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Development, Principles and Application of the Combustion Turbine.

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Synopsis.

The paper begins with an outline of the historical development up to about 1900. More recent developments are considered in some detail and against the contemporary background of technical progress. Descriptions of installations employing combustion at constant volume and at constant pressure are given and a comparison is made between the pressure and temperature conditions in the two types. The principles governing the operation of constant pressure installations are dealt with in detail, reference being made to modern types and applications. Comparisons are made, as regards fuel consumption and air utilisation, of eleven possible combinations of components. The conditions under which marine installations would operate are analysed, special consideration being given to air temperature, and methods of meeting these conditions are proposed, an installation for a merchant ship being taken as an example. Lastly, the design of components is considered and the requirements of satisfactory performance from each component are discussed, and, in particular, the influence of each on overall efficiency.

After a long period, during which progress in prime movers has followed a steady evolutionary course, the combustion turbine, as a new competitor to long-established types of power units, suddenly demands serious consideration by engineers. The recent symposium of papers read before this Institute provided evidence, in that important branch of mechanical engineering covering propelling machinery for ships, that the contest between reciprocating and rotary steam engines, initiated nearly fifty years ago when the late Sir Charles Parsons engined the *Turbinia*, is far from reaching a conclusion; further, that the secondary contest between steam engines of all kinds and internal combustion engines is similarly undetermined. The combustion turbine thus demands consideration at a time when the types of power units available for marine purposes are already more numerous than at any previous epoch and when, as a result, the minds of marine engineers are correspondingly less rigidly fixed upon a particular type.

The combustion turbine, since the combustion of the fuel necessary to provide the heat energy takes place within the structure of the engine itself, is an internal combustion engine. The name, turbine, indicates that the transformation of heat into mechanical energy takes place in a rotary machine of turbine form. From the fact that the working substance in the turbine is in the strictly gaseous state and is not, as in the steam turbine, in the state of vapour, such engines are sometimes called gas turbines. In gas engines of the reciprocating type, however, the word gas connotes the use of gaseous as distinct from liquid fuel. A possibility of ambiguity is thus introduced which renders this usage undesirable. The term combustion turbine, on the other hand, involves no kind of restriction to the fuel used, be this gaseous, liquid, or solid, and sufficiently clearly describes the machine.

In the present paper, the authors thought it well, as a preliminary, to give some account of the historical development of the

combustion turbine; they next give a statement of the principles governing the operation of potentially successful combustion turbines; finally, they set out the conditions to be met with and the problems to be solved in the application of such engines to marine practice.

A description of the development of the combustion turbine shares the same characteristic difficulty as that of the development of other machines, namely, of separating the various steps in progress due to individual workers from those derived from contemporary progress in other lines of work. Until late in the 17th century heat engines had either been rotary or had worked on the direct displacement principle, that is, without pistons, both types of which had been described by Hero of Alexandria in 150 B.C. His *æolipyle* was the first heat engine, and in this work was done by the reaction of jets of steam escaping tangentially from a rotary hollow vessel supported by hollow trunnions through which the steam was supplied—a jet reaction engine. The next important rotary engine was Branca's turbine, described in 1629 A.D., in which a jet of steam from a boiler heated over a fire was caused to impinge on the vanes of a wheel; this was the first impulse turbine. Father Verbiest, a French missionary to China, is said to have applied such a turbine to drive a model car. Newton, having enunciated his third law, that "Action and reaction are equal and opposite", appropriately produced, in 1680, a model of a car in which the motive power was a reaction jet of steam. At that time, however, there was no pressing demand for improvement to means of transport and thus little incentive to development.

Progress in mining, on the other hand, was retarded, in the 17th century, by the need for improved pumping facilities to deal with underground water and the attention of inventors was directed towards the possibility of mechanical power. The Abbé Hautefeuille, in 1675, knowing the effects of the expansion of gases in cannon, suggested an engine using gunpowder to raise water. Powder was to be burnt in a vessel communicating with a reservoir of water and the depression produced in the vessel by the cooling of the gases was to raise water at atmospheric pressure. In 1682, he suggested another form of gunpowder engine in which the water was to be raised by displacement, caused by the direct expansion of the burnt gases. Huyghens, the physicist, studied the same problem and in 1680 produced an engine with a piston, but Papin, in 1688, had the first practical success, also employing a cylinder and a piston. The gases were burnt above the piston which, on being forced downwards, discharged the water beneath it and allowed part of the gases above it to escape. The remaining gases were cooled, causing a depression above the piston which then moved upwards in the cylinder, and water was drawn into the space beneath the piston. With the low standard of construction, the depression set up in this way was not very satisfactory, while the limitation of the fuel used to gunpowder was a further disadvantage. Papin therefore sought another means of producing the desired depression and so came to use steam; he constructed a piston engine for this purpose, in which the cylinder was integral with the boiler. His successful use of steam to produce a depression was an important step in the development

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of power units. Nevertheless, the internal combustion engine, although proving less satisfactory, had preceded the steam engine and demonstrated that fire, or better, heat could be employed to do the mechanical work of pumping water. Papin's greater success with steam eventually led to the developments of Newcomen and Watt which initiated the long career of the reciprocating steam engine.

In addition to Watt's contributions to the reciprocating steam engine, the latter part of the 18th century witnessed three interesting developments, namely: John Barber's invention of a combustion turbine in 1791, William Murdoch's practical use of coal-gas in 1792, and Robert Street's invention of an internal combustion engine in 1794. Barber, in his patent specification, given in the Appendix, revealed a complete knowledge of the practical problems to be solved in the combustion turbine; the installation manufactured its own gas from coal, wood or oil, and the specification even indicated a method of ship propulsion by jet reaction. Murdoch's development of the production and use of coal-gas, and the subsequent distribution of coal-gas in towns, exerted a great influence on the later evolution of the gas engine, a development which was specially attractive in its early stages as a means of power production in smaller units than those then possible with steam. Street, in his specification, described a workable engine which used, as fuel, a volatile liquid, such as spirit of tar or turpentine, and of which he recommended ten drops to the cubic foot of air as a suitable mixture strength. Apart from the ingenuity of the inventions of Barber and Street, they demonstrated progress in the fuels available for internal combustion engines since Huyghen's and Papin's gunpowder engines, but there is no evidence of the practical application of either engine. Nevertheless, these inventions show that, although the reciprocating steam engine had a practical monopoly in power production for many years, it was never, even in its early stages, without competitors.

In the early part of the 19th century the possibilities of air as a working substance were explored, the Stirling and the Ericsson engines being the best known of actual applications. The former developed about 50 i.h.p. for three years about 1845 in a Dundee foundry, while the latter was fitted in a ship of the same name as the inventor. Both embodied heat regenerators to which heat was supplied externally and the burning out of these constituted the principal drawback of these engines; a second drawback, since they were piston engines, was the low m.e.p. and the correspondingly low mechanical efficiency. This was a period in which much attention was given to the fundamentals of thermodynamics, following the interpretation by Clapeyron in 1834 of Carnot's treatise of 1824. Joule and Kelvin were prominent among those in this country who made important contributions to this science, and, in 1851, Joule proposed his engine, which did not involve a heat regenerator. This worked on the constant pressure cycle, consisting of two constant pressure stages and two adiabatics. Its importance to the present subject follows from the fact that this is the ideal cycle of combustion turbines with continuous combustion at constant pressure, in their simplest form. It is described in most textbooks, but a short description here will aid explanation later.

Fig. 1 shows the pressure-volume diagram of the cycle and

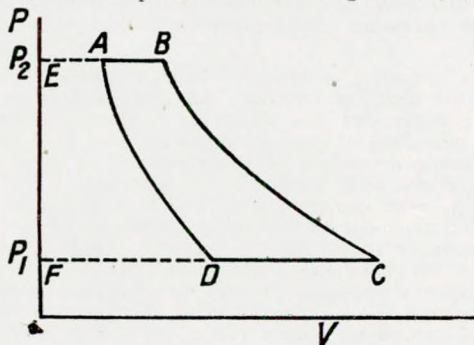


FIG. 1.

Fig. 2 shows diagrammatically Joule's air engine. At A and B the pressure is P_2 , while at C and D it is P_1 . Air is drawn from the cold chamber at temperature T_D and compressed from P_1 to P_2 and delivered at temperature T_A into the hot chamber. There it receives heat at constant pressure and its temperature is raised to T_B , with a corresponding increase in volume. The air is next passed through valve B to the expansion cylinder where it expands again to P_1 , the temperature attaining T_C . It passes through C into the cold chamber where it loses heat at P_1 and its temperature falls again to T_D , completing the cycle. The area E A D F gives the work done on the air in the compressor, while E B C F gives the work done by the air in

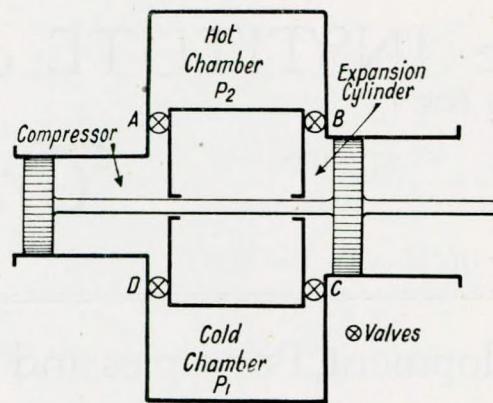


FIG. 2.

the expansion cylinder: the net work is thus given by the area A B C D. The heat received at P_2 is $C_p (T_B - T_A)$ units per lb., and that lost at P_1 is $C_p (T_C - T_D)$ units per lb., where C_p is the specific heat of the air at constant pressure, the net work done, in the ideal case, with adiabatic expansion and compression, being the difference of these two quantities. As Fig. 2 shows, Joule's engine was a piston engine, and, in spite of the earliness of Barber's invention of a rotary internal combustion engine, all attention in hot air and internal combustion engines seems to have been given to piston engines, probably owing to the accumulation of practical experience of cylinders and pistons gained with the successful steam engine.

The second half of the 19th century witnessed a great improvement in reciprocating steam engines, following step-by-step improvements to design, materials and workmanship. This epoch saw also the successful development of internal combustion reciprocating engines in most of their essential respects and to such an extent that many engineers prophesied the disappearance of steam engines, but the invention and practical application of steam turbines more than restored the balance. Until the work of Parsons and others, however, all successful power units had been of the piston type. Nevertheless, the patent literature during the half century furnishes interesting evidence of the attempts made to develop rotary machines.

In patent No. 1633, of 1861, for example, M. A. F. Mennons claimed an invention of "a new or improved combination of known machinery applied to the construction of locomotive, stationary, marine and other engines". The elements of this combination were: (1) a ventilating apparatus to project cold air into the body of the engine; (2) a furnace with a grate for fuel and grated apertures providing passage to the air from the ventilator, a portion of which passes to the furnace, while the remainder passes round the furnace and mixes later with the products of combustion; (3) a turbine driven by the dilated air mingled with the gaseous products of the combustion; (4) a regenerative apparatus, by which the cold air supplied by the ventilator is progressively heated to a given point, while the temperature of the hot air escaping from the turbine is reduced in the same proportion; (5) mechanical parts by which "the movement of the turbine is transmitted to the ventilator, the driving shaft, and other working parts of the engine". This combination, of course, gives the elements of a modern constant pressure combustion turbine, embodying a heat interchanger between the exhaust gases and the compressed air, and the patent even takes into account the necessity of supplying supplementary air to the hot gases after combustion. In patent No. 9685, of 1889, E. Hunt and J. Howden described a pure reaction explosion turbine. Four radial vanes in the turbine wheel are shaped to form a nozzle roughly tangential to the periphery of the wheel. An inlet port, through which the charge is admitted to the space between the vanes, is placed slightly offset to the centre of rotation and the charge is ignited by a gas flame or a heated tube. In order to leave the rotor, the gases are caused to pass at high velocity backwards through the nozzles and by reaction to drive the rotor.

