surface required} &= 5{,}230 \text{ sq ft}
\end{aligned}
$$

To maintain a vacuum of 28.5 in. with a sea-water temperature of 70° calls for an increase in condenser cooling surface of 26 per cent and an increase in capacity of the circulating pump of 83 per cent over those required with sea water at 60°. Since the water velocity through the tubes has been assumed the same for both cases, the increase in the horsepower of the circulating pump will be 83 per cent.

If the water velocity were increased in the second case, U would be increased, resulting in a smaller increase in cooling surface. The quantity of water required would be the same as before, but the increased resistance due to the greater water velocity would call for a larger pump horsepower than before.

Whether a vacuum of 28.5 in. or a lower vacuum should be carried with a sea-water temperature of over 70° can be determined only after the annual fuel costs, annual fixed charges on the condenser and circulating pump, and the operating costs of the circulating pumps have been compared for the two cases. Theoretically, at least, a decrease in the vacuum would call for a slight increase in the cost of boiler and turbine because of the higher steam consumption.

134. Scoops.—Scoops for supplying the condenser cooling water reduce the cost, weight, and space occupied by the machinery through the elimination of the circulating pump. When scoops are used, there is a small increase in the ehp of the ship through the added resistance of forcing water through the condenser; hence the power for pumping the water through the condenser is supplied by the main engine through the propeller. The efficiencies of the propeller and the circulating pump are approximately the same, but the water rate of the main engine is less than that of the engine required to drive the circulating pump. How much difference there is in fuel per shp-hour depends on whether the pump is driven directly by a turbine or by an electric motor receiving current from an auxiliary generating set. Scoops are universally used on destroyers and cruisers and have been fitted on a number of American passenger ships (see Fig. 59).

When scoops are fitted, a small circulating pump must be supplied for low-speed operation, for reversing, and for use in port. Figure 111[1] shows a single-pass condenser installation

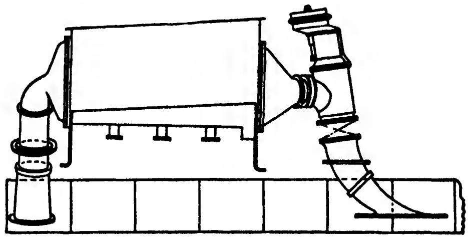


FIG. 111.—Condenser fitted with scoop and auxiliary circulating pump. (*Jour. Am. Soc. Naval Eng.*)

with a scoop. The turbine-driven auxiliary circulating pump shown at the right is of the propeller type, the impeller being installed in the injection piping. The impeller of the pump is allowed to rotate freely when the circulating water is being supplied by the scoop.

135. Auxiliary Condenser.—When steam auxiliaries and steam-deck winches are fitted, the main function of the auxiliary condenser is to take care of the exhaust steam from the cargo winches in port. As a rule, the vacuum carried in the auxiliary condenser in port is low. The winches exhaust at around atmospheric pressure, and hence the function of the condenser is to recover the condensate rather than to lower the back pressure. At sea, any excess of auxiliary exhaust steam that cannot be used for feed heating is often sent to the auxiliary condenser. Figure 112 shows an auxiliary condenser of the type usually fitted when steam engine-room auxiliaries and steam deck winches are used. This condenser is provided with a combined steam-driven circulating and air pump. The latter pump removes both condensate and air.

When the auxiliaries are driven by electric motors, as is the practice in most of the modern American ships fitted with steam turbines, the electric power is supplied by one or more turbine-

[1] Taken from a paper by H. J. Hanzlik, "Condenser Scoops in Marine Installations," *Jour. Am. Soc. Naval Eng.*, 1931, p. 250.

driven generators running parallel with the main unit, *i.e.*, using steam at the same initial pressure and temperature and exhausting at the same back pressure. Under these conditions, one or more auxiliary condensers carrying high vacua are fitted, and each is equipped with a separate circulating pump, condensate pump, and air ejector. When under way at sea, however, the main circulating pump and main air ejector are frequently used to serve the auxiliary condenser.

When motor-driven engine-room auxiliaries are used, the cargo winches are practically always motor-driven; so the auxiliary condenser performs the same function in port as at sea.

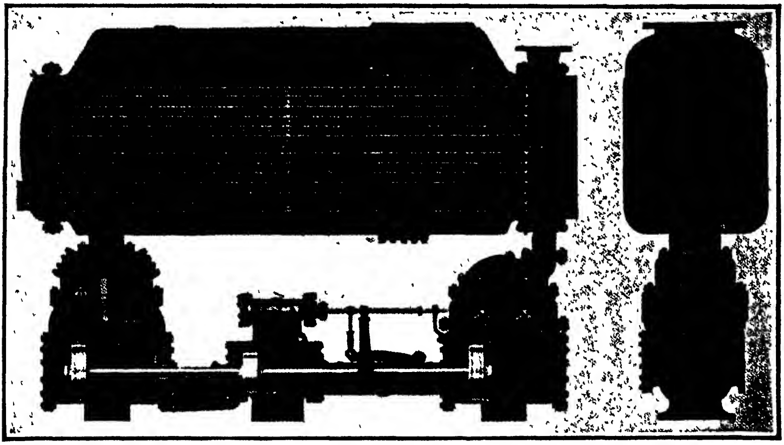


Fig. 112.—Auxiliary condenser with combined air and circulating pumps.

136. Air Pumps.—The function of the air pump is to remove the air and condensed steam from the condenser so that the condenser can continuously produce the desired vacuum. With a low vacuum, such as carried with reciprocating engines, a single "wet" air pump is sufficient to take care of the condensed steam and air leakage. When a vacuum of 28 to 29 in. is carried, the air pump becomes one of the most important auxiliaries. The higher the vacuum, naturally, the lower is the pressure in the condenser and the greater is the volume occupied by a given weight of air. Hence an air pump of large capacity is required at high vacua to take care of this large volume of air. As already pointed out, the amount of air to be removed depends not on the size of the plant or the amount of steam condensed but on the amount of air leakage into the system, although the larger the installation the greater is the tendency for air leakage.

By Dalton's law, it will be remembered, the total pressure within the condenser is the sum of the air pressure and the steam pressure. Table XXIII shows that at the entrance to the condenser the partial pressure of the air is very small, resulting in a very large volume occupied by the weight of air present—a volume altogether too large to be handled by an air pump. As the steam is condensed in its passage downward through the condenser, the volume occupied by a given weight of steam becomes smaller and smaller. Hence the ratio of air to steam by weight becomes larger and larger. This causes an increase in the partial pressure of the air and a decrease in the partial pressure of the steam. As already pointed out in the discussion of condensers, the reduction in the partial pressure of the steam results in a reduction of the temperature in the condenser. The increase in the partial pressure of the air and the decrease in the temperature cause compression of the air present in the condenser to a much smaller volume. Table XXIII shows this clearly. In this case, the temperature of air-pump suction has been depressed 20°, which is rather higher than necessary.

If the air leakage in a given installation increases and the air pump cannot handle the additional leakage, the partial pressure of the air will rise, causing the total pressure in the condenser to increase (and the vacuum to fall) until a balance is set up between the air pump and condenser. At times, under such conditions, the circulating pump is speeded up to decrease the temperature of the air and thus reduce its volume so that it can be handled by the air pump.

The air pump used with a reciprocating engine is usually of the reciprocating type. and is generally driven off one of the main crossheads by a lever (Fig. 113). This pump handles both the air and condensate.

With the introduction of the turbine and higher vacua, two reciprocating air pumps were employed in the early installations —a "dry" air pump to handle the air and a "wet" air pump to remove the condensate. This type of air pump has now been practically discarded in all new turbine ships. Steam-jet air pumps are used in most turbine installations for withdrawing the air from the condenser, and condensate pumps (usually of the centrifugal type) are used to remove the condensate from the bottom of the condenser. This type of equipment allows a higher vacuum to be attained, makes more economical use of the steam, and is lighter and more compact than the old twin reciprocating air pumps formerly used.

FIG. 113.—Engine-driven air pump and Weir condenser. Reciprocating engine installation.

137. Air Ejectors.—All modern ships propelled by geared turbines now use steam air ejectors in place of the reciprocating air pump used with reciprocating engines at one time employed with turbine installations. Steam air ejectors convert the potential heat energy of the steam into kinetic energy by means of steam nozzles. The design of the ejector is such that the high-velocity steam leaving the nozzles withdraws the air from the condenser, entrains it, and discharges it into a diffuser, where its pressure is raised.

Most air ejectors are of the two-stage type, with two sets of steam jets acting in series. The steam is condensed between stages in an intercondenser and after the second stage in an aftercondenser. The boiler feed water is now almost universally employed as the cooling water for the ejector condensers (Fig. 131). This allows most of the heat in the ejector steam to be recovered and sent to the boiler. By using a two-stage air ejector to raise the air pressure to that of the atmosphere and by condensing the steam between the stages the second stage handles only air, and the steam consumption of the ejector is only about half what would be required if it was attempted to raise the

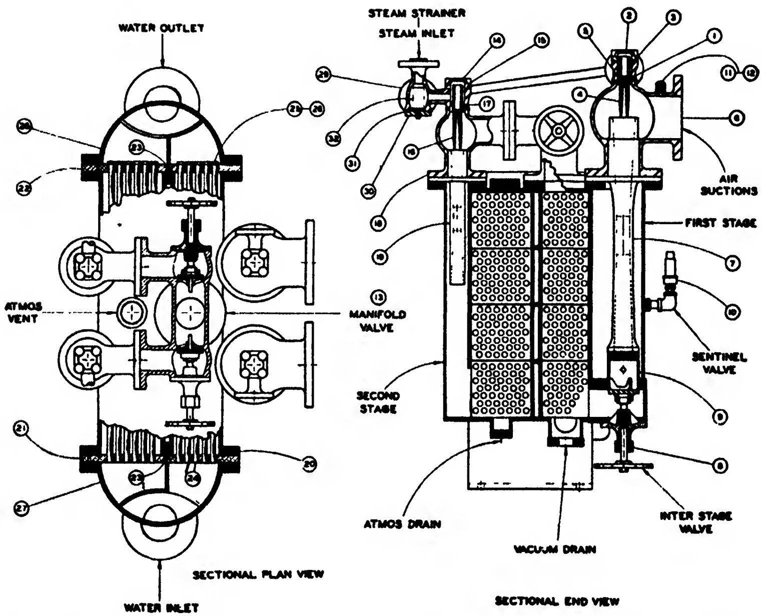


Fig. 114.—C. H. Wheeler air ejector with intercondenser and aftercondenser.

air pressure to that of the atmosphere in one step. Some British ships are fitted with three-stage air ejectors (Fig. 115).

There are a number of different makes of air ejectors on the market. Figure 114 shows a drawing of a C. H. Wheeler air ejector with intercondenser and aftercondenser. In the ejector shown in Fig. 114, live steam at reduced pressure is supplied to the steam strainer (29). From the strainer, the steam goes directly to the nozzles (4) of the two first-stage units and to the second-stage nozzles (16). The steam expands in the nozzles, leaving with a very high velocity. The steam in the first-stage nozzle, in passing across the suction chamber (6), entrains the air-vapor mixture coming from the condenser.

The mixture of air and steam passes into the diffuser (7). Here the kinetic energy of the mixture is reconverted into pressure energy, and the steam-air mixture is discharged at a higher pressure into the surface intercondenser (24). The steam used in the first-stage nozzle is condensed in the intercondenser. The vacuum in the intercondenser is from 3 to 5 in. below that in the

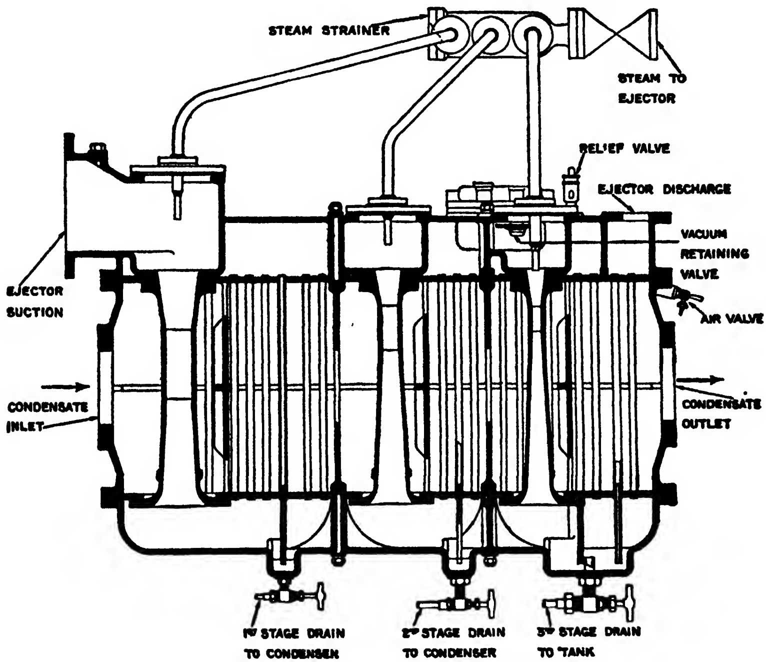


FIG. 115.—Weir three-stage air ejector with two intercondensers and aftercondenser.

main condenser, and the condensate from the intercondenser drains to the main condenser by means of a loop seal. The air left in the intercondenser passes out through the top of the condenser to the suction chambers (18) of the two second-stage ejectors. Here it is entrained by the steam leaving the second-stage steam nozzle (16) and passes to the diffuser, where the pressure is raised to that of the atmosphere. The steam and air now pass to the aftercondenser. Here the steam is condensed and drains out through the bottom by gravity to the atmospheric drain tank or passes through a drain trap to the main-condenser hot well. The air and a small amount of vapor escape to the atmosphere through an opening in the top of the aftercondenser. The condensate from the condenser to which the ejector is attached is circulated by the condensate pump through the tubes of the inter- and aftercondenser of the air ejector to condense the steam. A high percentage of the enthalpy of the steam used for operating the air ejector is thereby recovered. In Fig. 114, interstage valves (8) and manifold valve (13) permit isolating any one of the four ejectors for inspection and repair without interfering with the operation of the other elements of the unit.

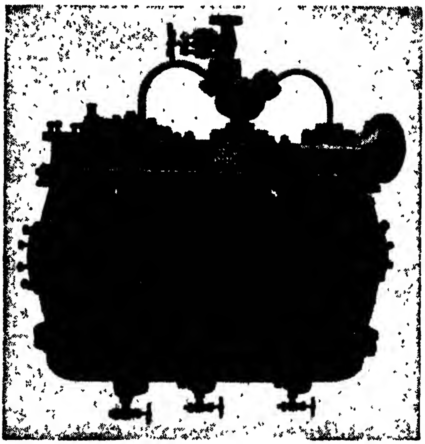


FIG. 116.—Weir three-stage air ejector.

The tubes used in the air-ejector condensers are usually either $\frac{5}{8}$ or $\frac{3}{4}$ in. outside diameter and are either expanded into both tube plates or expanded into one tube plate and made tight with packing and ferrules in the other tube sheet.

Figures 115 and 116 show a cross-sectional diagram and a photograph of a Weir three-stage air ejector with two intercondensers and an aftercondenser. It will be noted that the condensate is circulated around the outside of the tubes and diffuser nozzles, the steam-air mixture passing through the tubes of the condensers. The steam consumption of a three-stage air ejector is less than that of a two-stage air ejector and is especially suitable when vacua of over 28.5 in. are to be carried.

The air ejectors shown in Figs. 114 and 115 have the inter- and aftercondensers in the same shell. In some installations, the aftercondenser is separated from the intercondenser, and at times other heaters are placed in the feed circuit between the intercondenser and aftercondenser (see Fig. 130).

Figure 117 shows a Westinghouse air ejector with combined inter- and aftercondenser. It is interesting to observe that the condenser for the gland ejector (Art. 148) is included in the same shell as the air-ejector condensers. The various parts of the ejectors and condensers are clearly marked in Fig. 117, and a study of this drawing will make clear the operation of this piece of apparatus.

138. Quantity of Air Leakage.—The amount of air entering a condenser depends upon the air leakage and the amount of air entering the boiler in the feed water. It is not necessarily a function of the horsepower or the quantity of steam used.

The amount of dry air to be expected in a condenser can be estimated from the following authorities:

Orrok[1] gives the following figures from experiments on turbine units of 5,000 to 20,000 kw:

1 cu ft of free air per min (4.5 lb per hr) under best conditions
15-20 cu ft of free air per min (68–91 lb per hr) ordinary leakage
30–50 cu ft of free air per min (136–230 lb per hr) bad leakage

Hodgkinson[2] states that 3.5 cu ft of free air per min is considered good practice with a 40,000-shp turbine unit, and 7 cu ft is excessive.

Thaeler and MacMillan[3] give the following formula, based on a curve in Sterling's "Marine Engineers' Handbook":

$$A = 7.5 + 0.00025C$$

where A = pounds of dry air to be removed per hour.
C = pounds of condensate per hour.

James Sim[4] gives the following formula of the British Electric and Allied Manufacturers Association:

$$A = 3 + 0.0005C$$

The foregoing values are for turbine installations. The air leakage with a reciprocating-engine installation will be between two and three times the amounts given above, but, since the

[1] *Trans. A.S.M.E.*, 1910.
[2] *Trans. Soc. Naval Arch. & Mar. Eng.*, 1918.
[3] *Trans. Soc. Naval Arch. & Mar. Eng.*, 1938.
[4] "Steam Condensing Plant," 1925.

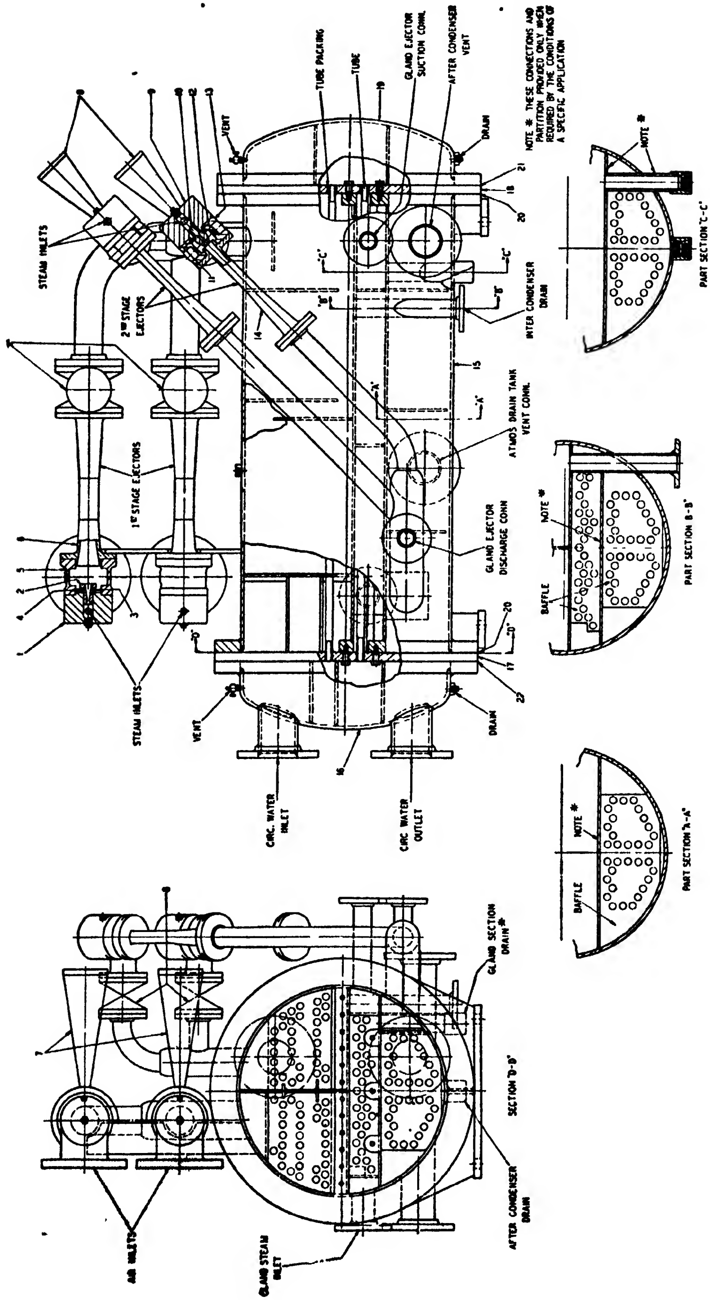


FIG. 117.—Westinghouse air ejector filled with combined intercondenser and aftercondenser.
(*For remainder of legend see opposite page*.)

vacuum with reciprocating engines seldom exceeds 26 or 27 in., the problem of air removal from the condenser is not so serious as with turbine installations.

In addition to the air, the water vapor associated with it must also be removed by the air pump or air ejector. The amount of water vapor per pound of air can be readily calculated by Dalton's law if the temperature is known.

Example:

Vacuum in bottom of condenser = 28.62 in.

$P_t = 1.38$ in.

Temperature of air-vapor mixture at air outlet = 75°F

P_s at 75°F (from steam tables) = 0.875 in. of mercury

$P_a = P_t - P_s = 1.38 - 0.875 = 0.505$ in. = 0.248 lb per sq in.

$$V_a = \frac{53.3 \times (460 + 75)}{144 \times 0.248} = 800 \text{ cu ft per lb}$$

From the steam tables, the volume occupied by 1 lb of dry steam at 75°F is 740 cu ft. But since $V_a = V_s$, the weight of vapor associated with 1 lb of air is $^{800}/_{740} = 1.08$ lb.

