PAPERS ON ENGINEERING SUBJECTS.

THE INFLUENCE OF THE AUXILIARY ENGINE ON FUEL CONSUMPTION.

In a warship in times of peace, when a considerable proportion of time is spent in harbour and when the sea-going speed is usually low, the efficiency or otherwise of the auxiliary machinery exercises a considerable influence on the consumption of fuel. Its efficient maintenance and operation is consequently a matter of importance, second only in fact in a modern installation to the economical generation of steam at source. The principles of efficient maintenance are generally well appreciated, but a detailed consideration of the best operating conditions to meet specific speeds or conditions, viewed in the light of the steam consumption of the various auxiliaries, will be found to reveal in many cases sources of improvement.

Information as to the performances of auxiliary engines under different conditions of working is not usually available on board, and it is proposed therefore to consider this matter first, and to give some typical data which will afford at least a basis for an investigation in any case.

The auxiliary engines usually fitted may be broadly divided into two types, viz., fast-running reciprocating engines, either simple or compound, and relatively slow running reciprocating pumps. As an alternative to the first named the turbine is finding an increasing use, more particularly for electric generating purposes.

Economy in Theory and Practice.

In theory, the steam consumption per unit of power delivered by a reciprocating engine depends upon the initial steam pressure, the initial condition of the steam, the final pressure and the ratio of expansion. On this basis the consumption decreases as the initial pressure and the ratio of expansion are increased. In practice, while the same leading considerations hold, the results to be expected on theoretical grounds are not realised, mainly as a result of steam leakage and of condensation in the clearance spaces and cylinder, the broad effect of which is to cause heat to be carried through with the exhaust without doing useful work. These losses increase with the pressure range over which the engine works, and, for the same pressure range, decrease with the size of the engine and generally as the speed of revolution is increased. The condensation loss also increases as the ratio of expansion is increased. The expression " loss " is to be taken to mean the ratio between the weight of steam due to the loss and the total weight of steam consumed by the engine.

A lesser loss in the steam circuit arises from wire-drawing of the steam at entry to and discharge from the engine, the tendency of which is to lower the pressure range within which the engine works, and so leads to a greater consumption of steam. This influence may become most marked in the case of a fast-running engine remote from the source of steam supply and from the place at which the exhaust steam is finally put to further use. Broadly speaking, this loss decreases as the size of the engine is increased and increases as the speed of revolution is increased.

The remaining important loss is the frictional loss in the mechanical parts of the engine. For engines of equal power the mechanical efficiency is greater for fast-running engines than for slow. It is less for small engines than for large engines and it decreases as the output of a particular engine is reduced.

From the above considerations, the fact emerges that, other things being the same, a large engine is necessarily more economical than a small engine and a fast-running engine is more economical than a slow-running engine.

The loss due to condensation and leakage is greatly mitigated by providing stage expansion as a result of the lower temperature and pressure range to which the cylinder surfaces are exposed, and this much lesser loss permits of much higher ratios of expansion being advantageously employed than in simple engines and so to better efficiency, notwithstanding a somewhat augmented loss from such influences as wiredrawing.

In this connection it is of interest to note in passing that the development of the so-called "uniflow" engine, in which the exhaust passage is arranged at the middle of the length of the cylinder, thereby eliminating direct valve leakage from steam to exhaust and reducing condensation, has permitted the theoretical advantage attending much higher ratios of expansion being more nearly realised in simple engines. By a comparatively small increase in size and weight, an improved economy of the order of 30 per cent. can be obtained in engines of comparatively small power. The application of this principle to engines of small power is comparatively recent.

Effect of running at reduced output.—Consider now the effect on the steam consumption per unit power of running the engine at a lower power than that for which it is designed. Except in the special case of electric generating plant or of a turbo-driven pump on a constant pressure service, which will be separately considered, the reduced power output is attended by a reduced speed of revolution, and consequently the condensation and leakage losses are in general increased, while the mechanical efficiency is lower. In all cases, other than pumps which maintain a constant discharge pressure (feed pumps, for example) at all outputs, the reduction of power is attended by a lower mean pressure; consequently a lower initial pressure, and theoretically a lesser weight of steam admitted to the cylinder per horse-power. Notwithstanding this and some gain in economy from other causes, *e.g.*, more complete expansion and lesser wiredrawing, the influence of leakage and of the speed factor on the condensation losses, together with the reduction in mechanical efficiency, combine to increase the consumption of steam per unit power delivered as the output of a specific engine is reduced, until at one quarter of full output, for example, the steam consumption per B.H.P. of a simple fast-running engine is increased by from 40 to 50 per cent. This relation applies approximately for any conditions of back pressure and both for constant speed engines and for engines in which the revolutions fall with the power within the limits generally experienced for such auxiliaries on service in warships. Actually the steam consumption per I.H.P. of a constant speed engine at low outputs does not fall off so rapidly as in an engine in which the revolutions decrease also with the output, but the mechanical efficiency is lower, which sets off this gain, and on the whole there is but little difference in the steam rate per B.H.P., as already observed.

The steam consumption of direct acting pumps delivering against a constant pressure does not in practice fall off so rapidly as would be expected in comparison with the fast-running engines, considering that the reduction in speed in relation to reduction of output is so much greater. The very late cut off entailed by the maintenance of a steady discharge pressure leads to a rather higher consumption at full output since the expansive property of the steam can only be used to a relatively small extent. It appears, therefore, that the condensation loss in engines of this type is less than in the fast-running reciprocating engine of similar power as a result of the late cut off and low temperature range, in conjunction with the larger sizes of the steam cylinder. Under these conditions it is probable that piston speed is not an important influence on the condensation.

On this assumption the falling off in economy at reduced output would be due mainly to the *rate* of steam leakage, which, in view of the sensibly constant steam pressures required to deal with the constant pressure of discharge, remains practically the same at all outputs, and therefore exercises a greater influence on the steam consumption per H.P. hour at the lower outputs.

The steam rate for a compound engine of the constant speed type as fitted for electric generation falls off at a greater rate than the simple engine, mainly as a result of the decrease in the ratio of expansion (generally the expansion in a simple engine of the types discussed is incomplete at full output) and at $\frac{1}{4}$ power averages about 70 per cent. increase in the steam consumption per B.H.P.

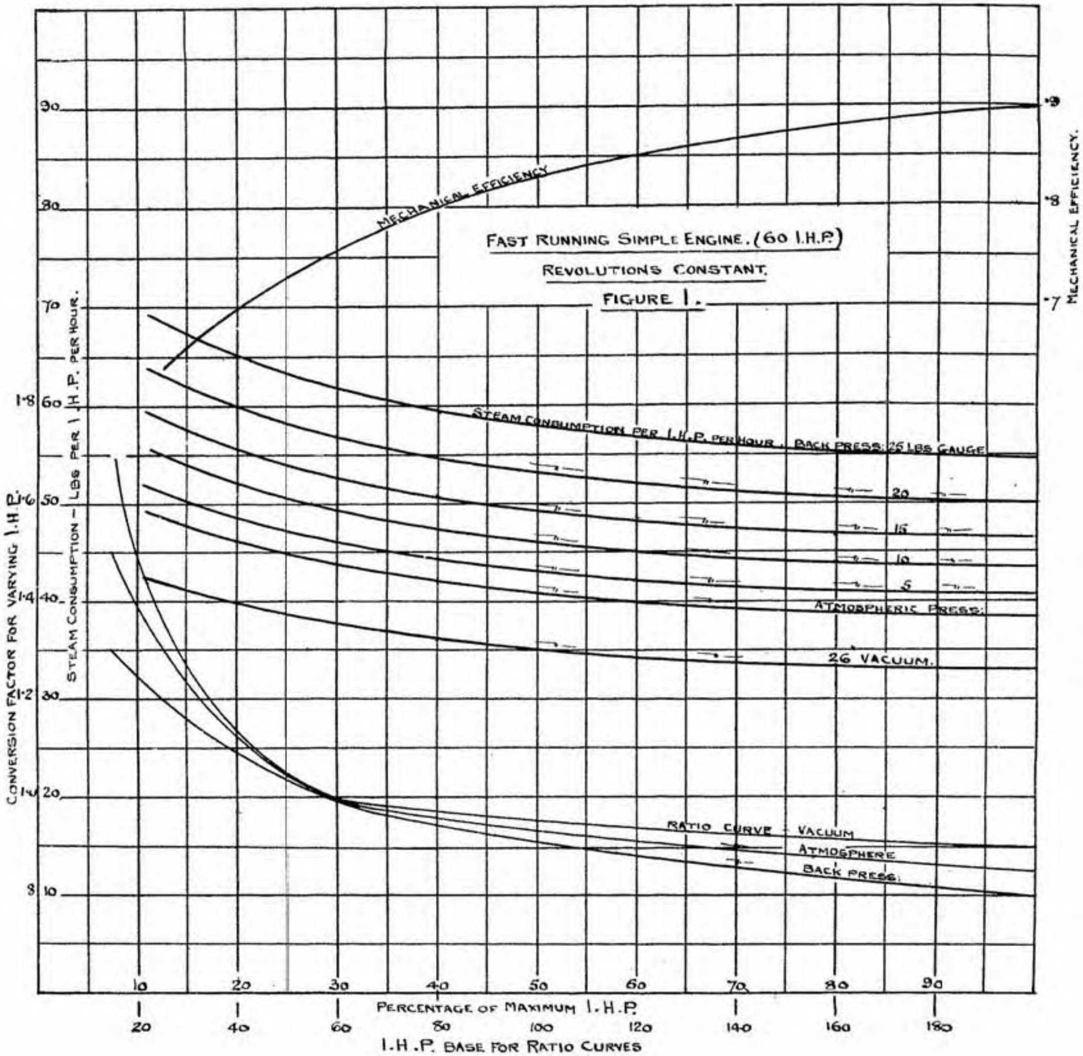
Turbine Auxiliaries.—The turbine auxiliary requires separate consideration. This type is, from its uniflow nature, not exposed to the effect of condensation, and it can therefore employ a higher ratio of expansion without incurring the heavy losses that attend a reciprocating engine. To realise the advantages of the higher expansion ratios, it is necessary, however, that the blade speeds shall be consonant, according to the type of the turbine, with the high steam velocities corresponding to the high expansion ratios. In turbines of moderate size and large size this essential condition can be met, in whole or in part, by the distribution of the pressure drop over a number of stages. But in the smaller sizes, such as are employed for the fast-running services in war ships, this requirement cannot usually be even approximately attained, unless the size and weight are made unduly great in relation to the power to be yielded. Even so, the other sources of losses, such as steam leakage and steam friction, &c., in the rotating parts, increase at a rapidly increasing rate as a turbine is made for smaller outputs.

For these reasons the turbine in its application to auxiliary services in small units cannot compete in steam consumption with the reciprocating engine. The comparison becomes even more unfavourable when working under back pressure conditions; but units of 200 K.W.s, such as are employed in the Capital Ships' turbo-generators, when fitted in association with their own condensing plant, are more economical than compound reciprocating engines.

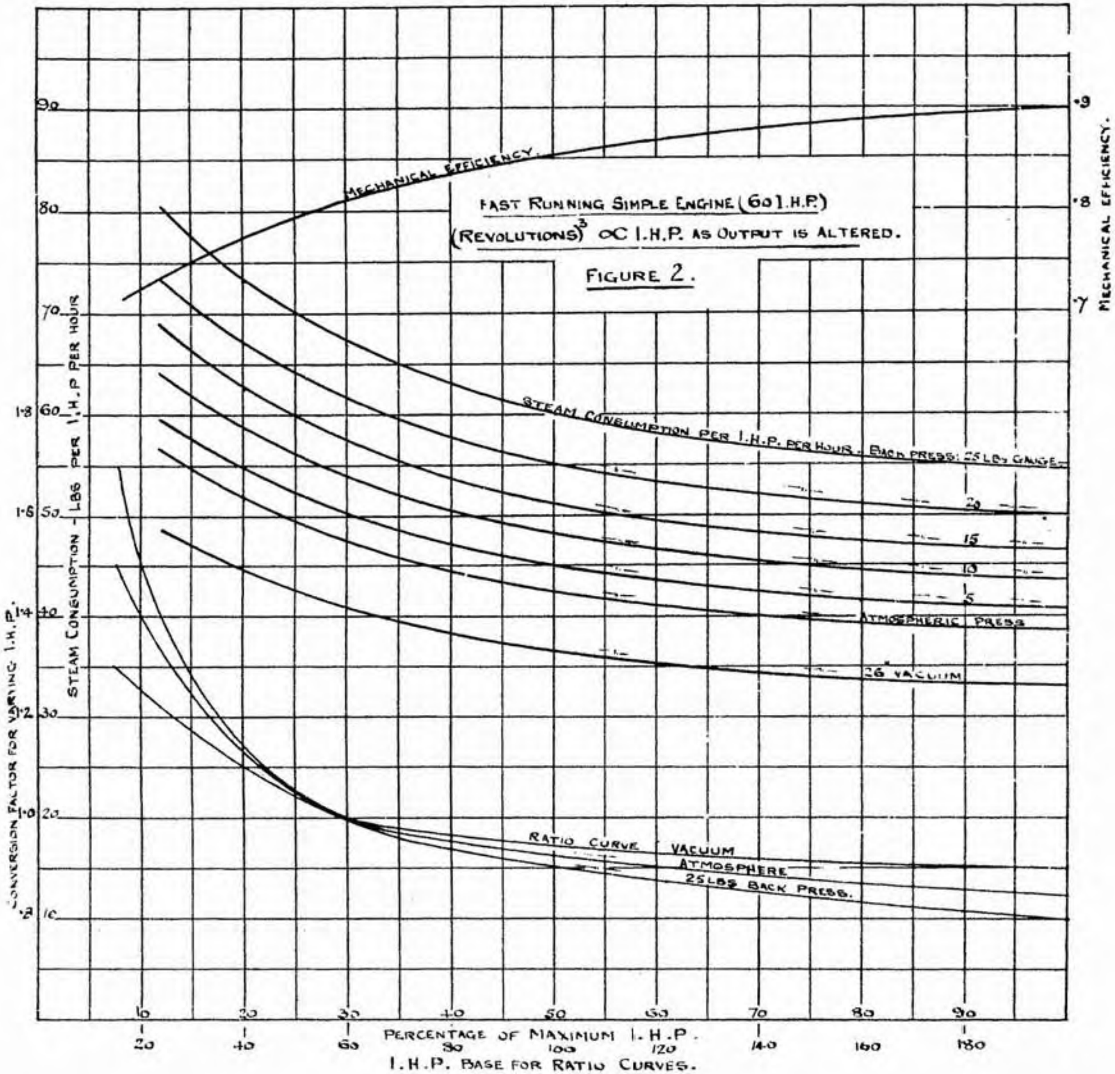
In the case of turbine-driven auxiliaries, a reduction of output is attended by an increase in the steam rate. The reduction may be attained by the throttling of the steam or by nozzle control. In the former case the pressure range is definitely lowered, leading directly to a higher consumption, and in either case the designed distribution of energy in the stages is altered, leading to additional shock losses due to unsuitable blade angles, which are generally greater when the reduction of output is attended also by a reduction in speed. On the other hand, in the case of a constant speed turbine, the power required to overcome the rotational losses (which varies, amongst other factors, with the cube of the peripheral speed) remains sensibly constant at all powers and may even increase in nozzle controlled turbines as a result of partial admission. Similarly, in a reaction turbine the steam leakage remains sensibly constant at all outputs when the initial pressure remains the same. As in the case of a reciprocating engine, the mechanical efficiency is lowered as the output of a turbine is reduced, although this factor is of less importance.