The progress in appreciation of thermodynamic principles is also noteworthy. In patent No. 10034, of 1894, H. J. Haddon expressed his view that, struck by the inefficiency of existing steam engines, even with condensers and superheated steam, many engineers for a long time past had directed their investigations towards the increased temperature drop resulting from the elevation of the initial temperature of the working substance, but this elevation presented them with vapour at too high a pressure, so they sought to make use of gases. He spoke of igniting any type of fuel, including pulverised coal, in a vessel, the products of combustion being trans-

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mitted through a jet to the blades of a turbine. Thinking that, for maximum efficiency, the speed of the blades of the rotor must be equal to the speed of the gases, he reduced the speed of the jet by discharging it into a vessel containing a second fluid, such as water or air, and discharged both fluids through a second jet to the turbine. The underlying idea was that, by increasing the mass, the speed was reduced, but without a decrease in the momentum. His mention of pulverised coal at that date is also interesting.

In patent No. 11955, of 1895, by L. B. Atkinson, instead of the heat energy being applied to raise the pressure of the working substance for use in a piston engine, it is used to increase the velocity of the gases and thus to augment their kinetic energy. The working substances, gas and air, at atmospheric or a higher pressure, are admitted to a receiver, from which they pass through a regenerator heated by the exhaust heat from the engine. Combustion then follows and the temperature is raised, the result of the rise in temperature, if the pressure is constant, being an increase in the specific volume of the gases, so that, in steady flow, additional velocity has been communicated to the gas, which then passes into an impulse wheel or a reaction turbine.

One other patent will be mentioned, although it appeared slightly later than the 19th century, namely, No. 16766, of 1909, by F. W. Dodd. This, summarised, states: "In a compound engine or turbine for a torpedo, fuel from a reservoir is burned in a first combustion chamber, and, after doing work in one engine, the gases (with an excess of air) are supplied to another combustion chamber and then to another engine". He then foreshadows the type of compound expansion with interheating used in some modern combustion turbines.

This brief historical survey has thus been brought to a time when the steam turbine had already been firmly established for some years in land practice. In 1905, eight years after the sensational performance of the *Turbinia* at the Diamond Jubilee Naval Review, an Admiralty Committee on Naval Design advised that, in future, turbine machinery should be used exclusively in all classes of warships, while the decision to engine the new Cunard ships, *Lusitania* and *Mauvetania*, with Parsons turbines had been made in the previous year. The steam turbine can thus also be said to have become practically established in high-duty marine practice. In the forty years which have passed since then, great advances have been recorded in many branches of applied science and technology. Improvement in steam turbine practice has been directly helped by the contemporary progress in metallurgy, workshop machines and methods, and aerodynamics. It would be attractive to continue and to try to unravel the intermingled threads of progress, such as, for example, the interaction between aircraft engine practice and steam turbine practice. In connection with the present paper, however, it suffices to say that, without the progress in steam turbine design and manufacture, which has followed from the mass experience gained during the sixty years since Parsons' first turbine patent was taken out in 1884, the present position of the combustion turbine could not have been reached.

The more recent development of combustion turbines has been dealt with by various writers*, but, nevertheless, consideration cannot be entirely omitted here if the various factors determining the present position of development are to be set out and discussed. A gaseous working substance, in order to perform mechanical work in a turbine, must undergo both a reduction in its total energy and a fall of pressure. It follows that, in a cycle of operations, the exhaust gases from a turbine must both receive energy and be raised in pressure before they can again do work under the given conditions in the turbine. The methods by which the working substance is prepared and supplied to the turbine provide a convenient standpoint from which the various designs and applications may be considered, as regards both their present forms and their development. This leads to a convenient classification of installations under the following subdivisions: (a) completely independent power units; (b) units incidental to industrial processes; (c) units that are auxiliaries in a principal installation; (d) units for which a fortuitous supply of gaseous working substance at a suitable pressure is available, as, for example, natural gas. While the present paper is concerned primarily with (a) the other subdivisions deserve mention either on account of the influence they may have had on (a) or from their own intrinsic interest.

Completely independent power units now worthy of consideration for practical application are of the two main kinds, namely, those

consisting essentially of combustion turbines and those in which a compression-ignition engine of the piston type is combined in one unit with the gas turbine; the parallel in steam practice is the complete turbine installation and a combination such as the Bauer-Wach system. Limiting consideration to combustion turbines proper, two principal types are used in practice, and are characterised by the manner in which combustion takes place. In the type now receiving most attention, combustion takes place continuously at constant pressure; in the other type, in which practical success is associated with the name Holzwarth, combustion takes place intermittently at constant volume.

The main line of development of the constant pressure type is associated with the names Stolze, Armengaud-Lemale and Brown

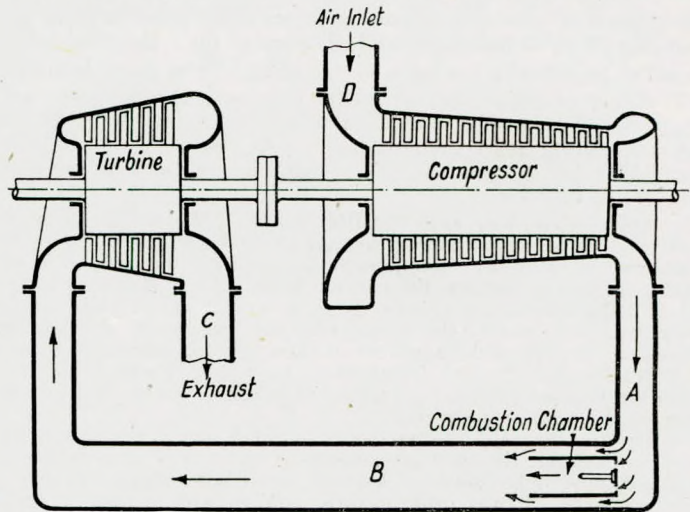


FIG. 3.

Boveri. In its fundamental form, it comprises three components—a compressor, a vessel in which heat is received by the gaseous working substance at constant pressure, and the working turbine itself. The organs are shown diagrammatically in Fig. 3. The pressure-

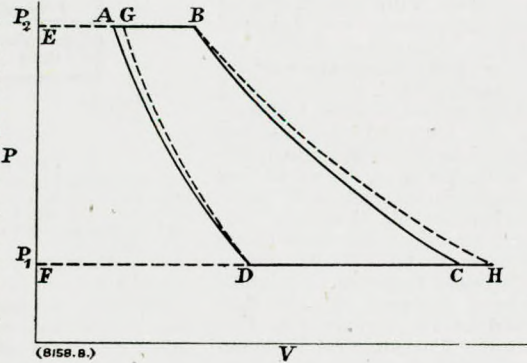


FIG. 4.

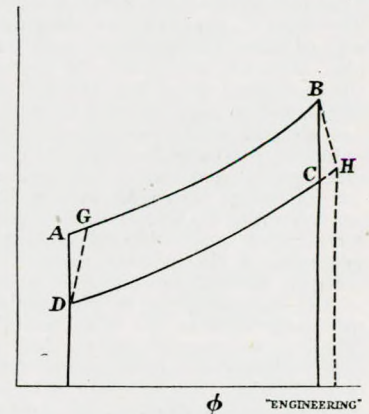


FIG. 5.

volume diagram, under ideal conditions with frictionless adiabatic compression and expansion, is that of the Joule's cycle shown in Fig. 1. In actual installations, such conditions are never realised and the operations involve internal losses both during compression and expansion. The ideal and actual conditions are shown on the pressure-volume diagram of Fig. 4, and on the temperature-entropy diagram of Fig. 5, but neglecting the changes in the specific heats of the gases with change of temperature. The adiabatic efficiency of compression, $\eta_c = \frac{T_A - T_D}{T_G - T_D}$, and the adiabatic efficiency of the turbine,

$\eta_t = \frac{T_B - T_H}{T_B - T_C}$. If the reception of heat is brought about by the combustion of fuel in air, instead of being supplied externally, and if w lb. of fuel of calorific value H be supplied per lb. of air, then the heat received per lb. of air = $wH = (1+w) C_p (T_B - T_C)$. The work done on the air in the compressor = $C_p (T_G - T_D)$; the work done by

*Stodola: Steam and Gas Turbines, Vol. II. 1927, probably gives the most comprehensive treatment up to 1927.

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the gases in the turbine = $(1+w) C_p (T_B - T_H)$. So that the overall efficiency of the cycle*

$$\eta = \frac{C_p (1+w) (T_B - T_H) - C_p (T_G - T_D)}{C_p (1+w) (T_B - T_G)}$$

$$= \frac{(1+w) \eta_t (T_B - T_C - \frac{1}{\eta_c} (T_A - T_D))}{(1+w) (T_B - T_G)}$$

The temperatures T_A , T_B , T_C and T_D are constant for a given case, w is small and rarely greater than 0.01, so that the magnitude of the numerator increases with increasing values of η_t and η_c . Moreover, the numerator is the difference of two quantities, the greater of which varies directly as η_t and the lesser inversely as η_c . Finally, the changes in value of the denominator are small with change of T_G and thus of η_c ; so that the overall efficiency of the cycle is extremely sensitive to individual changes in η_t and η_c . From three examples, all with a pressure ratio of compression and expansion = 4, with $T_B = 1,500^\circ$ F. abs., $T_D = 525^\circ$ F. abs. and with $\eta_c = \eta_t = 0.8, 0.85$ and 0.9 , respectively, the overall efficiency, η , has the values 0.116, 0.177 and 0.229; that is, approximately a doubling of η for a 10 per cent. increase in η_c and η_t .

Stolze, whose basic patent of 1898 was No. 7398, used air throughout and heated this externally instead of by burning fuel in it; he employed an axial flow compressor to compress the air and a reaction turbine to perform the positive work. As in Menon's patent of 1861, he led the hot exhaust gases to a heat interchanger where they gave up heat to the compressed air passing to the heating chamber. As far as the authors can trace, there is no record of any test results with Stolze's installation; he seems, however, to have been the first to construct a turbine installation of any kind and to employ the reaction principle.

The turbine of Armengaud and Lemale† was developed at the works of the Société des Turbomoteurs at Paris. The air compressor was a multi-cellular machine designed specially by Rateau to give a high efficiency. It was made in three sections with continuous cooling. It was said to deliver 1 cu. meter of air at 100 lb. per sq. inch with an efficiency of 65 per cent. It was not stated whether this efficiency is relatively to adiabatic or isothermal conditions, but, from the contemporary literature, probably the latter was taken. Continuous combustion of the fuel at a temperature of over 3,000° F. was arranged in a water cooled refractory combustion chamber, the pipe for the cooling water being embedded in the chamber walls. The nozzle supplying the working substance to an impulse wheel was of refractory material, also water cooled. Following a similar idea, proposed by Barber, the heated circulating water was delivered through small holes into the hot gases just before they entered the nozzle and, being converted to steam, reduced the temperature of the hot air and combustion gases. An alternative method of avoiding reduction of the high temperature of the combustion gases had been proposed by Armengaud but apparently was not tried. In this, two jets, one from each side, were to be applied to the single wheel, one jet conveying high temperature gases while the other conveyed low temperature steam derived from the water jacket and from a regenerator in the exhaust gases. The blades would thus pass successively through hot and cold zones, and, providing the temperature of the exhaust gases be sufficiently high to produce enough steam, the temperature of the wheel would be reduced to practicable limits. Since, in his installation, the best efficiency reported was less than 3 per cent., the surplus power developed by the turbine over that taken by the air compressor was very low. The practical problems involved were understood by Armengaud and his colleagues, but no further development of the unit for direct power production was reported. The design was, however, applied to torpedoes, in which compressed air from a reservoir was led to a combustion chamber from which the combustion gases and steam flowed to a turbine. The turbine, with an expansion ratio of 8.4:1, was said to develop 120 h.p. at 1,000 r.p.m.