Weight of air-vapor mixture per pound of air = 2.08 lb

---

1 = steam chest, first stage
2 = nozzle, first stage
3 = nozzle ring
5 = mixing chamber, first stage
6 = diffuser
7 = first stage discharge gate valve
8 = second stage air inlet gate valve
9 = steam chest, second stage
10 = nozzle, second stage
11 = nozzle ring
13 = mixing chamber, second stage
14 = diffuser
15 = intercondenser and aftercondenser shell
16 = inlet water box
17 = inlet end tube plate
18 = reverse end tube plate
19 = reverse water box
20 = gasket between tube plate and shell
21 = gasket between tube plate and reverse water box

CHAPTER XI

FEED-WATER HEATING AND FEED-WATER SYSTEMS

139. General.—The water pumped into a boiler to supply the water for producing steam is known as the "feed water." Practically all steam vessels navigating in salt water operate with a vacuum and use a surface condenser to save the condensate so that it can be returned to the boiler again as feed water. Thus the medium for operating the propelling engine is continuously recirculated through the system, and no new supply of feed water is necessary except to make up for the small losses. By this procedure a continuous supply of pure feed water is assured for the boiler, provided that there are no salt-water leaks into the system from the condenser.

Vessels operating on the American Great Lakes and other bodies of fresh water sometimes use jet condensers. When this practice is followed, the condensate is discharged overboard with the cooling water, and the feed water for supplying the boiler is picked up from overboard. This practice is usually satisfactory with Scotch boilers, but it is not to be recommended where high-pressure water-tube boilers are used (see Art. 39).

All ships carry a certain amount of water in their double-bottom tanks to supply the losses from the system due to leakage, galley steam, use of whistle and soot blowers, etc. This water is known as "make-up feed."

Feed-water heaters are used on all steamships for heating the condensate before it is pumped into the boilers. This is done to improve the efficiency of the plant, as pointed out later, and also to avoid local stresses in the boiler that would be caused by the injection of comparatively cold feed water. Feed heating increases the capacity of a boiler as well as reducing the fuel consumption. The feed water can be heated by one of the three following methods or by a combination of two of them.

1. **Auxiliary exhaust steam.**
2. **Steam extracted from the main turbine.**
3. **Economizers installed in the gases beyond the boiler generating tubes.**

140. The Feed-water System.—The feed-water system in its simplest form consists of the condensate pump, which removes the condensate from the condenser, a feed and filter tank or surge tank, one or more heaters for heating the feed water, the boiler feed pump, and the necessary piping, fittings, valves, and drains.

A supply or reservoir of water in the feed system between prime mover and boiler is absolutely necessary in every feed system. Such a supply is that contained in the feed and filter tank (Fig. 118). When the turbine is running at constant power, the amount of water leaving and water entering the condenser will be constant; hence the condensate pump would deliver a constant amount of water per minute. The boiler feed pump, however, does not always operate at a constant rate, and hence some tank capacity must be installed between the two pumps to take care of the difference in rates of the two pumps.

When a boiler is generating steam, the water in the boiler is full of steam bubbles, which makes its density considerably less than that of hot water at just boiling temperature. The density ratio *i.e.*, the ratio of the weight of 1 cu ft of water in the boiler under full-speed operating conditions to the weight of 1 cu ft of water at boiler pressure and zero evaporation (as obtained from the steam tables), varies with the rate of evaporation per square foot of heating surface. The greater the rate of evaporation the lower is the density ratio. Consider the extreme case in which the turbine is suddenly shut down while running at full power. The rate of burning oil will be greatly reduced and the rate of evaporation will drop to that required to supply the small amount of steam required by the auxiliaries, and the boiler water level, which was visible in the gage glass, rapidly drops because the amount of steam bubbles in the water decreases, *i.e.*, the density of the water increases. As soon as the turbine stops, the condensate pump loses its supply of water and hence can furnish no more for boiler feed until the turbine starts. The boiler feed pump, however, must be run at full capacity until the boiler water level has been brought back again to its proper level in the gage glass. To accomplish this, the boiler feed pump draws on the supply of water in feed tank. When the turbine is started again and the burners are lighted, the steam bubbles again fill the water in the boiler, and the water level rises. Hence the boiler feed pump will not start until some of the water has been evaporated and the boiler water level has dropped slightly below its operating level. In the meantime, however, the condensate pump has been taking water from the condenser and delivering it to the feed tank, which brings the supply of water in this tank back to the level existing before the turbine was stopped. Variations in demands for steam will, of course, change the rate of evaporation and cause the water level in the feed tank to change. This simple description of boiler surge will show the need for a reserve capacity in the feed system and will make clear why the modern name "surge tank" has been substituted for the former name "feed tank."

The capacity of the surge tank for merchant ships varies from 6 to 10 min supply. The theoretical minimum required capacity can be readily calculated if the quantity of the water in the boilers and its density at maximum evaporation are known.

141. Feed Heating by Auxiliary Exhaust Steam.—Up to about 1925, the usual practice on shipboard was to drive the auxiliaries by steam that was exhausted at a pressure between 5 and 10 lb above atmospheric pressure. This auxiliary exhaust steam, which was at a temperature of 228 to 240°F and contained a great deal of latent heat, was used for heating the boiler feed water. Noncondensing auxiliaries are lower in first cost and require less attention than condensing auxiliaries. By using the heat in the exhaust steam from the auxiliaries, there is a direct gain, since heat that would otherwise be thrown away in the condenser is recovered and sent back to the boiler in the feed water. Live steam has been used in the past for heating, but this practice has now been almost universally discarded.

The usual practice when the auxiliary exhaust was employed for feed heating was to use a single closed-type heater (Fig. 119), which, with the exhaust steam at 5 lb per sq in. gage pressure (228°F), would usually heat the feed water to around 218°F. A diagram of a plant using steam auxiliaries and auxiliary exhaust for heating the feed is shown in Fig. 118.

142. Feed Heating by Extracted Steam.—On many of the modern steamships, especially those built in the United States and propelled by geared turbines or turbo-electric drive, the auxiliaries are driven by electric motors. The current for the auxiliaries is supplied by one of the following methods:

1. Auxiliary generating sets driven by steam turbines operating under the same initial steam conditions and the same vacuum as the main turbine.
2. Auxiliary generating sets driven by Diesel engines.
3. An auxiliary generating set geared to the main propelling turbine.
4. A-c current supplied by the main propelling generator (turbo-electric drive installation only), as done on the *U.S.S. Pontchartrain* class (Art. 96).

Whichever of the preceding methods is used, only small amounts of auxiliary exhaust steam are available (from the air

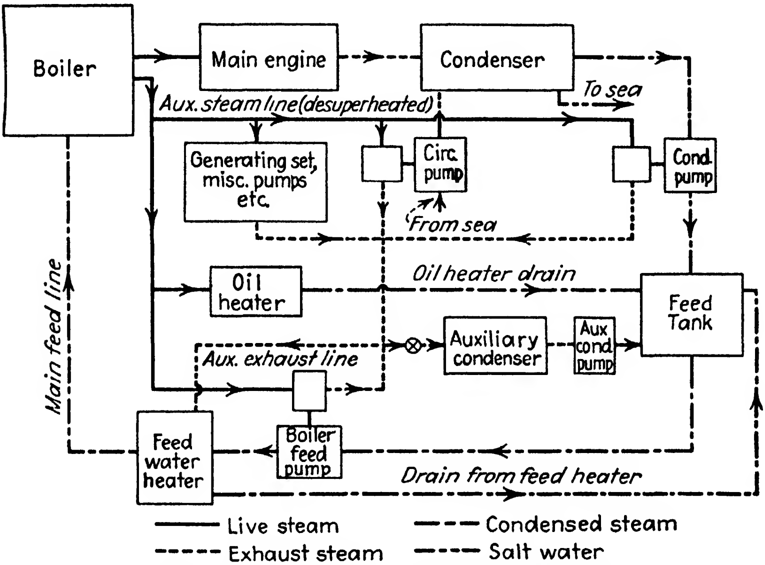


FIG. 118.—Diagram of power plant using steam auxiliaries. (In order to simplify the diagram, the air ejector and some other pieces of auxiliary apparatus have been omitted.

ejectors and miscellaneous small steam pumps, such as fuel oil transfer and bilge pumps) unless the feed pump is steam-driven. Of course, when motor-driven auxiliaries are employed, a turbine-driven auxiliary generating set could be operated noncondensing and thus furnish a supply of exhaust steam for feed heating. However, for reasons that will be made clear later, generating sets in such installations are practically never operated noncondensing at sea, although they may operate noncondensing in port to supply exhaust steam for feed heating.

With the introduction of motor-driven auxiliaries, the steam for heating the feed water was extracted or bled from the main turbine at the proper point in the expansion to give the desired steam temperature. Many installations are using series feed heating, two and three feed heaters being used to heat the feed water by stages from condenser temperature to around 300 to 350°F. The steam for each heater is extracted from a different point in the turbine.

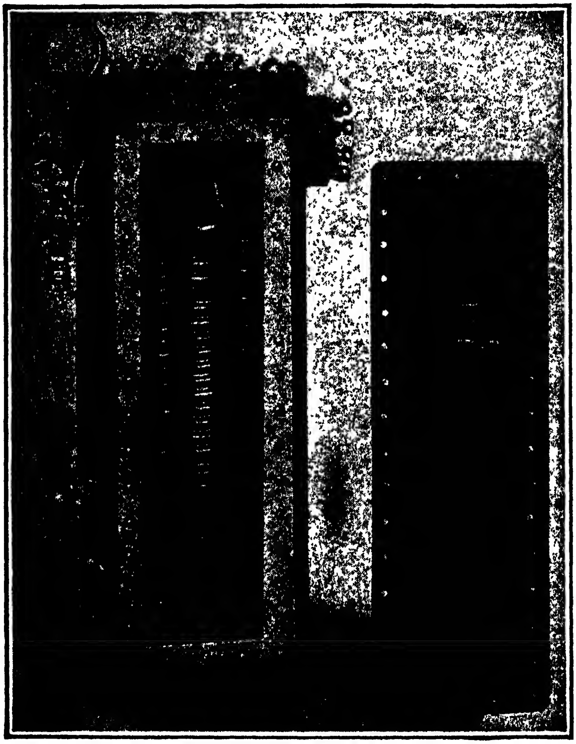


FIG. 119.—Reilly feed-water heater.

143. Feed-water Heaters.—Two types of feed-water heaters are in common use, the closed heater and the direct-contact heater, often called an "open" heater. In a closed heater (Figs. 120 and 121), the feed water and heating steam do not mix. The feed water passes through tubes in the heater, and the heating steam is admitted to the shell of the heater and surrounds the tubes. The closed heater acts in the same way as a surface condenser working without a vacuum. The feed water passing through the tubes condenses the steam outside the tubes and absorbs the enthalpy of evaporation h_{fg}; the condensed steam drains out of the bottom of the heater, carrying with it the enthalpy of the liquid h_f. When only one feed heater is used and it is of the closed type, it is placed between the boiler feed pump and the boiler (Fig. 118); hence its tubes are under full boiler pressure. When more than two feed heaters are used, closed heaters frequently are placed on the suction side of the boiler feed pump (see Figs. 128 and 130).

Feed heaters with coiled tubes, such as are shown in Fig. 119, have been widely used in the past in plants using low boiler pressures, especially when steam auxiliaries have been employed and the feed is heated by exhaust steam. In modern steam ships with high boiler pressures and hence high feed-water pressures, feed heaters with straight or U-shaped tubes are usually employed, such as are shown in Figs. 120 and 121.

Figure 120 illustrates a Bethlehem four-pass heater designed for use as the highest stage heater in a three-stage heater installation. In Figure 130 the third-stage heater is shown at the extreme left of the diagram between the boiler feed pump and the boiler. In this heater the tubes are arranged in two groups. The feed enters at the right-hand end of the heater, passes through one group of tubes, returns to the right-hand end, and then makes a second double pass through the second group of tubes. The heating steam that surrounds the U tubes enters the heater at the

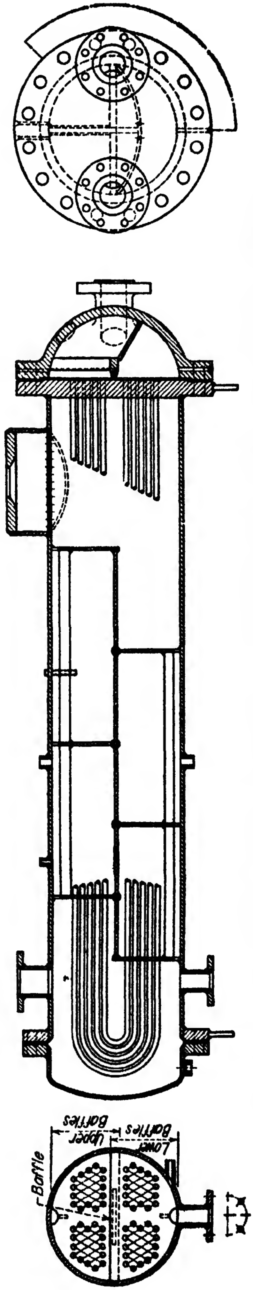


FIG. 120.—Four-pass closed feed-water heater. (*Bethlehem Steel Company, Shipbuilding Division.*)

top, and the condensed steam leaves through the drain at the bottom.

Figure 121 shows a diagrammatic sketch of a Griscom-Russell straight tube heater for low pressure service. This is also a four-pass heater. This heater is fitted with patented partition sealing plates in the water heads, which employ the difference in the pressure of the feed water at inlet and outlet to keep the gaskets of the partition plate tight. The floating head allows for the expansion of the tubes.

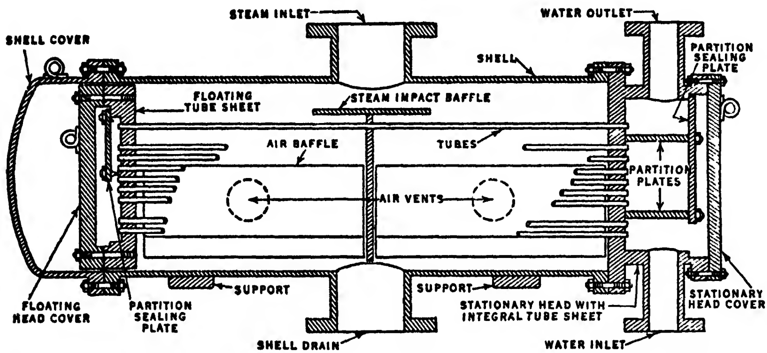


FIG. 121.—Diagrammatic sketch of a typical Griscom-Russell straight tube closed feed-water heater, showing partition sealing plate in water heads.

In the direct-contact heater (Fig. 124), as its name implies, the feed water and heating steam are mixed together, the heater acting much like a jet condenser. The water is pumped into the heater in the form of a spray and mixes with heating steam. The heating steam condenses and gives up its enthalpy of vaporization to the water. The feed water and condensed steam drop to the bottom of the heater, and the water is pumped off by the boiler feed pump. The steam pressure of the heating steam used in direct-contact heaters is seldom over 8 or 10 lb per sq in. gage.

Direct-contact (D.C.) heaters must always be placed on the suction side of the boiler feed pump. If the temperature of the feed-pump suction exceeds 170°F, there is danger of the pump becoming vapor-bound unless there is a positive head on the suction side of the pump. Hence, when the temperature in a direct-contact heater approaches or exceeds this value, it must be placed in an elevated position in the ship in order to give a positive hydraulic head on the boiler feed pump. In the U. S. Maritime Commission's ships, the direct-contact heaters are located in the upper engine-room casing above the main deck, around 30 ft above the tank top. If an elevated position is impossible, as in naval vessels, a booster pump must be used to deliver the water from the heater to the boiler-feed-pump suction (Fig. 131). Direct-contact heaters allow the feed water to be raised to a temperature that is practically that of the heating steam, and no drain connections are necessary. When closed heaters are employed, the feed leaves the heater at a temperature from 10 to 15° below that of the heating steam. In stationary plants, where more space is available for larger feed heaters, the feed water is sometimes heated to within 7° of the heating steam.

Direct-contact heaters occupy more space than closed heaters and because of the required elevated position are not so conveniently located. A customary practice today is to incorporate the boiler surge tank (feed tank) in the bottom of the direct-contact heater, which results in a unit of large size (see Fig. 123).

One of the main advantages of direct-contact heaters is their deaerating characteristics. It was pointed out in Art. 39 that boiler feed water must be free of dissolved oxygen and carbon dioxide in order to prevent corrosion in the boiler and economizer tubes. The feed water is sprayed into a direct-contact heater, where it mixes with steam which is at a pressure somewhat above atmospheric pressure. By this process the air in the water is liberated and escapes through the vent provided in the top of the heater. Deaerating direct-contact heaters, such as that shown in Fig. 122, have been found in service to deliver water to the boilers in a practically air-free condition, and they are being widely adopted on many modern ships.

A sectional view of a Cochrane marine deaerating heater manufactured by the Cochrane Corporation is shown in Fig. 122, and an illustration of one of these installed on a recently constructed ship is shown in Fig. 123. The shell is of welded rolled plate and is made large enough to provide sufficient water-storage capacity to permit it to serve also as a feed-water surge tank. The spray preheater shown in the upper part of Fig. 122 is of the direct-contact type and is designed for operation under low water pressures. The water is sprayed outward and upward in contact with steam supplied by the atomizer. The water preheated and partly deaerated by this process is collected and distributed uniformly over the atomizer shown in the center of the figure. The heating steam coming from the atomizer breaks the water up into a fine mist, thereby exposing a large aggregate liquid surface. The atomizer consists of a bronze

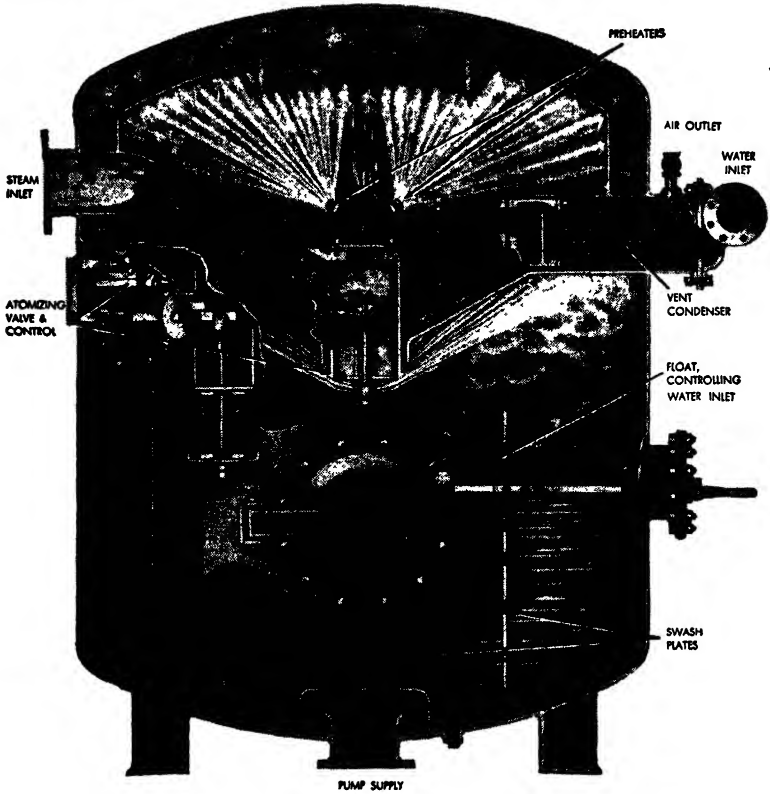


Fig. 122.—Cochrane direct-contact deaerating heater.

valve, ordinarily in the form of an inverted cap seating against the end of the downwardly projecting steam pipe. It is held on its seat by an external spring loading. The incoming steam discharges through the atomizer at practically constant differential pressure in a conical jet, and the flow of steam agitates and scrubs the water surfaces and carries away the liberated oxygen and other gases. With a differential pressure of ½ lb per sq in., the velocity of the steam jet exceeds 300 ft per sec, and its kinetic energy atomizes the water, creating a fog or mist. The deaerated water falls into the storage compartment at the bottom. The steam and liberated gases pass up through the preheater and then to a vent condenser, shown at the right-hand side of Fig. 122, where nearly all the remaining steam is condensed and the gases are discharged to the atmosphere. The feed water

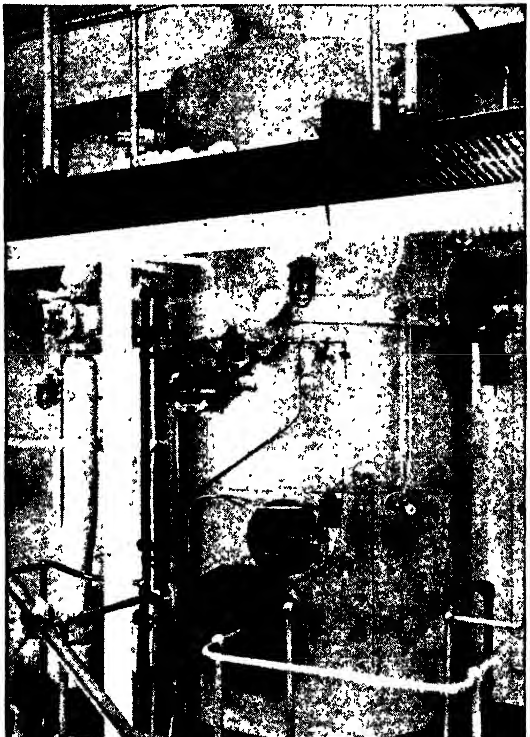


FIG. 123.—Cochrane deaerating heater installed on a modern ship.

is passed through the tubes of the vent condenser as the cooling medium. The vent condenser thus serves as a feed-water heater for the feed water as well as a condenser for the steam not condensed in the main part of the heater. The feed water is heated in the heater to a temperature corresponding to the steam pressure carried in the heater shell. Both the water sprayer and the atomizer of this heater operate at full efficiency, regardless of the motion, trim, or list of the ship.