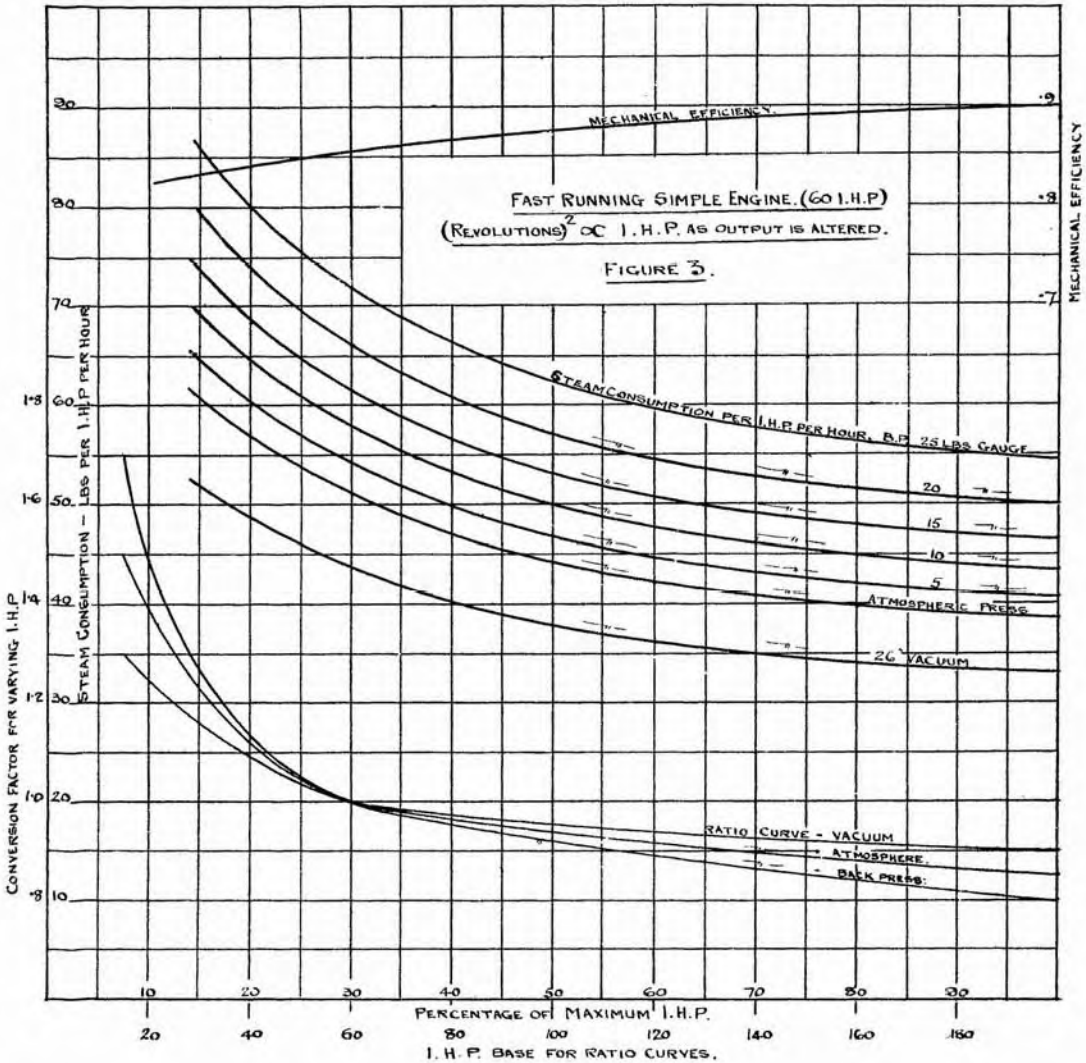
The shock losses exercise on the whole the greatest influence on the economy at reduced outputs, and a turbine such as is fitted for electric generation does not therefore fall off so rapidly as one which is fitted for a variable speed service.

Effect on Consumption of Back Pressure.—Another influence of the steam consumption to be considered is that arising from changing back pressure. The effect of an increase in the back pressure against which an engine works leads, of course, to a higher initial pressure if the same power is to be attained and so to a greater weight of steam admitted to the cylinder per stroke. The steam consumption does not, however, increase in the same ratio as the weight of steam, as would be indicated by theoretical considerations, but rather less, since the temperature



FAST RUNNING SIMPLE ENGINE. (60 I.H.P.)
REVOLUTIONS CONSTANT.
FIGURE 1.
MECHANICAL EFFICIENCY
STEAM CONSUMPTION PER I.H.P. PER HOUR. BACK PRESS: 25 LBS GAUGE
20
15
10
5
ATMOSPHERIC PRESS:
26 VACUUM.
RATIO CURVE - VACUUM
ATMOSPHERE
BACK PRESS.
CONVERSION FACTOR FOR VARYING I.H.P.
STEAM CONSUMPTION - LBS PER I.H.P. PER HOUR.
MECHANICAL EFFICIENCY.
PERCENTAGE OF MAXIMUM I.H.P.
I.H.P. BASE FOR RATIO CURVES
90
80
70
60
50
40
30
20
10
1·8
1·6
1·4
1·2
1·0
·8
·9
·8
·7
10 20 30 40 50 60 70 80 90
20 40 60 80 100 120 140 160 180


FAST RUNNING SIMPLE ENGINE (60 I.H.P.)
(REVOLUTIONS)^3 ∝ I.H.P. AS OUTPUT IS ALTERED.
FIGURE 2.
MECHANICAL EFFICIENCY.
STEAM CONSUMPTION PER I.H.P. PER HOUR. BACK PRESS: 25 LBS GAUGE.
20
15
10
5
ATMOSPHERIC PRESS
26" VACUUM
RATIO CURVE VACUUM
ATMOSPHERE
25 LBS BACK PRESS.
STEAM CONSUMPTION - LBS PER I.H.P PER HOUR
CONVERSION FACTOR FOR VARYING I.H.P.
MECHANICAL EFFICIENCY.
PERCENTAGE OF MAXIMUM I.H.P.
I.H.P. BASE FOR RATIO CURVES.



FAST RUNNING SIMPLE ENGINE. (60 I.H.P)
(REVOLUTIONS)² ∝ I.H.P. AS OUTPUT IS ALTERED.
FIGURE 3.
MECHANICAL EFFICIENCY
STEAM CONSUMPTION PER I.H.P. PER HOUR, B.P. 25 LBS GAUGE
20
15
10
5
ATMOSPHERIC PRESS
26" VACUUM
RATIO CURVE - VACUUM
ATMOSPHERE
BACK PRESS.
CONVERSION FACTOR FOR VARYING I.H.P
STEAM CONSUMPTION - LBS PER I.H.P PER HOUR
MECHANICAL EFFICIENCY
PERCENTAGE OF MAXIMUM I.H.P.
I.H.P. BASE FOR RATIO CURVES.

range is somewhat reduced and condensation is, therefore, less. So, conversely, the effect of employing a vacuum in a reciprocating engine does not realise the gain that would at first sight be expected. Other considerations than those mentioned enter into this question, wiredrawing losses for example, but it will be sufficient for the purpose of these notes to indicate only the most important factors.

For turbines of any size, the economy falls off at a much greater rate than the reciprocating engine when passing from vacuum conditions to back-pressure conditions, and particularly at low outputs. This follows partly from the unsuitability of the areas for the passage of steam, but mainly from the relatively greater effect of leakage losses and rotational losses occurring at the high pressure end of the turbine, which exercises a considerable influence under back-pressure conditions.

Curves of Steam Consumption per I.H.P.—Figures 1, 2 and 3 represent the steam consumption per I.H.P. of simple engines at varying proportions of their full output and at varying back-pressures. They show the results for a fast-running engine of about 60 I.H.P., and curves are also embodied giving the approximate differences in consumption which might be expected from engines of smaller and larger sizes, but designed for the same speeds, pressure and cut off. For example, referring to the lower curves on Figure 2, an engine of 150 I.H.P. would require ·88 times the amount of steam per I.H.P. shown on the upper curves for the 60 I.H.P. engine, *i.e.*, under atmospheric back-pressure, $\cdot 88 \times 39 \cdot 7 = 34 \cdot 9$ lbs. per I.H.P. per hour. Similarly, an engine of 15 I.H.P. would require 1·24 times, *i.e.*, $1 \cdot 24 \times 54 \cdot 3 = 67 \cdot 5$ lbs. per I.H.P. per hour.

It will be appreciated that the figures necessarily represent average results and that fairly wide variations may be found for engines by different makers. The cut off is about ·65 and the engines are designed to give their full output for initial pressures of about 170 lbs. gauge; the consumptions are given on the assumption that at all outputs a boiler pressure of 225 lbs. is available and the lower pressure required to operate the engine is obtained by throttling.

A curve showing the approximate mechanical efficiency at varying outputs for the 60 I.H.P. engine is included, from which the steam rate per B.H.P. can be obtained. The mechanical efficiency would be rather better for engines of larger size and *vice versa ;* thus the mechanical efficiency of a 300 I.H.P. fast-running engine would be about ·92 and a 15 I.H.P. engine about ·87.

These curves are applicable to the engines employed on such services as condenser circulating engines and fan engines.

The steam consumption for other auxiliaries is more conveniently given on an overall basis and will be referred to in the following section.

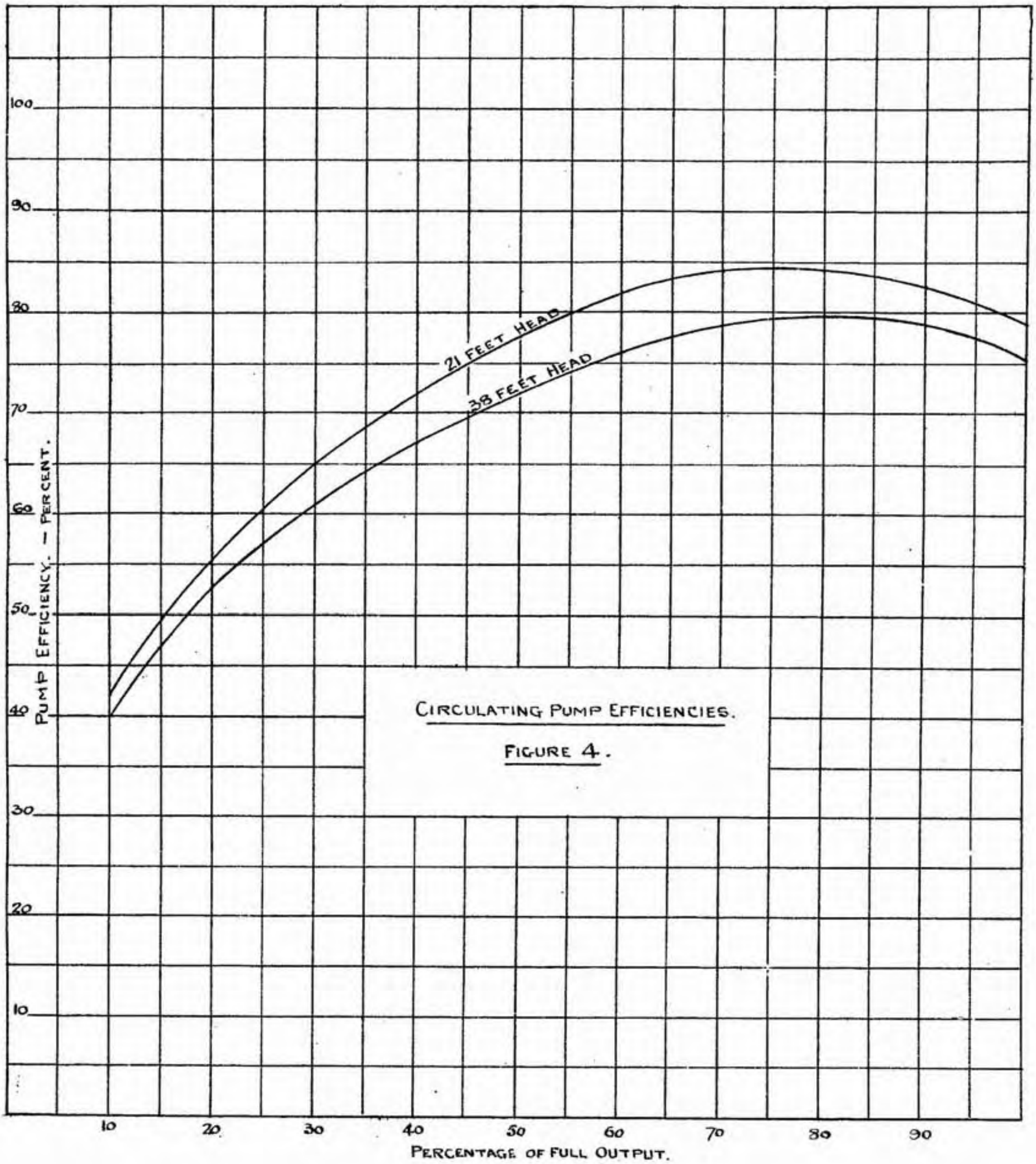
EFFICIENCY OF PUMP OR OTHER DRIVEN SERVICE.