While in the Lemale-Armengaud design the effects of the high temperature of the combustion gases had been reduced by the use of steam, so that an excessive amount of dilution of these by cooler air, to reduce the temperature of the gases leaving the combustion chamber, had been avoided, the work of compression, nevertheless, was too great for the turbine to be a success. Experimenters at that time were naturally greatly impressed by the need for reducing this negative work of compression, and several interesting attempts were made in this direction. By far the most successful of these was that

of Holzwarth*, and, in his type of installation, combustion was arranged to take place at constant volume, so as to raise the pressure of the working substance to the desired extent without excessive pre-compression of the charge. An explosive charge is introduced into a combustion chamber fitted with the necessary valves and is ignited electrically. The gases suffer a rise of pressure and temperature and are then led through controlled valves to the turbine from which, after performing positive work, they pass to a steam generator in which steam is raised to drive a steam turbine. In the first installation, the explosive charge was at atmospheric pressure, but, with newer designs, pre-compression of the air charge has been progressively increased, and now reaches 45 lb. per sq. inch. The work of compression is conveniently performed by the steam turbine. In comparison with modern constant pressure machines, the air-fuel ratio is low and thus the air quantity is much smaller. The whole of the gas side of the installation is jacketed, so that as

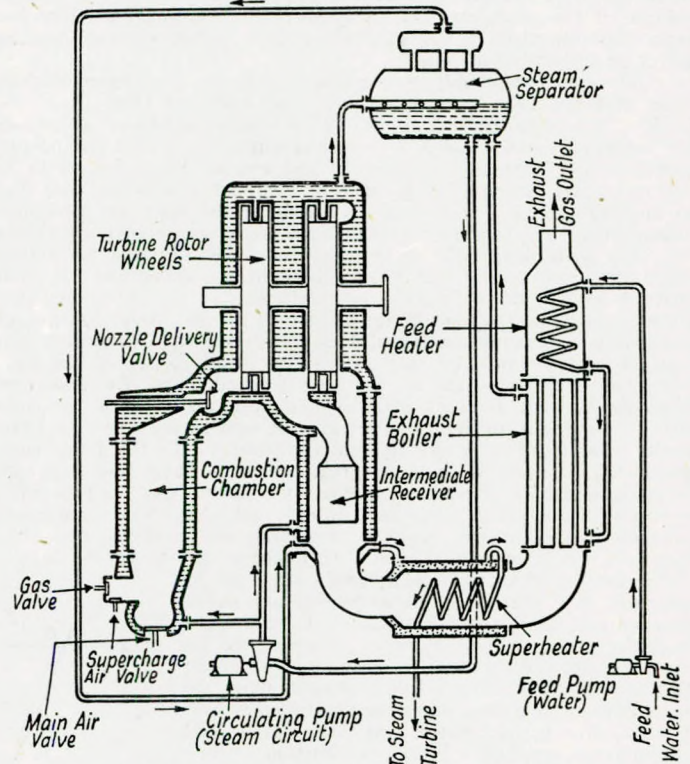


FIG. 6.—Diagram of Holzwarth installation.

- (1) Combustion chamber; (2) Gas valve; (3) Main air valve; (4) Supercharge air valve; (5) Nozzle delivery valve; (6) Turbine rotor wheels; (7) Intermediate receiver; (8) Feed pump (water); (9) Feed heater; (10) Exhaust boiler; (11) Steam separator; (12) Circulating pump (steam circuit); (13) Superheater.

much waste heat as possible is transferred to the steam side. The installation, as the diagram in Fig. 6 shows, with oil-pressure-valves controlling the entry of gas, combustion air and supercharge air to the combustion chamber, and the exit of the exhaust gases, and with its complicated steam cooling system, involving feed pump, water pre-heater, exhaust boiler, steam separator, steam circulating pump and superheater, together with the steam turbine and condensing plant, is extremely complicated. In spite of this, considerable success in practice has been achieved; a unit of 2,000 h.p. using blast furnace gas, to drive the pressure blowers of the furnace has been in service since 1933, while one of 5,000 h.p. was put into service in 1940. This type can thus be said to have justified the efforts expended in its development.

The constant pressure type of installation, with its manifest simplicity, is naturally receiving most attention to-day. Nevertheless, in the present early stage of combustion turbines, the authors take the view that a study of both successful types is desirable in order to guide future development. While restricting their treatment to the Holzwarth and constant pressure types, they feel also that, given future progress in such matters as heat resisting materials, it may prove profitable, at some later date, to re-examine other proposals for rotary engines which, probably for reasons connected with the contemporary state of technical skill at the times they were made, did not achieve practical success.

*See Stodola's book for other interesting types.

*This and other similar matters yet to arise have been treated at length elsewhere: cf. "Engineering", Vol. 156, pp. 401 and 421 (1943) and Vol. 157, pp. 181 and 221 (1944).

†cf. Cassier's Magazine, 1907-8, pp. 187 and 617.

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The ideal cycles of constant pressure and constant volume types will first be considered and then the practical conditions causing departure from these will be discussed. In the pressure-volume and temperature-entropy diagrams of Figs. 7 and 8, A B C D gives the cycle of a constant pressure installation and L M S N D that of a constant volume installation. In A B C D, T_B and T_D are, respectively, 1,500 and 525° F. abs., while the upper and lower pressures are 60 and 15 lb. per sq. inch, respectively. In L M S N D, T_D is 525° F. abs. and T_M is 3,600° F. abs. The pressure at D is 15 lb.

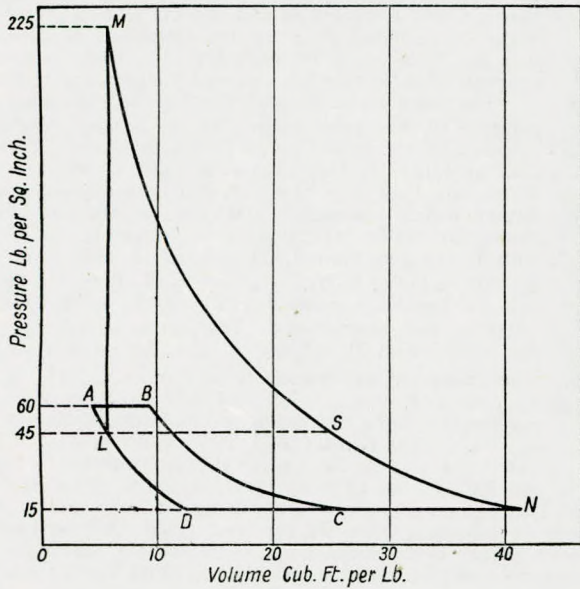


FIG. 7.

per sq. inch and that at L is 45 lb. per sq. inch. These are all values from successful practical installations. Compression and expansion are assumed to be adiabatic, while, for the working substance, $C_p = 0.238$ and $C_v = 0.17$, giving $\gamma = 1.4$, throughout. With these assumptions, the assumed and calculated values of pressure, temperature and volume are:—

Points.	A	B	C	D	L	M	S	N
Pressures, lb./sq. in.	60	60	15	15	45	225	45	15
Temperatures, °F. abs.	780.3	1,500	1,007.6	525	718.7	3,600	2,280	1,666.6
Volumes, cu. ft./lb.	4.81	9.23	24.84	12.96	5.91	5.91	18.8	40.85

The efficiency of the ideal constant volume air cycle is given by:

$$\eta_v = \frac{C_v(T_M - T_L) - C_p(T_N - T_D)}{C_v(T_M - T_L)} = \frac{0.17(3,600 - 718.7) - 0.238(1,666.6 - 525)}{0.17(3,600 - 718.7)} = 0.443;$$

similarly, the efficiency of the ideal constant pressure air cycle is given by:

$$\eta_p = \frac{C_p(T_B - T_C) - C_p(T_A - T_D)}{C_p(T_B - T_A)} = \frac{(1,500 - 1,007.6) - (780.3 - 525)}{1,500 - 780.3} = 0.329.$$

The constant volume cycle is, of course, passed through periodically, while the constant pressure cycle is continuous. For the combustion chamber of the former to begin to receive a fresh compressed charge, the pressure, on the expansion curve, must fall below S, where the pressure equals that at L. At high speeds of revolution, however, the pressure of the gases up-stream of the turbine wheel may not fall greatly below this value. If the temperature conditions of the turbine runner are to be similar to those of the first stages of the constant pressure turbine, that is, at about 1,500° F. abs., it is clear from Fig. 8 that the gases after the instant of reaching maximum pressure and temperature must lose considerable quantities of heat, since the difference between T_M and T_B is no less than 2,100° F. This necessary loss of heat is so great that the steam side of the installation must be of vital importance to the overall efficiency and does not figure merely as an auxiliary part of the installation. Further, the temperatures of the gases losing this heat are very high, so that "high-grade" heat is available. The discharge of the gases from the combustion chamber takes place periodically, and, although there must be a considerable pressure drop through the

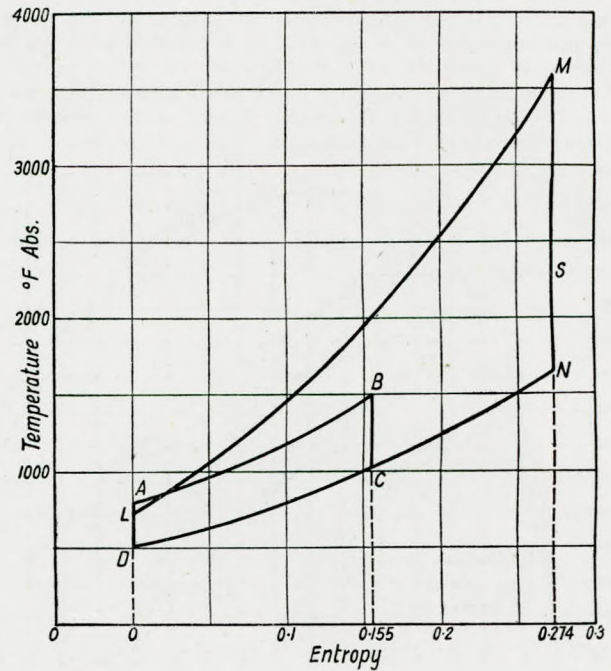


FIG. 8.

nozzle valve, the pressure fluctuations at the nozzles must be very marked; the release of the gases, of course, takes place from about the maximum pressure when the valves are opened and not, as in a reciprocating engine, after the gases have expanded behind the piston to a considerably lower pressure. The gas speeds through the turbine wheels will be subjected to periodic variations, so that, at constant speed of revolution of the turbine, the relationship between gas speed and blade speed must deviate from the optimum for efficiency.

Summarising the conditions for the constant volume type, it may be said: (1) that the basic advantages of the cycle are greatly reduced by the necessary loss of vast quantities of heat; (2) that, on account of the fluctuations of the gas discharge to the turbine, the turbine efficiency must be reduced from the normal maximum value; (3) that the high temperature heat loss is utilised in steam production, from which additional work is derived; (4) that, since these two sides of the installation, employing respectively gases and steam as working substances, are essential, the value of the overall efficiency of the installation, rather than that of the gas side alone, should be used for practical comparison. It will be interesting, when circumstances permit, to learn more concerning the test performance of the two large Holzwarth installations in Germany, on which the authors find no information.

As regards the relationship between the actual efficiency of the cycle of the constant pressure installation and the ideal efficiency of the cycle A B C D, the deviations from ideal conditions are much smaller. Neither compression nor expansion are adiabatic, but adiabatic efficiencies of 0.85 are practicable in each case. Variations of the specific heat of the gases have a small effect and, assuming the heat to be added by burning the fuel in the compressed air, the mass of the gas to expand is greater than that of the air compressed, by the weight of the fuel. On this basis, the efficiency becomes 0.177, which gives an efficiency ratio of 0.537 compared with the "air standard" efficiency value of 0.329. The authors have dealt elsewhere* with the efficiencies of constant pressure combustion turbines, so that it is only necessary here to give the more important matters emerging from their calculations.

The characteristics of such constant pressure machines are different from those of other power units with which designers have to concern themselves. In ordinary reciprocating internal combustion engines, an increase of compression ratio, other things being equal, leads to an increase in thermal efficiency and in the specific output per lb. of air, and, since the constant pressure combustion turbine works on a Joule cycle, it would be reasonable to expect the same conditions to apply. On page 4 the efficiency was given as:—

$$\eta = \frac{C_p(1+w) \eta_t (T_B - T_C) - C_p \frac{1}{\eta_c} (T_A - T_D)}{C_p(1+w) (T_B - T_C)}$$

*Engineering: loc. cit.