Figure 124 shows a sectional view of a direct-contact feed heater manufactured by G. & J. Weir, of Glasgow.

R. C. Roe, in a paper before the Society of Naval Architects and Marine Engineers, in 1932,[1] sums up well the advantages

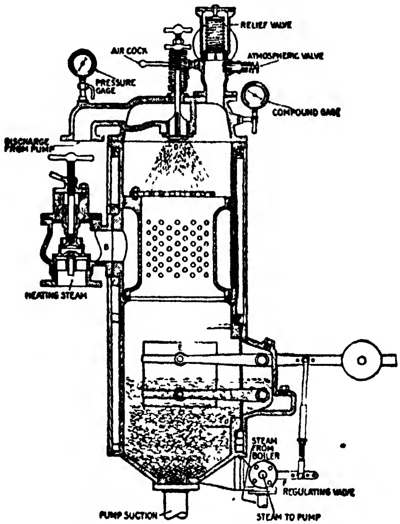


FIG. 124.—Weir direct-contact feed-water heater.

of using direct-contact heaters in place of closed heaters as follows:

> The ability to build heater types which would be unaffected by the motion of the ship, the absence of tubes and tube sheets to cause failure, the absence of drip pumps or drip cascading, the ability of the turbines to operate with or without bleeder heaters in service, with bleeder heater shells acting merely as part of the pipe line, the ability to get good deaeration and flexibility and the high efficiency of this system are points in its favor.

Besides economizers and the two types of feed heaters described above, other devices for heating the feed water are installed in the modern steamship, such as drain coolers, air-ejector condensers, and gland exhaust condensers. These are shown in Figs. 128 to 131 and are discussed later in detail.

144. Economizers.—An economizer is a feed heater placed in the path of the hot boiler gases, either in the boiler casing beyond

[1] "Modern Power Plant Practice and Its Application to Marine Work," p. 389.

the main boiler tubes or in the uptake outside the boiler proper. Economizers have long been used in power plants ashore, but they have been introduced on shipboard only in recent years. They can be used alone for heating the feed water or can be placed in series with one or more feed heaters that utilize steam as the heating medium. The boilers of many modern merchant ships are fitted with economizers alone or with both economizers and air heaters (see Figs. 14 and 16). Practically all marine installations that employ economizers use them in connection with one or more feed heaters and incorporate the economizers within the boiler casing, as shown in Figs. 14 and 16. In such

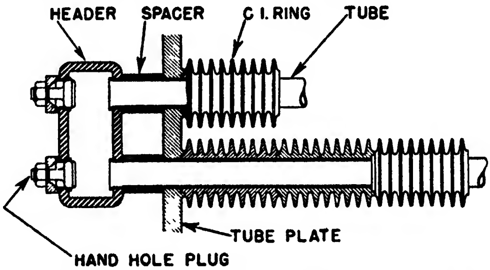


FIG. 125.—Section of Foster Wheeler economizer, showing cast-iron gilled rings.

cases, the amount of economizer surface installed is rather small, and the rise in feed temperature in the economizer is of the order of 75 to 100°F. Cases are on record where economizers have been installed to improve the economy of existing installations which had a high uptake temperature.

The Foster Wheeler economizer is probably the best known economizer in the United States. This design has an extended surface in the form of gilled rings fitted over the economizer tubes. These gilled rings greatly increase the heating surface for absorbing heat from the gases and also protect the economizer tubes from the corrosive action of the gases. A section of a Foster Wheeler economizer with cast-iron rings is shown in Fig. 125. The gilled rings are made of either cast iron or aluminum. The former are used mostly on merchant vessels, where low first cost is essential, and the latter where a low machinery weight is desired.

145. The Regenerative Cycle.—In the ideal regenerative cycle, heat is withdrawn from the prime mover at an infinite number of points between the throttle and condenser, and this heat is transferred to the main condensate to raise its temperature to that corresponding to the boiler pressure. If the steam at the beginning of the expansion is dry and saturated or wet, such a cycle would have an efficiency as high as that of the Carnot cycle. The proof of this statement can be shown with the aid of the temperature entropy diagram (Fig. 126).

Let the steam at the beginning of the expansion be dry and saturated (point 1 in Fig. 126). The heat withdrawn from the turbine along line 1-2 (represented by area 8-2-1-9 in Fig. 126) is just sufficient to heat the feed water from condenser temperature up to the boiling point; hence area 8-2-1-9 must be equal to area 6-3-4-7, and line 1-2 must be parallel to the liquid line 3-4. The boiler adds only the enthalpy of evaporation, namely, the area 7-4-1-9.

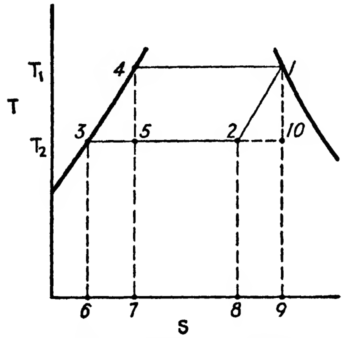


FIG. 126.—Temperature-entropy diagram of the regenerative cycle.

The work done in the regenerative cycle is the area 1-2-3-4 and the heat rejected in the condenser is the area 3-2-8-6. Since area 3-4-5 equals area 2-1-10, the work done (area 1-2-3-4) is equal to the area 5-4-1-10. Now the area 5-4-1-10 is the work done in the Carnot cycle working between the same absolute temperature limits T_1 and T_2; hence the ideal regenerative cycle has the same efficiency as the Carnot cycle.

It will be noted from Fig. 126 that the area 2-1-10 represents that part of the work done in the Rankine cycle that does not appear as work in the regenerative cycle. By using this heat 2-1-10 to heat the feed water (3-4-5) instead of converting it into useful work, the heat 2-10-9-8 formerly lost in the condenser is now saved, and the heat represented by the total area 8-2-1-9 is returned to the boiler in the feed water (6-3-4-7).

In addition to the practical difficulties encountered in such a cycle, another serious objection would be the low quality of the steam during the last stages of expansion near point 2. Instead of transferring the heat from all the steam by using an infinite number of heaters, it is customary in actual practice to heat the feed water by means of small quantities of steam extracted at one to three points during the expansion. The major part of the steam is permitted to expand in the turbine and do work without excessive condensation.

146. Use of Extracted Steam for Heating the Feed Water.—As pointed out in Art. 145, practical difficulties prevent the use of the ideal regenerative cycle. However, the principle is made use of in the modern power plant by extracting small amounts of steam from the main turbine at two or three different points and sending this steam to feed heaters to heat the feed water in a series of steps from the temperature of the condensate leaving the condenser up to a temperature of around 300 to 350°F. As will be shown later, the amount of steam bled from the turbine at each point is small, and the bulk of the steam passes through the turbine to the condenser.

Steam that has been expanded to 28½ in. vacuum (91.7°F) is at too low a level to be used for any useful purpose on shipboard, and hence all its enthalpy of evaporation is lost. However, by withdrawing 1 lb of steam from the turbine at a higher temperature (say, at 228°F), all the enthalpy of evaporation can be made use of for feed heating.

The following simple example will make clear the advantage of extracting steam from the main turbine for feed heating.

Assume a turbine operating under the following conditions:

Throttle pressure P_1 = 375 lb abs
Superheat at throttle = 250°F
Vacuum in condenser P_2 = 28½ in. (91.7°F)
Condensate temperature = 87°F
Steam extracted for feed heating at P_e = 20 lb abs (228°F)
Engine efficiency (excluding gears) = 70 per cent

For convenience we shall assume that the fractional efficiencies of the turbine between P_1 and P_e and between P_e and P_2 are each 70 per cent:

$$h_1 = 1357, \text{ from the Mollier diagram}$$

After an isentropic expansion, $h_e = 1095$ and $h_2 = 902$.
The actual enthalpies are

$$h'_e = 1357 - [0.70(1357 - 1095)] = 1174 \text{ Btu per lb}$$
$$h'_2 = 1357 - [0.70(1357 - 902)] = 1039 \text{ Btu per lb}$$

If the steam expands in the turbine from 20 lb abs to 28½ in. of mercury, the useful work done by each pound is 1174 − 1039 = 135 Btu. In addition, 55 Btu per lb is recovered in the condensate leaving the condenser and returned to the boiler. Thus the total enthalpy made use of per pound of steam is 135 + 55 = 190 Btu. The heat lost in the condenser is 1039 − 55 = 984 Btu per lb.

Now, if 1 lb of steam is withdrawn from the turbine at the point in the expansion where the pressure has reached 20 lb abs and is used in a direct-contact heater to heat the feed water, 1174 Btu is returned to the boiler. Thus the 190 Btu mentioned above plus that formerly lost in the condenser (984 Btu) are returned to the boiler. The percentage of the enthalpy converted into work in the turbine by not extracting is 190/1,174 = 16.2 per cent of that used when it is extracted for feed heating.

Unfortunately, only a very small percentage of the steam passing through the turbine is needed to heat the feed water up to 228°F, and most of the steam passes along through the turbine to the condenser. If steam extracted from the turbine can be used for heating the ship, obviously all its enthalpy will also be made use of.

When steam auxiliaries are employed, the auxiliary exhaust (at about 20 lb abs) is used for feed heating. Usually there is more auxiliary exhaust steam than needed for this purpose, and in such cases the excess of auxiliary exhaust is frequently sent to the low pressure stages of the turbine instead of to the auxiliary condenser. This, of course, is the direct opposite of extracting steam. However, this procedure is better than sending the excess of auxiliary exhaust to the auxiliary condenser. In the latter case, all the enthalpy in the steam is lost except that of the condensate, whereas in the former case, about 16 per cent of it is recovered in useful work in the turbine.

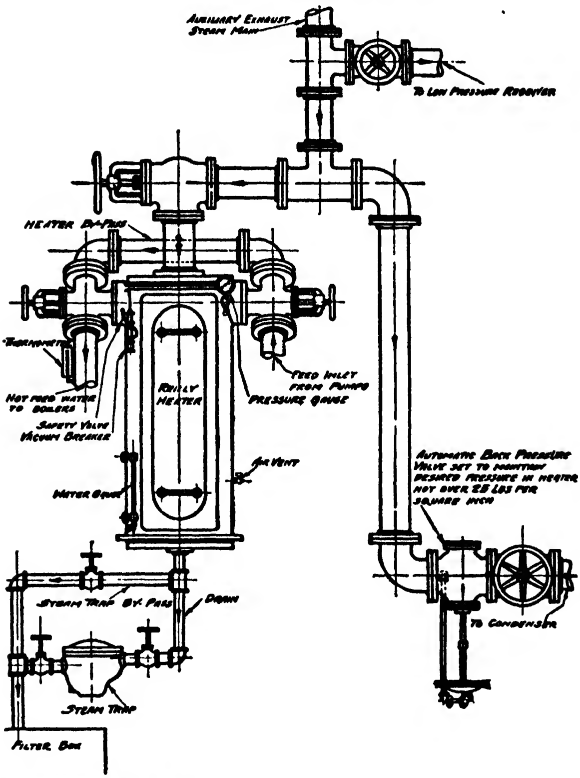
In addition to the advantage discussed above, the use of extracted steam for feed heating reduces the size of the condenser and the size and power of the condenser auxiliaries, decreases the size of the blading in the low pressure end of the turbine, increases the quantity of steam flowing through the high pressure blading (and hence improves slightly the efficiency of the high pressure stages, especially in pure reaction turbines), and, by improving the over-all plant efficiency, reduces the size of the boiler slightly.

Marine feed systems that use extracted steam for heating the feed water have from one to three heaters in series. Economizers are often used in series with one or more heaters, but economizers are practically never used alone in marine installations as the only means of heating the feed water.

In designing an installation for using bled steam for feed heating with several heaters, the definite steam temperatures desired at each heater are established, and the turbine is tapped at the proper points in the steam expansion to give steam pressures corresponding to the desired temperatures. If the bleeder points are established for full power conditions, when the ship is slowed down and the pressure drop through the turbine is changed, the steam temperatures are changed and are incorrect to give the proper operation of the feed-heating system. The falling off of the boiler feed temperature is usually not serious, however, unless the power is reduced below 50 per cent.

147. Feed System Using Auxiliary Exhaust Steam.—Figure 127 shows an installation of a two-pass closed feed-water heater as fitted in a plant using steam auxiliaries. Such a plant is shown in Fig. 118. The auxiliary exhaust line is connected to the shell of the heater and also through an automatic back pressure valve to the auxiliary or main condenser. If the supply of exhaust steam is greater than needed for feed heating, the back pressure valve will open and discharge steam to the auxiliary or main condenser as soon as the pressure builds up in the exhaust line. The condensate from the exhaust steam is drained from the shell of the feed heater to the filter end of the feed and filter tank. The drain to the feed and filter tank either is direct or is through a trap. In the former case, a small amount of water is kept in the bottom of the heater as a water seal. A gage glass is fitted near the bottom of the heater so that the water level can be observed, and the correct level is maintained by regulating a valve in the drain line. Air vents are fitted in the heater just above the water surface so that the air that comes in with the exhaust steam can be blown off. This air collects in the lower part of the heater, and unless it is periodically blown off the heater will fill up with air and fail to function. An accumulation of air in the heater will cut down the effective tube area, resulting in a lowering of the temperature of the feed water leaving the heater.

Heaters are sometimes connected with the automatic back pressure valve on the discharge side of the heater. All the steam in the exhaust line passes through the heater, but if there is an excess of heating steam part of it will be discharged to the auxiliary condenser instead of being condensed in the heater.



BY PERMISSION GRISCOM-RUSSELL CO.

FIG. 127.—Diagram of two-pass induction type installation.

The thoroughfare type has the advantage that there is no tendency for the feed heater to become air-bound, as might be the case with the induction type if the air vent was not given proper attention.

148. The Closed Feed System.—The feed system shown in Fig. 118 has an open feed and filter tank into which the condensate is pumped from the main condenser and which receives the drains from feed heater, evaporator, oil heater, and other atmospheric drains. In some of the earlier turbine installations, where both condensate pumps and air ejectors without after-condensers were used, the steam and air leaving the air ejector were often discharged through a perforated pipe submerged in the feed tank. Obviously, the water in this feed tank contained a considerable amount of air in solution, and since the tank was open to the atmosphere the water could readily absorb air.

An open feed system of this type has proved satisfactory with Scotch boilers and was used to a considerable extent with early water-tube boiler installations, where comparatively low steam pressures were the rule. When water-tube boilers of small tube diameter using high steam pressures and temperatures were introduced (especially when economizers were installed), it was necessary to reduce greatly the air entering the boiler in the feed to eliminate tube corrosion. A feed-tank arrangement such as is shown in Fig. 118 allowed too much air to enter the boilers.

To obviate this difficulty, various types of closed feed systems have been introduced in which the condensate was freed of as much air as possible in the condenser (and sometimes again in a deaerating heater) and was not exposed again to the atmosphere on its path from condenser to boiler. One of the earliest of these feed-water systems was the Weir closed feed system. The diagram of the *S.S. Viceroy of India's*[1] installation shown in Fig. 128 is of a Weir system. In the Weir system, the lower part of the condenser shell serves as a feed tank (surge tank), and the condensate pump (water-extraction pump) delivers directly to the boiler feed pump in a closed circuit. An open reserve feed tank is used in the system to receive the atmospheric drains and drains from some of the heaters. The feed-water supply in the condenser shell is under full vacuum, and since the temperature of this water is maintained very close to the temperature corresponding to the condenser pressure, the amount of oxygen in the water will be very small. When the water level in the condenser falls, the float-operated valve (see Fig. 128) opens, and some of the water that has been accumulating in the reserve feed tank from the drains is drawn up by the vacuum into the condenser. This water is sprayed into the top of the condenser and hence deaerated as it enters. The circuit of the feed water in Fig. 128 and the various heaters and connections will be more

[1] Taken from *The Marine Engineer & Motorship Builder*, vol. 52, p. 135.

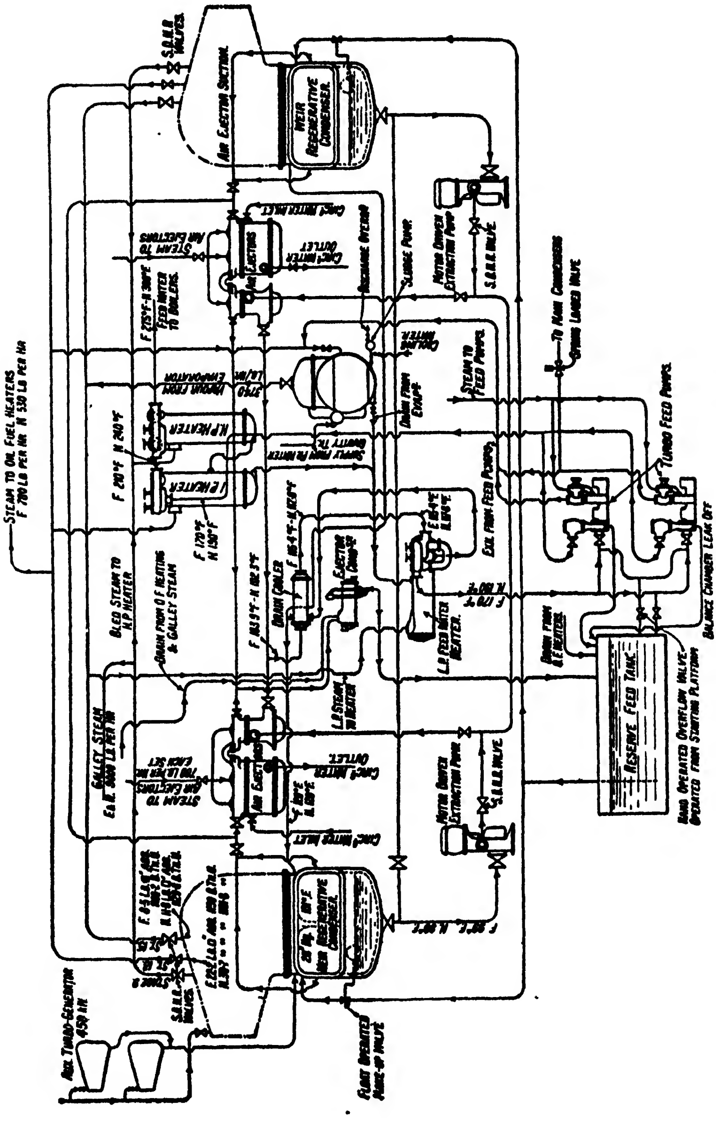


FIG. 128.—Diagram of Weir closed feed system on the S.S. *Viceroy of India*. (*The Marine Engineer & Motorship Builder*.)

easily followed after a study of the following articles. Extracted steam with three heaters is used for feed heating in the installation shown in Fig. 128.

Practically all modern feed-water systems are of the closed type, in contrast to the "open" system shown in Fig. 118, although they are not always referred to as such. Most of the systems described in the following articles are closed, in that the feed is not exposed to the atmosphere after leaving the condenser.

149. Modern Feed-water Systems.—In modern steamships, where an effort is made to secure a low fuel consumption, all the possible sources of heat are made use of for heating the feed water. Three feed-water heaters in series utilizing steam bled from the main turbine at three different pressures are now frequently installed. The auxiliaries are driven by electric motors.

In a typical modern installation, the main condensate from the condenser is sent through the coils of the inter- and after-condenser of the main air ejector and thence through a drain cooler and the feed heaters. The feed water in its passage through the ejector condensers acts as the circulating water for these condensers and absorbs the enthalpy of vaporization of the ejector steam. The feed temperature is raised 3 to 10° in its passage through these two condensers. If a closed feed-water heater is used operating with low pressure steam or auxiliary exhaust, the drain from the heater is sent to a drain cooler, and the feed water is passed through the coils of this cooler feed heater, where its temperature is raised 5 to 10°. Drain coolers operating in conjunction with low pressure feed heaters are shown in Figs. 128 and 130. In each of these cases, the pressure in the heater that drains to the cooler is below atmospheric, and hence the drain from the drain cooler is led to the condenser. Drain coolers can be used in connection with a three-heater installation or with a single closed-heater installation, where the heating steam is either auxiliary exhaust or steam-extracted from the turbine. A simple feed system with one feed heater, working as described above, is shown in Fig. 129. It will be noted that this is a closed feed system, and, to give a reservoir of feed water similar to that formerly contained in the feed and filter tank, a surge tank is installed in the line between the condenser and boiler feed pump. The feed water on its way to the boiler does not pass through this surge tank. The surge tank is connected to the feed line and acts as a reservoir supply when needed. Surge tanks sometimes are placed directly in the feed line, and all the feed water passes through the tank.