In addition to the performance of the steam end, it is necessary, in order to find the steam required for a particular auxiliary service, to take into account the efficiency of the pump or other agent which the engine actuates and also to the conditions under which the pump works in respect to external influences, that may affect either the required output or the efficiency.

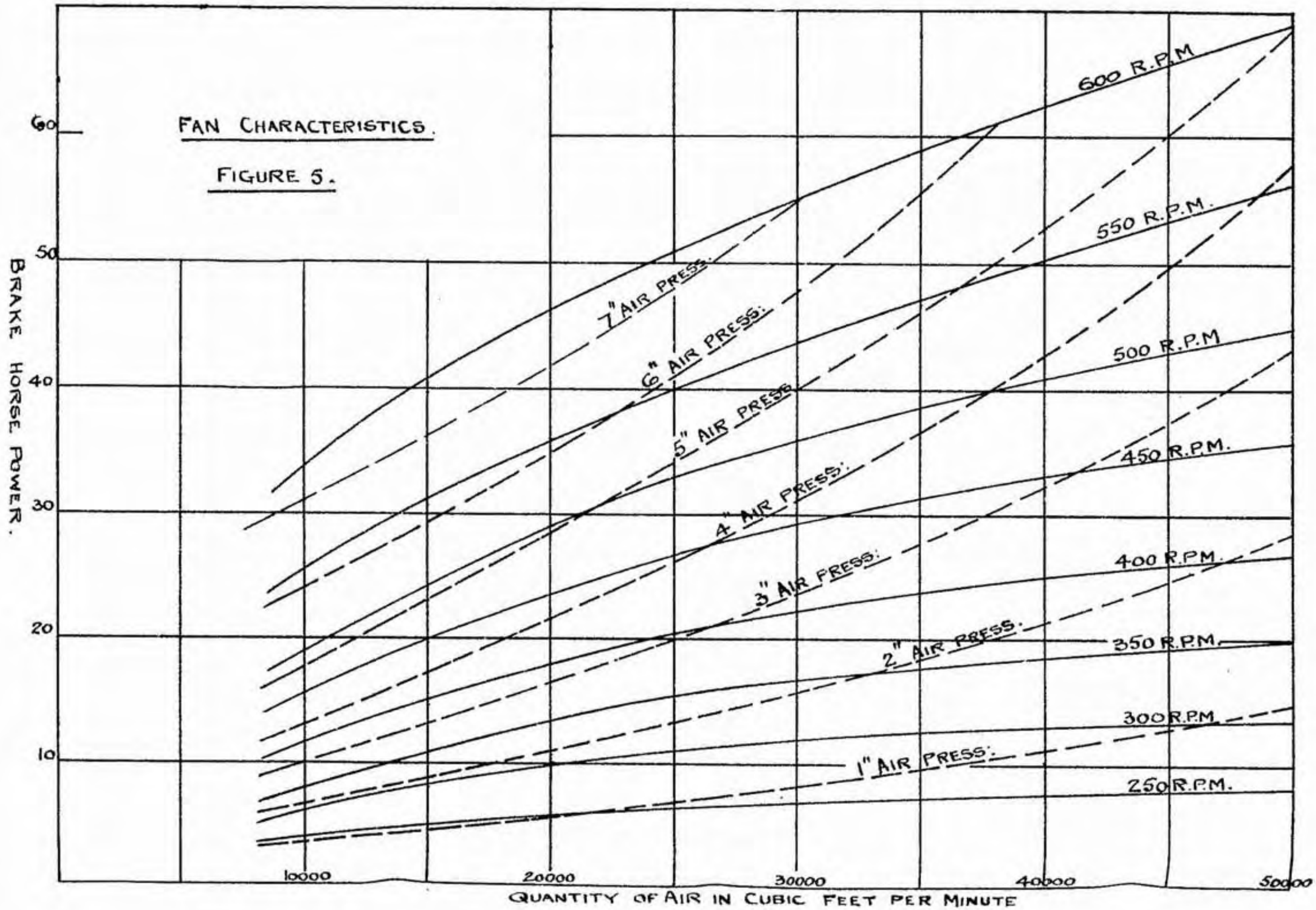
Consider, first, a pump of the impeller type. The impeller is designed to deliver a certain quantity of fluid against a certain head, when running at certain revolutions, and the blade angles are determined in such a way that the minimum amount of shock, and therefore the maximum efficiency, arises when the above-named variables are in agreement with the design figures. As a rule, the pumps are designed to give the best efficiency at an output somewhat below the full output. A reduction in output is usually attained by reducing the speed of revolution and this is attended by a reduction in the resistance head; under these conditions the design relations between these variables on which the blade angles were based will not be realised and the efficiency of the pump will fall off. Other influences affect the efficiency, as, for example, frictional losses in the blade and pump channels, leakage losses between the disc and casing, &c., and, while some of these in general tend to improve the efficiency at low outputs, the effect of the inaccurate blade angles is overwhelming and on the whole, therefore, the efficiency of the pump falls off as the output is reduced.

Two cases occur in service installations, viz., those in which the external conditions remain the same, as in the case of a circulating pump employed on a condenser, and those in which the external conditions vary, as in the case of a fan engine employed for forced draught, in which the area of opening through which the delivery is finally passed may be varied (different numbers of boilers and burners employed).

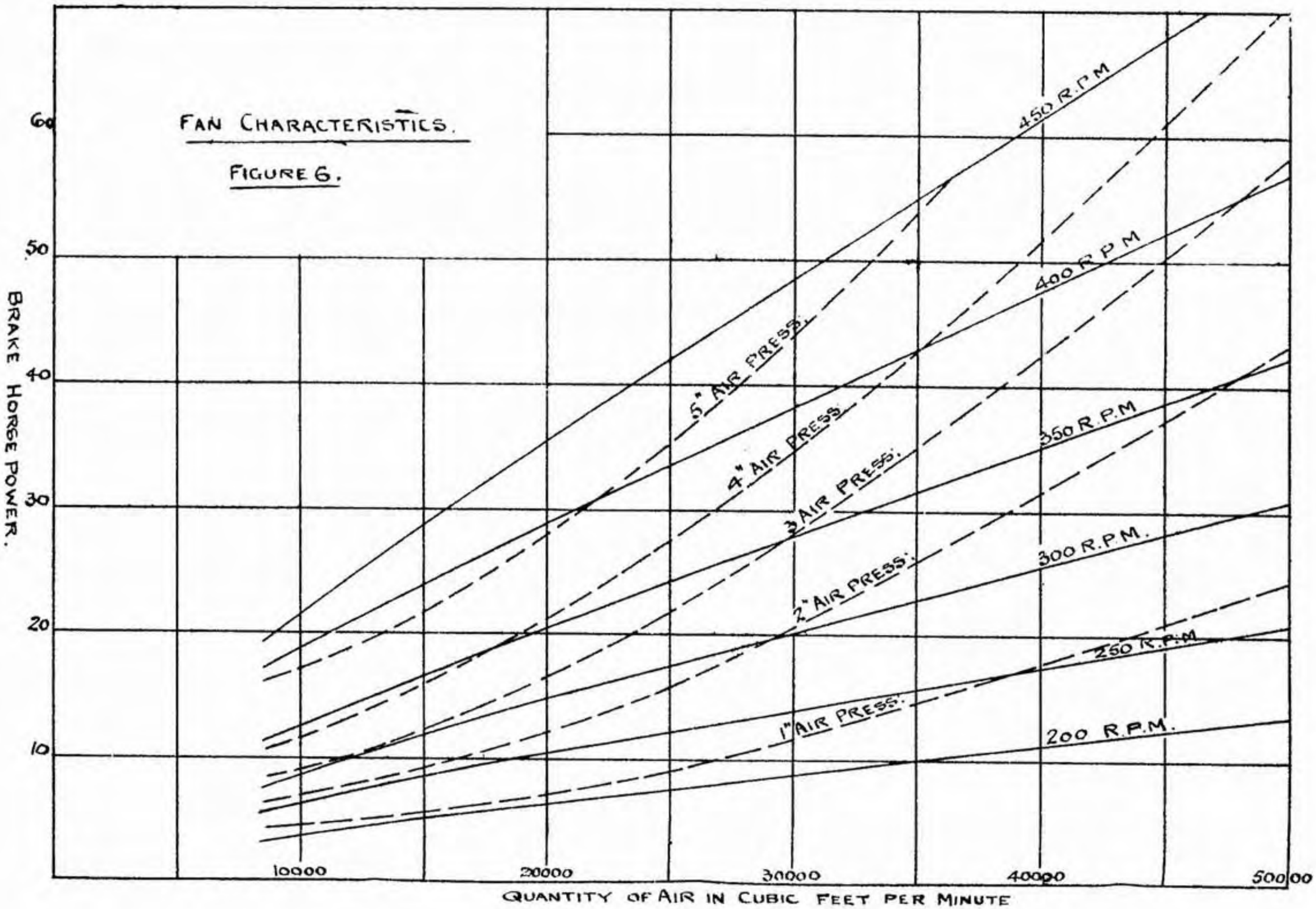
Circulating Pumps.—In this case the output of the pump may be taken to vary roughly as the speed of revolution, and the resistance head in the condenser to vary as the square of the output (*i.e.*, the square of the velocity of flow). Thus, the water horse-power of the pump will vary approximately as the cube of the revolutions and as the cube of the quantity of water delivered. So, at half power of the main engines, the power required from the circulating pump would be one-eighth of that which is required for full power of the main engines. This relation cannot be closely pressed to very low outputs of the main engines because the amount of steam to be condensed becomes higher per unit of power developed by the main engines and the very low velocity of flow through the condenser attending a low output from the circulating pump would lead to a lesser efficiency of the heat transmission in the condensers; this fact entails a greater amount



CIRCULATING PUMP EFFICIENCIES.
FIGURE 4.
21 FEET HEAD
38 FEET HEAD
PUMP EFFICIENCY. - PERCENT.
PERCENTAGE OF FULL OUTPUT.
100
90
80
70
60
50
40
30
20
10
10
20
30
40
50
60
70
80
90



FAN CHARACTERISTICS.
FIGURE 5.
BRAKE HORSE POWER.
60
50
40
30
20
10
10000
20000
30000
40000
50000
QUANTITY OF AIR IN CUBIC FEET PER MINUTE
600 R.P.M
550 R.P.M.
500 R.P.M
450 R.P.M.
400 R.P.M.
350 R.P.M.
300 R.P.M.
250 R.P.M.
7" AIR PRESS.
6" AIR PRESS.
5" AIR PRESS.
4" AIR PRESS.
3" AIR PRESS.
2" AIR PRESS.
1" AIR PRESS.


FAN CHARACTERISTICS.
FIGURE 6.
BRAKE HORSE POWER.
60
50
40
30
20
10
10000
20000
30000
40000
50000
QUANTITY OF AIR IN CUBIC FEET PER MINUTE
450 R.P.M
400 R.P.M
350 R.P.M
300 R.P.M.
250 R.P.M
200 R.P.M.
5" AIR PRESS.
4" AIR PRESS.
3" AIR PRESS.
2" AIR PRESS.
1" AIR PRESS.

of circulating water per pound of steam discharged from the main engines.

Typical efficiency curves of circulating pumps at varying outputs are shown in Figure 4. The curves give average results for a double flow condenser and single flow condenser respectively and represent the relation between the water horse-power delivered by the pump and the brake horse-power required to drive the pump, the corresponding resistance head on the pumps at full output being 38 feet and 21 feet.

A special case of the circulating pump is afforded by a rotary feed pump, whether driven by a turbine or motor, employed for boiler feeding. Here, the discharge pressure is sensibly constant at all outputs of the pump and the speed of revolution therefore remains constant. So the rotational and leakage losses remain sensibly constant and, taken in conjunction with the unsuitable blade angles entailed by the lesser speed of flow through the pump, lead to a very low pump efficiency at low outputs. In the limit, when the discharge is reduced to the minimum possible, say 5 per cent., the steam consumption of a turbo combination would be of the order of 60 per cent. of that when the pump was yielding its full output.

These remarks on the falling off in the efficiency of a constant speed rotary pump at low outputs have a bearing on the question of whether it is of advantage to employ motor-driven rotary pumps for a continuous low output requirement, such as a sanitary pump, rather than to employ a steam pump, keeping in view also the fact that the efficiency of the motor is also reduced.

Boiler Room Fans.—The changing conditions to be met on service by fans can be best expressed by characteristic curves, examples of which, suitable for air pressures up to 7 in. and 5 in. respectively, are shown in Figures 5 and 6. From these curves, when the quantity delivered and the air pressure is known, together with the speed of revolution, the B.H.P. required can be obtained. These curves, which are typical rather than representing any particular design, may at first sight appear rather complicated, but they are based on somewhat simple relations, qualified by experimental observations. A brief explanation of the underlying principles may be of assistance.