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The numerator gives, in heat units, the work done, and the denominator the heat supplied per lb. of air. In a practical case, T_D is the temperature at which the air is supplied and T_B is the highest permissible temperature of the cycle. With given values of T_B and T_D , η_o and η_t , the values of T_A , T_C and T_G depend on the pressure ratio, R , of the compression and expansion stages. R is thus a natural

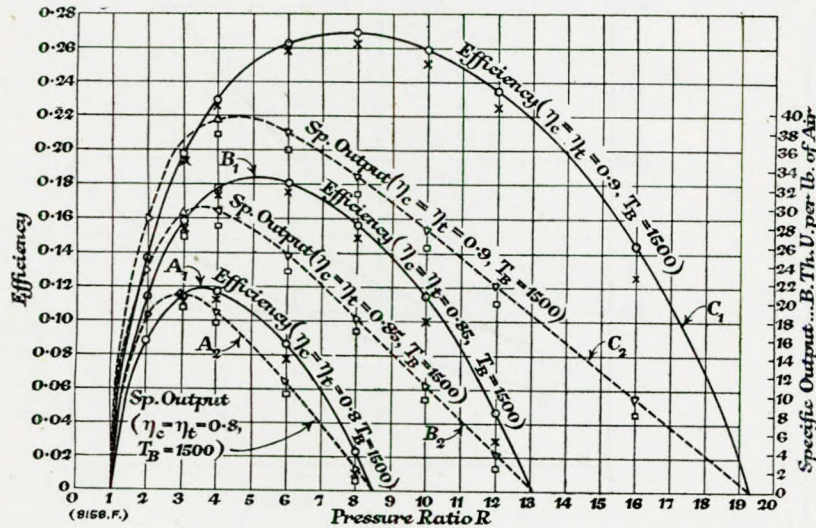


FIG. 9.

base on which to plot the values of efficiency and output per lb. of air under the given conditions. Fig. 9 shows three pairs of curves, A_1A_2 , B_1B_2 , C_1C_2 , respectively, corresponding to three pairs of values: $\eta_o = \eta_t = 0.8$, $\eta_o = \eta_t = 0.85$ and $\eta_o = \eta_t = 0.9$, changes of specific heat being taken into account; the firm curves give efficiencies and the broken curves give specific outputs. All curves begin at zero for $R=1$, increase to a maximum and again decrease to zero; further, the maximum value for efficiency for a pair always occurs at a higher value of R than the maximum value for output. These peculiar characteristics follow from the constant value taken for T_B the highest temperature of the cycle.

A designer, fixing the temperatures T_B and T_D and knowing the respective values to be expected in his design for η_o and η_t , may then fix his pressure ratio to give either maximum efficiency or maximum specific output, depending on the relative importance of fuel consumption and engine size, or he may compromise between these two values.

The three pairs of curves in this figure show the marked influence upon overall efficiency and the somewhat smaller influence upon the specific output resulting from improvements to the component efficiencies η_o and η_t . In the calculations, it was convenient to use equal values of η_o and η_t for each pair, but the separate influences of

the same percentage increments to η_o and η_t respectively are not equal; under the broad conditions given, the effect of a given increase of η_t is about 60 per cent. greater than that of an equal increase of η_o .

The crosses and squares plotted in the figure give, respectively, the values of efficiency and of specific output resulting from calculations made with air only, assuming constant specific heats and $\gamma=1.4$, that is, "standard air" throughout, with which the calculations are greatly simplified. Comparison shows that the actual efficiencies and outputs to be expected rule higher than those given by the simplified treatment, so that the latter may be regarded as a safe method of approximation within this order of temperature range.

The value to be adopted for T_B , the maximum temperature of the gases passing to the turbine component, is generally recognised as of vital importance. $1,500^\circ$ F. abs., or $1,040^\circ$ F. (560° C.) is a value which, according to the late Professor Stodola*, proved satisfactory in the Brown Boveri Company's 4,000 kW. installation. Fig. 10 shows the results of calculations, assuming $\eta_o = \eta_t = 0.85$, with T_B ranging from 1,400 to $1,700^\circ$ F. abs. Four pairs of curves, G_1G_2 , H_1H_2 , J_1J_2 , and K_1K_2 , show the efficiencies and specific outputs for $T_B = 1,400$, 1,500, 1,600 and $1,700^\circ$ F. abs., respectively. The pair of curves C_1C_2 give the results with $T_B = 1,500^\circ$ F. abs., but with $\eta_o = \eta_t = 0.9$. The maximum efficiencies from curves G_1 , H_1 , J_1 , and K_1 , are 0.16, 0.183, 0.205 and 0.224 respectively, while the corresponding maximum specific outputs, from curves G_2 , H_2 , J_2 and K_2 , are 23.3, 29.7, 36.8 and 44.2 B.Th.U. per lb. of air; so that, while the increments of efficiency per 100° F. rise of temperature decrease, those of output increase. The percentage improvements to output are relatively higher than those to efficiency, being 24 per cent. from 1,500 to $1,600^\circ$ F. abs. against 12 per cent. Comparison of curves C_1C_2 with the four pairs of curves emphasizes that greatly increased overall efficiency follows an increase of η_o and η_t , but that to increase output a higher maximum temperature is more effective.

All these results relate to the constant pressure installation in its simplest form. Since pressure ratios, and not actual pressures, are employed, the results apply whatever the actual pressure and density of the working substance. One other variable requires consideration, especially in relation to marine practice, namely, the temperature T_D of the air supply. Calculations show that the changes to the positive work per lb. of air performed in the turbine are negligible with increase of T_D , but that the negative compression work increases directly with T_D . In Fig. 11, curve A_1 , shows the variation of overall efficiency and curve A_2 the variation of specific output from $T_D = 500$ to 600° F. abs., that is, from 40 to 140° F., for the case in which $\eta_o = \eta_t = 0.85$ and $T_B = 1,500^\circ$ F. abs. Both curves, if produced, cross the base line at $T_D = 738^\circ$ F. abs. Considering the change from 40 to 140° F., the falling off of output is from 33.5 to 19.5 B.Th.U. per lb. of air, a reduction of 41.7 per cent., and the falling off of overall efficiency is from 0.187 to 0.148, a reduction of 20.8 per cent.

But, assuming no change of revolution speed of the compressor, the weight of air taken in would also be reduced from 100 at 40° F. to 83.3 at 140° F., and the output of the installation would be reduced further in this proportion. Curve B shows the overall equivalent output, assuming constant speed, taking as datum, 100 at 500° F. abs. If the speed ratio of driving shaft to propeller shaft is constant, there would, of course, be a decrease in propeller speed, following the reduced torque, and a still further reduction in power output, until equilibrium between torque and propeller speed had been attained. These changes are obviously of great importance in connection with the application of combustion turbines to ship propulsion.

The simple combustion turbine working with constant pressure combustion may, with the addition of various components, be improved as regards overall efficiency and specific output, or both combined. Such improvements involve, in various degrees, increased complexity of the installation, with increased cost, weight, space and upkeep charges, and, for a given case, the best compromise will be sought. In Fig. 5 it is seen that the exhaust gases

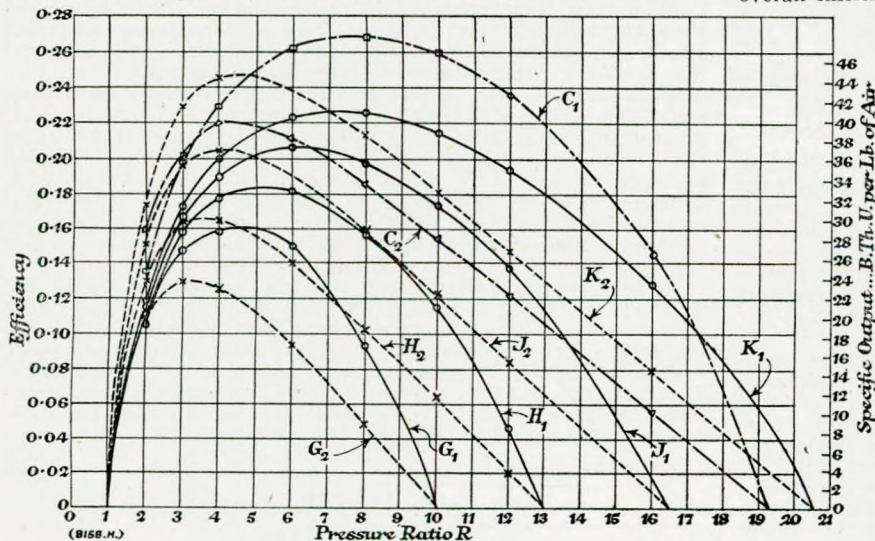


FIG. 10.

*"Engineering", Vol. 149, p. 1 (1940).

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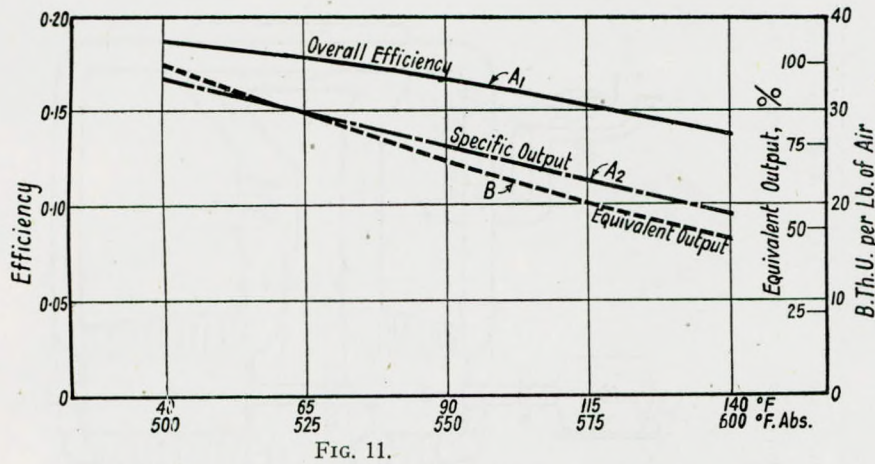


FIG. 11.

leave the turbine at a temperature exceeding 1,000° F. abs. (540° F.), and, in the simple case, the heat in these gases is wasted. Two ways in which some of this heat may be utilised are: (a) by fitting a heat interchanger so that the compressed air on its way to the combustion chamber receives heat from the exhaust gases; (b) by passing the exhaust gases through a waste heat boiler where some of the heat is used to raise steam, this steam being used to drive a steam turbine, to develop additional power. The overall efficiency and specific output of the simple machine may also be improved by improvement to the separate components, that is to say, by (c) compounding the compression and intercooling between the stages and by (d) compounding the expansion and interheating between stages. The former involves an intercooler and a cooling medium as additional complications; in the latter, it is necessary to provide a second combustion chamber, into which the gases at the end of the first stage of expansion may pass and in which more fuel is burnt, in the ample surplus oxygen still present in the gases, to raise the temperature of the gases to its initial value, before the gases pass to the second stage of the complete expansion to atmospheric pressure. Methods (c) and (d), either separately or together, may be combined with either of methods (a) or (b) for utilising the heat of the exhaust gases, but not with both. Nevertheless, the gases, on leaving the heat interchanger and the exhaust boiler, are at temperatures of the order of 450 and 350° F. respectively.

The authors have made calculations of a kind similar to those above for eleven combinations, and Table I gives the results of this study. Fig. 12 shows diagrammatically the organs and circuits of combinations, 1, 2, 3, 9, 10 and 11. The following conditions and limitations apply to the data upon which these results are based. Each individual set of conditions results in pairs of curves similar to those

of Fig. 9, from which a particular value of R, the pressure ratio, is taken so as to give a compromise between optimum efficiency and optimum specific output. Intercooling is defined as 100 per cent. efficient when the initial temperature is recovered by the air; since this is only exceptionally realised, in addition to 100 per cent., the values for 50 per cent. cooling are also given, this percentage relating to a recovery of one half the increase of temperature. In heat interchange between exhaust gases and compressed air, contra-flow conditions are assumed and the minimum difference of temperatures is taken as 120° F., radiation losses being neglected. In heat rejection from the exhaust gases to the exhaust boiler, steam is produced dry saturated at 30lb. per sq. inch abs. (15.3 gauge), at which the temperature is 250° F., and the gases leave at 350° F. Of the heat lost in the exhaust boiler, 90 per cent. was assumed to be received on the steam side.

Compounding the compression reduces the work of compression, and so increases the net output, by an amount which depends on the efficiency of intercooling, so that the improvement gained bears some proportion to the capacity of the intercooler. Since the temperature at the discharge from the compressor is lower, correspondingly more fuel must be used and this nearly offsets the gain in efficiency resulting from the increase in net output; the overall efficiency is thus only slightly improved. Compounding the expansion increases the output considerably, but more fuel is necessary, although, nevertheless, the improvement in overall efficiency is also worthy of consideration. Altogether, this measure is well justified if the complication of a second combustion chamber does not introduce excessive difficulties.