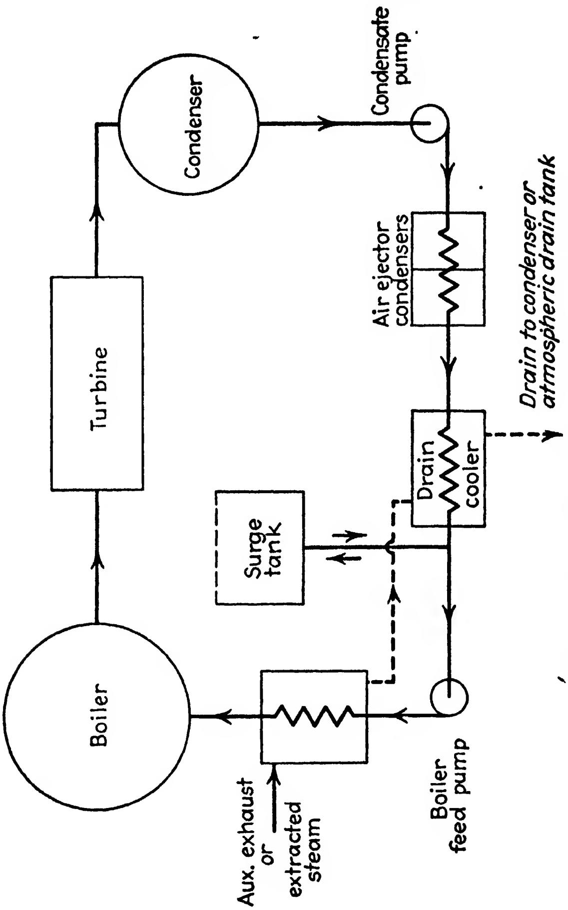


FIG. 129.—Diagram of feed system with open surge tank.

The tank is then given an elevated position and is usually vented to a low pressure feed heater or to the condenser.[1]

If three feed heaters are used, the feed water, after leaving the drain cooler, passes in series through the three heaters on its

[1] For descriptions and diagrams of various types of feed-heating systems and surge tanks, see the paper by John E. Burkhardt, "The Present Trend in Marine Engineering in the United States," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1938.

way to the boiler (Figs. 128 and 130). Thus the feed water that left the main condenser at a temperature around 85 to 90°F (for a 28.5-in. vacuum) is successively raised in temperature until it leaves the high pressure heater at a temperature between 300 and 350°F. Economizers are sometimes fitted to raise the feed to a higher temperature before it enters the boiler drum. Gland leak-off condensers (Figs. 130 and 131) are sometimes used in the circuit, and in some cases the inter- and aftercondensers are placed in separate shells, and the drain cooler is placed between them (Fig. 130). This gives a slightly more efficient use of the heating mediums but also a slightly more complicated installation.

When three feed heaters are employed, the steam pressure in the low pressure heater is usually below that of the atmosphere (around 8 to 11 lb absolute). When the other feed heaters are of the closed type, the drains are usually led to the next lowest pressure heater. Since the feed and heating steam mix together in a direct-contact heater, no drain is required for this type of heater. In shore power plants, drain pumps are sometimes used to pump the drain from a heater into the main feed line leaving the heater in question. This complicates the installation but avoids dropping the temperature of the drains ("cascading") to a lower level, with its consequent loss in heat availability. One of the advantages of the direct-contact heaters is that no cascading of drains is necessary.

The extracted steam used in the highest pressure feed heater is usually superheated as it leaves the turbine, but the feed can be heated only in a closed heater to within 10 or 15° of the temperature corresponding to dry steam at the extraction pressure. This causes a slight loss in heat availability, and to secure the full gain from this high-temperature steam, desuperheating heaters are sometimes installed in stationary plants beyond the high pressure heater. The extracted steam passes first through this desuperheating heater before going to the high pressure feed heater. A slightly higher feed temperature is thus secured without any increase in the amount of extracted steam. The Griscom-Russell closed feed heaters used with superheated steam have a patented baffle arrangement that encloses the last water pass of the heater so that the incoming steam first comes in contact with the outgoing end of feed-water

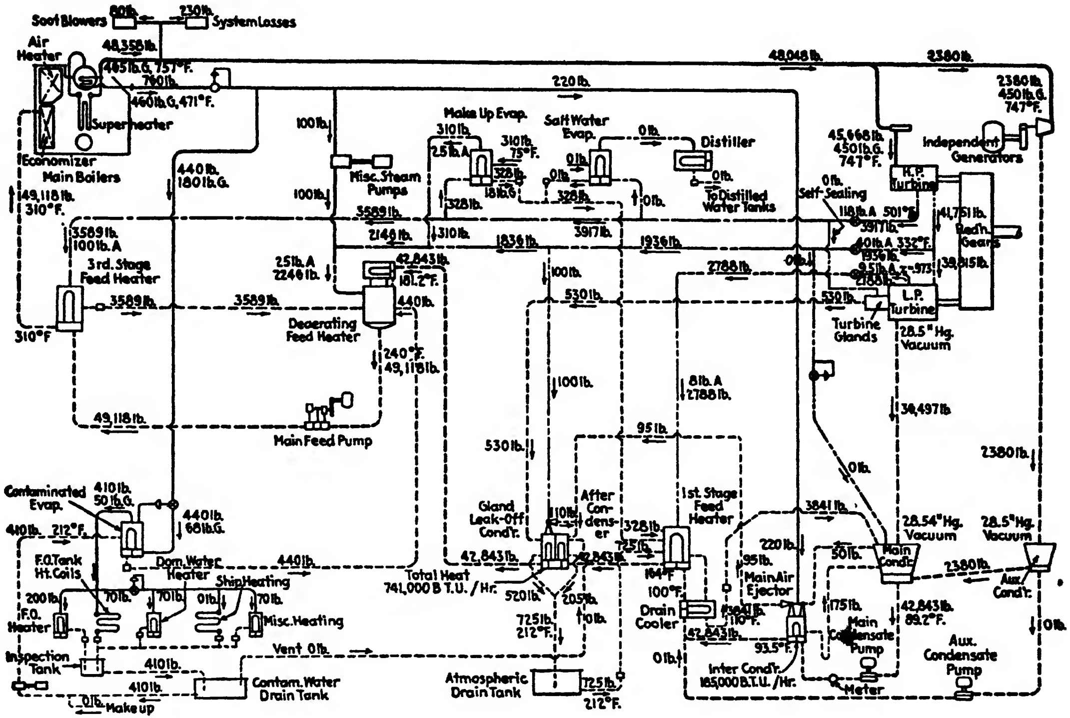


FIG. 130.—Heat balance diagram of *S.S. Challenge* class ships (C-2) of U.S. Maritime Commission. (*From Trans. Soc. Naval Arch. & Mar. Eng.*)

tubes and thus heats the water a few degrees above the saturated steam temperature. This would not be possible in an ordinary type of feed heater.

Most modern feed-water systems incorporate a single-effect evaporator with the system (Figs. 128 and 130). The steam for operating the evaporator is taken off one of the bleeder lines, and the vapor produced is delivered to the bleeder line of the next lowest pressure. These evaporators are generally used only for evaporating the make-up feed carried in the double bottom to ensure its pureness. An evaporator of this type operates at very high efficiency, the main loss being that of heat availability in dropping the pressure from that of the steam in the coils to that of the vapor in the shell.

A very complete diagram of a modern feed-water system (1939) together with a heat balance is shown in Fig. 130. This diagram, published by D. C. MacMillan,[1] is worthy of careful and detailed study.

Many of the cargo and passenger steamships constructed in the United States during the period 1938–1941 were equipped with a direct-contact feed heater because of its excellent deaerating characteristics (see Art. 143). The direct-contact heater may be one of several feed heaters, as shown in Figs. 130 and 141, or a single direct-contact heater may be employed, as shown in Figs. 131 and 139. Figure 131[2] shows a diagram of a naval feed-water system employing a deaerating feed heater. In Fig. 130, a low pressure closed feed heater is placed in the circuit before the direct-contact heater and a high pressure closed heater after the direct-contact heater. This arrangement allows the direct-contact heater to heat the water to the desired temperature with a steam pressure of around 5 to 10 lb gage. High steam pressures are not desirable in large heaters of this type, although they have been used in some land installations. It is also essential for proper deaeration that the pressure in the direct-contact heater be 5 to 10 lb above that of the atmosphere.

[1] Presented by Mr. MacMillan in his discussion of a paper by J. E. Schmeltzer, "Engineering Features of the Commission's Program," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1940.

[2] Taken from a paper by George B. Emerson, "Feed Systems for Naval Vessels," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1940.

When two or three heaters are used in series, one of these is usually a direct-contact heater, and the others are closed heaters.

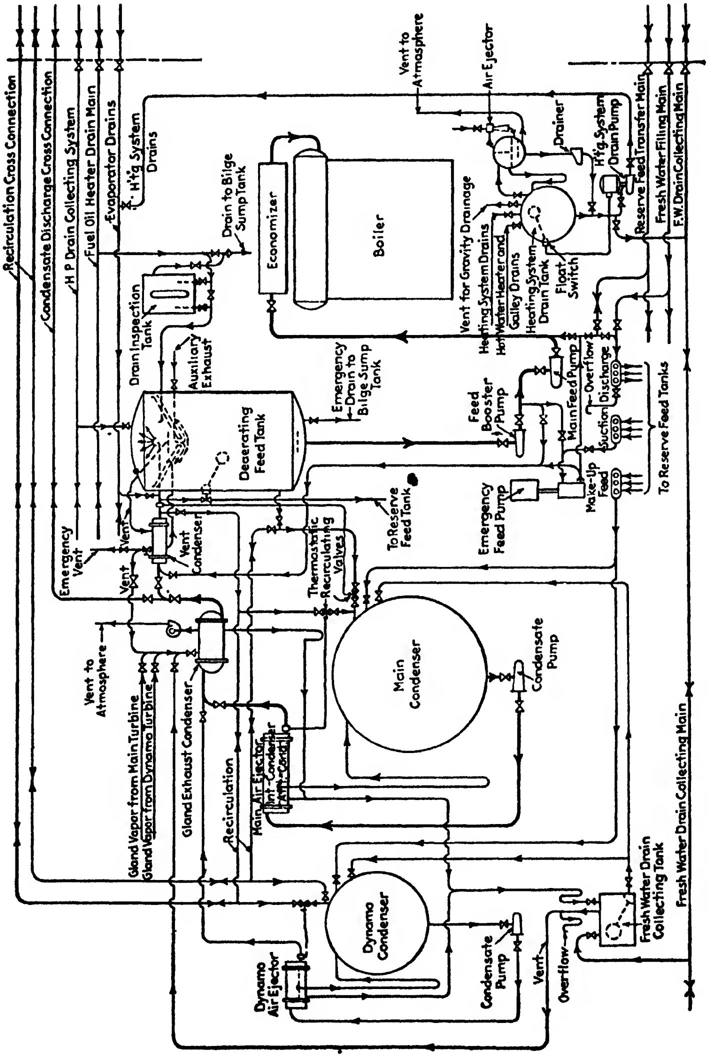


FIG. 131.—Diagram of a naval closed feed system. (*From Trans. Soc. Naval Arch. & Mar. Eng.*, 1940.)

Some plants ashore have used three direct-contact heaters in series, with booster pumps between the heaters.

When direct-contact heaters are employed, the surge tank is now usually incorporated within the heater shell so that the heater serves as a heater, deaerater, and surge tank combined. The feed water in the circuit and the reserve for boiler surge are thus not exposed to the atmosphere at any time in passage from condenser to boiler. As pointed out in the foregoing articles, the reserve water in the feed-water circuit may be contained in a feed and filter tank (open system), as shown in Fig. 118, in the bottom of condenser (Fig. 128), in the direct-contact heater (Figs. 130 and 122), or in an open-top surge tank floating on the feed line (Fig. 129).

150. Fuel Oil Heating.—As pointed out in Art. 45, the fuel oil requires heating in a fuel oil heater before being delivered to the burners and also heating in the settling tanks and frequently in the double-bottom tanks. The heating is done by steam. There is always a possibility that the drains from the fuel oil heater and other fuel oil heating piping will become contaminated by oil leakage and allow oil to get into the boiler with disastrous results. In many of the older ships, the drain from the fuel oil heater was allowed to go into the bilge to obviate this danger. When the oil heater drain is allowed to return to the main feed system, it is usually sent to an inspection tank (Fig. 131), through a filter, or is otherwise checked up.

In the ships constructed by the U.S. Maritime Commission (1939–1941), an ingenious system was used to avoid the contamination of the main feed-water supply by oil leakage. A contaminated-vapor evaporator (see the lower left-hand diagram in Fig. 130) is installed that acts in a way as a low pressure boiler. Steam is supplied to the coils of this evaporator through reducing valves at a pressure around 65 to 70 lb gage. Vapor is produced in the shell of this evaporator at a pressure of around 50 lb gage, which is used for fuel oil heating, ship heating, etc. The drains from the fuel oil heating coils and other miscellaneous heaters that use this steam are returned to the contaminated-water evaporator. Thus this evaporator and its piping form a circuit independent of the main system, and no danger exists of oil ever getting into the main feed system.

CHAPTER XII

POWER PLANT LAYOUTS

Arrangements of machinery for a few representative modern ships are shown in Figs. 132 to 138. The arrangement of the machinery for the triple-screw motorship *M.S. Oranje* is given in Fig. 97. The various pieces of machinery are clearly marked on these arrangement drawings, and they should be helpful in giving the student a better understanding of the layout of ships' power plants.

Figure 132 shows the plan view and Fig. 133, the sections of the machinery arrangement adopted on a number of the U.S. Maritime Commission's C-type single-screw steam cargo ships. A representative diagram of the feed-water system for this type of ship is shown in Fig. 130. This is an extremely compact machinery layout, outstanding because of the short fore-and-aft length devoted to the machinery. In the C-3 class of steamships, 8,500 shp is installed in a length of only 50 ft. In the layout shown in Figs. 132 and 133, the boilers and engines are placed in the same compartment. The boilers are located on the upper level, *i.e.*, on the first deck above the tank top, and the propeller shaft passes under the boilers. The two-pass condenser is placed athwartship under the L.P. turbine. High- and low-level sea suctions are provided for the condenser circulating water. The low-level sea chest is for use at sea and the high-level sea chest for use in rivers and shallow water, where there is danger of drawing in muddy water. The various pieces of auxiliary machinery are clearly marked, and this plan should be studied in connection with the calculations carried out in the next chapter. Particular attention is called to the direct-contact feed heater (16) shown in the upper engine-room casing in the right-hand section (Fig. 133). The auxiliary generating sets (17) are located on the upper level, and their condensers (11) are hung below them on the underside of the deck.

Figures 134*a* and 134*b* show the plan and elevation of the turbo-electric machinery of the *S.S. President Coolidge* (twin screw). It will be noted from Fig. 134*b* that the main turbo-

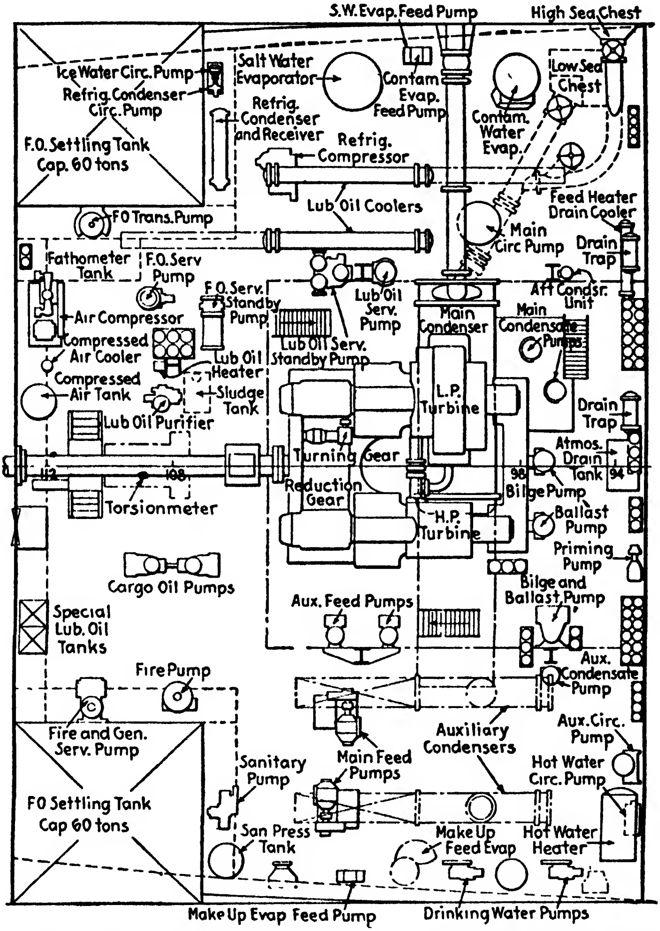


FIG. 132*a*.—Machinery arrangement used on some of the U.S. Maritime Commission's C-type steamships. Plan at lower level. (*Marine Engineering and Shipping Review.*)

generators are placed on the deck above the tank tops. The auxiliary generators are also located on this deck.

Figure 135 shows the plan view of the machinery of the *S.S. Delbrasil* (single screw) of the Mississippi Shipping Company.

Here the more usual arrangement is adopted, with the engines and boilers in separate rooms. The condenser is of the single-pass type and is placed athwartship under the L.P. turbine. The sea suction is on the starboard side, and the overboard

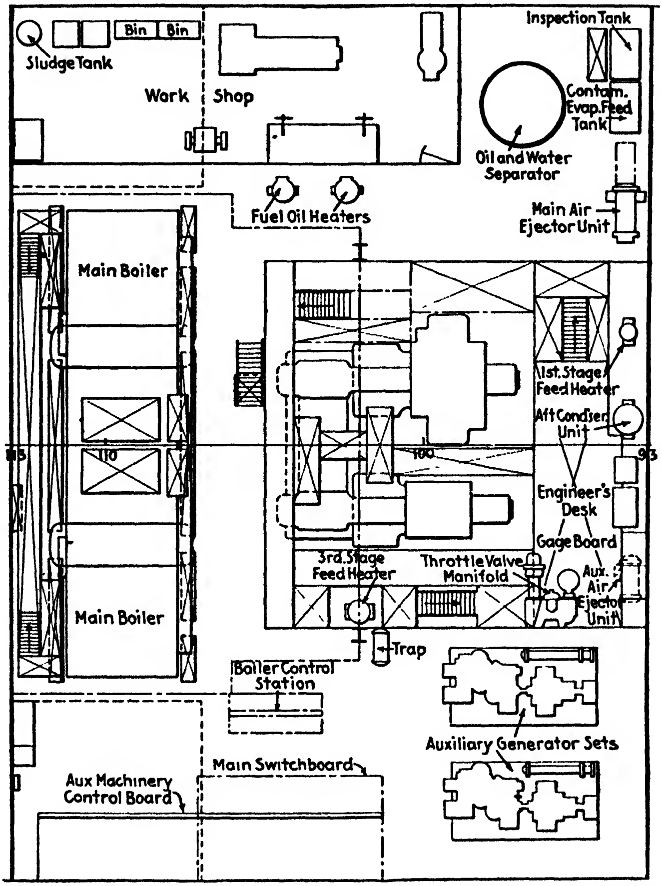


FIG. 132b.—Machinery arrangement used on some of the U.S. Maritime Commission's C-type steamships. Plan at upper level. (*Marine Engineering and Shipping Review.*)

discharge is on the port side. The power of this installation is 7,800 shp. Two Babcock & Wilcox single-pass boilers are fitted similar to that shown in Fig. 11.