If a fan be allowed to deliver into a chamber closed except for one orifice, and the area of this orifice be kept constant, the following relations will hold approximately, provided the size of the chamber is sufficiently large in relation to the fan to ensure that all the air is brought to rest :—

(1) Air pressure is proportional to the square of the peripheral speed.

(2) Quantity of air delivered is proportional to the peripheral speed.

(3) The power delivered to the fan is proportional to the cube of the peripheral speed.

If the size of the orifice be altered the relations will maintain the same character but the constants will be different. That is to say, for one particular orifice, the air pressure will be expressed by a formula $h=C_1 R^2$, C_1 being a constant, h the air pressure, and R the revolutions per minute; while for another orifice the constant in this formula will have a different value, C_2 say. Similarly for the other two relations given under (2) and (3), the constants in their equations would change as the area of the orifice was altered.

The air horse-power required to deliver a quantity of air Q cubic feet per minute against an air pressure equivalent to h inches of water is given by :—

$$\frac{Q\ h\ 5\cdot 2}{33000}$$

This represents the actual power delivered by the fan, and the difference between it and the power delivered to the fan, *i.e.*, the B.H.P., is made up of losses in the fan such as friction, eddying, and leakage. The fan efficiency is the ratio between the two quantities and, so far as the relations given under (1), (2) and (3) are valid, it follows from them that this efficiency is constant for any particular orifice and is independent of the number of revolutions at which the fan is run. At any one speed of revolution there is one orifice for which the efficiency is a maximum.

In their practical application to boilers the same general relations hold and an analogy with the changing exit orifice holds, this being represented by the changing resistance to flow in the air and furnace circuit and by the varying area of the orifices in part of the circuit (*e.g.*, number of burners or number of boilers in use).

The character of the curves expressing the relations (1), (2) and (3) is shown in Fig. (7(*a*)) for one specific orifice in the case of a particular fan. The effect of variations in the area of the orifice (or, what amounts to the same thing in an actual installation, varying the resistance in the circuit, whether it be the result of a changing orifice or a changing speed of flow, or again a combination of both) is shown in Figures 7 (*b*),(*c*) and (*d*). From Figures 7 (*b*) and (*c*), taking one particular speed of revolution at a time, say R, the relation between Q and h can be plotted as shown in Figure 8.

From the values for the power P delivered to the fan as given in Figure 7 (*d*), and the relations between Q and h from Figure 8, an efficiency curve can be drawn and this has also been indicated on Figure 8. The form of this curve in an actual installation depends upon the design of the fan, *i.e.*, upon the number, form and disposition of blades, ratio between external diameter and eye, &c., and upon the environment in which the fan works, including such influences as size and form of downtakes, leakage, effect of natural funnel draught, eddying effects and possibly interference from other fans.

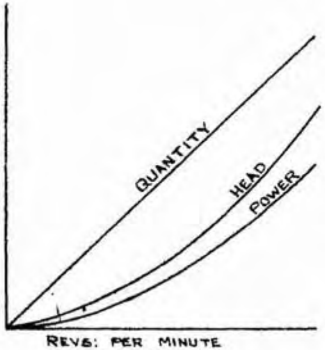


FIGURE 7(a).

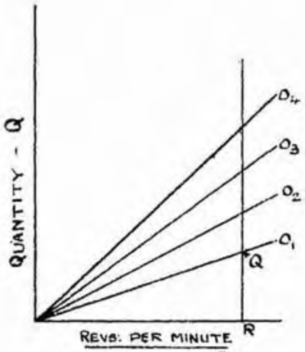


FIGURE 7(b)

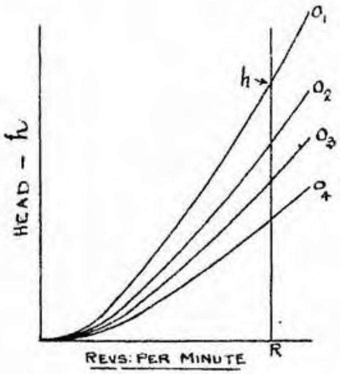


FIGURE 7(c)

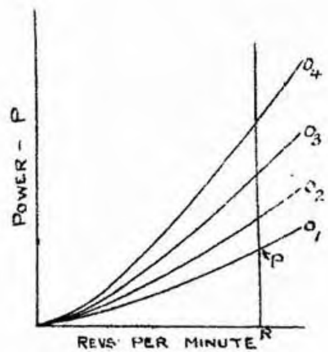


FIGURE 7(d)

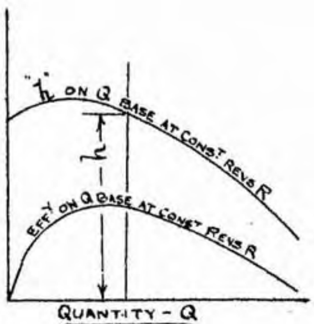


FIGURE 8

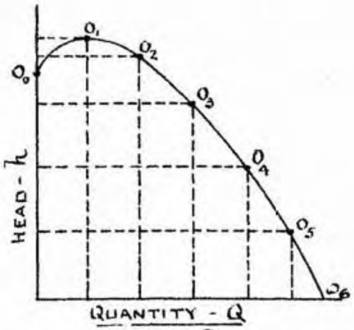


FIGURE 9.

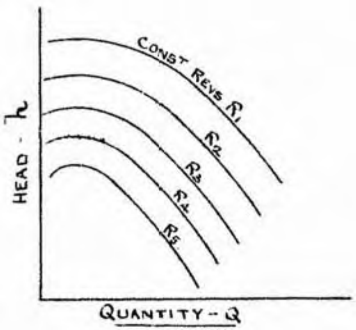
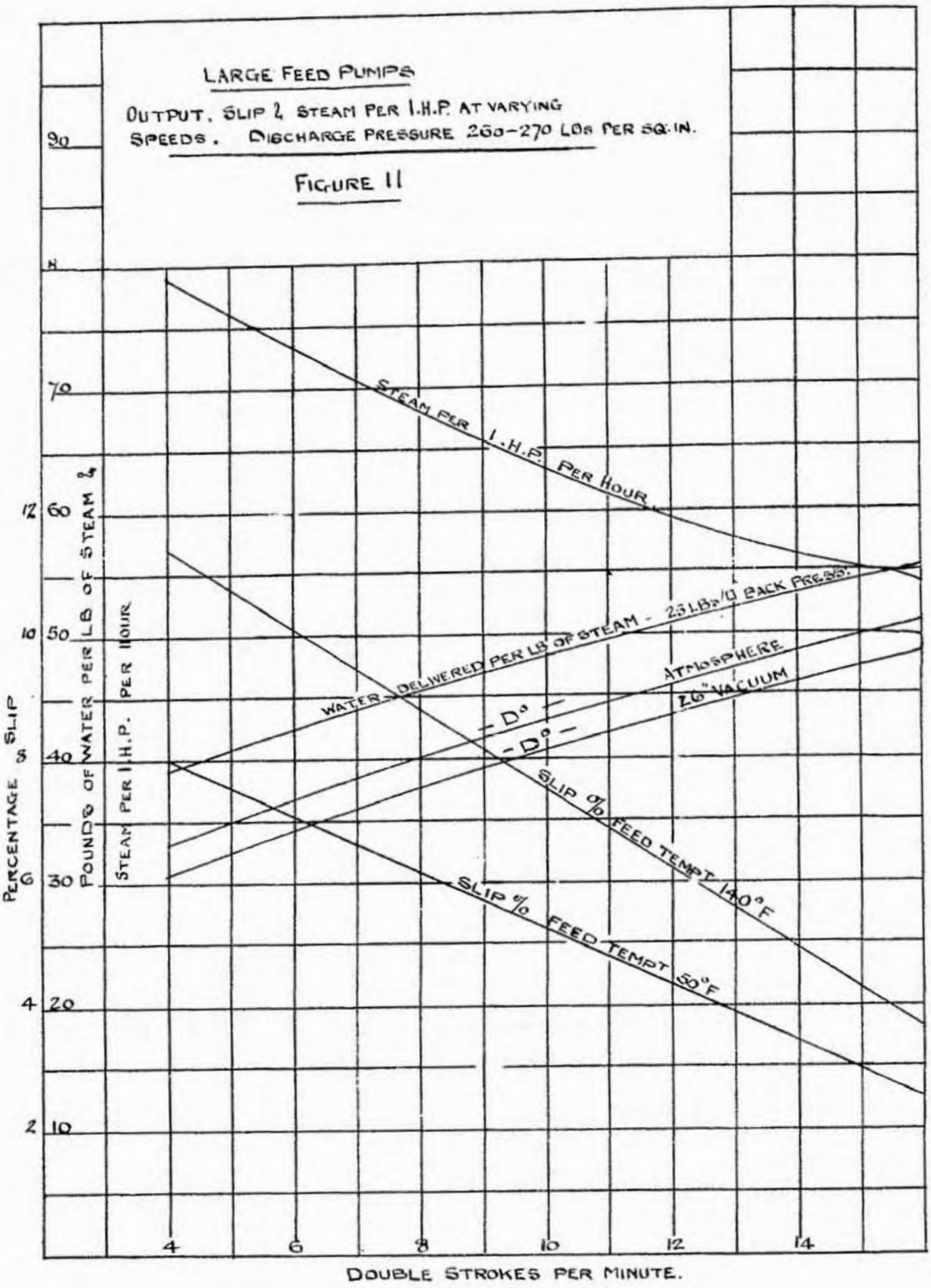


FIGURE 10.

FAN CHARACTERISTICS.


LARGE FEED PUMPS
OUTPUT, SLIP & STEAM PER I.H.P. AT VARYING SPEEDS. DISCHARGE PRESSURE 260-270 LBs PER SQ. IN.
FIGURE 11
STEAM PER I.H.P. PER HOUR
WATER DELIVERED PER LB OF STEAM - 25 LBs/□ BACK PRESS.
- D° - ATMOSPHERE
- D° - 26" VACUUM
SLIP % FEED TEMPT 140°F
SLIP % FEED TEMPT 50°F
PERCENTAGE SLIP
POUNDS OF WATER PER LB OF STEAM &
STEAM PER I.H.P. PER HOUR
DOUBLE STROKES PER MINUTE.
4
6
8
10
12
14
2
4
6
8
10
12
10
20
30
40
50
60
70
90

ERRATA.

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Figure **11,** ***Page*** **11.**

Upper curve showing water delivered per lb. of steam should be marked 26″ vacuum instead of 25 lbs. per square inch, and lower curve should be marked 25 lbs. per square inch instead of 26″ vacuum.

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Figure 9 shows the Qh curve for one speed of revolution extended each way to extreme conditions. At the point O_0 the area of the orifice is zero and the discharge is zero; it represents churning conditions. Points O_1, O_2 represent gradually increasing orifice areas until at O_6 the room is entirely open, the head is zero and the discharge is a maximum. This curve indicates the trend of the relation between Q and h in an oil-fired boiler room when an increasing number of sprayers is put into use, the fan being meantime run at a steady speed.

Repeating the given procedure for other speeds of revolution, the Qh curves shown in Figure 10 can be obtained.

Applying also to other speeds of revolution, the method given for obtaining the efficiency, the horse-power required to be delivered to the fan under the varying conditions can be found. From such data the characteristic curves given in figs. 5 and 6 are constructed, enabling all the essential data to be shown in a convenient form. It is to be specially noted that the characteristic curves represent typical results as taken during shop trials, and are not therefore applicable to ships' installations without some correction. In a reasonably air-tight installation on board in which there is but little interference, it would be sufficiently accurate to correct for the suction draught in the downtake in the vicinity of the fan eye, this being added to the water gauge reading in the stokehold to give the effective head produced by the fan. This correction can be ascertained by water pressure gauges. With CO_2 recorders or other means of finding the quantities of air delivered per lb. of fuel, trials to ascertain the characteristics of fans in actual installations can be carried out and these would afford a more satisfactory basis than the typical curves given, which, although probably reasonably accurate so far as the fans go, may be wide of the mark in practice as a result of the variations due to their environment on board.

This data, however obtained, in conjunction with the curves showing steam consumption per I.H.P., affords information which enables the most profitable means of operating the fans to be determined for intermediate ship's speeds. Generally speaking, the object to be aimed at is to deliver the necessary quantity of air required for the combustion of the fuel at as small a steam consumption on the part of the fans as possible, consistent with satisfactory combustion of the fuel and the working of the boilers at particular outputs which experience shows to be the most efficient from the evaporation point of view.