Utilisation of the exhaust heat in the two ways described has, respectively, two quite different effects. Transferring the heat to the compressed air has no appreciable influence on the output, since the positive work in the turbine and the negative work in the compressor each remain practically unaltered, but, since the temperature of the air entering the combustion chamber is higher, less fuel is necessary and the overall efficiency is correspondingly improved. The efficiency with which the heat is transferred depends on the size and good design of the interchanger. In all applications except the simplest, this method is to be recommended. Using the heat to raise steam introduces considerable complications. The case is not exactly similar to that of the Holzwarth installation, since the heat available is at a relatively low temperature, and high pressure steam cannot conveniently be produced; on the other hand, with the much greater weight of gases per lb. of fuel available with the constant pressure type, the gross heat may, nevertheless, be considerable. The ideal case for this method is one in which low pressure steam is necessary for other purposes. The effect on overall performance is that, without changing the quantity of fuel consumed, the utilisation of fuel is improved; with a steam turbine, for instance, additional work is available. Thus, following this increase of specific output, there is a proportionate increase in the overall efficiency of the installation, but the complexity of the installation is considerably increased.

Combinations of these methods are not matters of simple addition. For example, one incidental result of compounding the expansion is that the temperature of the exhaust gases is raised, so that more heat is available for transfer, and a new set of conditions is to be met with in the heat interchanger or in the exhaust boiler. Each combination, therefore, in the calculations in Table I is considered separately on its merits. The balance of conditions which enable a proper compromise to be made may be somewhat delicate; running costs are clearly not a simple matter of capital charges and running charges, while maintenance charges may also depend on the complexity of an installation. It should also be pointed out that the picture presented by Table I would be completely changed if, for example, it were found possible to work with $T_B = 1,600°$ F. abs., and a new set of calculations would then be necessary.

To complete this discussion of principles, mention should be made of the interesting proposals of Professor J. Ackeret and Dr. C. Keller for a constant pressure installation employing a closed cycle. Air or any more convenient gaseous substance may be used as the working substance, but, in describing the operations of the cycle, the working substance will, for brevity, be referred to as

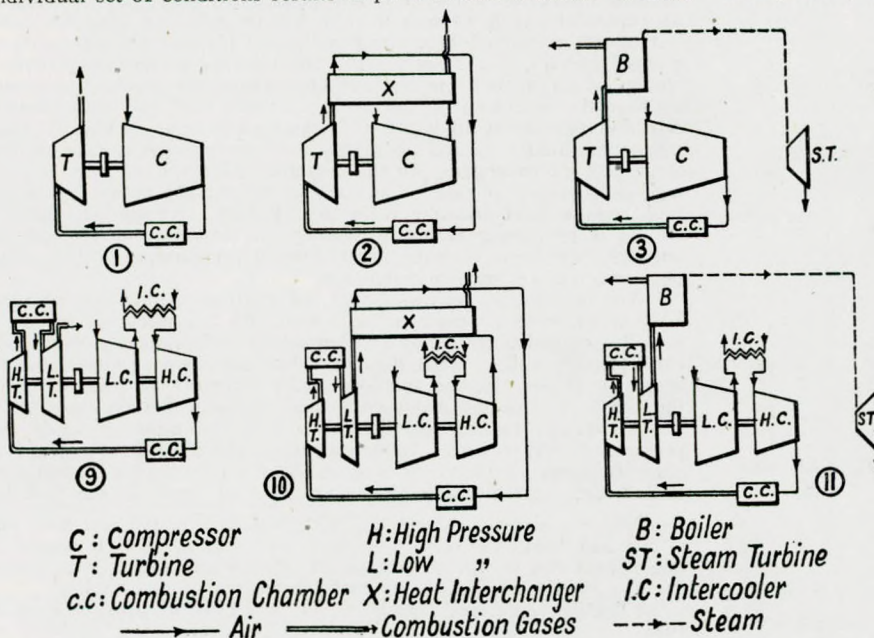
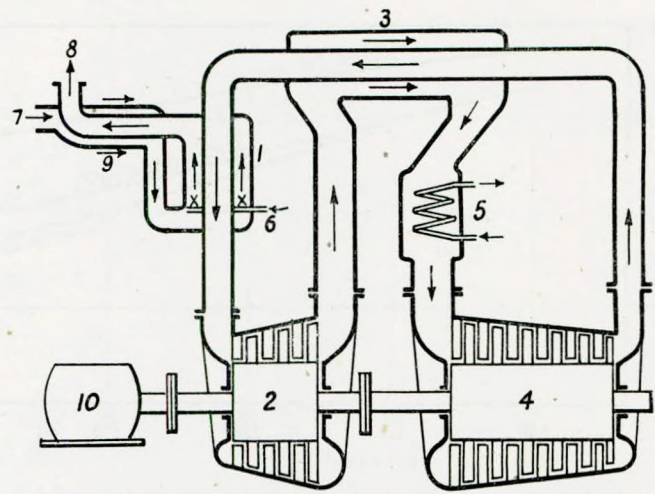


FIG. 12.

TABLE I.
 $T_B = 1,500$ deg. F. abs., $\eta_c = \eta_t = 0.85$.

Type of Installation.	1	2	3	4	5	6	7	8	9	10	11
Simple.	Simple.	Simple, with Heat-Interchanger.	Simple, with Exhaust Boiler and Steam-Turbine.	Compressor Compounding.	Turbine Compounding.	Turbine Compounding with Heat Interchanger.	Turbine Compounding with Exhaust Boiler and Steam-Turbine.	Compressor Compounding with Exhaust Boiler and Steam-Turbine.	Compressor and Turbine Compounding.	Combined Compounding with Heat Interchanger.	Combined Compounding with Exhaust Boiler and Steam-Turbine.
Extra components added to installation	Nil.	Heat Inter-changer.	Exhaust boiler. Steam-Turbine and auxiliaries.	Intercooler. Compressor in two stages.	Second comb. Turbine in two stages.	Second comb. chamber. Heat inter-changer. Turbine in two stages.	Second comb. chamber. Exhaust boiler and auxiliaries. Compressor in two stages.	Intercooler. Exhaust boiler. Steam-Turbine and auxiliaries. Compressor in two stages.	Intercooler. Second comb. chamber. Turbine and compressor in two stages.	Intercooler. Second comb. chamber. Heat inter-changer. Turbine and compressor in two stages.	Intercooler. Second comb. chamber. Exhaust boiler and auxiliaries. Turbine and compressor in two stages.
Efficiency	0.177	0.248	0.216	0.181	0.189	0.248	0.249	0.209	0.212	0.289	0.253
Fuel consumption, lb./h.p./hr.	0.787	0.562	0.645	0.770	0.737	0.562	0.560	0.667	0.657	0.482	0.551
Specific output, B.Th.U./lb. of air	29.6	29.8	37.6	31.4	39.1	38.8	44.7	40.5	45.6	42.3	53.0
Air, lb./h.p./hr.	86.00	85.4	67.7	81.1	65.1	65.6	56.9	62.8	55.8	60.1	39.4
Air, cu. ft./h.p./hr. (at 60° F. and 14.7 lb./sq. in.)	1103.8	1096.1	869.0	1041	885.6	842.0	731.0	806.1	716.8	771.9	506.3
Relative efficiency, simple case taken as 100	100	140	112	102	107	140	141	118	117	163	143
Relative output per lb. of air, simple case taken as 100	100	100.6	127	106	132	131	151	137	152	143	179



1. Combustion Chamber
 2. Turbine
 3. Interchanger
 4. Compressor
 5. After-Cooler
 6. Fuel Supply
 7. Combustion Air Supply
 8. Exit for Combustion Gases
 9. Air Preheater
 10. Elect. Gen'r

FIG. 13.

"gas". This is being developed at the works of the Escher Wyss company and a diagrammatic arrangement is given in Fig. 13. In passing around the cycle, the gas, at the upper region of pressure, receives heat from an external combustion chamber (1), and attains its highest temperature while on its way to the turbine (2), where it performs positive work and, of course, suffers a fall of pressure to the lower region of pressure. The used gas from the turbine passes to the heat interchanger (3), where it gives up heat to the compressed gas being delivered from the compressor (4), after giving up heat in (3), the used gas passes through the after-cooler (5), and having reached its lowest temperature, is then ready to pass to the compressor (4), from which, after compression, it is delivered, by way of the heat interchanger (3) and the combustion chamber (1) again to the turbine (2). The fuel supply to the combustion chamber is at (6), and the fuel may be solid, liquid or gaseous. The combustion air enters at (7), while the products of combustion leave at (8). In leaving, these gases pass through an air pre-heater (9) in which they lose heat to the combustion air. (10) represents an electric generator driven by the installation.

The advantages, in addition to the possibility of using solid fuel, are: the working substance undergoes no change of composition and remains clean; since there is no communication between the working substance and the atmosphere, any desired pressure may be adopted, an expedient which, as seen in p. 6, has no influence upon efficiency, but allows improved heat transmission in (3) and (5) and with increased density of the gas reduces the machine dimensions; further, change of output is brought about by changing the level of pressures, so that the efficiency is independent of the load and thus remains sensibly constant at all loads. A drawback is that all heat is transmitted externally to and from the working substance, necessitating large heat interchangers, but the inventors counteract this by employing special pipes of small diameters in those components, for which high overall heat transfer rates are claimed. Altogether, there is so much promising technical novelty in these proposals that the authors look forward with great interest to learning of the actual performances of such installations.

The position as regards actual and possible applications of combustion turbines in practice, following the classification given on p. 3, is somewhat as follows: Completely independent power units, class (a), are still in what may be called the "service experimental" stage for power stations and locomotive practice*, but, for aircraft, they have reached the production stage in Great Britain, the U.S.A. and Germany; marine installations are in the stage of study and preliminary experiment. Industrial units (b) have been applied to blast furnaces in Germany and in America to supply compressed heated air, blast furnace gas being used as fuel; in the U.S.A.†

*Meyer; Proc. I. Mech. E., 1939, Vol. 142, p. 197 and Vol. 143, p. 393 and 1944, Vol. 150, p. 1, and Vol. 151, p. 179. It would be ungenerous not to pay tribute to Dr. Meyer and the Brown Boveri Co. for their contributions to many branches of this subject.

†Tucker; Am. Soc. Mech. E., O. and G.P. division, May 8-10, 1944.

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some 36 sets, each of 2,000 h.p. and upwards, have been applied in petrol refining; in the U.S.S.R. a combustion turbine set, designed by Kapitzka for the production of the necessary oxygen from liquid air, is embodied in a plant for the direct gasification of coal*. Auxiliary units (c), are well-known in the form of turbo pressure-chargers for reciprocating oil engines and as turbo superchargers for conventional aircraft engines; the Velox boilers have found application both in Europe and in America, and, similarly have passed successfully through the experimental stage. Under (d), turbine units using natural gas available under pressure have been employed in the U.S.A. to generate small powers. In such cases the working substance is at ordinary temperature and is inflammable, so that precautions must be taken, on the one hand against the effects of the low gas temperatures after expansion upon the materials of construction and the possibility of ice formation and, on the other hand against the risk of explosion. This and similar applications are thought to be capable of much extension, since the possibilities of power production from gases, available under quite low pressures, in turbines, and especially in small units, are not yet realised in industrial circles.

There is one matter in the general comparison of class (a) with the other classes (b), (c) and (d) to which attention may be drawn, namely, that while the former must completely justify itself in direct competition with existing power units, the employment of the latter often involves no competition of any kind, since they constitute an addition or auxiliary representing a direct gain to the principal installation. As a result, the order of their efficiencies is secondary, whereas in independent units this is usually of primary importance; the development of (b), (c) and (d) has thus not been competitive. This has been fortuitous as regards its relationship to (a), but it is certain that the experience gained in practice with units (b) and especially (c) has had, and will have, a beneficial influence on the development of (a).