Figures 136 and 137 show the elevation and plan view of the geared-turbine machinery installation of the *S.S. America* (twin

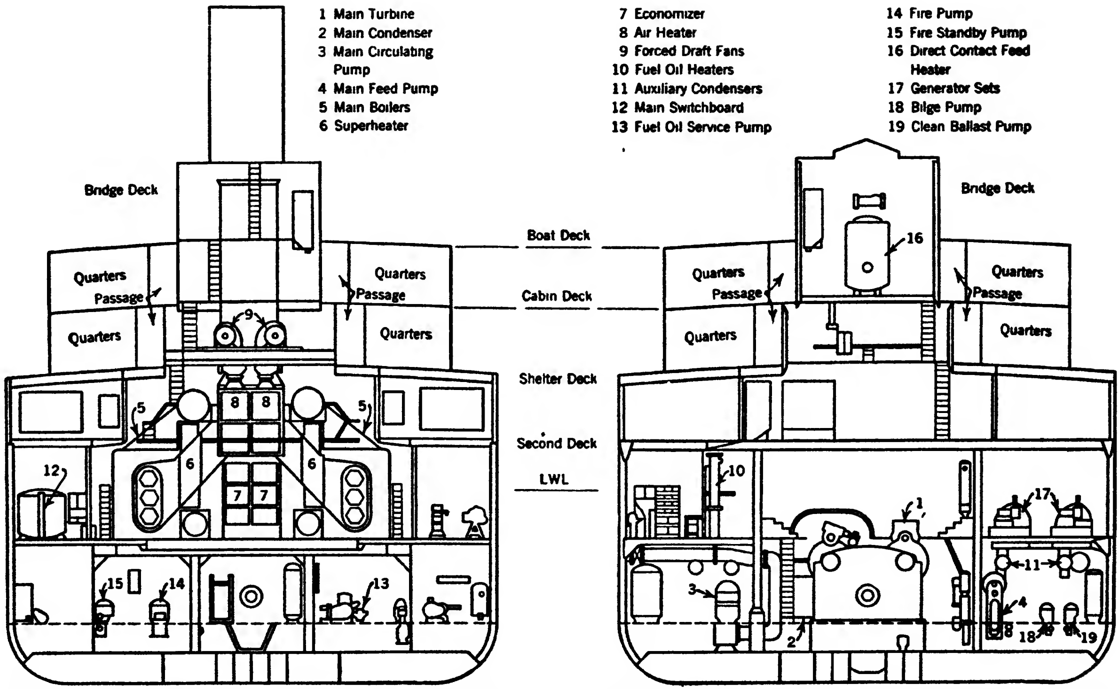


Fig. 133.—Cross sections through the machinery space of *S.S. Frederick Lykes* (C-3 type). Right-hand section looking forward and left-hand section looking aft. (See Figs. 132*a* and 132*b*.) (*The Nautical Gazette.*)

screw). In this ship the boilers are placed in two boiler rooms, one forward and one aft of the engine room. It will be noted that a single-pass condenser is used and that it is placed athwart-

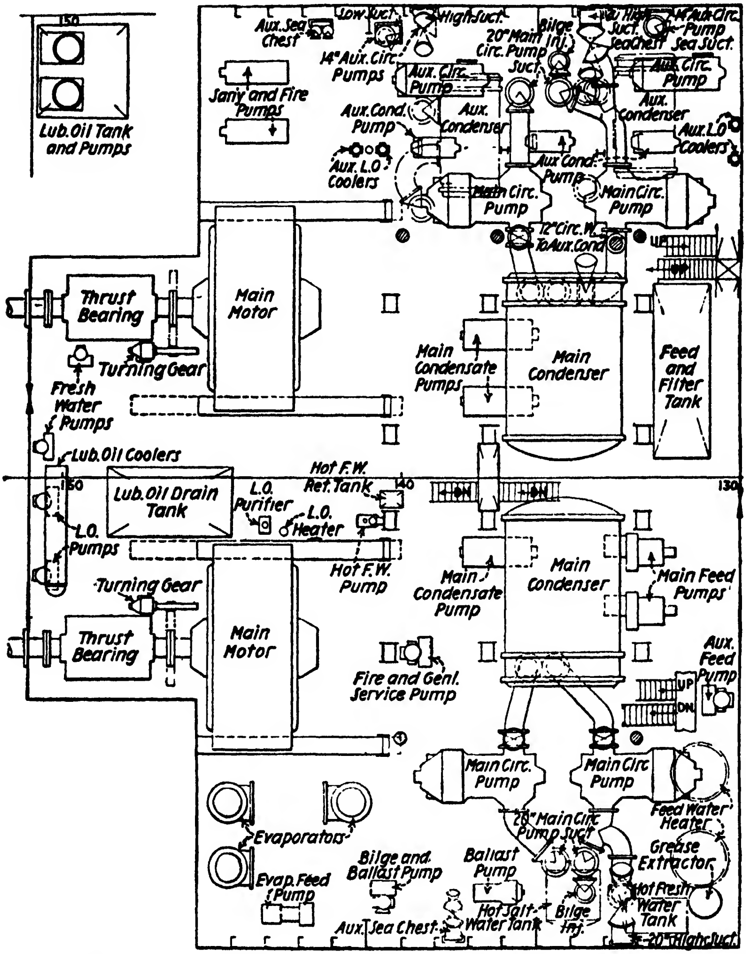


FIG. 134a.—Plan of engine room of *S.S. President Coolidge*. Twin-screw turbo-electric installation of 26,000 shp. (*From Trans. Society of Naval Arch. and Mar. Engineers.*)

ship under the L.P. turbine. Scoops are used for the condenser circulating water, the injection piping extending through the bottom of the ship. Two small circulating pumps are provided for use during maneuvering. Attention is called to the lubricat-

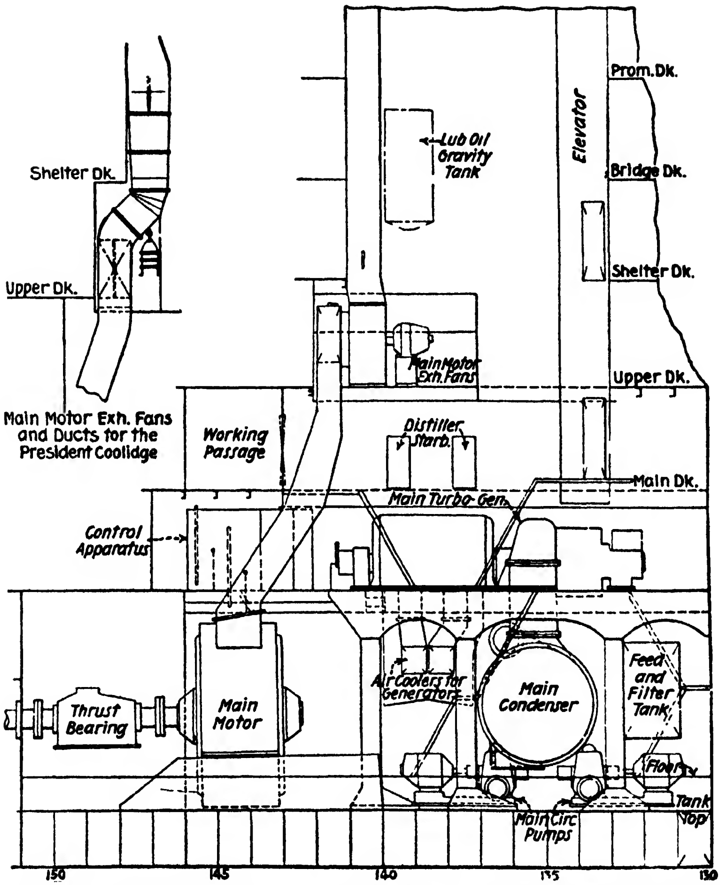


FIG. 134b.—Longitudinal section through engine room of *S.S. President Coolidge.* (*From Trans. Soc. Naval Arch. & Mar. Eng.*)

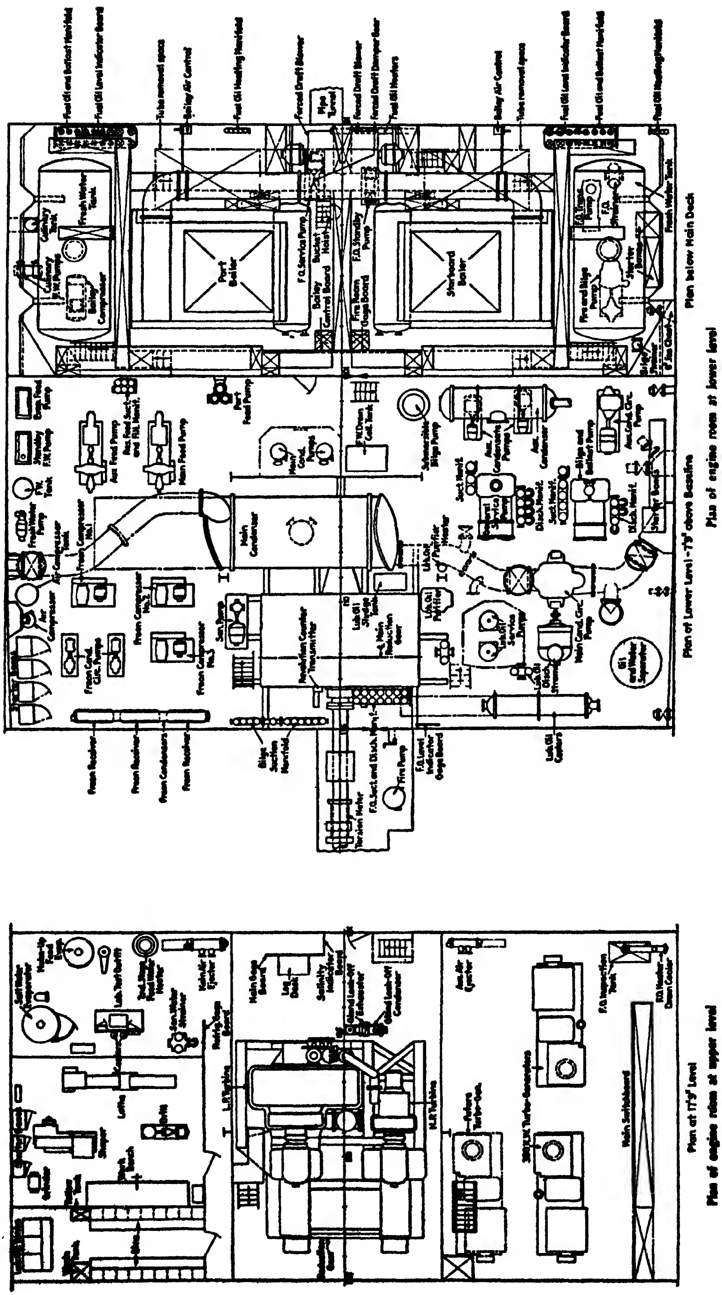


FIG. 135.—Plan of machinery arrangement of *S.S. Delbrasil.* 7,800 shp geared-turbine installation. (*Marine Engineering and Shipping Review.*)

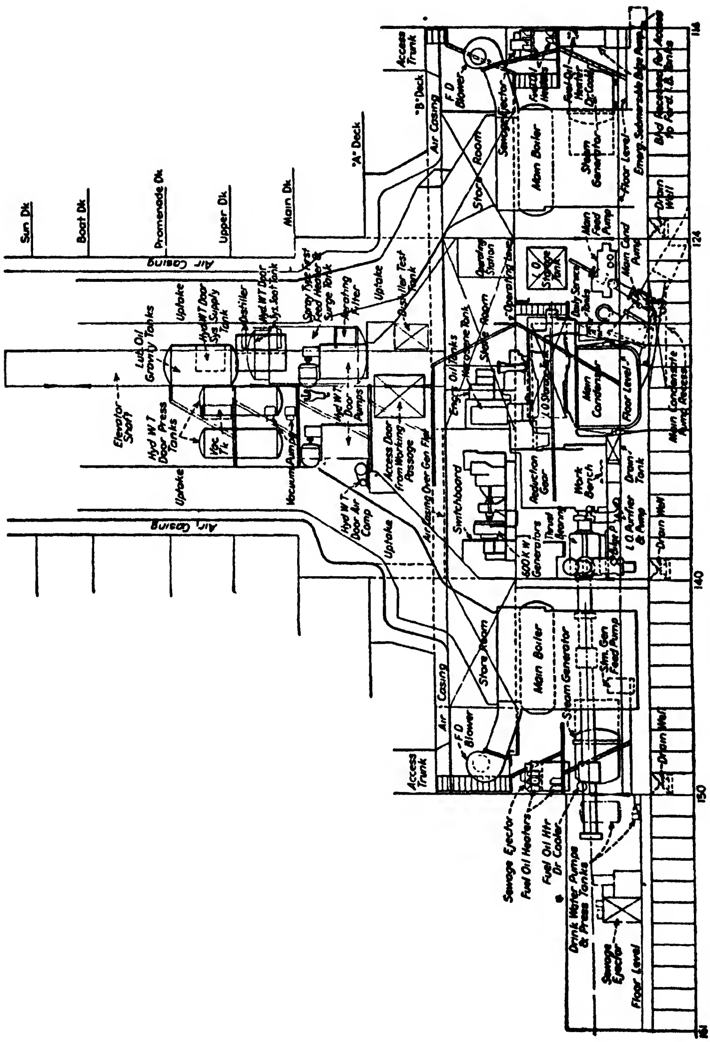


FIG. 136.—Longitudinal elevation of machinery arrangement of S.S. America. (*Trans. Soc. Naval Arch. & Mar. Eng.*)

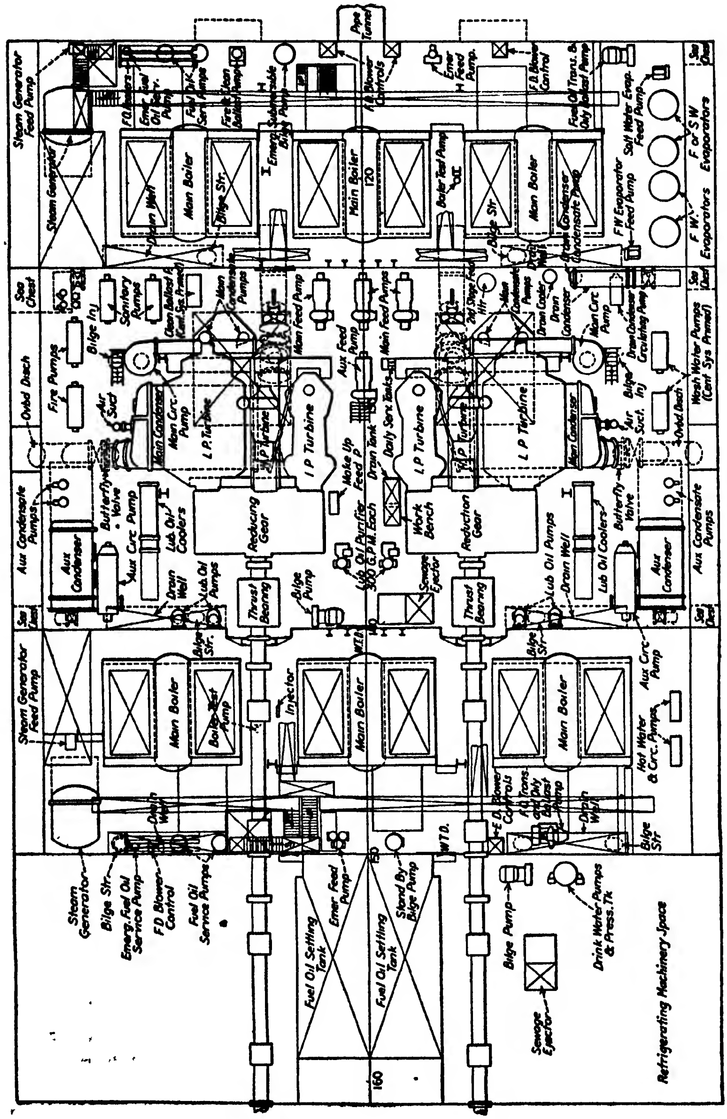


FIG. 137.—Plan of machinery arrangement of S.S. America. (*Trans. Soc. Naval Arch. & Mar. Eng.*)

ing-oil gravity tanks and the combined direct-contact feed heater and surge tank in the upper engine-room casing (Fig. 136).

Figure 138 shows the plan view of the lower level of the geared Diesel machinery as installed on the U.S. Maritime Commission's

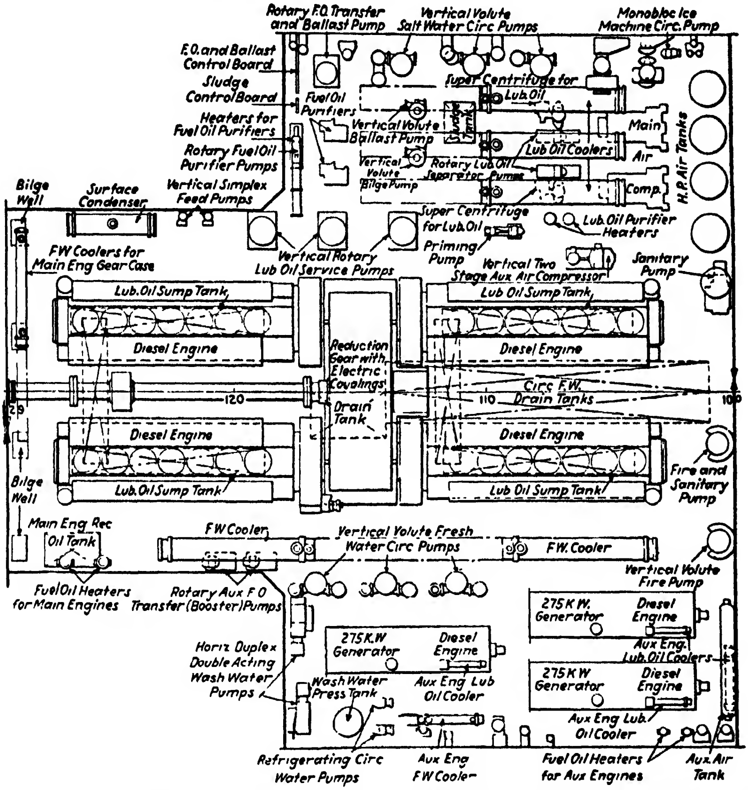


FIG. 138.—Arrangement of machinery of U. S. Maritime Commission's C-3 geared Diesel ships. (*Marine Engineering and Shipping Review.*)

C-3 single-screw cargo motor ships. These ships are fitted with four Busch-Sulzer Diesel engines of the type shown in Fig. 90 (see also Table XV). Electro-magnetic clutches are interposed between the engines and the gearing. The various pieces of auxiliary machinery that are located on the lower level are clearly marked in Fig. 138.

CHAPTER XIII

COMPUTATIONS FOR THE POWER PLANT OF A MERCHANT SHIP

In order to illustrate the procedure for calculating the fuel consumption, boiler heating surface, and horsepower of the various auxiliaries for marine power plants, computations for two simple installations are given in this chapter: (I) with steam auxiliaries and (II) with motor-driven auxiliaries, with the feed water heated by steam extracted from the main turbine. To simplify the work for the student who is beginning his study of marine engineering, some of the refinements of the modern marine installation have been omitted.

Case I.—*Single-screw cargo ship with the following machinery installation:*

Double-reduction-geared turbines, oil-fired water-tube boilers.
Steam auxiliaries.
Service shp = 5,400.
Turbines: compound (H.P. and L.P.) driving through double reduction gears.
Two water-tube boilers fitted with air heaters and economizers.
Propeller rpm = 90.
Boiler pressure (steam drum) = 415 lb gage.
Pressure of steam leaving superheater P_b = 400 lb gage.
Temperature of steam leaving superheater t_b = 700°F.
Pressure of steam at turbine inlet P_1 = 380 lb gage.
Temperature of steam at turbine inlet t_1 = 680°F (237°F superheat).
Vacuum at entrance of condenser P_2 = 28½ in. of mercury (30-in. barometer).
Pressure in auxiliary steam line = 150 lb gage
Pressure in auxiliary exhaust line = 10 lb gage
Feed-water temperature entering economizer = 240°F
Feed-water temperature leaving economizer = 340°F
Engine efficiency (assumed) = 73 per cent (turbine and gears)

Efficiency of boiler, air heater, and economizer (assumed) = 84 per cent in service
Heating value of fuel = 18,500 Btu per lb

Feed-heating system: direct-contact heater, inter- and after-condensers of air ejector and vent condenser (see Fig. 139); auxiliary exhaust steam used for heating.

151. Estimated Steam Consumption at Service Speed.—The estimate for steam requirements when operating at full power

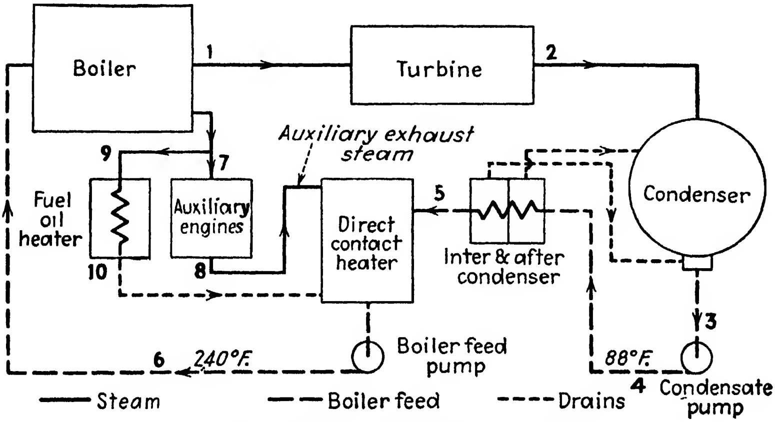


FIG. 139.—Diagram of feed system for Case I.

must be broken down into that required for the main turbine and that required for the auxiliary machinery and other auxiliary purposes.

For cargo ships without refrigerated holds, the amount of steam used for auxiliary purposes other than driving the main engine auxiliaries and boiler auxiliaries is so small that it is customary to make an over-all assumption to cover steam for propelling auxiliaries and ship's use. For passenger ships, separate estimates must be carefully made to cover auxiliary steam for propelling auxiliaries and "hotel" load. The hotel load is made up of:

1. Lighting.
2. Fresh water and sanitary systems.
3. Refrigeration.
4. Ventilation.
5. Heating.
6. Galley.

When cargo ships carry refrigerated cargo, a separate allowance must be added for the steam consumption of the refrigerating machinery.

The estimated auxiliary steam requirements for a cargo ship fitted with steam auxiliaries may be arrived at by assuming a water rate for the auxiliaries in terms of the main turbine shp or by taking it as a percentage of the main throttle steam. When a percentage allowance is used for the auxiliaries, it must be borne in mind that as the economy of the main turbine is improved by increasing initial steam pressure and temperature, better turbine design, higher vacuum, etc., the steam consumption of the auxiliaries decreases but little; hence the percentage of steam used by the auxiliaries will increase. Twenty years ago, when water rates of turbines were of the order of 9 and 10 lb per shp-hr, the percentage of steam used by the main engine auxiliaries for ships fitted with turbines was around 20; today the percentage is higher than this because auxiliary steam consumptions have remained fairly constant and the steam consumption of the main turbine has decreased. The auxiliary steam for the propelling auxiliaries of turbine-driven merchant ships is in the neighborhood of 23 to 32 per cent. If scoops are used instead of a circulating pump, the percentage is appreciably lower because of the relatively high power of this pump. When the steering gear is operated by an electric motor and some of the smaller pumps are motor-driven, as in the following example (see the table on page 378), the amount of auxiliary steam is naturally less than when all auxiliaries are steam-driven.

Ships driven by reciprocating engines have a lower percentage for auxiliary steam, mostly because of the higher water rate of the main engine. The percentages given above for auxiliary steam include the losses, *i.e.*, the make-up feed.