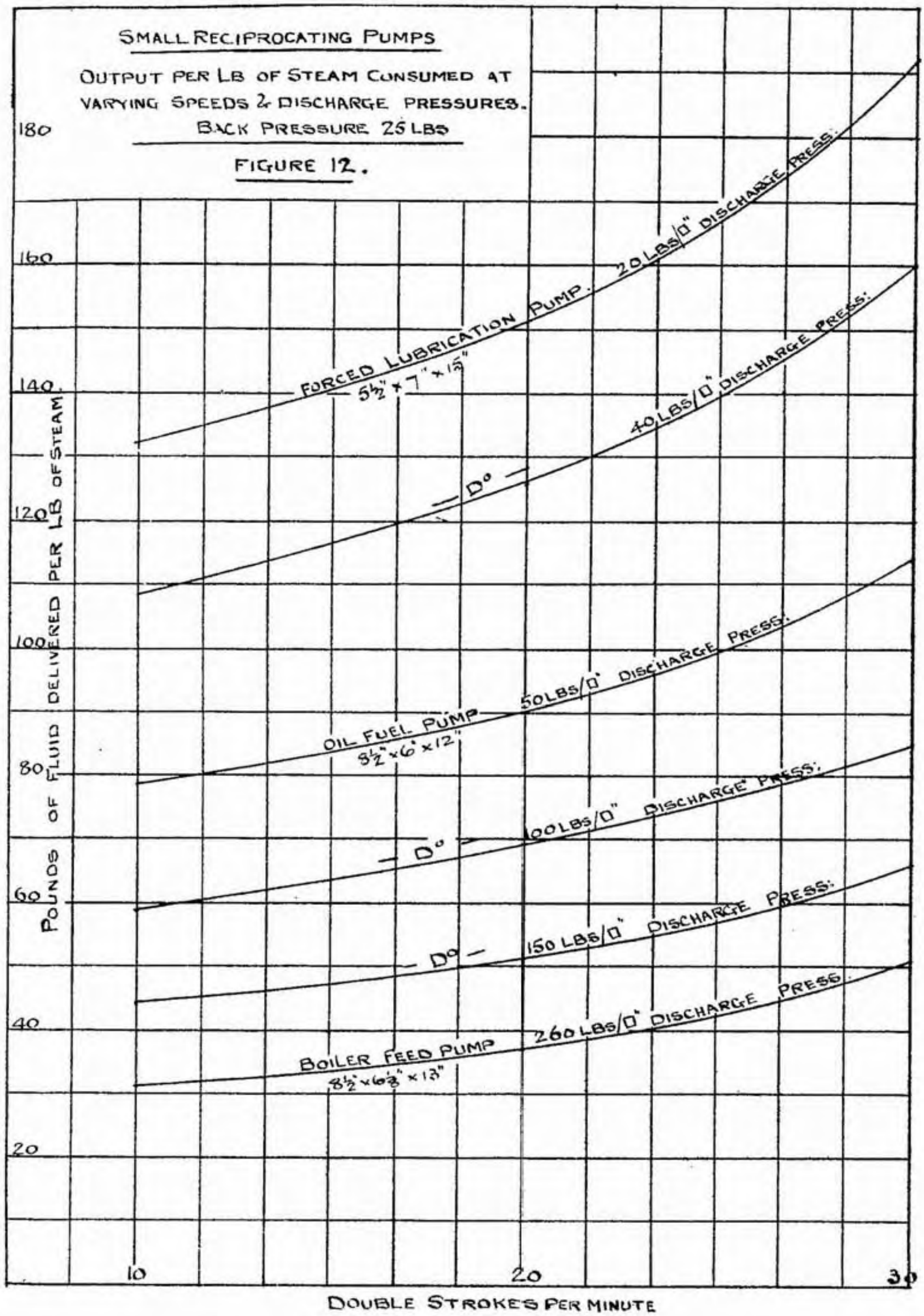
The principles given apply also to ventilating fans generally.

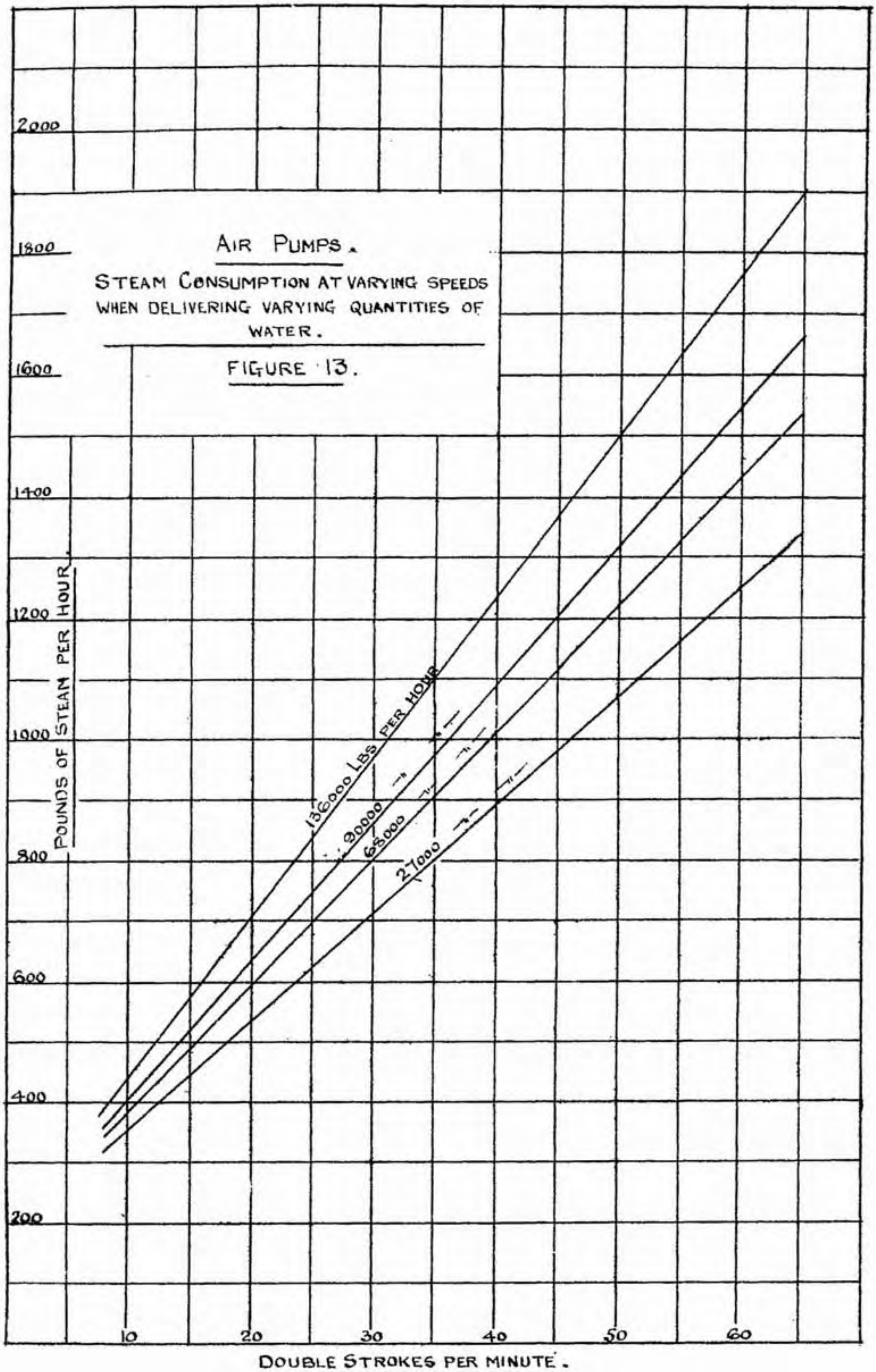
Reciprocating Pumps.—Typical curves showing the average performance of a large feed pump ($10\frac{3}{4}$-inch by 24-inch stroke) are given in fig. 11. The steam consumption per I.H.P. has been indicated to illustrate the general remarks made earlier as the effect of reducing output, but it is more convenient to express the consumption by the quantity of feed delivered for the expenditure of one pound of steam. and this relation has been indicated for varying back pressures. The steam consumption increases, of course, with the pressure against which the pump discharges; the curves given are for a discharge pressure of about 260 lbs.

The falling off in efficiency of pumps of this description is much less than in rotary pumps; the loss is mainly due to leakage through plungers and valves which increases as the speed of the pump is reduced. The mechanical efficiency is sensibly constant. Slip curves are shown on the diagram. The slip increases as the temperature of the feed increases, but the steam consumptions given are based on a feed water temperature of about 100° F., corresponding to average ship conditions. Depending upon the length, diameter, and arrangement of the feed suction pipe and upon the temperature of the feed water, the effect of increasing the speed of a pump would in the limit give rise to cavitation and lead to a slip much in excess of that given on the curves. This condition should, however, never arise in the installations fitted if their designed speeds are not exceeded, although it may be approached in the case of the faster running pumps fitted in the T.B.D. Classes when running at their full output.

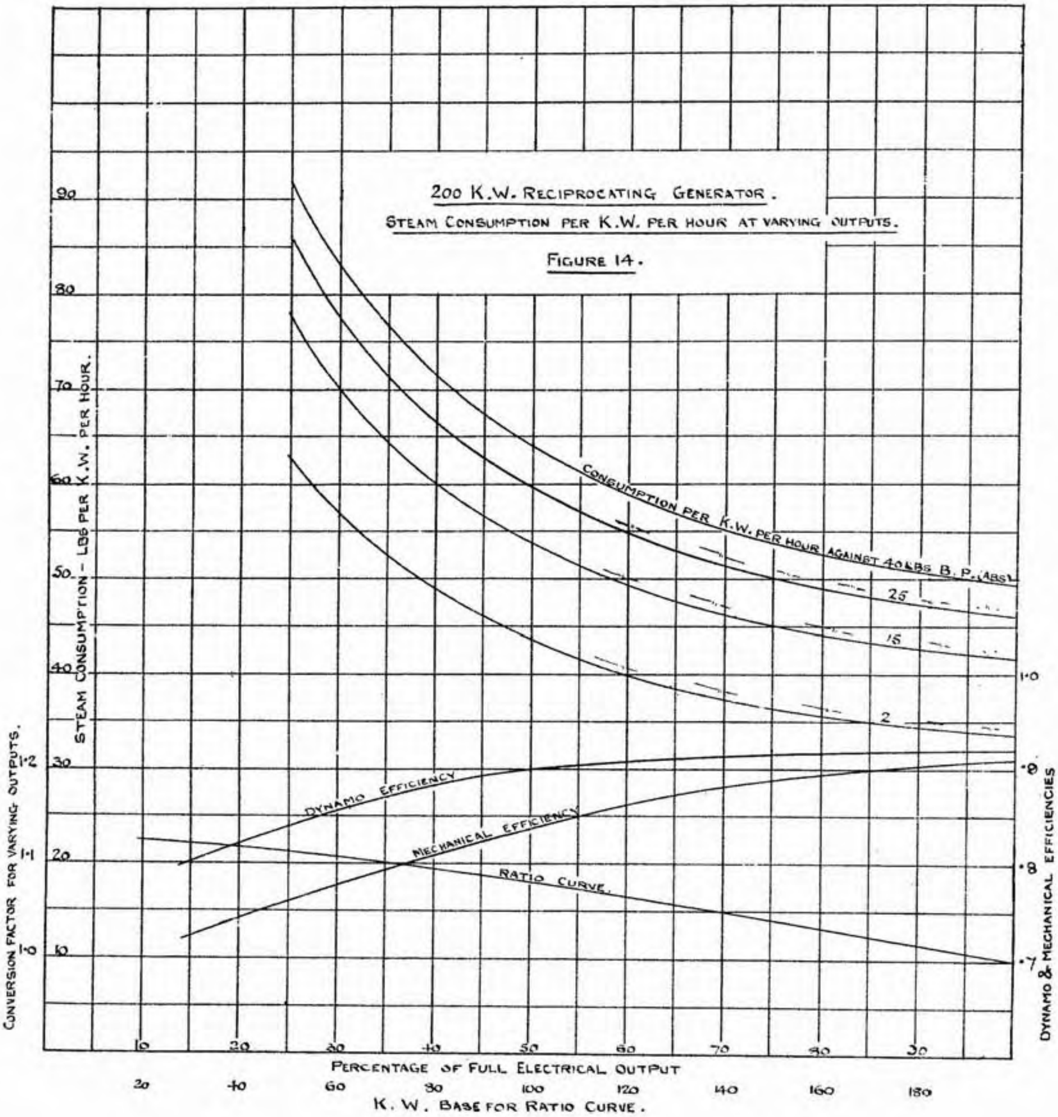
Figure 12 gives approximate data for the steam consumption of small feed pumps and for fuel and forced lubrication pumps at particular discharge pressures and under back pressure conditions. Water service pumps would give slightly better results, pressure for pressure, than the forced lubrication pumps. The slip of oil fuel pumps, assuming plungers and valves are in good condition, would average about 1·2 double strokes per minute at a discharge pressure of 150 lbs. and ·25 double strokes per minute at 50 lbs., when working at any speed within the range usually worked to. In the case of forced lubrication pumps, for an oil temperature of 100° F. the slip would range from 1 double stroke at 60 lbs. discharge pressure to ·4 double strokes at 30 lbs. discharge pressure.