The authors propose to limit themselves, as regards the discussion of independent power units, to their application to marine propulsion. As Table I shows, there is a considerable latitude in choice of arrangement, and, of the combinations given there, No. 10, for the purpose of concentrating the discussion, will receive special consideration. The arrangement of components, too, offers various

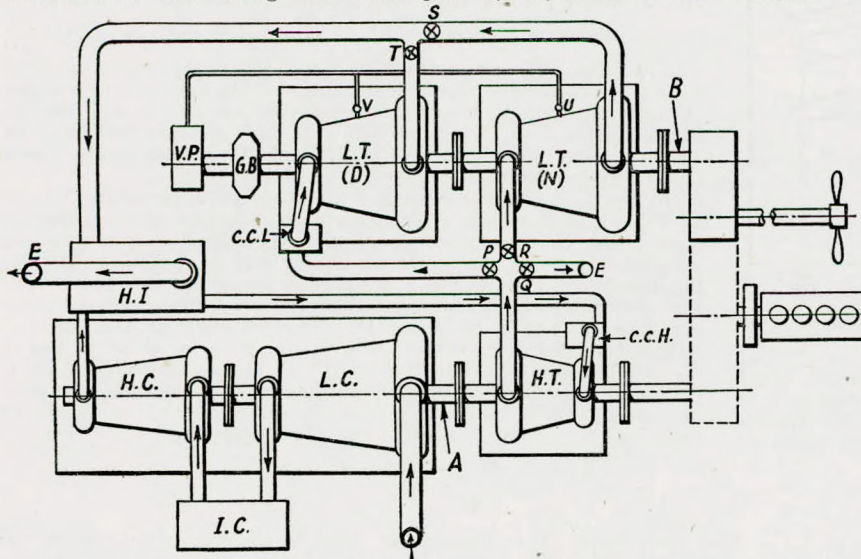


FIG. 14.

possibilities and, with No. 10, a two-shaft lay-out will be the principal subject of study; such a layout is shown in Fig. 14. One shaft, A in Fig. 14, will carry both stages of the compressor L.C. and H.C., and one stage of the turbine, H.T., which, under certain conditions, will provide just enough power to drive the compressor; this shaft will be called the "compressor shaft". The second shaft, the "power shaft", will carry the other stage of the turbine and will deliver the power. L.T.(D) is the ahead turbine and L.T.(N) is the astern turbine, if such is fitted. I.C. is the intercooler of the compressor; H.I. is the heat interchanger; C.C.H. and C.C.L. are combustion chambers. Other components, etc., will be referred to later. As an

*Dr. W. Davies, of the Royal School of Mines, will read a paper on this subject to the S. Wales Branch of the I.Mech.E. at Swansea, on 14th November, 1944.

example, a unit to deliver 3,500 s.h.p. will be taken, in which the power shaft will make 3,000 r.p.m., reduced at the propeller shaft to 125 r.p.m. This may be the only unit for a single-screw ship or one unit of a multi-screw ship. The power available at the power shaft can be transmitted to the propeller shaft by three principal methods: (a) by electric transmission, under which the speed of the power shaft will be constant and uni-directional in motion; (b) by reduction gears and a variable-pitch reversible propeller*, with which the power shaft is unidirectional and the speed changes of the power shaft are much less proportionally than those of the propeller shaft; (c) by reduction gears, with which all necessary changes of speed and reversals of the propeller shaft must take place proportionately in the power shaft. The unit is considered to be fitted in a merchant ship, so that the full load design conditions are those normally demanded of it.

The principal advantages of a two-shaft arrangement are:—

(1) The overall length of the installation is greatly reduced and there is considerable flexibility in the disposition of the components in the engine room space. This applies with all methods of transmission.

(2) With method (c), only one astern turbine is necessary and only one turbine is running idly. The inertia of the rotating masses of the power shaft is correspondingly reduced.

(3) Speed changes of the propeller shaft only affect the power shaft with methods (b) and (c) and do not directly affect the compressor shaft.

In a single shaft arrangement, with method (c), all units would be duplicated for ahead and astern working and all would follow the speed changes and reversals of the propeller-shaft, and with method (b) all would undergo the same speed changes; so that a two shaft arrangement is essential with (c) and offers considerable advantages with (a) and (b). The selection of the stage of the turbine, high-pressure or low-pressure to be placed on the power shaft and of the corresponding pressure ratios to be adopted for each stage are, however, matters which demand careful consideration.

Fig. 11 revealed the remarkable degree of sensitiveness, as regards output and efficiency, of the simplest combination No. 1 in Table I to changes of the temperature of the air supplied to the compressor. After selecting combination No. 10, it appeared desirable,

therefore, before determining the final arrangement of the components, to investigate similarly the effects of changes in supply temperature. Fig. 15 shows the results of this analysis, for three values of the total pressure ratio, R , plotted as curves of specific output and overall efficiency on a basis of air supply temperature for the range 500° F. abs. (40° F.) to 600° F. abs. (140° F.). Curves B and B' show the specific outputs and curves B₁ and B'₁ the overall efficiency with $R=4$, the pair B and B₁ being for 100 per cent. intercooling and the pair B' and B'₁ for 50 per cent. cooling. Curves C are for $R=6$ and curves D for $R=8$, the dashes and suffixes following the same conditions as for curves B. Curves A and A₁ are the curves of the simple case of Fig. 11, reproduced for comparison. In all cases $\eta_c = \eta_t = 0.85$ and $T_B = 1,500^\circ$ F. abs.

In compounding the compression, the intercooler pressure is the geometric mean of the upper and lower pressures, giving the highest efficiency of compression; in compounding the expansion, the intermediate pressure has been selected so that, with an air supply temperature of 525° F. abs. (65° F.), the work in the high pressure stage of the turbine just balances the work of compression. The compression pressure stages were thus: 1-2.4; 1-2.45-6; 1-2.83-8, in the three cases B, C, and D, respectively; the corresponding expansion pressure stages were found to be 4-1.75-1, 6-1.87-1, and 8-2-1, respectively, assuming 100 per cent. cooling throughout, stages which, as an approximation, are sufficiently close also for 50 per cent. cooling.

The curves show that, as regards specific output, the lower the overall pressure ratio R , the less the falling off of specific output with increase of supply temperature; further, the reduction of specific output and efficiency following from a decrease of the efficiency of intercooling from 100 to 50 per cent. is least with B, with the lowest pressure ratio. Nevertheless, when the actual values of specific output and efficiency are considered, it is seen that the curves of group C, with $R=6$, show the best compromise, and all further calculations were made using this pressure ratio. Curve C', with 50 per cent. intercooling, shows a reduction of specific output of about 35 per cent. over the extreme range of temperature from 40 to 140° F., and curve C', about the same reduction in overall efficiency.

*"The Motor Ship", April, 1943, p. 6, and April, 1944, p. 8.

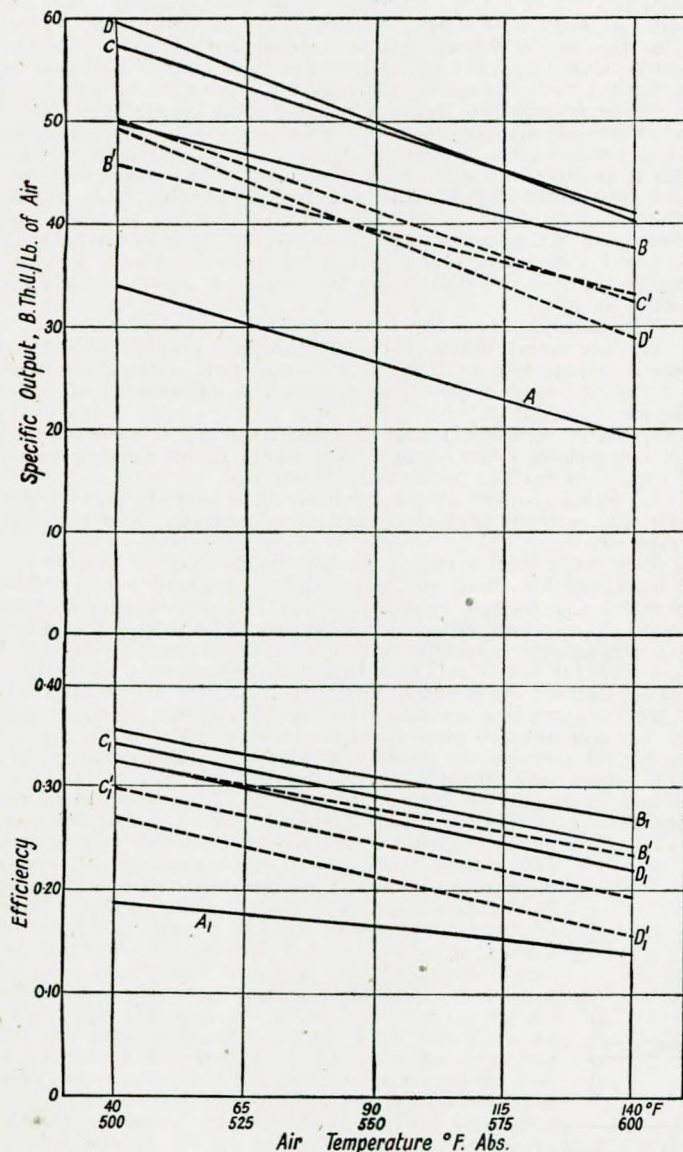


FIG. 15.

A further result of changing the air supply temperature, which is, however, not revealed by the curves, is a serious disturbance of the balance of work between the different stages of the turbine and the compressor. As was stated earlier, the primary cause of the falling off of specific output and efficiency with increase of the supply temperature is the fact that the work of compression, in any stage, varies very closely in direct proportion to the absolute supply temperature. It follows that, as soon as the absolute temperature of supply rises above the value 525° F. abs., at which the work done by the high pressure stage of the turbine exactly balances the work done in both stages of the compressor, two things happen: the weight of the air charge, at constant speed, is reduced inversely as the absolute temperature of the supply, which leads to a proportionate reduction in the power output from each stage of the turbine; the work of compression, per lb. of air, increases in direct proportion to the

absolute temperature of supply, but, since the air charge is reduced in the same proportion, the total work of compression remains constant. It follows, therefore, that the power output from that stage of the turbine arranged to be exactly that required to drive the compressor at 525° F. abs. (the high pressure stage in Fig. 14) is insufficient, and the more so, the higher the temperature is raised above this value. Similarly, if the temperature falls below the equilibrium temperature, 525° F. abs., the stage in question will develop more power than is necessary for compression. Under these conditions a two-shaft arrangement, in the absence of special measures, can only work if the shafts are geared positively together, so that, as regards the distribution of power, this becomes equivalent to a single-shaft arrangement of the same combination. In this case the deficiency of power of the stage driving the compressor is taken from the other stage, or the surplus power of the former is added to that from the other stage, and the results given in Fig. 14 are exactly realised. In that figure, the outputs are specific outputs of the installation per lb. of air, and the actual output of the installation will be reduced in proportion to the air density. It was shown that, on curve C', the specific output falls proportionately by 35 per cent. as the temperature changes from 500 to 600° F. abs., that is, from 40 to 140° F.; the actual net output of the installation will thus fall to $(100-35) \frac{500}{600} = 54.2$ per cent. of its value at 40° F., assuming no change in the efficiencies of the individual components and no change of speed of revolution. Accompanying this large fall of output there is also the reduction of overall efficiency, already noted in curve C', of the order of 35 per cent.

The range of air supply temperature selected is wide but more than half this range might be experienced in travelling from Great Britain to India, and the consequent reductions of output and of efficiency are thus of an order which could never be accepted. The authors, therefore, have considered whether it is in any way possible to improve this state of affairs. A range of temperature from 40 to 110° F., formed the basis of their calculations and, at the lowest temperature, the speed of the compressor shaft was taken to be the same as that of the power shaft, 3,000 r.p.m. It is possible, by speeding up the compressor shaft in direct proportion to the absolute supply temperature, to ensure that the same weight of air, per unit time, is aspirated; for the upper temperature 110° F., the speed will be $3,000 \times \frac{570}{500} = 3,420$ r.p.m., it being assumed, as a convenient approximation, that the conditions remain otherwise unchanged. The results for four sets of conditions have been calculated and these are set out in Table II, and numbered I to IV. In all sets the high pressure turbine is on the compressor shaft. In Sets II, III and IV, supplementary power at the high temperatures is supplied by an oil engine.