1. *Steam Consumption of Main Turbine.*—From the Mollier diagram we find that the enthalpy of the steam at the turbine inlet for a pressure of 395 lb abs and 680°F is 1352 Btu per lb, and the entropy is 1.63. After an isentropic expansion to 28.5 in., the total enthalpy is 895 Btu per lb.

$$\begin{aligned} h_1 &= 1352 \\ h_2 &= \underline{895} \\ h_1 - h_2 &= 457 \end{aligned}$$

The theoretical water rate $w_t = 2{,}545/457 = 5.58$ lb per shp-hr. With an engine efficiency of 73 per cent, the actual water rate is

$$w_a = \frac{5.58}{0.73} = 7.65 \text{ lb per shp-hr}$$

2. *Auxiliary Steam.*—Assumed = 2.0 lb per hr per shp of main turbine = 26.2 per cent of the main throttle steam. This includes a make-up feed allowance of 2.5 per cent (1,300 lb per hr) to cover losses.

3. *Total Steam Consumption.*

$$7.65 + 2.0 = 9.65 \text{ lb per shp-hr}$$
$$9.65 \times 5{,}400 = 52{,}100 \text{ lb per hr}$$

Later in the chapter separate calculations will be made for the horsepower and steam consumption of each piece of auxiliary machinery and a more accurate value obtained for the auxiliary steam consumption. The preceding value will give us a first estimate for obtaining the fuel consumption and the horsepower of the various pumps and auxiliary generating set, etc.

152. Fuel Consumption.—Under the assumed conditions, the auxiliary engines are operating under a steam pressure of 150 lb. This steam will be obtained by taking the steam as it leaves the superheater at 400 lb gage pressure and 700°F temperature (252°F superheat) and desuperheating it by passing it through a coil submerged in the water of the boiler steam drum. If we assume that this steam leaves the desuperheater at 400 lb pressure and 15 degrees of superheat and is passed through two reducing valves to reduce its pressure to 150 lb per sq in. (gage), the superheat in the auxiliary steam line will be approximately 33°F (from a constant enthalpy expansion on the Mollier diagram). Much of this superheat will be lost by the time the steam reaches the various steam auxiliaries, so the steam entering the auxiliaries will be only slightly superheated. The enthalpy in the auxiliary steam as it leaves the desuperheater in the steam drum of the boiler is that corresponding to steam at 415 lb per sq in. abs and 15 degrees superheat, namely, 1216.3 Btu per lb. No appreciable change in enthalpy takes place in dropping the pressure to 165 lb abs (Art. 11).

The fuel consumption of the ship per shp per hour for all purposes (a.p.) is obtained from the following formula:

$$f = \frac{w_a(h_1 - h_f) + w_{aux}(h_1' - h_f)}{\text{Btu per pound of fuel} \times \frac{e_b}{100}}$$

where w_a = water rate of main turbine per shp-hour.
w_{aux} = water rate of auxiliaries per turbine shp-hour.
e_b = service boiler efficiency in per cent.
h_1 = total enthalpy of main turbine steam leaving superheater.
h_1' = total enthalpy of auxiliary steam leaving boiler.
h_f = enthalpy of feed water entering economizer.
f = fuel consumption per shp-hour (a.p.).

$$f = \frac{7.65(1{,}361.9 - 208.3) + 2.0(1{,}216.3 - 208.3)}{18{,}500 \times 0.84}$$

f = 0.698 lb of oil per shp per hr.

Total fuel per hour = 5,400 × 0.698 = 3,800 lb.

153. Estimate of Boiler Heating Surface.—From the preceding estimated fuel oil consumption, the required boiler heating surface can be readily obtained by assuming a rate of combustion, *i.e.*, oil burned per square feet of boiler heating surface per hour.

As pointed out in Art. 34, these rates are given today in terms of boiler generating surface alone and in terms of combined boiler generating surface and economizer heating surface. The former practice is the more logical and will be used here. Rates of combustion used today in merchant ships with boilers fitted with economizers vary between 0.45 and 0.60 lb of oil per sq ft of boiler generating surface per hr. When boilers are not fitted with economizers and the heating of the water must be done by the boiler tubes, the rates of combustion will generally be lower, usually varying between 0.30 and 0.50. The rate of combustion is tied up with the rate of evaporation (Art. 34), and the rate of evaporation is, of course, not constant for all the boiler heating surface. It is highest for the furnace heating surface and lowest for the tubes farthest from the furnace. Hence, when an economizer is not used and additional boiler tubes are substituted, the evaporation per square foot of generating surface will be lower.

In this example, we shall use a rate of combustion of 0.50 lb of oil per sq ft of boiler generating surface. Hence

$$\text{Boiler heating surface} = \frac{3{,}800}{0.50} = 7{,}600 \text{ sq ft}$$

This will be divided equally into two boilers of 3,800 sq ft each.

154. Quality of Steam Entering Condenser.—From the Mollier diagram, the moisture in the steam after an isentropic expansion from 395 lb abs and 680°F to 28.5 in. of mercury is 19.8 per cent. Because of the reheating of the steam in its passage through the turbine, the actual percentage of moisture is much less than this. If we assume that the loss in the gearing and turbine bearings is 3 per cent, the engine efficiency, based on the internal shp of the turbine, is 0.73/0.97 = 0.752. The loss due to radiation can be assumed to be 3.0 Btu per lb of steam. The actual enthalpy of the steam converted into useful work in the turbine is

$$(1352 - 895) \times 0.752 = 344 \text{ Btu per lb of steam}$$

The total enthalpy of the exhaust steam is therefore

$$1352 - 344 - 3 = 1005 \text{ Btu per lb}$$

At a pressure of 1.5 in. of mercury, the enthalpy of the liquid is 59.7; hence the actual quality of the steam entering condenser is

$$\frac{1005 - 59.7}{1042.0} = 0.907$$

The percentage of moisture can also be obtained from the Mollier diagram at 1005 Btu and 1.5 in. of mercury pressure, namely 9.3 per cent. The entropy at this point is 1.83 instead of 1.63 after an isentropic expansion.

The percentage of moisture (9.3) is entirely satisfactory from the point of view of turbine blade erosion discussed in Chap. VI—in fact, a somewhat higher percentage could have been allowed that would have permitted a higher steam pressure to be used with this steam temperature and engine efficiency.

If the condensate leaves the condenser at a temperature 3.7° below that corresponding to the vacuum, or 88°F, the heat rejected in the condenser is

$$7.65(1005 - 56) = 7252 \text{ Btu per shp per hr}$$

and the percentage of the heat received by the power plant that is rejected in the condenser is

$$\frac{7,252}{18,500 \times 0.698} = 56.3$$

155. Blower Computations.—If we assume a fuel of the following analysis:

$$\begin{aligned} C &= 86.2 \text{ per cent} \\ H &= 10.3 \text{ per cent} \\ O &= 0.7 \text{ per cent} \\ N &= 0.4 \text{ per cent} \\ S &= 2.3 \text{ per cent} \end{aligned}$$

the theoretical amount of air required per pound of fuel is, by the equation given in Art. 41,

$$(11.6 \times 0.862) + 34.8\left(0.103 - \frac{0.007}{8}\right) + (4.32 \times 0.023) = 13.64 \text{ lb}$$

If 20 per cent excess air is allowed, the actual amount of air required per hour is $13.64 \times 1.20 \times 3,800 = 62,000$ lb per hr. If we assume the air to be at 60°F, the required capacity of the blower is

$$\frac{62,000}{60 \times 0.076} = 13,600 \text{ cu ft per min}$$

The air pressure required at the blower for a forced-draft installation can be estimated by assuming certain losses through the burners, boilers, superheater, air heater, economizer, etc. The actual draft losses through any given boiler installation must be obtained from the boiler manufacturer.

In this case we shall assume that the draft losses are as follows:

	In. of Water
Air through air heater	2.2
Air through burner air registers	2.1
Gases through boiler and superheater	0.3
Gases through air heater	0.2
Total draft loss	4.8

In order to see how much draft the stack will produce, we shall make the following assumptions:

Height of stack above furnace = 70 ft
Temperature of air leaving air heater t_s = 290°F
Temperature of outside air t_a = 60°F

$$\text{Draft in inches} = 0.80 \times 70 \left(\frac{7.63}{520} - \frac{8.05}{750}\right) = 0.22 \text{ in.}$$

Obviously the draft produced by a stack of this height, with uptake gases at 290°F, is of little account and hence may as well be neglected in calculating the horsepower of the forced-draft blower.

If we allow a blower efficiency of 60 per cent, the service horsepower input to the blower is

$$\frac{62{,}000 \times (4.8 \times 68.5)}{33{,}000 \times 60 \times 0.60} = 17.1 \text{ hp}$$

where $68.5 = \dfrac{62.5}{12 \times 0.076}$ = feet of air at 60°F corresponding to 1 in. of water pressure.

The required area of the stack will be

$$\frac{3{,}800}{200} = 19.0 \text{ sq ft}$$

156. Condenser Design.—The following data are assumed for the main condenser:

Vacuum at steam inlet to condenser = 28.5 in. of mercury (91.7°F)
Vacuum in bottom of condenser = 28.5 + 0.12 = 28.62 in. of mercury
Barometer = 30 in. of mercury
Steam through condenser = 7.65 × 5,400 = 41,300 lb per hr
Sea temperature = circulating-water-inlet temperature t_1 = 70°F
Overboard discharge t_2 = 91.7 − 9.7° = 82°F
Condensate temperature = 92 − 4° = 88°F
Tubes, cupro-nickel, ¾ in. outside diameter (18 B.W.G.)
Velocity of circulating water = 5 ft per sec
Cleanliness coefficient C_1 = .85 (service condition)
Material coefficient C_2 = .80 (for cupro-nickel)
Air richness ratio $\dfrac{P_s}{P_t} = 0.975$

Enthalpy of steam entering condenser = 1005 Btu per lb (Art. 154)

Air leakage = 17.8 lb of free air per hr at 60°F

The heat transfer in Btu per sq ft per hr per deg difference is

$$U = 350 \times C_1C_2\left(\frac{P_s}{P_t}\right)^2 \sqrt{V}$$

$$U = 350 \times 0.85 \times 0.80 \times 0.95 \times \sqrt{5} = 503 \text{ Btu per sq ft per deg difference per hr}$$

Mean temperature difference is

$$t_m = \frac{t_3 - t_4}{\log_e \dfrac{t_3}{t_4}}$$

where t_3 = initial temperature difference = 91.7 − 70 = 21.7°.
t_4 = final temperature difference = 91.7 − 82 = 9.7°.
t_m = 14.85°F.

The heat to be removed by the cooling water is

$$7.65 \text{ lb} \times (1005 - 56) = 7252 \text{ Btu per shp per hr}$$

$$\text{Required condensing surface} = \frac{7252 \text{ Btu} \times 5{,}400 \text{ shp}}{503 \times 14.85} = 5{,}230 \text{ sq ft}$$

This surface is the amount required for condensing the steam, and an additional allowance of 5 to 10 per cent should be added for air cooling (Art. 131); hence the total condenser cooling surface required = 5,230 × 1.07 = 5,600 sq ft.

$$\text{Cooling water required} = \frac{7252 \times 5{,}400}{82 - 70°} = 3{,}260{,}000 \text{ lb per hr}$$

For a ship operating in salt water,

$$\text{Cooling water required} = \frac{3{,}260{,}000}{60 \times 8.55} = 6{,}360 \text{ gal per min}$$

$$= \frac{3{,}260{,}000}{64 \times 60 \times 60} = 14.15 \text{ cu ft per sec}$$

$$\text{Ratio of cooling water to steam condensed} = \frac{3{,}260{,}000}{41{,}300} = 79$$

$$\text{Cooling surface per shp} = \frac{5{,}600}{5{,}400} = 1.04$$

$$\text{Condensation per square foot} = \frac{41{,}300}{5{,}600} = 7.38 \text{ lb. per hr}$$

Condenser Tubes:

$$\text{Outside diameter} = \tfrac{3}{4} \text{ in. (18 B.W.G.)}$$
$$\text{Inside diameter} = 0.652 \text{ in.}$$
$$\text{Inside cross-sectional area} = 0.00232 \text{ sq ft}$$
$$\text{Cooling surface per foot of length} = 0.196 \text{ sq ft}$$

$$\text{Number of tubes required in each pass} = \frac{14.15 \text{ cu ft per sec}}{0.00232 \times 5 \text{ ft per sec}} = 1{,}220 \text{ tubes}$$

$$\text{Total length of tubes} = \frac{5{,}600 \text{ sq ft}}{0.196 \times 1{,}220} = 23.4 \text{ ft}$$

For a two-pass condenser

$$\text{Length between tube sheets} = 11 \text{ ft} - 9 \text{ in.}$$
$$\text{Number of tubes} = 2{,}440$$

The number of tubes varies inversely as the velocity of the cooling water, and hence an increase in water velocity decreases the required cross-sectional area of the condenser. However, in order to secure the necessary cooling surface, the condenser must be made longer. A study of the preceding calculations will show that the length of the condenser changes as the square root of the cooling-water velocity. Increasing the temperature of the overboard discharge reduces the required cross-sectional area and increases the length of the condenser.

There is a small drop in pressure from the turbine exhaust to the bottom of a condenser, usually varying between 0.02 and 0.20 in. The vacuum in the exhaust nozzle near the turbine is thus less than that existing in the condenser, and that at the top of a condenser is less than that at the bottom. Care must be taken in computing engine efficiencies to use the correct back pressure. Usually the vacuum at the steam inlet of the condenser is used.

157. Vacuum in the Tropics.—If a ship is to operate in tropical waters, it is well to make some investigations of the influence of an increase in sea temperature on the vacuum before fixing the capacity of the circulating pump. In order to illustrate this point, we shall carry through an investigation for the ship with the condenser cooling surface that was calculated in the foregoing article. We shall assume a sea-water temperature of 80°F. If we install a circulating pump with a reserve capacity of 20 per cent, or 7,650 gpm, the rise in cooling-water temperature could be reduced from 12 to 10°.

Allowing an overboard discharge temperature of 10° lower than the vacuum temperature, the vacuum temperature would be 80 + 10 + 10 = 100°F. This corresponds to a vacuum of 28.1 in.

$$\text{Overboard discharge temperature} = 100 - 10 = 90°\text{F}$$

$$t_m = \frac{(100 - 80) - (100 - 90)}{\log_e \cdot 2.0} = 14.45°\text{F}$$

$$U = 503 \times \sqrt{1.20} = 550$$

If the same amount of heat entering the condenser as before is assumed

$$\text{Required cooling surface} = \frac{7{,}252 \times 5{,}400 \times 1.07}{550 \times 14.45} = 5{,}280 \text{ sq ft}$$

This cooling surface is slightly less than the amount installed; and hence the condenser can carry a vacuum of 28.1 in., with sea water at 80°F, and an increase in water velocity of 20 per cent.

The horsepower required for the circulating pump is a function of head times quantity. The quantity of water handled for a fixed area through the tubes will vary directly with the water velocity. The total lost head in a condenser varies approximately from the 1.7 to the 1.9 power of the velocity; hence if we assume that the lost head varies as $V^{1.7}$, an increase in pump capacity of 20 per cent will call for a pump power of $(1.20)^{2.7}$ times the original power, or an increase of 63.5 per cent.

A complete analysis of this problem would, of course, require a more exact determination of the amount of heat entering the condenser. A decrease in vacuum will result in an increase in the water rate, and unless more fuel is burned per hour there will be a falling off in horsepower of the main turbine.

158. Circulating Pump.—From the foregoing articles, the required capacity of the circulating pump is

$$\text{Service capacity} = 6{,}360 \text{ gpm}$$
$$\text{Maximum capacity} = 7{,}650 \text{ gpm}$$

The total head against which the circulating pump must deliver the water is made up of (*a*) static head, (*b*) the pipe friction and losses in valve, strainers, bends, etc., (*c*) the friction head in the condenser tubes, (*d*) the loss at the entrance and exit of the tubes, and (*e*) the loss due to turbulence in the condenser water boxes. The static head to be overcome is the difference in height between the overboard discharge and the water level. This is only from 5 to 8 ft at the light draft, and when the ship is loaded it will usually be close to zero.

The loss due to strainers, valve, and friction in piping to and from the sea must be calculated from a layout of the piping or estimated from another design. These losses for the condenser in question are estimated as follows:[1]

	Ft	
a and *b*	8 0	
c	7.0	(two pass, 11.75 ft long; ¾-in. tubes)
d	0 4	
e	1.6	
Total	17.0	

If an efficiency of 82 per cent for the centrifugal circulating pump at normal load and 80 per cent at maximum power is allowed, the service horsepower input to the pump becomes

$$\frac{(6{,}360 \times 8.55) \times 17.0}{33{,}000 \times 0.82} = 34.2$$

For tropical conditions, an increase in capacity of 20 per cent is to be allowed. This will increase the water velocity from 5 to 6 ft per sec and increase the head from 17 to about 23 ft.

$$\text{Maximum horsepower} = \frac{(7{,}650 \times 8.55) \times 23.0}{33{,}000 \times 0.80} = 57.0$$

159. Condensate Pump.—The quantity of water to be handled by the condensate pump under normal operating conditions is

$$(7.65 \times 5{,}400 \text{ shp}) + 350 \text{ (ejector drain)} + 1{,}300 \text{ (make-up)} = 42{,}950 \text{ lb per hr}$$

This pump must lift the water from a vacuum slightly less than 28.5 in. (depending on the distance from pump to water

[1] Sterling's "Marine Engineers' Handbook," p. 979, gives curves showing total lost head ($c + d + e$) through two-pass condensers of various lengths and for various water velocities and the Standards of the Heat Exchange Institute gives the head lost in tubes of various diameters.

level in condensers) to the level of the direct-contact heater 50 ft above the pump and also must overcome the lost heads through the piping, the tubes of the inter- and aftercondensers, of the air ejector, and vent condenser, and the direct-contact-heater spray valve. This will call for a two-stage centrifugal pump with a total head of around 120 ft. If we allow a pump efficiency of 46.5 per cent, the service horsepower input to the pump is

$$\frac{42{,}950 \times 120}{60 \times 33{,}000 \times 0.465} = 5.6$$

160. Air Ejector.—The probable maximum air leakage can be estimated from the equation in Art. 138.

$$\begin{aligned} A &= 7.5 + 0.00025C \\ &= 7.5 + (41{,}300 \times 0.00025) = 17.8 \text{ lb per hr} \end{aligned}$$

If we assume that the air-vapor mixture is withdrawn from the condenser at 75°, as shown by the example in Art. 138, the amount of air-vapor mixture to be withdrawn per hour is

$$17.8 \times 2.08 = 37.0 \text{ lb per hr}$$

If the air-vapor outlet temperature is taken at 7.5° below the temperature of the steam at inlet to the condenser, or 84.2°F, as recommended by the Heat Exchange Institute Standards, the amount of air-vapor mixture to be withdrawn is

$$17.8 \times 3.8 = 67.5 \text{ lb per hr}$$

If we allow 70 lb of air-vapor per hr and a steam consumption of 5 lb of steam per lb of air-vapor mixture, the steam consumption of the air ejector is

$$70 \times 5 = 350 \text{ lb per hr}$$

161. Boiler Feed Pump.—The following types of pumps are available for boiler feed:

1. Direct acting reciprocating steam driven.
2. Centrifugal.
    a. Driven by steam turbine.
    b. Driven by electric motor.
3. Plunger pump motor-driven through gears.
4. Aldrich-Groff controllable capacity pump.
    a. Motor driven (a-c or d-c).
    b. Turbine driven.

In this case, where steam auxiliaries are employed, we have a choice of type 1, 2*a*, or 4*b*. The plunger-type pump and the Aldrich-Groff pump have the advantage of higher pump efficiencies and the latter has the advantage of variable capacity at constant rpm. The plunger pump and the Aldrich-Groff pump have pump efficiencies of the order of 80 to 85 per cent, against around 55 to 60 per cent for a four-stage centrifugal pump. In this case, a centrifugal pump driven by a steam turbine will be selected and a direct-acting (reciprocating) pump will be installed as a stand-by.

The maximum capacity of the boiler feed pump should be about 25 per cent greater than the normal so that it can take care of any sudden overload or bring the level of the water in the boiler back to normal in a reasonable time when it falls because of boiler surge or for any other reason. Boiler feed pumps should be fitted in duplicate, one for normal service and one for a stand-by. The feed pump must have sufficient pressure to overcome the boiler pressure, the static head between pump and boiler drum, the heads lost through pipe friction, bends, valves (and closed heater, when one is installed) and, in addition, produce a head sufficient to create a velocity of flow into the boiler.

The required normal service capacity of the feed pump is 52,100 lb per hr. The boiler-drum pressure is 415 lb gage (corresponding to a head of 1,010 ft of water at 240°F). An estimate of the head lost through pipe and valve friction plus the static head indicates that about 100 ft of additional head should be allowed, making the total pumping head 1,110 ft. In this case, a direct-contact feed-water heater is installed on the suction side of the feed pump; hence there is no lost head through the feed-water heater to be taken into account, which often would be the case. From this head should be subtracted the positive net suction head of about 40 ft due to the elevated position of the direct-contact feed heater.

Allowing an efficiency of 55 per cent for the four-stage centrifugal pump, we have

$$\text{Service horsepower input to pump} = \frac{52{,}100 \times 1{,}070}{33{,}000 \times 60 \times 0.55} = 51.2$$

The specifications of the required pump are:

$$\text{Capacity} = 104 - 130 \text{ gpm}$$
$$\text{Discharge head} = 456 \text{ lb per sq in.}$$

162. Main Lubricating-oil System.—This system handles the lubricating-oil supply for the main reduction-gear sprays, turbine and gear bearings, thrust bearing, etc. (see Fig. 79). The following calculation is for a gravity system as used on merchant ships.