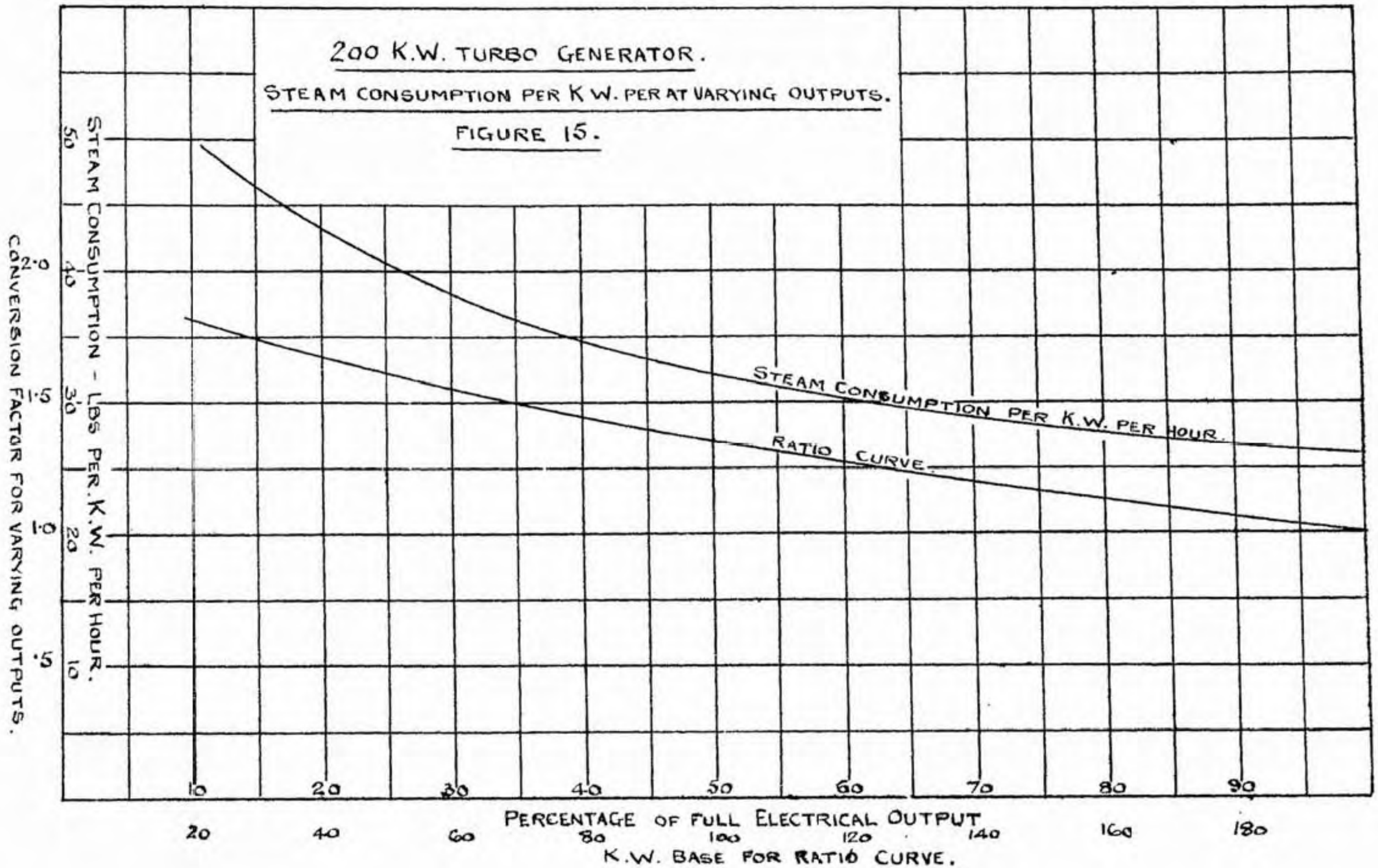
Reciprocating Air Pumps.—The reciprocating air pump is a special case not falling in one of the above categories. The discharge pressure here is relatively low and the engine works at a rather lower cut-off than in the case of the other pumps and the economy of the steam end is higher. As a result, however, of the high frictional loss and the large water valve surfaces, the pump efficiency is low and, measured in terms of useful work represented by the water delivered, the steam consumption is high. The conditions as to the relative amount of water and vapour in the pump system may be very variable and it is not possible therefore to give any conclusive figures of steam consumption at varying outputs. In general, for an average amount of air leakage in relation to the water dealt with, the steam consumption at full output should not exceed about 1½ per cent. of the feed water handled, assuming that the pump is run at a suitable speed.


SMALL RECIPROCATING PUMPS
OUTPUT PER LB OF STEAM CONSUMED AT VARYING SPEEDS & DISCHARGE PRESSURES.
BACK PRESSURE 25 LBS
FIGURE 12.
FORCED LUBRICATION PUMP. 20 LBS/□" DISCHARGE PRESS.
5½" × 7" × 1⅝"
— D° — 40 LBS/□" DISCHARGE PRESS.
OIL FUEL PUMP 50 LBS/□" DISCHARGE PRESS.
8½" × 6" × 12"
— D° — 100 LBS/□" DISCHARGE PRESS.
— D° — 150 LBS/□" DISCHARGE PRESS.
BOILER FEED PUMP 260 LBS/□" DISCHARGE PRESS.
8½" × 6⅜" × 13"
POUNDS OF FLUID DELIVERED PER LB OF STEAM
DOUBLE STROKES PER MINUTE
180
160
140
120
100
80
60
40
20
10
20
30



AIR PUMPS.
STEAM CONSUMPTION AT VARYING SPEEDS WHEN DELIVERING VARYING QUANTITIES OF WATER.
FIGURE 13.
2000
1800
1600
1400
1200
1000
800
600
400
200
POUNDS OF STEAM PER HOUR.
136000 LBS PER HOUR
90000 " "
68000 " "
27000 " "
10
20
30
40
50
60
DOUBLE STROKES PER MINUTE.


200 K.W. RECIPROCATING GENERATOR.
STEAM CONSUMPTION PER K.W. PER HOUR AT VARYING OUTPUTS.
FIGURE 14.
STEAM CONSUMPTION - LBS PER K.W. PER HOUR.
CONVERSION FACTOR FOR VARYING OUTPUTS.
DYNAMO & MECHANICAL EFFICIENCIES
CONSUMPTION PER K.W. PER HOUR AGAINST 40 LBS B.P. (ABS)
25
15
2
DYNAMO EFFICIENCY
MECHANICAL EFFICIENCY
RATIO CURVE.
90
80
70
60
50
40
30
20
10
1·2
1·1
1·0
1·0
·9
·8
·7
10
20
30
40
50
60
70
80
90
PERCENTAGE OF FULL ELECTRICAL OUTPUT
20
40
60
80
100
120
140
160
180
K. W. BASE FOR RATIO CURVE.


200 K.W. TURBO GENERATOR.
STEAM CONSUMPTION PER K W. PER AT VARYING OUTPUTS.
FIGURE 15.
STEAM CONSUMPTION PER K.W. PER HOUR.
RATIO CURVE.
STEAM CONSUMPTION - LBS PER K.W. PER HOUR.
CONVERSION FACTOR FOR VARYING OUTPUTS.
50
40
30
20
10
2·0
1·5
1·0
·5
10
20
30
40
50
60
70
80
90
PERCENTAGE OF FULL ELECTRICAL OUTPUT
20
40
60
80
100
120
140
160
180
K.W. BASE FOR RATIO CURVE.

The results of makers' trials for the Dual Air Pumps of the size (22-inch barrel by 17-inch stroke) and type fitted in certain T.B.D.'s and Light Cruisers, when working against atmospheric pressure in the exhaust pipe are shown in Figure 13. It will be seen from these curves that, when handling any particular quantity of water, the steam consumption varies as the speed of the air pumps. That is to say, for any particular power of the main engines, the steam consumption of the air pumps may be taken to vary as the speed at which they are run. For larger or smaller pumps than referred to above, the consumption could be taken without any great error to vary in proportion to the respective products of the cylinder diameter and stroke. In the case of pumps fitted in Capital Ships the steam consumption would be rather higher than given in the curves, say about 10 per cent., as a result of the greater head and resistance on the discharge side. Also some addition would require to be made to provide for back pressure in the exhaust pipe.

Dynamo Engines.—The efficiency of a dynamo becomes less as its output is reduced; this follows mainly from the fact that certain of the losses are dependent solely on the speed of rotation and are therefore constant at all outputs. It is more convenient to express the steam consumption per K.W. hour, and this is indicated in figs. 14 and 15, for a compound reciprocating engine and a turbine driven set respectively, of 200 K.W.'s in each case, both employing saturated steam. The variations for changing back pressure are shown for the reciprocating engine whereas in the case of the turbo generator the vacuum is taken to average 27 inches, in general agreement with the installations fitted.

The approximate variation in steam consumption for generating units of smaller output is also indicated by the lower curves. In Figure 15, for example, the consumption of a 100-K.W. set would be 1·36 times that indicated for the 200 K.W. set.

Relative Consumption of Auxiliaries.

Figures 16 and 17 represent respectively the estimated consumption of the more important auxiliary engines of a Battle Cruiser and a T.B.D. under trial conditions, at varying powers of the main engines. The total consumption for all auxiliaries in use, represented by the upper curve in each case was obtained by tank measurements during trials at various powers ranging from 5 per cent. to full power; an adjustment has been made as necessary, to provide for the auxiliaries working against a suitable back pressure applicable to the total power, assuming that the exhaust steam was used for feed heating to 140° F. and in the main turbines. During the water measurement trials the closed exhaust could not, of course, be so employed in view of the fact that the main and auxiliary consumptions were separately measured.

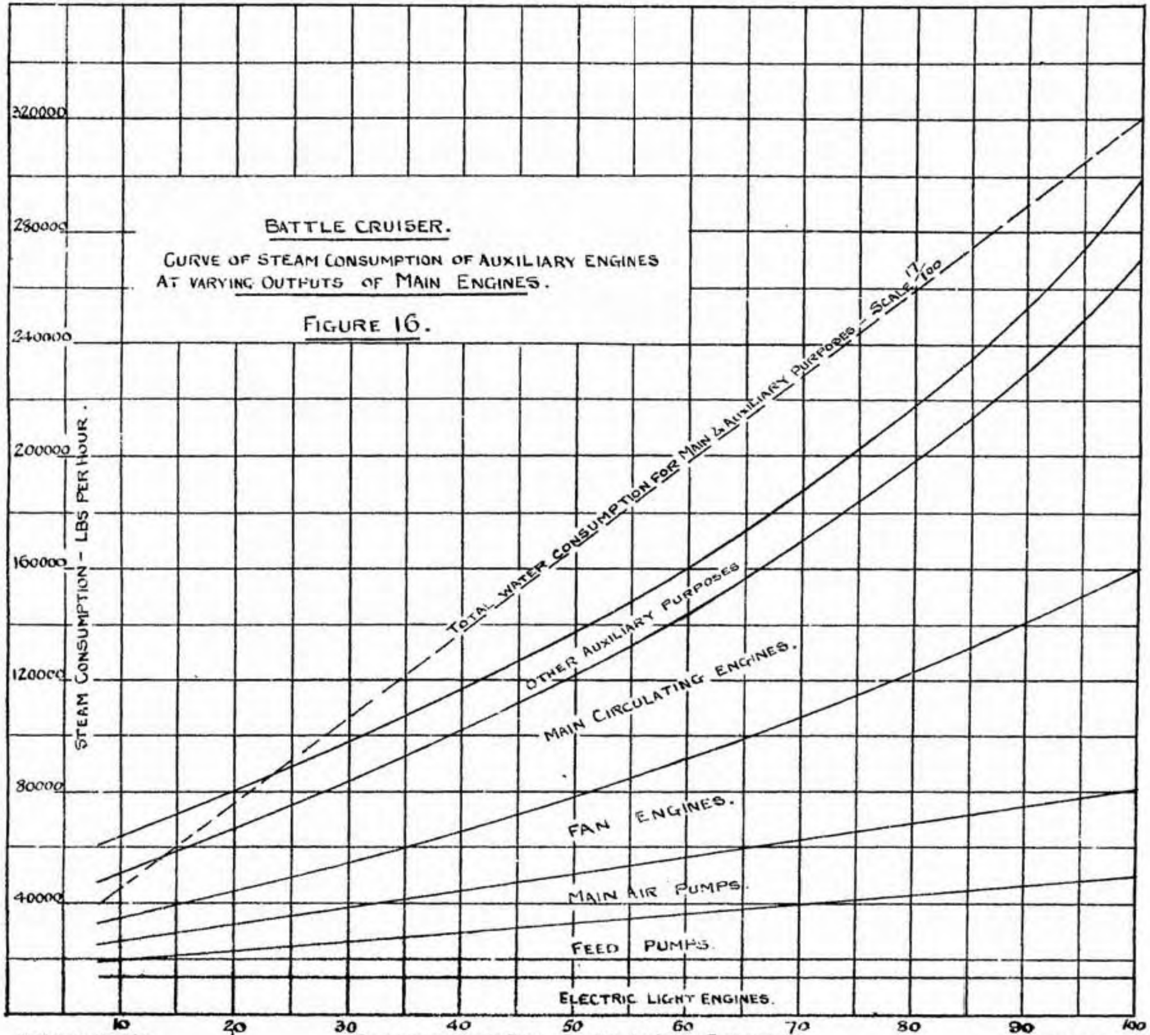
The estimated consumptions of the auxiliaries mentioned are calculated from the observations of the speeds of revolution and other relevant data usually taken on trial, considered in relation to the known performance of engines of similar type from data similar to that already given herein and cannot therefore make pretence to absolute accuracy, although it can safely be assumed that the relative divisions of the auxiliaries given are sensibly in accordance with the actual results.

The " other " auxiliaries, for which no detailed consumptions are estimated, are such services as forced lubrication and cooling, steering, oil fuel engines, ship pumping services, drainage, and so on. The evaporators were not employed nor were any auxiliaries pertaining to the armament in use.

These curves of relative consumptions only apply to the Classes mentioned and the relations between the respective auxiliaries would vary somewhat for other classes of vessel, the main factors influencing differences being :—the steam consumption of main engines at different speeds, the steam consumption required for standing requirements (electric power, &c.), closed exhaust pressure, boiler room air pressure, degree of vacuum and type of condenser (whether single or double flow).

The data given earlier will, however, afford a basis for a similar estimate of the consumption of the more important auxiliaries under varying conditions in any ship. The quantity of *total* feed can be estimated by means of observations of the speed of the feed pump, taking care to see that the designed stroke is being maintained and that the water valve, &c., leakage is not excessive; the latter may be checked from a creep test with the discharge valve shut. Similarly for the other supply pumps. In the case of the fan engines, the air delivered might be deduced from the known amount of fuel being burnt, making allowance for the excess air over that required for the perfect combustion of the fuel. This cannot be precisely determined without an analysis of the funnel gases, but if the oil is being burnt efficiently and means are taken to check the speed of the fans until incomplete combustion becomes evident, it can be assumed that the excess air is within the amount which is known to be necessary under the best conditions; this amount varies, of course, with the output of the boiler and increases as the output is reduced. At full power it will generally be found to average from 20 per cent. to 30 per cent. and rather less at intermediate outputs, but in the limit when the output of the boiler is very small, and, say one burner is employed, the opportunities for air leakage become very great, and 100 per cent. may easily occur.