In Set I, the only components are those of combination No. 10 of Table I, so that, with independent shafts, the high pressure turbine must drive the compressor under all circumstances; it must work satisfactorily under the worst conditions, that is at 570° F. abs., or the installation will not run at all at high temperatures. This condition determines the pressure ratios of the two turbine stages, which are given in line 2 of Table II. The efficiency at the "chosen" temperature, 0.218 at 570° F. abs. in this case, is given in line 3. As the temperature falls from 570° F. the compression work is decreased, and the high pressure turbine tends to produce a surplus of power, thus disturbing the equilibrium. This may be dealt with in two ways; (a) by supplying less fuel to

TABLE II.

Design condition	I		II		III		IV	
	Plant designed for 570° F. abs. No additional components.		Plant designed for 500° F. abs. Additional compressor power supplied by auxiliary oil engine.		Plant designed for 525° F. abs. Additional compressor power supplied by auxiliary oil engine.		Plant designed for 550° F. abs. Additional compressor power supplied by auxiliary oil engine.	
Pressure Ratio of Turbines	6 : 1.5 : 1		6 : 1.85 : 1		6 : 1.59 : 1		6 : 1.71 : 1	
Overall Efficiency at chosen Temperature	0.218		0.293		0.271		0.249	
Overall Efficiency at 500° F. abs.	(a)	(b)			(a)	(b)	(a)	(b)
	0.236	0.261	0.293		0.272	0.291	0.264	0.285
Overall Efficiency at 570° F. abs.	0.218		0.235		0.234		0.233	
Specific Output of Installation, B.Th.U. per lb. of air	36.0		50.4		45.2		41.4	
Supplementary Power by Oil Engine	0		864 h.p.		628 h.p.		304 h.p.	

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the combustion chamber C.C.H., Fig. 14, the inlet temperature to the high-pressure turbine can be so reduced that there is no excess of power; (b) by bleeding the compressor at a pressure equal to the inter-stage pressure of the turbines and supplying this part of the air, through the heat interchanger, direct to the low pressure turbine on the power shaft, thus reducing the work of compression and the power output of the high pressure turbine. Both methods involve a reduction of the power given by the high pressure turbine, leaving the low pressure turbine unchanged. The values of the overall efficiencies at 500° F. abs. are given in line 4, and are seen to be 0.236 with (a) and 0.261 with (b). Line 5 gives the values of the overall efficiencies at 570° F. abs.; in I, of course, it is 0.218, the same as in line 3. Line 6 gives the values of the specific output of the installation in B.Th.U. per lb. of air; these values are inversely proportional to the necessary sizes of the compressors and turbines. Line 7 gives the supplementary power to be supplied by the auxiliary oil engine; in I, of course, the value is zero.

In Set II, the plant is designed to give equilibrium between the power developed by the high pressure turbine and the compression work at the lowest temperature, 500° F. abs. In order to provide the

high. In IV, at a small sacrifice of efficiency, the supplementary power is reduced to 304 h.p. The engines both in III and IV can run at fairly high speeds, 500 r.p.m., say, since their load factors should normally be very low, and their weights and costs could be reduced relatively below those for similar engines called on for continuous running. Of the comparison between methods (a) and (b) of Sets I, III and IV, (b) gives the better efficiency, but (a) is simpler to carry out.

Summarising the results, it is clear that changes of temperature, and, in particular, high temperatures, have a marked effect on the performance of combustion turbines. Comparison between the figures in I at 570° F. abs. and those in II at 500° F. abs., at which the oil engine does not need to be coupled, indicates that the differences of efficiency and corresponding size for the same powers are considerable and render the combustion turbine much less attractive at the higher temperatures.

The operation of combustion turbines involves the passage of large volumes of gas at only moderate pressures through the components, and consideration must be given to what may conveniently be called the "hydraulics" of the design. In the installation under discussion the working substance passes through the air compressor, with its intercooler, the air side of the heat interchanger, the first combustion chamber, the high pressure turbine, the second combustion chamber, the low pressure turbine and the gas side of the heat interchanger. While it is obviously necessary to reduce fluid friction losses as much as possible, equal pressure drops in the various components do not exert equal influences upon the specific output and efficiency. This matter of the relative influences has been studied in relation to the installation of Fig. 14, and the results are plotted as curves of specific output, in B.Th.U. per lb. of air, and of overall efficiency in Fig. 16, on bases of pressure drop in lb. per sq. in. Five curves are shown, marked X=0, X=1, etc., the vertical differences between them giving the decreases caused by the corresponding pressure drops in the intercooler. The top curve of each set shows the results either of pressure drop, Y, between the compressor outlet and the intake to the high pressure turbine, or, on a base of three times the scale, of Z, the pressure drop between the high and low pressure turbines including the second combustion chamber. A drop of 1lb. in the latter has three times the effect of an equal drop in the former. With the two bases given, the effects of Y and Z may be added together. Thus the point B, on X=0, compared with point A, the highest point of all, may show the combined reductions of output and efficiency of, say, 3lb. per sq. inch on Y plus 1lb. per sq. inch on Z; equally, it would give the combined effects of 1lb. per sq. inch on Y plus 1lb. per sq. inch on Z, or 2lb. per sq. inch on Y plus 1lb. per sq. inch on Z. Similarly the point C, on X=2, compared with point A shows the reduction of output and efficiency caused by, say (2lb. per sq. inch in X) plus (3lb. per sq. inch in Y) plus (1lb. per sq. inch in Z). The pressure drop, on the gas side of the interchanger W, is 1.9 times as effective as Z in reducing specific output and efficiency, that is, a drop of 1lb. per sq. inch there is equivalent to a drop of 1.9lb. per sq. inch between the high and low pressure turbines. Thus, these relative effects are W : Z : Y = 5.7 : 3 : 1, so that, for example, in the heat interchanger, reduction of pressure drop on the gas side, Z, is vastly more important than reduction of pressure drop on the air side, Y. The orders of the reductions in specific output and efficiency is seen to be considerable, and careful design is obviously of great importance.

Although there is in the design of the components of a combustion turbine little that lies outside existing practice, it is desirable to discuss certain of the conditions of design. As regards turbo compressors, the axial type shows to better advantage than the radial type as regards efficiency, but the space taken is much greater. Whereas the Ljungström type of design would offer little advantage in space for the turbine component, it might be found worth investigating with inward flow for the compressor, since, apart from the saving in space, the reduction of the length of path of the air might lead to decreased losses and higher adiabatic efficiencies. Duplicating the driving turbine might, as regards blade speed, also be advantageous in large units. There is indirect evidence, too, that both as regards improved adiabatic efficiency and good delivery/speed characteristics, the continuous delivery rotary type of displacement compressor is worthy of consideration. The need to obtain the highest possible adiabatic efficiency of compressing air or other gases has not been generally realised until recently, and this becomes most important in compressors for combustion turbines; the total experience in this direction is still comparatively limited, and improvements might be expected to follow the intensive attack which would accompany a more general application of combustion turbines. Compressing in stages and intercooling between the stages leads to improved efficiencies and specific outputs, but only in proportion to the efficacy of the intercooling. The intercooler is a heat inter-

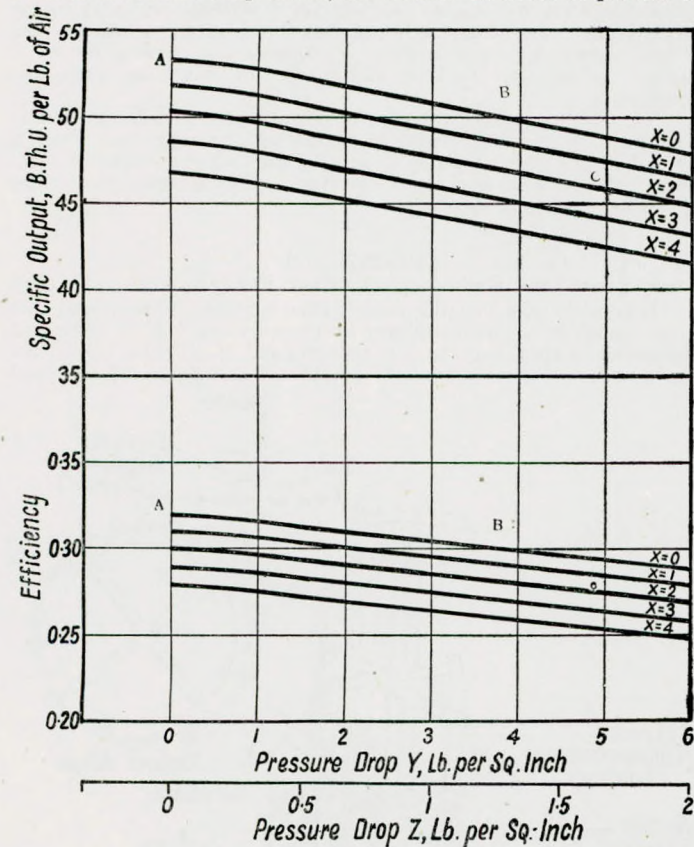


FIG. 16.

additional work necessary for compression as the supply temperature is raised, an oil engine is provided. At the highest temperature 570° F. abs., the supplementary power to be developed by the oil engine is no less than 864 h.p., that is, nearly one-quarter of the total power of 3,500 h.p. The overall efficiency at 500° F. abs. is high, namely 0.293, and this value decreases to 0.235 at 570° F. abs. (including the fuel supplied to the oil engine for which an individual efficiency of 0.30 is assumed). Comparing Sets I and II, it is seen that II gives much higher values of efficiency than I; the relative sizes of the turbines and compressors of I are 40 per cent. greater than those of II, but the large oil engine must be added to the latter. Altogether, the conclusion must be drawn that neither is very satisfactory: I, on account of its low overall efficiency and II, on account of the excessive supplementary power necessary.

Sets III and IV represent conditions which provide two examples of compromise between I and II. In III, equilibrium of the compressor conditions is taken at 525° F. abs. (65° F.), an average temperature, while in IV the equilibrium temperature is 550° F. abs. (90° F.). In III, the overall efficiencies are only slightly lower than in II, and the supplementary power necessary at 570° F. abs. from the oil engine is only 628 h.p., a more reasonable figure, but still somewhat

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changer in which air or gas at moderate pressure gives up heat to a cooling fluid. It has been shown that an excessive pressure drop in the gas flow may readily off-set these gains, so that the design of the intercooler should be aimed at extracting the greatest amount of heat from the gas without incurring excessive pressure drop; it is thus better to expend energy in circulating the coolant than in causing excessive hydraulic losses in the gas.

Apart from the pressure drop, the combustion chamber presents two problems: for good combustion, the fuel must be burnt in a region of high temperature, and then the hot gases must be mixed with colder air in the high pressure chamber, and with gases in the low pressure chamber, so that the temperature of the mixtures passing to the turbines shall not exceed the maximum permissible value for the blades. The operation is at sensibly constant pressure so that the hot gases of relatively low density must be uniformly mixed with the air of probably twice the density, and good uniformity is essential to avoid local overheating of the blades. The principles upon which the mixing process takes place have not, as far as the authors are aware, been worked out, and the matter remains one for trial and error.

As regards the rotors and blades of the turbine, considerably more reliable data concerning the creep strength of materials at high temperatures is desirable. Extended researches in this field would justify themselves. Possibly more information on this matter will be available in the near future.

These are matters which concern this type of power unit and are independent of its application. There are other matters demanding consideration when it is applied to ship propulsion, and especially those of speed and power control. Method (c) on p. 9, involving transmission by reduction gears with a fixed bladed propeller, will be considered, since the other methods of transmission involve relatively easier conditions. This arrangement requires an astern turbine, L.T.(N.), as shown in Fig. 14, and the windage losses, when this or the ahead turbine L.T.(D). are running idly, present a special problem. The authors would shut off the idle turbine completely and connect the casing to a simple rotary vacuum pump, V.P. Such a vacuum pump would be working continuously in one direction and the valves connecting it to the respective casings and the gear lever of the gear box G.B. would be interlocked with the control gear. Shaft A, when full power is or may be demanded, would be working at normal speed. If the ahead turbine, working at full speed, should be slowed down temporarily or stopped, its fuel would be correspondingly reduced or shut off, and valve P would be closed somewhat or shut completely. Valve Q, a spring loaded relief valve set just above the intermediate pressure, would then blow, the air being released to the exhaust E. To reverse, valve R would be opened, admitting the intermediate pressure gases, kept available by the action of valve Q, to the astern turbine, which might have no combustion chamber; with the exhaust gases at the intermediate pressure alone, the astern turbine would develop about 60 per cent. of the power of the ahead turbine. The fitting of a combustion chamber would, of course, allow this to develop 100 per cent. of the ahead power. Valve S would also be opened, connecting the astern turbine with the heat interchanger H.I. and the exhaust E, and valve T on the ahead turbine would be closed. Valve U connecting the astern turbine casing with the vacuum pump V.P. would be closed and valve V connecting the ahead turbine casing to the pump opened. A similar sequence of operations would be carried out to go ahead again. If it should be necessary to run for long periods at reduced speeds, either ahead or astern, the speed of shaft A would be reduced until valve Q just ceased to blow. There would then always be air available to accelerate shaft B, if it were desired to increase speed, but shaft A would, of course, also be increased in speed to correspond with shaft B.