Assumed operating data:

Type of pump for lubricating oil = direct acting
Type of lubricating-oil cooler pump = direct acting
Top of gravity tank above pump = 35 ft
Specific heat of oil = 0.48
Temperature of oil entering cooler = 135°F
Temperature of oil leaving cooler = 120°F
Temperature of water entering cooler = 70°F
Temperature of water leaving cooler = 100°F

In order to obtain the required capacity of the lubricating-oil cooler and pump, it is customary to allow 5 per cent of the turbine shp carried away as heat by the lubricating oil. The combined efficiency of the reduction gears and bearings has been assumed as 97 per cent. To this should be added the heat transmitted along the shaft from the turbine casings and the friction loss in the thrust bearings, which, for a bearing of the Kingsbury type, will be of the order of 3 to 5 shp.[1]
The heat to be carried away is

$$\frac{(5{,}400 \times 0.05) \times 2{,}545}{60} = 11{,}500 \text{ Btu per min}$$

The lubricating oil required per minute is

$$\frac{11{,}500}{(135 - 120) \times 0.48} = 1{,}600 \text{ lb per min}$$

$$= \frac{1{,}600}{7.5} = 213 \text{ gpm}$$

This quantity to be supplied to the lubricating-oil headers should be increased about 20 per cent to allow for sufficient overflow from the gravity tanks. Hence the required capacity of the pump for a rise of oil temperature of 15°F is 256 gpm. A pump of 250 to 275 gpm should be supplied.

[1] H. A. Stevens and N. Ogden, "Friction Tests of Propeller Thrust Bearings," *Jour. Am. Soc. Naval Eng.*, November, 1922.

The static head between pump and overflow of gravity tank has been assumed to be 35 ft, and to this should be added the friction head in the suction and discharge piping, loss through valves, strainers, etc., and head lost through the oil cooler, amounting to about 45 ft, making a total head of oil of 80 ft, which is equal to a pressure of 31 lb per sq in.

The service hydraulic horsepower of the lubricating-oil pump is

$$\frac{(1{,}600 \times 1.20) \times 80}{33{,}000} = 4.66 \text{ hp}$$

The usual practice today is to place the lubricating-oil coolers in circuit parallel with the main condenser and to use a branch line from the main circulating pump to cool the oil. In this case, however, we shall use a direct-acting lubricating-oil cooler pump for supplying the salt water for cooling the lubricating oil.

The quantity of water required is

$$\frac{11{,}500}{100 - 70} = 384 \text{ lb per min}$$

With a discharge pressure of 25 lb (56 ft), the hydraulic horsepower of the lubricating-oil-cooler pump is

$$\frac{384 \times 56}{33{,}000} = 0.65$$

163. Fuel Oil System.

Assumed operating data:

Oil burned per hour = 3,800 lb
Fuel oil temperature in bunkers = 60°F
Fuel temperature at burners = 200°F
Oil pressure at burners = 300 lb per sq in.
Lost head between service pump and burner = 30 lb per sq in.
Service pump suction pressure = 10 lb per sq in.
Combined suction and discharge heads of fuel oil transfer pump = 60 ft
Steam pressure at heater (gage) = 150 lb per sq in.[1]

[1] Usually the 150 lb pressure in the auxiliary steam line would be reduced to around 50 lb before sending it to the fuel oil heater in order to avoid oil deposits on heater tubes.

Quality of steam at heater = dry
Specific gravity of oil, 8.1 lb per gal = 0.975
Type of fuel oil service pump = direct acting
Type of fuel oil transfer pump = direct acting
The steam required for heating the oil is

$$\frac{3{,}800 \times (200 - 60) \times 0.48}{857} = 300 \text{ lb per hr}$$

The head on the service pump is 340 lb per sq in. = 810 ft. The service hydraulic horsepower of the fuel oil service pump is

$$\frac{3{,}800 \times 810}{33{,}000 \times 60} = 1.55$$

The hydraulic horsepower of the fuel oil transfer pump, if in continuous operation, would be

$$\frac{3{,}800 \times 60}{33{,}000 \times 60} = 0.11$$

Actually the fuel oil transfer pump is run not continuously but intermittently to pump up the fuel oil settling tanks. If we assume that each settling tank holds an 8-hr supply of oil and that the tanks are to be pumped up in 2 hr, the required capacity of the pump would be

$$\frac{3{,}800 \times 8}{2 \times 60 \times 8.10} = 31 \text{ gpm}$$

The fuel oil pump should have a maximum capacity considerably in excess of this. A pump of about 100 gpm capacity would usually be supplied for a ship of this horsepower.

164. Surge-tank Capacity.—The ship in this example is fitted with a direct-contact feed heater that also functions as a surge tank, and hence it should have sufficient water capacity in the bottom to take care of boiler surge and any additional supply of boiler feed water that may be desired.

When open feed and filter tanks were fitted (Fig. 118), the tank was made of sufficient size to hold around a 15-min supply of boiler feed water. Surge tanks, as noted below, seldom have capacities as large as this.

Surge-tank capacities should be at least sufficient to take care of boiler surge when the engines are suddenly shut down. To fix the capacity of the surge tank in this manner requires a knowledge of the density of the water in the boiler under normal operating conditions (*i.e.*, when the water is filled with steam bubbles). This density can be determined only by tests of the boiler in question, and, of course, the density will vary with the rate of evaporation. If the density of the water in the boiler under normal operating conditions is known, the surge tank should have at least sufficient volume of water at 240°F to take care of the difference in weights of water in the boiler (at steaming level) at shut-down or zero evaporation and maximum evaporation. The density of the water for zero evaporation can be obtained from the steam tables by looking up the specific volume of water corresponding to the boiler pressure. When information on densities is lacking, the capacity of the surge tank for a merchant ship should be designed to hold a supply of from 6 to 10 min.

If we allow an 8-min supply, the surge-water capacity at the bottom of the direct-contact heater would be

$$\frac{52{,}100 \times 8.0}{60 \times 8.33} = 835 \text{ gal}$$

165. Final Estimate of Steam Consumption.—The detailed estimate of steam used by the auxiliaries in Table XXIV gives the following results:

	Lb of Steam per Hr
Main turbine, 5,400 × 7.65	41,300
Auxiliary engines	10,425
Air ejector	350
Fuel oil heater	300
Make-up feed	1,300
Total	53,675

The preliminary estimate for the auxiliary steam (2.0 lb per shp-hr) was 10,800 lb per hr. The detailed estimate is 12,375 lb per hr.

This is 30 per cent of the main throttle steam, or 2.29 lb per shp per hr. Our original estimate was out by 1,575 lb.

The available enthalpy in the ejector exhaust is 350 lb × 1000 Btu per lb = 350,000 Btu per hr. The rise in the temperature of the feed in passing through the inter- and aftercondensers of the air ejector will be

$$\frac{350{,}000}{42{,}950} = 8.2°\text{F}$$

TABLE XXIV.—POWER AND STEAM CONSUMPTION REQUIREMENTS OF AUXILIARIES
Back pressure = 10 lb per sq in. gage

Auxiliary	Type of drive	Type of auxiliary	Service hp	Steam per hp-hr	Total steam consumption	Remarks
Circulating pump	Geared turbine	Centrifugal	34.2	54	1,850	
Boiler feed pump	Geared turbine	4-stage centrifugal	51.2	48	2,462	
Air ejector				350	350	70-lb air-vapor mixture removed per hr
Condensate pump	Direct-connected turbine	2-stage centrifugal	5.6	85	475	
Blower	Direct-connected turbine	Blower	17.1	70	1,200	
L.O. pump	Direct acting	Reciprocating	5.0	150	750	
L.O. cooler pump	Direct acting	Reciprocating	0.70	160	112	
L.O. purifier	Motor					Not operating
F.O. service pump	Direct acting	Reciprocating	1.55	160	248	
F.O. transfer pump	Direct acting	Reciprocating	0.11	160	18	Calculations allow for continuous operation
Sanitary pump	Direct acting	Reciprocating	1.0	160	160	
Fresh-water pumps	Motor	Rotary			†	
Refrigerating set	Motor				†	
Steering gear	Electric				†	
Generating set	Turbine	Direct current	75	42	3,150	See footnote †
Galley	Electric					
Evaporator pumps	Direct acting	Reciprocating				Not operating
Fire pump	Direct acting	Reciprocating				Not operating
Bilge pump	Direct acting	Reciprocating				Not operating
F.O. heater					300	
Losses*					1,300	
Ship heating						Not operating (400 lb per hr)
Total					12,375	

See facing page for footnotes.

The temperature of the feed water entering the direct-contact heater will be $88 + 8.2 = 96.2°F$.

The revised fuel consumption is

$$f = \frac{7.65(1{,}361.9 - 208.3) + 2.29(1{,}216.3 - 208.3)}{18{,}500 \times 0.84}$$
$$= 0.72 \text{ lb per shp per hr}$$

If the excess of the auxiliary exhaust discussed on the following page is injected into the low pressure end of the main turbine, the preceding figure will be reduced slightly.

166. Calculations for Feed Heating.—If all the auxiliary exhaust, as shown by the preceding estimates, were used for heating the feed water, the results would be as follows:

Main condensate at 96°F = (41,300 + 1,300 + 350) × 64 Btu per lb =	2,750,000
Fuel oil heater drain = 300 × 338.5 Btu per lb	101,600
Auxiliary exhaust at 10-lb gage = 10,425 × 1,160.6	12,150,000
	15,001,600

$$\frac{15{,}001{,}600}{53{,}675} = 279 \text{ Btu per lb}$$

This corresponds to a temperature of 309°F. Since the temperature of the auxiliary exhaust steam at 25 lb abs is only

---

Note: In the tables, L.O. indicates lubricating oil and F O. indicates fuel oil.

* Whistle, soot blower, gland leak-off, leakage, etc. = 1,300 lb per hr make-up feed.

† Motor driven. The load on the auxiliary generating set is made up as follows:

	Hp Input to Auxiliary
Fresh-water pumps	1.5
Refrigerating set	5.0
Steering gear	8.0
	14.5
Allowances for lighting, navigation equipment, etc.	27.5 kw
Allowance for galley	6.5 kw
	34 kw

Allowing 85 per cent generator efficiency and 80 per cent as the average motor efficiency, the required horsepower input to the generator set is

$$\frac{14.5}{0.85 \times 0.80} + \frac{34 \times 1.34}{0.85} = 75 \text{ shp}$$

Data on steam consumption of auxiliaries will be found in the following books and papers: "Water Rates and Steam Consumption of Marine Machinery," by H. E. Brelsford and E. A. Stevens, Jr., Simmons-Boardman, 1926. "Modern Marine Auxiliary Plant," by H. Hillier, *Trans. Inst. Eng. and Shipbuilders in Scotland*, 1936. "Heat Balance Calculations for Marine Steam Plants," by A. S. Thaeler and D. C. MacMillan, *Trans. Soc. Nav. Arch. & Mar. Eng.*, 1938.

240°F it is evident that there is more auxiliary exhaust than can be used for feed-water heating, and part of the auxiliary exhaust steam will have to be injected into the low pressure end of the main turbine or sent to the main or auxiliary condenser, where its enthalpy of vaporization will have to be thrown away. Obviously the former method of using the excess exhaust steam is more economical.

Only 6,450 lb per hr of auxiliary exhaust steam are required to heat the feed from 96 to 240°, as shown by the following calculation:

$$
\begin{array}{rrcrl}
\text{Main condensate,} & 42{,}950 & \times & 64 & = 2{,}750{,}000 \\
\text{Fuel oil heater drain,} & 300 & \times & 338.5 & = \quad 102{,}000 \\
\text{Auxiliary exhaust,} & \underline{6{,}450}\text{ lb} & \times & 1{,}160.6 & = \underline{7{,}500{,}000} \\
 & 49{,}700 & & & \quad 10{,}352{,}000
\end{array}
$$

$$\frac{10{,}352{,}000}{49{,}700} = 208 \text{ Btu per lb}$$

The enthalpy of the liquid at 240°F is 208 Btu per lb; hence 10,425 − 6,450 = 3,975 lb per hr of auxiliary exhaust will have to be sent to the low pressure end of the main turbine.

Practically all marine power plants using steam auxiliaries have an excess amount of auxiliary exhaust steam that cannot be used for feed-water heating as shown by the preceding example. Usually the excess is somewhat larger than that in the illustrative example, especially when steam steering gears are employed and all the small pumps and refrigerating machinery are steam-driven instead of motor-driven as in the example. When a closed feed heater is used, the auxiliary exhaust pressure is generally around 5 lb gage (228°). Under these conditions the feed can be heated to only about 218°F.

When there is more auxiliary exhaust than can be used for feed heating, the excess is sent to the auxiliary condenser, where its enthalpy of evaporation is thrown away; or it is sent to one of the low pressure stages of the main turbine. With a reciprocating-engine installation, it could be sent to the receiver between the I.P. and L.P. cylinders if the pressure of the auxiliary exhaust steam was higher than that in the receiver. Obviously, injecting the excess steam into the turbine is more efficient than sending it to the auxiliary condenser. However, this process is just the reverse of bleeding steam from the main turbine, discussed in Art. 146, and the amount of heat converted into work before the steam enters the condenser is small, as shown by the example in Art. 146.

It is the excess of auxiliary exhaust steam that exists with practically all installations using steam auxiliaries that is largely responsible for making the use of steam auxiliaries so inefficient. If all the auxiliary exhaust could be used for feed heating and no excess steam existed, the auxiliaries in such an auxiliary system would be operating under efficient conditions, since all the enthalpy of the steam not converted into useful work (or lost by radiation and leakage) would be returned to the boiler in the feed water. Of course, any one of the noncondensing auxiliary units considered by itself outside the system is very inefficient, since the enthalpy in the exhaust under such conditions would be thrown away.

When steam auxiliaries are to be employed, estimated water rates of each unit should be obtained from the manufacturer and calculations carried through as above before the ship is built. If the calculations show more auxiliary exhaust steam than can be used for feed heating it can be reduced by replacing some of the proposed auxiliaries by more efficient ones. Some of the direct-acting steam pumps can be replaced by pumps driven by reciprocating engines or turbines, turbines with more stages can be employed, compound or unaflow reciprocating engines used, etc. Another means of reducing the auxiliary steam consumption is to drive some of the smaller auxiliary pumps by electric motors. The water rate of the generating set is appreciably lower than small steam-driven pumps as shown by the preceding table. A steam steering gear uses a large amount of steam and should be replaced by an electric one, even when practically all the engine-room auxiliaries are steam-driven. The main circulating pump should be used for cooling the lubricating oil, as is now the usual practice.

Naturally, as much of the excess exhaust steam as possible should be used for heating the ship, for heating water, and for the galley.

Little gain can be realized by increasing the auxiliary back pressure in order to obtain a greater steam temperature and thus utilize more steam by heating the feed to a higher temperature, for an increase in the back pressure raises the water rate of the auxiliaries and results in an increase in the amount of auxiliary exhaust steam.

Unless attention is given to the upkeep of the auxiliaries, the steam consumption will increase over that existing when the units are new and in good repair. There is no doubt that steam auxiliaries require more constant attention than motor-driven auxiliaries in order to maintain a low fuel consumption.

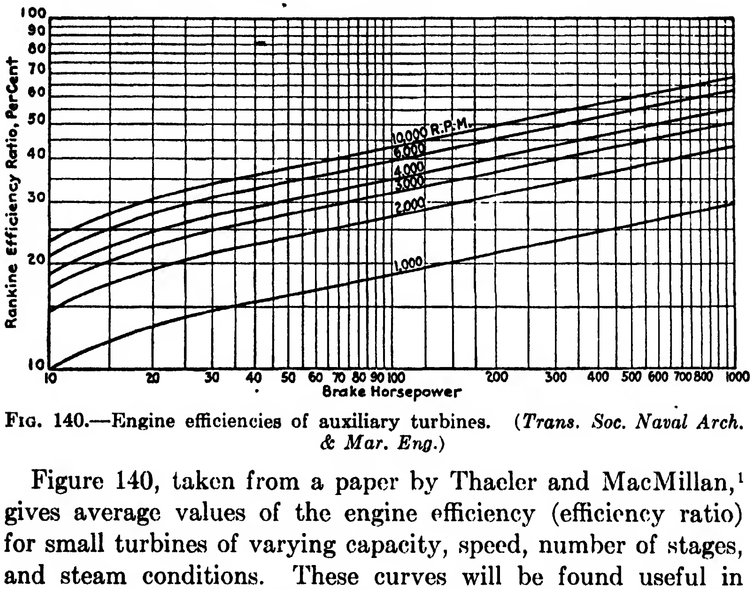


FIG. 140.—Engine efficiencies of auxiliary turbines. (*Trans. Soc. Naval Arch. & Mar. Eng.*)

Figure 140, taken from a paper by Thaeler and MacMillan,[1] gives average values of the engine efficiency (efficiency ratio) for small turbines of varying capacity, speed, number of stages, and steam conditions. These curves will be found useful in estimating the approximate steam consumption of auxiliary turbines.

Case II.—Cargo ship with the following machinery installation:

Service shp = 5,400.

Double-reduction-geared turbines; 2 oil-fired water-tube boilers fitted with air heaters and economizers.

Boiler pressure = 465 lb gage.

Pressure of steam at superheater outlet = 450 lb gage.

Temperature of steam at superheater outlet = 725°F.

Pressure at turbine inlet = 435 lb gage.

Temperature of steam at turbine inlet = 710°F.

[1] A. S. Thaeler and D. C. MacMillan, "Heat Balance Calculations for Marine Steam Plants," *Trans. Soc. Naval Arch. & Mar. Eng.*, 1938.

Vacuum at steam inlet of condenser = 28½ in. of mercury (30-in. barometer).

Motor-driven auxiliaries, Diesel-driven generating set.

Feed-water system: extracted steam used for feed heating in two heaters (a direct-contact heater and a closed heater), inter- and aftercondensers of air-ejector, gland condenser, and a vent condenser.

Engine efficiency (excluding gears)[1] = 0.772

Engine efficiency (including gears and bearings) = 75 per cent.

Temperature of feed entering boiler = 320°F.

Boiler efficiency (assumed) = 84 per cent in service.

Heating value of fuel = 18,500 Btu per lb.

Efficiency of gears and bearings (assumed) = 97 per cent.

Temperature of condensate leaving condenser = 87°F.

Make-up feed (assumed) = 780 lb per hr.

Turbine glands self-sealing; gland leak-off = 100 lb per hr.

Turbine horsepower (*i.e.*, before reduction gears) $= \dfrac{5{,}400}{0.97}$
$= 5{,}570$

Feed-heating System.—In order to illustrate the calculations for a plant using steam extracted from the main turbine for feed heating, a simple layout with two feed heaters has been used, so that the student beginning his study of marine engineering can readily follow the calculations. After the fundamental principles are understood, calculations for more complicated arrangements can readily be made.

An arrangement of the feed-heating system is shown in Fig. 141.

In this case two heaters of two different types are used—a direct-contact or "open" heater for the lower pressure and a closed heater for the higher pressure. The feed water is passed through the inter- and aftercondenser of the main air ejector and the gland steam condenser on its way from the main condenser to the low pressure heater *B*. A Diesel-driven generating set has been used. This simplifies the problem, since it avoids the necessity of estimating the horsepower of all the auxiliary motors in order to find the required horsepower and hourly steam consumption of the auxiliary generating set. To simplify

[1] Obtained from sectional efficiencies. (Art. 167.)

the calculations further, it is assumed that the rise in temperature through the inter- and aftercondenser and gland condenser is 13°F. The steam-heating system for heating the accommodations is assumed inoperative, and no auxiliary steam pumps are running, such as bilge and fuel oil transfer pumps.

The drain from the closed heater is led direct to the direct-contact heater, and since only two feed heaters are used no drain cooler is required in this hookup.

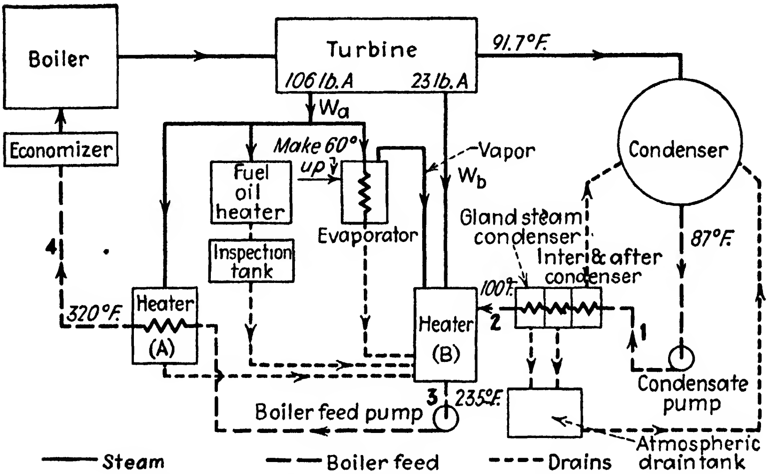


FIG. 141.—Diagram of feed-system for Case II.

A make-up evaporator is used operating on steam from the high pressure bleeder outlet of the turbine; the vapor from evaporator is sent to the direct-contact heater. The fuel oil heater also receives steam from the high pressure bleeder outlet, and the drain goes to an inspection tank and thence to the direct-contact heater. The atmospheric drains are led to a drain tank. The water accumulated in this tank is drawn into the main condenser.