The rule given for the circulating pumps as to the output varying with the speed of revolution will enable the water horse-power to be calculated (observing that this varies roughly as the cube of the output), and so from the pump efficiency and other curves, the Brake Horse Power and the steam consumption. In



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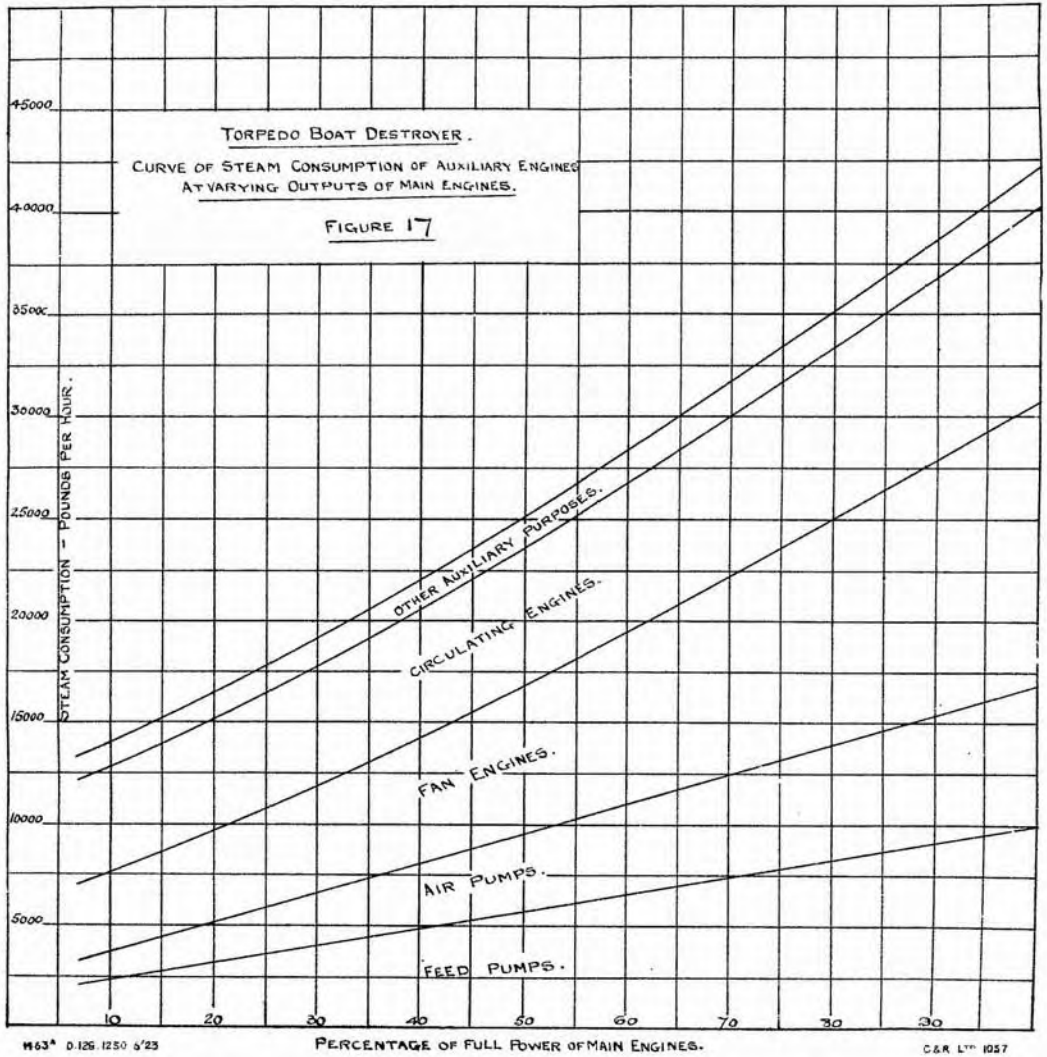

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general, particulars of the full designed output and the head at which it is realised are shown on the ship's drawings.

The observations of electrical output afford a ready means of ascertaining the output of the electric generators and the results of the Makers' trials, which give the water consumption rates at full output at least, are also usually available on board. The curves given earlier will enable the steam rate at outputs below full output to be assessed.

Due regard should be given in each case to the back pressure conditions under which the engine is working and the necessary corrections made for the steam consumption rate, for which the curves given afford a means in the cases of the more important engines. In the case of engines very remote from the ultimate position at which the exhaust steam is put to further use or from the pressure gauge recording the pressure in the exhaust steam, it is desirable that an observation of the exhaust pressure should be made in the vicinity of the engine itself; at high outputs this pressure will be sensibly higher in many cases than that recorded on the main exhaust range.

In this way an approximate estimate can be made of the consumption of the principal auxiliaries, the aggregate of which considered in relation to any water measuring data available for Contractors' trials of similar ships, will give an idea of the efficiency of this part of the installation and lead to any large losses of economy in maintenance or operation to be detected. Admittedly, the method is not by any means exact, but it will be found sufficient for the end in view, in the absence of steam meters or other means of separately measuring the consumption of individual auxiliaries; in the larger ships it is sometimes possible to measure the steam consumption of at least, a group of auxiliaries by direct means with but little alteration to existing fittings.

A consideration of the data given will also in any specific case afford a guide to the most advantageous means of operating the machinery under varying conditions and, above all, to the employment of the exhaust steam in the most profitable manner.

Use of Auxiliary Exhaust Steam.

It will be evident that the steam consumption per unit of power in the auxiliary engines as given by the curves and which exceeds 50 lbs. per B.H.P. in the case of a simple engine working against a back pressure, is very much in excess of that realised in the main engines. This excessive consumption is, however, greatly mitigated by the further employment of the exhaust steam. The arrangements provide for the use of the exhaust steam for boiler feed heating, for supplementing the power obtained by the use of boiler steam in the main engines, and for use as primary steam in the evaporators.

Feed Heating.—Considering these applications in order :— By employing the exhaust steam for feed heating, the whole of the surplus heat rejected by the auxiliaries may be returned to the boilers and the net charge in heat units to be debited to the auxiliary is the sum of the thermal loss in the boiler (attending the evaporation of the steam passing through the auxiliary) and the heat units abstracted from the steam on its passage through the auxiliary; in effect, therefore, the net consumption of the auxiliary is rather less than one half the consumption indicated in terms of steam per B.H.P.

A simple calculation will illustrate this point. The calorific value of 1 lb. of fuel is, say, 18,500 B.T.U's. When this fuel is burnt in the boiler, assuming a boiler efficiency of 70 per cent., 5,550 B.T.U's are lost per lb. of fuel, and the remaining 12,950 B.T.U's go to produce steam. For a pressure of 250 lbs., and a feed temperature of 130°, each lb. of fuel will produce 12 lbs. of steam. Suppose this steam to be used in the auxiliaries and to be exhausted against a back pressure of 20 lbs. gauge, with a dryness fraction of ·85, the total heat per lb. being therefore: $267 + \cdot85 \times 934\cdot5 = 1{,}061$ B.T.U's. Of this, $1{,}061 - 140 = 921$ B.T.U's are available for heating the main feed and assuming a loss of 5 per cent. in the process $875 \times 12 = 10{,}500$ B.T.U's are returned to the feed circuit for every pound of fuel used.

The expenditure for the auxiliaries is thus not 18,500 but $18{,}500 - 10{,}500 = 8{,}000$ B.T.U.s, and the net steam consumption per B.H.P. is $\dfrac{8{,}000}{18{,}500} = 43$ per cent.

of that which would be represented by the steam passing to these engines.

Feed heating clearly represènts the most advantageous application for further use of the exhaust steam, but the application is limited by the relatively large quantity of exhaust steam available, considered in relation to the total amount of feed passing and the limitation in the temperature to which the feed can be conveniently raised before entry into the boiler. In Naval installations provision has hitherto only been made for heating the feed on the suction side of the feed pumps, and the limit of temperature is therefore decided by the question of the reliability and durability of the feed pump, *i.e.*, overheating or difficulties with the suction, and cannot be much greater than about 140° F.

In the merchant service and power-plant practice the feed is usually heated to about 220° F. in modern installations, and the arrangement usually entails feed heating on the discharge side of the pump with a certain amount of additional complication. This high temperature feed-heating appears likely to find an increasing use in any future Naval installations.

For a back pressure of 10 lbs. the proportion (in the form of auxiliary exhaust steam) of the total feed required to raise the temperature of the main feed from, say, 85° F., at which it leaves the air pump, to 140° F. is 7 per cent., and to 200° F. is 17 per cent., making allowance for radiation losses, and in the latter case to the loss of heat in the exhaust condensate which, for reasons that need not be entered into here, could not be employed for feed heating in a convenient way. Under full power conditions the proportion that the auxiliary exhaust steam bears to the total feed falls within the 17 per cent. referred to above in the case of a capital ship, but is beyond it at very low powers as is indicated in Figure 16, so that the whole of the exhaust steam could not be employed for feed-heating, even to high temperatures, at extremely low powers. In the limiting case, under harbour conditions, only 17 per cent. of the exhaust steam can be used for feed heating, even with high temperature feed, and 6 per cent. with the feed temperatures at present general in Naval practice.

For seagoing conditions it is necessary therefore to provide an alternative employment for the exhaust steam in the form of connections to the main turbines, seeing that the use of the exhaust steam for distilling is an intermittent service and cannot in any case utilise the whole of the available supply when at sea.

Use in Main Engines.—The use of the auxiliary exhaust in the turbines is a much less economical course than for feed heating, because the heat in the steam at the pressure corresponding to the condenser vacuum is entirely lost; this, for a vacuum of 28 inches, would correspond to a loss of about 80% of the total heat required to generate the steam in the boiler from, say, 140° F.

As compared with an auxiliary discharging direct to a condenser, the gain due to the use of the exhaust in the main turbines is represented by the further employment of the steam down to the high vacuum possible in the main condenser, observing that it is not practicably possible to attain a high vacuum in the auxiliary exhaust system, and even if it were, the use of extremely low exhaust pressure in a reciprocating engine would not (as earlier pointed out, and indicated in figs. 1 to 3) be attended by the same saving that occurs in a turbine, mainly as a result of the increased condensation losses.

Aside from the less profitable use of the heat than for feed heating, the employment of the exhaust in the main turbines somewhat handicaps their performance, as compared with that to be expected from the sole use of boiler steam. The auxiliary exhaust steam is in general wetter than obtains in the turbine when using boiler steam alone, and this leads to a lesser efficiency of the later stages as a result of the increased frictional losses; this would become of even greater importance in an installation employing super-heated steam. Further, the additional steam to be passed in the ultimate stages, in which real difficulty exists in the high-power designs in obtaining the necessary areas to prevent an unduly high leaving velocity, leads to higher leaving losses and to a lesser efficiency of the turbine as a whole.

As a rough working rule, it may be assumed for the pressure limits within which Naval turbines work, that a given weight of exhaust steam used in the later stages of the main turbines will produce about one-half the shaft horse power that would be realised by the use of the same weight of boiler steam, employed throughout all the stages. Thus if w be the steam consumption of the turbines in lbs. per S.H.P. per hour, when using boiler steam when the turbines develop a horse power of P, and if A lbs. of exhaust steam are available per hour, then the total horse power attained when the exhaust steam was admitted to the turbines would be $P + \frac{A}{2w}$.

Gain from Feed Heating.—As illustrating the advantage of the exclusive use of the closed exhaust for feed-heating as compared with its use for feed-heating and for turbines, the following calculation showing the improvement estimated for the case of a Battle Cruiser, will be of interest. In existing circumstances, at full power, the consumption of steam for auxiliary purposes is about 15 per cent. to 16 per cent. of the total steam produced from the boilers when the engines are working against a back pressure of 10 lbs. gauge. Of the auxiliary exhaust steam, about 40 per cent. is used for heating the boiler feed from the temperature at which it is discharged from the air pumps to 130° F. at which it enters the boilers, and the remainder is used in the main turbines, augmenting the power by 5 per cent. If the whole of the exhaust steam were used for feed-heating, the feed temperature would be raised to 185° F., but, on the other hand, to produce the same horse-power, an additional amount of boiler steam would be required in the main turbines; this, taking everything affected thereby into account, would lead to an increase of 4 per cent. in the amount of the steam to be produced from the boilers. But, although the steam consumption increases, the effect of the lesser amount of heat to be given to each pound of steam in the boilers is to yield a saving of fuel represented by $100\left(1 - \frac{104}{100} \times \frac{1045}{1100}\right) = 1 \cdot 2$ per cent.

This represents the calculable saving, and may be assumed to be a minimum in view of the fact established in shore practice that high temperature feed-heating is accompanied by an increase in the efficiency of heat transmission in the boiler, and so to a lesser amount of fuel burnt.

Use for Distilling.—The use of the exhaust steam for distilling provides an economical application, second only to the use for feed-heating, provided the vacuum maintained in the evaporator shell is sufficiently high, and the coil drain condensate is returned to the feed system. A high vacuum entails, of course, greater difficulty in operating, more care in maintenance, and a greater liability to priming, but its advantages from the economical aspect are sufficiently weighty to justify this greater attention in operation and maintenance. Of the available heat in the steam supplied to an evaporator, there are unavoidable losses due to radiation, blowing down, and the heat required to produce the vapour. All these losses are susceptible of reduction by a reduction in the shell pressure at which the evaporator is operated. In any case the heat in the condensate from the primary steam is available for boiler feed heating as is also a proportion of the heat in the condensate from the secondary steam for heating the evaporator feed.

Most Advantageous Use.—Under seagoing conditions it would be profitable to employ all the exhaust steam for boiler feed-heating where this course is possible, but this ideal is impracticable of attainment in existing vessels, in view of the rather low increase of feed temperature permissible by the arrangements. In any case, at low speed, when the proportion of auxiliary exhaust is high, only a relatively small amount of the exhaust steam available can be employed for feed-heating even were a higher degree of feed-heating permissible. The question of the choice of service for the surplus exhaust steam then arises, and, where speed of production, or similar considerations, are not of first importance, it is of advantage to employ as much of this surplus steam for distilling as is possible, provided evaporation at low pressure can be realised; the surplus heat in the coil drain being also employed for heating the boiler feed.

Under extreme conditions, such as obtain in harbour, the use of only 7 to 17 per cent. of the exhaust steam (depending on whether the feed is heated to 140° or 200°) is possible for feed-heating, and there is then no alternative than to use the closed exhaust for distilling. Under most conditions, in large ships at least, it is possible and of advantage from the economical aspect to produce the bulk of the distilled water required when in harbour, and this course is to be commended. Even in ships fitted with auxiliary condensers it is not possible to maintain a high vacuum and in other ships in which the main condensers are used in harbour no vacuum is possible. The fuel cost, then, of producing the distilled water is that due to the extra steam consumption required by the auxiliary engines when working against the somewhat higher back pressure entailed by the use of exhaust steam for distilling, plus the steam required to operate the distilling pump. In some cases it is possible, too, to dispense with the use of the circulating engine when passing the bulk of the auxiliary exhaust steam through the distillers and to use instead a connection from the fire main which is sufficient for these lesser cooling requirements and leads to a further saving. The typical diagrams of steam consumption for varying back pressures in figs. 1, 2 and 3, will enable an idea to be formed of the extra steam consumption entailed by change in the back pressure. Clearly, the lower the exhaust pressure the lower is the extra steam consumption, thus indicating the desirability of working the evaporators under low-pressure conditions.

Another point which may be emphasized here is the desirability of limiting, so far as is possible, the number of engines working against the back pressure to the particular requirements of the moment and to arrange for any auxiliaries which have duplicate connections to the condenser and the exhaust system to be run direct to the condenser in cases where the supply of exhaust steam is beyond the needs at the time. Also, in respect to the operation of the evaporating plant, the tendency of operators is to run the evaporating pump at too high a speed. Considerable savings in the cost of distilling can be effected by giving greater attention to the maintenance of the efficiency of this pump and so permitting its being run at a speed commensurate with the output. As a result of its multiple functions, its mechanical efficiency is on the low side and the wear and tear is considerably augmented when run at the higher speeds. In particular, the efficiency of the fresh-water pump is of first importance if it is intended to run the plant successfully under vacuum conditions.

Operation and Maintenance of Auxiliaries.

Turning now to the operating and maintenance aspect of auxiliaries generally, it is of interest first to trace the full effect of one extravagant engine. It leads directly to an increase in the supply of steam from the boilers to satisfy its own needs, but its influence does not end here. More power is called for from the boiler and condenser auxiliaries to deal with the extra steam, which leads to further demands for steam. So the effect is a cumulative one which may lead, in the extreme case—full power conditions, for example — to a reduction in the boiler efficiency due to a greater degree of forcing. These remarks apply to an extravagant engine, whether due to design, maintenance, or operation, and in respect to faulty operation still other influences may be noted. A fan engine run at a greater speed than the conditions strictly demand, leads, not only to extra steam being consumed, but also to an excess of air supplied to the boiler, and to a reduction in boiler efficiency due to the greater amount of heat carried away *viâ* the funnel, on which the *whole of the fuel* expended in the boiler has to pay toll. A similar effect, but to a lesser degree, may be noted in a circulating pump which is run unduly fast, when the effect is to lower the temperature of the feed below that necessary to yield the desired vacuum, with an unnecessary loss of heat that should be conserved in the circuit. In this connection it may perhaps be recalled that the attainment of a vacuum is influenced by two considerations, viz., the condensation of the steam and the removal of the air and vapour, and, with this in view, discrimination in the relative operation of circulating pumps and air pumps respectively is called for if the best results are to be attained.

The importance of reducing the heat losses due to radiation and leakage of steam hardly needs emphasis. It may be mentioned in passing, however, that the former can be considerably reduced, and with it the consumption of fuel, by lowering the boiler pressure to the exact needs of the case when working under conditions of low output as when in harbour. Also that the performance of a turbine falls off disproportionately to that calculable from the lesser heat content of the steam when supplied with wet steam; this follows from the increased frictional losses and the maintenance of the efficiency of the lagging of the steam ranges is desirable for this reason, no less than from the direct losses of fuel, as well as the inconvenience and discomfort due to the heat lost by radiation. The improvement in the performance of the machinery of T.B.D's at high outputs when a steady low-water level is maintained in the boilers, will be within the knowledge of all who have running experience of this class of machinery, and will provide support for the statement as to the effect of supplying wet steam to turbines.

A matter sometimes lost sight of is the radiation losses from the feed suction pipe on its way from the engine-room to the boiler-room in cases where the feed-heater is fitted in the engine-room. Unless adequately lagged, the feed-water may easily fall in temperature for 10° to 20° before it reaches the boilers in the case of a large ship; this means at least an additional 1 per cent. of heat to be added in the boiler.

Leakage.—It need hardly be said that leakage of steam may become a very fruitful source of loss. External leakage entails a dual loss, represented by the loss of heat as such, and the loss of water, which has to be made good at the expense of additional fuel and wear and tear. But the insidious effects of internal leakage may be of even greater significance, and, being out of sight, it is more difficult to trace. It can only be kept within reasonable bounds by a regular examination of internal parts, such as piston and valve rings, for example, and by a judicious choice of clearances which, while sufficient to safeguard moving parts from undue friction and wear, is no more than experience shows can be safely employed—having in view any effects of differential expansion in those cases where the materials of ring and cylinder are different.

Leakage of steam through slide valves and pistons is probably the most potent cause of falling off in efficiency of auxiliary engines on service, observing that its effect is enhanced when the auxiliaries are run at low outputs, so much so, that the leakage loss may conceivably exceed the amount of steam usefully employed in the cylinder.

The results of various trials carried out to determine the influences governing leakage of this nature, given in abstract below, will enable some idea to be formed of the extent to which this loss may attain and, in particular, of the effect of clearance. It will be noted that the observations and conclusions of these investigators are not always in agreement in respect to the significance of some of the factors, but this need not detract from their value for the purpose in view.

*Callendar and Nicholson's Trials for Piston and Valve Leakage (1895).

Preliminary trials showed the direct leakage of steam from the steam chest into the exhaust to be by far the largest and most important, and that stationary trials were of little value. This was measured as nearly as possible under running conditions by blocking the cylinder ports with lead (engine tried was a simple engine, 10½ diameter by 12-inch stroke, with a balanced D-slide valve) and running the valve by an electric motor, the piston being disconnected. In the first trial 112 lbs. was condensed in 25 minutes at a gauge pressure of 91 lbs., and the rate of leak appears to increase slightly as the oil film was gradually dissipated. In a second longer trial 317·5 lbs. was condensed in 66 minutes at a gauge pressure of 80½ lbs.

Similar trials were run on a quadruple expansion engine for the H.H.P. and L.P. cylinders with unbalanced D-slide valves. The H.H.P. gave a leak of 38 lbs. per hour with a pressure difference of 100 lbs. between steam chest and exhaust pipe. The L.P. valve gave 41 and 29 lbs. per hour with pressure differences of 34 lbs. and 21 lbs. respectively. The fitting of the valves was said to have been of a high order and the L.P. valve was proved to be absolutely steam tight when stationary.

By the law of transpiration of liquids, assuming that the leakage takes place chiefly in the form of water, the leakage of a liquid through a fissure of nearly uniform thickness, depending on the nature of the water and oil film on the surfaces, should be proportional directly to the difference of pressure and the perimeter of the port and inversely to the width of the bearing surfaces. The last named factor is of course somewhat difficult to assess for a moving slide valve, but an average value was taken in each case.

Thus, if the fissure through which the leakage takes place is of nearly the same thickness in each case, the leakage should be in agreement with a formula of the form :—

$$K = \frac{C.p.}{l.},$$

where K is the rate of leakage per pound pressure per hour, p the perimeter of the port, l the mean overlap, and C a constant depending on the nature of the film. Actually, in the four trials referred to on these two types of engines, the constant C varied from ·019 to ·021. It was concluded from them that the leakage does take place in the form of water and is proportional to the difference of pressure on the two sides. It was also considered probable that such leakage is the normal state of things with a moving valve.

An independent verification of the Constant C was obtained by other experiments (the leakage trials were only part of a

* Proceedings I.C.E., vol. cxxvi.

series investigating the laws of condensation) in which the leakage into the cylinder after cut off was measured, showed values of C equal to ·022 and ·020.

In this connection also, other trials showed that of the leakage past the piston at a mean pressure of 33 lbs., that occurring *after* cut off (·25) was only a small proportion.

From an analysis and comparison of the indicated condensation in these trials, it was also concluded that the rate of leakage past the valves was nearly independent of the speed of reciprocation of the valve.

Callendar advanced the following rationale for leakage of this nature :—

> " So long as the valve is stationary, the oil film may suffice to make a perfectly tight joint, but as soon as it begins to move, the oil-film becomes broken up and partly dissipated. Water is being continually condensed on the colder parts of the surface exposed by the motion of the valve. This water works its way through and breaks up the oil-film under the combined influence of the pressure and the motion. The continual re-evaporation taking place in the exhaust tends to keep the valve and the bearing-surfaces of the seat cool and to maintain the leaking fluid in the state of water. The exhaust steam from the cylinder has the same tendency. The coefficients of viscosity of steam and water at the temperatures which occur in a steam-engine are not accurately known. But whereas that of steam increases with rise of temperature, that of water diminishes very rapidly. It is not improbable that the quantity of water which can leak through a given crack under a given difference of pressure, may be from twenty to fifty times greater than the quantity of steam which can leak under similar conditions. This agrees with well-known facts in regard to leakage, and explains how it is that the leakage in the form of water is so great. A few simple experiments were made with regard to the transpiration of water and steam under the conditions in question, and the leakage in the form of water was more than twenty times as great, the water being at a temperature below boiling point. The motion, both of the water and the steam, owing to the high velocity was certainly turbulent or eddying, which would have the effect of greatly increasing the resistance as compared with that due to viscosity, if the motion were steady. For the case of steady motion, comparative tests were made of the relative values of the viscosity of water, cold and hot. The measurements were not sufficiently accurate to give the law of the variation of the viscosity with temperature above 212°; but it appeared that the viscosity at 212° F. was only one quarter of that at 62° F., and that it continued to diminish very rapidly. Under the actual conditions of the valve-leak

experiments, the water leak is more likely to have been between forty or fifty times the steam leak."

Sankey carried out trials on the leakage of a Piston valve in a Willans engine (1895) and found that while the provisional law for leakage suggested by Callendar fairly represented his experiments, the leakage from a piston valve with the ordinary rings and springs would be much less than given by Callendar's constants. The value of the coefficient C was considered to depend on the design of the valve and was not a constant; it depended also to a considerable extent on the condition of the particular valve itself, whether the surfaces were properly polished or not. In piston valves fitted with rings and springs he found that C was equal to ·004, or about $\frac{1}{6}$th of that given by Callendar. But with a plug piston, ground to fit the trunk of an experimental Willans engine so tight that it was on the point of seizing, the value of C was ·09, which was twenty times as great as that obtaining with rings and springs; if the clearance was increased to $\frac{4}{1000}$th inch, the value of C increased to ·6, or 150 times greater than with rings and springs. This confirmed the surmise that the weight of water that could leak through a given crack under a given difference of pressure might be between 10 and 50 times greater than the quantity of steam.

*Lobley** carried out trials of leakage for a 4-in. piston valve fitted with four ramsbottom rings; this being reciprocated in a specially constructed dummy liner by means of an electric motor. The trials were run with steam pressures up to 200 lbs., and steam temperatures up to 588° F., and the liner was jacketed.

It was found that, other things being the same, the leakage was independent of the speed and that the standing leakage was about one-half of that which occurred when the valve was moving. While the total leakage per hour was found to increase with the pressure difference between the two sides of the valve, the rate of increase with increase of pressure difference was much less than that ascertained by Callendar, and did not follow the straight line law predicted by him. The average value of the coefficient obtained, which was ·0022, corresponded more nearly with that obtained by Sankey. The foregoing results refer to saturated steam, but in experiments with superheated steam Lobley also found that for the same steam pressures and speeds the leakage per hour fell off rapidly as the steam temperature was increased until 500° F. is reached, after which the distortion of the rings causes it to increase slightly. He concluded from this that the large difference found between the leakage from D slide valves and piston valves might be explained by the fact that the former warp when heated and so lift off the face.

Rycroft† recently investigated the leakage through a piston drop valve 9-in. in diameter, similar in all essentials to one designed

* "The Engineer." Vol. cxiii., No. 2,928.
† "The Engineer." Vol. cxxiv., No. 3,473.

to supply steam to a uniflow engine. The piston valve was fitted with solid rings $\frac{7}{32}$-in. deep, which had a working tolerance of ·004 inches when measured cold.

For saturated steam at a constant pressure of 149 lbs. gauge at inlet and with variable back pressures up to 80 lbs. gauge (obtained by throttling the discharge), it was found that the leakage was sensibly constant. For varying inlet pressures from 149 downward and a constant atmospheric back pressure, the leakage per hour varied almost directly with the inlet pressure. From this it is considered that under these conditions, *i.e.*, with the valve at rest, the leakage varies with the initial pressure, and not the pressure difference and the value of the coefficient C on this basis is ·0022. These experiments are being continued to ascertain the effect of other variables, including superheat and a moving valve.

Finally, as an example of the important effect of what might at first sight be assumed to be a trivial leak, the case may be cited of a merchant service installation of 2,700 S.H.P. in which the results of water measurement trials on service indicated an increase of 6 per cent. in the steam used by the main turbines as compared with Contractors' trials under parallel conditions. It was ascertained that this increase was mainly due to a leaky drain valve between the H.P. turbine first expansion and the condenser. After refitting the drain valve seat, a further water measurement trial showed that the increased consumption had been reduced to 1½ per cent. above the Contractors' result, and proved beyond question that this one defective fitting was responsible for an increase of 4½ per cent. in the steam consumption.