A pressure of 23 lb abs has been used for the heating steam for the direct-contact heater *B*. This will raise the feed temperature to 235°F. The closed feed heater *A* raises the temperature of the feed from 235 to 320°F. As shown in Fig. 141, the condensate leaves the condenser at 87°, which is 4.7° below the temperature corresponding to 28½ in. of vacuum. The feed is assumed to enter the direct-contact heater *B* at a temperature of 87 + 13 = 100°F. This arrangement does not give an equal rise in feed temperature through each of the heaters, which is desirable. However, a pressure of 8 to 10 lb above the atmosphere is necessary in the direct-contact heater to ensure complete deaeration of the feed water. A pressure of 23 lb abs has been taken for heater *B*, which fixes the temperature of the feed leaving heater *B* at 235°F.

Since the temperature of the feed water leaving heater *A* has been fixed at 320°F and since about 12° difference is necessary between the heating steam and the feed leaving the heater, the required extraction temperature is 332°, which corresponds to 106 lb abs pressure.

167. Estimate of Auxiliary Steam Consumption.—The auxiliary steam is desuperheated in the boiler drum and leaves the desuperheater at 425 lb gage and 465°F.

	Lb per Hr
Miscellaneous steam pumps	None operating
Main air ejector (70 lb air vapor) × 5.0 lb steam per hr	350
Gland steam ejector	150
Soot blowers	230
Whistle, galley, and losses	550
Total auxiliary steam	1,280

$$\frac{1{,}280}{5{,}570} = 0.23 \text{ lb per turbine hp per hr}$$

Steam required by evaporator:

Make-up feed = 780 lb per hr
Evaporator feed = 60°F (28.1 Btu per lb)
Vapor pressure in evaporator = 25 lb abs
Pressure in evaporator coil = 106 lb abs
Enthalpy of extracted steam at 106 lb abs (see Fig. 142)
= 1260 Btu per lb

$$\text{Steam required} = \frac{780 \times (1160.6 - 28.1)}{(1260 - 302.8)} = 925 \text{ lb per hr}$$

(Assuming 100 per cent efficiency for evaporator)
Enthalpy of evaporator vapor at 25 lb abs = 1160.6 Btu per lb
Enthalpy of evaporator coil drain = 302.8 Btu per lb

Steam required by fuel oil heater:

Temperature of oil at burners 227°F
Temperature of oil at heater 232°F

Oil burned per hour = 5,400 × 0.60 = 3,240 lb
(Fuel consumption assumed = 0.60 lb per shp-hr)
Extracted steam at 106 lb abs required per hr

$$= \frac{3{,}240 \times (232 - 60°) \times 0.48}{1260 - 302.8} = 280 \text{ lb per hr}$$

Enthalpy of heater drain = 302.8 Btu per lb

Before calculations can be made for the amount of steam to be extracted from the main turbine for heating the steam to the desired temperatures in heaters *A* and *B* (see Fig. 141), the condition of the steam at the extraction points must be determined.

The following sectional efficiencies of the turbine have been assumed:

Throttle to bleeder point *A* (e_1) = 0.69
Bleeder point *A* to bleeder point *B* (e_2) = 0.80
Bleeder point *B* to L.P. exhaust (e_3) = 0.735

These efficiencies are internal efficiencies and are not comparable to our standard engine efficiency, which includes the gear and bearing losses. They should be used in connection with the turbine horsepower, which is $\frac{\text{shp}}{\text{gear efficiency}} = \frac{5{,}400}{0.97} = 5{,}570$.

H.P. inlet pressure P_1 = 450 lb abs and 710°F
Bleeder point *A* (P_a) = 106 lb abs
Bleeder point *B* (P_b) = 23 lb abs
L.P. exhaust (P_2) = 1.5 in. of mercury

From the Mollier diagram

$$\begin{aligned} h_1 &= 1365 \\ h_2 &= \underline{\ \ 895} \text{ (after isentropic expansion)} \\ h_1 - h_2 &= \ \ 470 \end{aligned}$$

Actual enthalpy of the steam entering the condenser is

$$h_2' = 1365 - (470 \times 0.772) = 1002 \text{ Btu per lb}$$

from the Mollier diagram; the percentage of moisture at 1½ in. abs pressure and 1002 Btu is 9.5.

A diagrammatic condition curve from the Mollier diagram is shown in Fig. 142. The enthalpy at the various pressures for an isentropic expansion from P_1 is designated by h without any primes. The enthalpy for the actual expansion in the turbine is designated by h'; and the enthalpy after an adiabatic expansion from the actual condition at the pressure at entrance of the section, by h''. The enthalpy values for the various points, as taken from the Mollier diagram, are shown on the diagram in Fig. 142.

Sectional efficiency e_2 from P_a to P_b, is

$$e_2 = \frac{h'_a - h'_b}{h'_a - h''_b}$$

The enthalpy converted into work $= h'_a - h'_b = e_2(h'_a - h''_b)$

$$h'_b = h'_a - e_2(h'_a - h''_b) = 1260 - 0.80(1260 - 1132) = 1158 \text{ Btu per lb}$$

$$h'_a = h_1 - e_1(h_1 - h_a) = 1365 - 0.69(1365 - 1213) = 1260 \text{ Btu per lb}$$

$$h'_2 = h'_b - e_3(h'_b - h''_2) = 1158 - 0.735(1158 - 945) = 1002 \text{ Btu per lb}$$

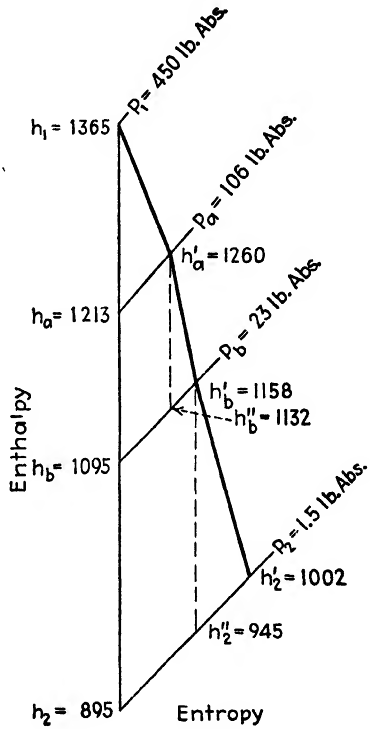


FIG. 142.

168. Calculation for Steam to Be Extracted from Turbine for Feed Heater A.—As shown on the diagram, part of the steam extracted from the turbine at 106 lb abs (point A) is used for operating the make-up evaporator and the fuel oil heater. This amount is (925 + 280) lb or 1,205/5,570 = 0.216 lb per turbine hp per hr.

If we let w_a equal the total steam per turbine horsepower per hour bled from the turbine at 106 lb abs, the steam going to heater A is $(w_a - 0.216)$ lb per turbine hp per hr.

Let w_x = steam required at the main turbine throttle per turbine horsepower per hour.

w_a = steam bled from the turbine per turbine horsepower per hour at bleeder point A.

The feed passing through heater A per hour is the total amount fed to the boiler and is equal to 5,570 w_x plus 1,280 lb of auxiliary steam, which is equal to $(w_x + 0.23)$ lb/turbine hp per hr.

The heat equation for the heat entering and leaving heater A is

$$(w_a - 0.216)(h'_a - h_{f_a}) + (w_x + 0.23)h_{f_3} = (w_x + 0.23)h_{f_4}$$

where h'_a = actual enthalpy of the steam extracted for heater A.
h_{f_a} = enthalpy of the liquid at pressure P_a (drain to heater B).
h_{f_3} = enthalpy of the liquid of the feed entering heater A (see Fig. 141).
h_{f_4} = enthalpy of the liquid of the feed leaving heater A.
0.216 = the extracted steam in pounds per horsepower per hour used by evaporator and fuel oil heater.

Substituting the known values in the preceding equation, we have

$$(w_a - 0.216)(1{,}260 - 302.8) + 203.3(w_x + 0.23) = 290.3(w_x + 0.23)$$

$$957.2w_a = 87w_x + 224$$

$$w_a = 0.091w_x + 0.235$$

169. Calculation for Steam to Be Extracted from Turbine for Direct-contact Heater B.—As noted in the diagram in Fig. 141, feed heater B, in addition to receiving extracted steam from the turbine, receives steam and hot water from the following sources:

Evaporator coil drain	= 925 lb per hr = 0.166 lb per turbine hp per hr
Evaporator vapor	= 780 lb per hr = 0.140 lb per turbine hp per hr
Fuel oil heater drain (assumed)	= 280 lb per hr = 0.050 lb per turbine hp per hr
Drain from heater A	= $(w_a - 0.216)$ lb per turbine hp per hr

The feed water entering heater B in line 2 (Fig. 141) at 100°F is $\left(w_x - w_a - w_b + \dfrac{500}{5{,}570}\right)$ lb per turbine hp per hr, and the water leaving heater B in feed line 3 is $(w_x + 0.23)$ lb per turbine horsepower per hr.

where $\dfrac{500}{5{,}570} = 0.09$ = condensate from the inter- and aftercondensers of the air ejector and from the gland steam condenser per turbine horsepower per hour.

w_b = steam extracted from the turbine for heater B in pounds per turbine horsepower per hour.

The heat equation for heater B is

(Heat extracted from turbine) + (heat from evaporator coil drain) + (vapor from evaporator) + (drain from fuel oil heater) + (drain from heater A) + (heat entering in feed line 2) = heat leaving in feed line 3

$$w_b h_b' + (0.166 \times 302.8) + (0.140 \times 1{,}160.6) + (0.05 \times 302.8) + (w_a - 0.216)h_{f_a} + (w_x - w_a - w_b + 0.09)h_{f_2} = (w_x + 0.23)h_{f_3}$$

Substituting the known values for h_b', h_{f_a}, h_{f_2}, h_{f_3}, and w_a (in terms of w_x), we have

$$w_b = 0.1045w_x - 0.161$$

170. Calculation for Steam Requirements.—From the general equation for steam consumption per shp per hour

$$w_{\text{actual}} = \frac{2{,}545}{\text{engine efficiency } (h_1 - h_2)} = \frac{2{,}545}{h_1 - h_2'}$$

We can set up an equation for the heat converted into useful work per turbine horsepower per hour in each section of the turbine as follows:

$$2{,}545 = w_x(h_1 - h_a') + (w_x - w_a)(h_a' - h_b') + (w_x - w_a - w_b)(h_b' - h_2')$$

Substituting, we have

$$2{,}545 = 105w_x + 102w_x - 9.3w_x - 24.0 + 156w_x - 14.2w_x - 37 - 16.2w_x + 25$$

Steam required at throttle,

w_x = 7.93 lb per turbine hp per hr

$w_a = (0.091 \times 7.93) + 0.23 = 0.953$ lb per turbine hp per hr

$w_b = (0.1045 \times 7.93) - 0.161 = 0.667$ lb per turbine hp per hr

Amount of steam to heater A = 0.953 − 0.216 = 0.737 lb per turbine hp per hr

Steam to condenser = 7.93 − 0.953 − 0.667 = 6.31 lb per turbine hp per hr

Total steam from boiler = 7.93 + 0.23 = 8.16 lb per turbine hp per hr

Total hourly steam consumption = 5,570 × 8.16 = 45,400 lb per hr

$$\text{Steam per shp-hr} = \frac{45{,}400}{5{,}400} = 8.41 \text{ lb}$$

Check calculation.

Heater A:

Heat received from extracted steam = 0.737 × (1,260 − 302.8)
= 709 Btu per turbine hp per hr

Feed through heater = 8.16 lb per turbine hp per hr

$$\text{Increase in enthalpy per pound} = \frac{709}{8.16} = 87 \text{ Btu}$$

This is the difference in enthalpy of the liquid between temperatures of 235 and 320°F (290.3 − 203.3 $\doteq$ 87).

Heater B: The total heat entering heater *B* per turbine horsepower per hour is made up as follows:

Extracted steam *B*, 0.667 × 1,158	=	772
Evaporator drain, 0.166 × 302.8	=	51
Evaporator vapor, 0.140 × 1,160.6	=	161
Drain from heater *A*, 0.737 × 302.8	=	225
Oil heater drain, 0.05 × 302.8	=	15
From main condenser, (6.31 + 0.09) × 68	=	436
Total	=	1660 Btu per shp per hr

$$\text{Enthalpy of water in heater} = \frac{1660}{8.16} = 203.2 \text{ Btu per lb}$$

This corresponds to a temperature of 235°F and checks the calculations.

If we allow 1000 Btu per lb of air-ejector and gland-ejector exhaust steam and gland leak-off, the rise in temperature in the ejector and gland steam condensers will be

$$\frac{(350 + 150 + 100) \times 1000}{(5{,}570 \times 6.40)} = 16.8°\text{F}$$

instead of 13°, as assumed in Fig. 141, and hence the steam to be extracted from the turbine for heater *B* will be slightly less than calculated if all the assumptions regarding ejector steam, make-up feed, etc., are correct.

171. Calculations for Fuel Consumption.—If we assume that the shp of the auxiliary generating set is 4 per cent of the power required for propulsion, the fuel consumption of the plant is

$$\frac{(7.93 \times 5{,}570)(1{,}365 - 290) + (0.23 \times 5{,}570)(1{,}213.5 - 290)}{18{,}500 \times 0.84}$$
$$+ (5{,}400 \times 0.04 \times 0.38) = 3{,}130 + 82 = 3{,}212 \text{ lb, of oil per hr}$$

or

$$f = \frac{7.93(1{,}365 - 290) + 0.23(1{,}213.5 - 290)}{18{,}500 \times 0.84 \times 0.97} + \frac{(5{,}400 \times 0.04) \times 0.38}{5{,}400} = 0.595 \text{ lb per shp per hr}$$

The required boiler heating surface will be

$$\frac{3{,}130}{0.50} = 6{,}260 \text{ sq ft}$$

The condenser calculations will be carried through in the same manner as illustrated for Case I. The steam entering the condenser will be

$$5{,}570 \times 6.31 = 35{,}200 \text{ lb per hr}$$

If a turbine-driven generator set is used, an estimate will have to be made for the shp and steam consumption of this set. When the steam consumption of the auxiliary generating set per horsepower of the main turbine (w_{aux}) has been determined, this can be added to the water entering heaters A and B, and heat equations may be set up as before.

Thaeler and MacMillan[1] have given the following equation for the kilowatt load of the auxiliary generating set:

$$\text{kw} = (c_1 \times N) + c_2 \times \text{shp}$$

where kw = average generator load in kilowatts.
N = number of persons on board.
shp = shaft horsepower of main propelling unit.
$c_1 = 0.75$.
$c_2 = 0.025$.

Applying this equation to the preceding example and assuming a crew of 40 men, we have

$$\begin{aligned}\text{kw} &= (0.75 \times 40) + (0.025 \times 5{,}400) = 165 \text{ kw}\\ &= 245 \text{ shp (generator eff.} = 90 \text{ per cent)}\end{aligned}$$

[1] See the footnote on p. 382.

A value of 216 bhp (4 per cent of 5,400) was used above for the horsepower of the Diesel generating set.

A detailed calculation of the horsepower of each auxiliary unit should now be carried through similar to those shown for Case I (see Table XXIV). This will give a more exact determination of the required service shp of the auxiliary generating set. The foregoing calculations can then be modified if the more exact determination of the horsepower of the auxiliary generating set differs materially from the preliminary estimate.

If in the preceding example a turbine-driven generating set had been used, operating under the same steam conditions and vacuum as the main turbine, the water rate would be 10.85 lb per shp-hr for an engine efficiency of 0.50. For a generating set of 216 shp, this would give

$$\frac{10.85 \times 216}{5,570} = 0.42 \text{ lb of steam per turbine shp-hr}$$

The air ejector for the auxiliary condenser would increase the steam consumption approximately 81 lb per hr. A new solution of the preceding equations with these values added will give a total fuel consumption for the ship of 0.62 lb per shp-hr.

172. Methods of Driving Auxiliary Generating Sets.—When the machinery installation consists of geared turbines and motor-driven auxiliaries, several methods of driving the auxiliary generating sets are in use, as pointed out in Art. 142. When the auxiliary generating set is driven by a steam turbine, in order to obtain a high over-all plant efficiency, the auxiliary turbines operate with the same initial steam pressure, superheat, and vacuum as the main propelling units. With motor-driven auxiliaries, auxiliary generating sets do not operate noncondensing, since steam extracted from the main turbine is usually used for series feed heating and no use could be made of any large amount of auxiliary exhaust.

The methods of supply power for the auxiliary motors are as follows:

A. Geared turbine operating parallel with the main unit.

B. Auxiliary generator driven off the main reduction gears.

C. Diesel engines.

D. A-c auxiliary motors receiving current from the main generator, with turbo-electric drive only (see Art. 96).

E. Geared turbine operating on steam extracted from main propelling unit.

The following table gives a comparison of the first four methods for a 20,000 shp passenger steamship and also for a motor ship.

TABLE XXV.—COMPARISON OF METHODS OF DRIVING AUXILIARY GENERATING SETS FOR A PASSENGER SHIP

Shp = 20,000. All auxiliaries motor-driven (steam used for air ejector and fuel oil heater)

Auxiliary load (output of motors) = 750 kw (1,000 shp)

No	*A*	*B*	*C*	*D*	
Type of propelling machinery.	Geared turbine	Geared turbine	Geared turbine	Turbo-electric	Diesel engines (direct connected)
Type of auxiliary drive	Turbine-driven generators	Generator driven off main gears	Diesel-driven generators	A-c motors driven by main generator	Diesel generating sets
1. Fuel for main engines per shp-hr (main turbines, air ejector, and fuel oil heater only), auxiliary motors excluded	0.557	0.557	0.557	0.598[b]	0.37
2. Auxiliary f = auxiliary fuel per hr ÷ shp output of auxiliary motors	1.00[c]	0.78[d]	0.533[e]	0.633[f]	0.533
3. Auxiliary f × cost of fuel in dollars per bbl	1.00	0.78	0.905	0.633	0.905
4. Relative standing of auxiliary drives	4	2	3	1	3
5. Fuel per shp-hr for all purposes	0.607[g]	0.597	0.585	0.630	0.396
6. Cost of fuel per hr.	$47.94	$47.06	$47.92	$49.60	$43.30

For geared-turbine installations the fuel per turbine horsepower per hour for main engines, air ejectors, and fuel oil heaters has been assumed as 0.54. The turbine horsepower is taken before the gears and is equal to shp ÷ gear efficiency.

Gear efficiency assumed = 97 per cent
Auxiliary generator efficiency assumed = 87 per cent
Average efficiency of d-c auxiliary motors assumed = 82 per cent
Average efficiency of a-c auxiliary motors assumed = 90 per cent
Generator output = 750 − 0.82 = 915 kw
Efficiency of generator in turbo-electric installation = 97 per cent; efficiency of motor = 93 per cent
Diesel fuel assumed = $1.70 per bbl
Bunker *C* fuel assumed = $1.00 per bbl

[a] $f = \frac{0.54}{0.97} = 0.557$ lb per shp-hr.

[b] $f = \frac{0.54}{0.97 \times 0.93} = 0.598.$

[c] Based on two 450-kw sets: $P_1 = 450$ lb; $t_1 = 750°F$; $P_2 = 28.5$ in. vacuum; $e_b = 85$ per cent; feed = 300°F; engine efficiency of auxiliary turbine = 52 per cent; $w_a = \frac{2,545}{0.52(1,390-907)} = 10.15$ lb per shp-hr; $f = \frac{10.15 \times (1,390 - 269)}{18,800 \times 0.85 \times 0.87 \times 0.82} = 1.00$ lb per shp-hr.

[d] $f = \frac{0.54}{0.87 \times 0.82 \times 0.97} = 0.78.$

[e] $f = \frac{0.38}{0.87 \times 0.82} = 0.533.$

[f] $f = \frac{0.54}{0.97 \times 0.90 \times 0.98} = 0.633$ (electric transmission losses = 2 per cent).

[g] $\frac{(20,000 \times 0.557) + (1,000 \times 1.00)}{20,000} = 0.607.$

The fuel consumptions given in line 5 of the preceding table shows that the Diesel generating set stands first. However, for the assumed fuel costs, line 6 shows that the ship with generating set driven off the main reduction gears is slightly cheaper to operate at sea. For the assumed conditions, the advantage of one drive over the other is very small for operation at full power at sea. The larger the horsepower of the generating sets, compared to the main turbine, the greater will be the influence of the generating-set fuel consumption on the total fuel consumption.

First costs and maintenance expenses, of course, must be given consideration along with fuel costs in any comparison of generator drives. When the generating set is driven off the main shaft (col. *B*), a steam stand-by set must be supplied to carry the auxiliary load when the main engine is shut down, during maneuvering, and in port. This, of course, increases the first cost of the installation.

For a cargo ship, the auxiliary load would be smaller than that used in Table XXV; hence the engine efficiency of the turbine-driven set would be smaller and its fuel per shp-hour considerably higher. On the other hand, the smaller horsepower of the auxiliary set would exert less influence on the total fuel per shp-hour (line 5). The fuel per shp-hour for all purposes for a cargo ship might thus be larger or smaller than that given in Table XXV.

A cargo ship with electric cargo winches would have a larger load on the auxiliary generating set in port than at sea, and here the advantages of the Diesel-driven generating set might make it superior to the turbine-driven set. It should be borne in mind that the fuel consumption of a Diesel engine is about the same for a small horsepower set as for a large horsepower set, whereas the steam consumption of a geared turbine is much higher for small horsepowers than for large.

Note: The arrangements used in Figs. 139 and 141 are not intended to show the merits of any particular piping layout but have been used only to make the illustrative calculations instructive and as simple as possible. A study of the feed heating diagrams in Chap. XI will show the numerous modifications in use today.

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