The oil engine, if one were fitted, would be placed as shown, and would convey power through a clutch to the compressor shaft as necessary. If a second clutch were fitted to enable shaft B to be disengaged from the propeller shaft, the oil engine alone could propel the ship at low speed when coupled to the propeller shaft, in which case the turbine installation would remain at rest.

Finally, in the particular application of combustion turbines to marine practice, there are certain circumstances needing full discussion. It may be asked why the authors have based their calculations on a maximum temperature of 1,500° F. abs. (1,040° F.) in spite of the fact that there is common talk of much higher working temperatures. The answer is that, with the extreme reliability essential in propulsion units at sea, this is a proven practicable value. Combustion turbines, in common with most innovations, number their worst enemies among their over-optimistic friends, and it is better to advance slowly. It may, however, be pointed out that when, as a first step, it is practicable to permit, say, 1,600° F. abs. for limited intervals of time, this will result in an increase in output of 13 per cent., a useful overload value. As was pointed out earlier, any practicable raising of the maximum working temperature will greatly

improve the picture outlined here. The advantages of the combustion turbine may be summarised as: saving in weight and space; flexibility in arrangement of plant in the engine room space; simplicity, especially in the absence of engine auxiliaries; no need for feed water; few spare parts, necessary; reduced engine room staff and rapidity of getting under way. One drawback is the need to arrange for the supply of a vast quantity of air—over four times that necessary for an oil or steam installation of equal power—and to get rid of the corresponding volumes of exhaust gases. The latter, it may be noted, are themselves innocuous since they contain only about 2 per cent. of CO₂. On the other hand, engine room temperature should be lower if the air is allowed to pass freely to the compressors. For efficient control, the engine room staff should be given the fullest information concerning the running conditions of the machinery, and suitable thermometers should be fitted in all circuits, since with this type of power unit the usefulness of thermometers may be regarded as equivalent to that of pressure gauges with steam plants.

In the present subject we are mostly beginners. This country has already contributed worthily to the application of combustion turbines to aircraft; given suitable co-operation between metallurgists, engine designers, and engine users it can contribute similarly in the marine branch, and it may be hoped that this Institute, when the time becomes opportune, will initiate or support the setting up of a Marine Combustion Turbine Committee to stimulate such co-operation.

The authors acknowledge with thanks the permission kindly given them by the Editors of "Engineering" to reproduce Figs. 4, 5, 9 and 10 and part of Table I, and the loan of the blocks of these figures. They wish also to thank those friends who contributed helpful criticism and data.

APPENDIX I.

Extract from John Barber's specification, Pat. No. 1833, A.D. 1791.

"It consists of a metallic vessel called a retort, so contrived that (when heated by a circumambient fire) coal, wood, oil, or any other combustible matter may be put therein, and the smoke or vapour therein collected may be brought out by a small pipe, and conveyed

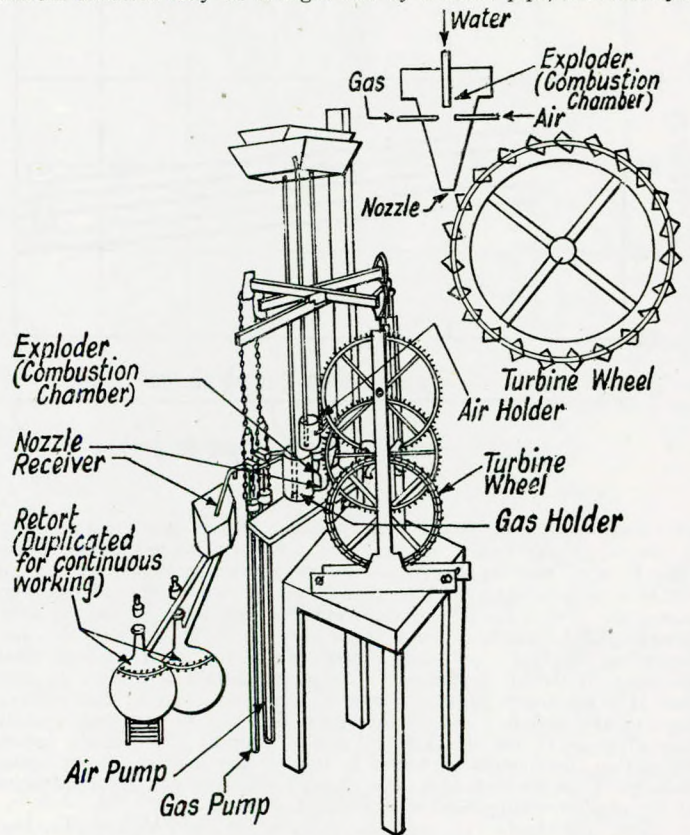


FIG. 17.

in a regular stream into another metallic vessel called an exploder, by means of an air pump and a compressor or regulating bellows, which pipe opposing its orifice to another similar pipe which enters the exploder (on the opposite side from the pipe which brings in

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the inflammable vapour from the retort) and injects by similar means a proper quantity of atmospheric or common air, causing an admixture of the two airs, which, so mixed, will take fire on application of a match or candle to the mouth of the exploder, and rush out with great rapidity in one continued stream of fire, so long as the exploder is supplied with proper quantities of the respective airs. The fluid stream is also considerably augmented both in quantity and velocity by water injected into the exploder by means of another small pipe entering therein, which water is also intended to prevent the inward pipes and the mouth of the exploder from melting by the velocity and intensity of the issuing flame. This water, as well as the airs, are forced into the exploder by means of a pump, which in lieu of common crank carries upon the axis of one of its

wheels two eses or double portions of circles, whereby a more regular motion is procured than can be done by any crank work. This engine is wrought by the stream issuing from the mouth of the exploder, and may be applied to grinding, rolling, forging, spinning, and every other mechanical operation; and the fluid stream may be injected into furnaces for smelting metallic ores, or passed out at the stern of any ship, boat, barge, or other vessel, so as by an opposing and impelling power directed against the water carrying such vessel, the vessel with its contents may be driven in any direction whatsoever".

For brevity, Barber's descriptive matter has been modified and the enlarged views of the combustion chamber and the turbine wheel have been added to Fig. 17.

Discussion.

The President, in opening the proceedings, said that to-night they were to hear a paper on a development which was assuming increasing importance and which was attracting ever greater attention in the engineering world, viz., that of the gas, or as the authors preferred to call it, the combustion turbine.

Professor Davies, who was well known to them, held the important position of Professor of Mechanical Engineering at the University of London, whilst before that he had a very extensive and varied career during which he had obtained much practical experience with aero and automobile engines. He was a D.Sc. and a Ph.D.

Dr. Fawzi was a Lecturer at Fouad University, Cairo. In addition to being a Bachelor of Engineering of that University he took his Ph.D. degree at King's College for research into two stroke oil engines. He had also carried out research work at the Research Laboratory of the Institution of Automobile Engineers and had had practical experience at the Associated Equipment Company, Southall.

In short, both authors were experienced in the theory and practice of internal combustion engines and members would listen to their opinions with great interest. For himself, this new development had brought home to him how new methods and new devices made things practicable which at one time seemed of little use. He remembered how many years ago one had looked on the hot air engine as an interesting toy and little more, largely because of the bulkiness of the reciprocating compressor and expansion cylinders. He remembered thinking that it would never be of any use, but rotary compressors and turbines had since arrived and had shown him how very wrong he was.

He was sure that those present would attend with the greatest interest to the practical problems that were discussed in the paper; personally, he wondered what was the authors' estimate of the vacuum which would permit of these machines being driven in the reverse direction. He mentioned this point because of the experiments made with steam turbines in this connection.

He had great pleasure in calling on Professor Davies to read the paper.

Dr. S. Livingston Smith (British Shipbuilding Research Association), who opened the discussion, said he was sure that marine engineers would regard the paper as very opportune, because the time had arrived, at any rate in his opinion, when they should give their attention to the possibilities associated with the application of the gas turbine to marine propulsion. It was constantly being said that new ideas nearly always emanated from abroad, and although, as the authors pointed out, this country had made a worthy contribution to the application of gas turbines to aircraft, the marine field had hardly been touched, and there was considerable danger, he thought, of its exploitation by other nations.

With regard to the paper itself, the various combinations of compressor, turbine, heat exchanger and so on, together with their possible efficiencies and fuel consumption, were not new to some members, and on the whole he agreed roughly with the authors' figures. He thought, however, that they were a little optimistic for the conditions chosen, due to the neglect of such factors as change in the specific heat of the gases and losses due to pressure drop through the plant, which, however, they pointed out. On the other hand, he thought that they erred on the conservative side in basing their calculations on a maximum temperature (which they called T_B) of $1,500^\circ$ F. abs., though he was fully aware of the several factors which imposed a temperature limitation.

As stated in the paper, the picture presented in Table I would be completely changed if it were found possible to work with a temperature T_B of $1,600^\circ$ F. abs. Personally, he would not class that as over optimistic by any means, and he thought it was a pity that the authors had not given complete figures for $1,600^\circ$ F. abs.,

and even for $1,700^\circ$ F. abs. By doing so they would have given a fairer picture of the possibilities of the application of the combustion turbine to marine practice, notwithstanding the fact that $1,500^\circ$ F. abs. might be, as was stated in the paper, "a proven practical value".

An all-important factor when comparing one form of marine propulsion with another was the power/weight ratio, and it was here that the gas turbine might show to great advantage. He asked whether the authors could give such figures relative to the example chosen, which was to deliver 3,500 shaft h.p. at 3,000 r.p.m. on the power shaft.

Dr. Smith said he would like to compliment the authors on a very interesting and stimulating paper.

Dr. H. Roxbee Cox (Chairman of Power Jets (Research and Development), Limited), expressed the hope that the paper would stimulate thought on the application of marine gas turbines, but said that his own contribution would be of a general rather than of a technical nature. Because he believed—and he thought the belief would be widely shared—that the application of the gas turbine in aircraft, in marine work, in land work and for traction purposes generally was of vital importance, he felt that it might be of interest to describe what had been arranged by H.M. Government to make it possible to carry out the research and development which they considered necessary if this country was to keep itself ahead, as he had reason to think that it was at the moment, in what was a very vital development which would have revolutionary effects on aircraft design and would be of the utmost importance in most other engineering industries.

As was well known, considerable progress had been made in this country in the last few years. It was in 1936 that activity really seriously started; at that time a considerable amount of work began at the Royal Aircraft Establishment on axial compressors, and in the same year Air Commodore (then Flight-Lieutenant) Whittle set to work on ideas which he had had in mind for several years previously. In both those cases the Government encouraged the activity, and when the war started a considerably greater impetus was given to the work. The result in the case of the Whittle type of engine, which had a centrifugal impeller, was now well known to the whole world. An aeroplane fitted with his engine flew in May, 1941, and since then developments had been very rapid indeed. It was also well known that a number of firms in this country had been engaged in aircraft gas turbine developments, but unfortunately it was not yet possible, for security reasons, to refer to them by name. It might be said, however, that they had done a remarkable job.

The Government concluded, on seeing how the gas turbine was being developed for aircraft, that it was necessary for the country to have a national gas turbine establishment for research and development, and that establishment was formed in a somewhat unusual way. It was decided that it should control the work of Power Jets Limited, the Company which had sponsored the Whittle engine, and early this year it purchased the business of that company. Then the turbine division of the engine department of the Royal Aircraft Establishment was added, both organisations coming under the control of a State-owned company known as Power Jets (Research and Development), Limited.

That Company was the Government's national organ for the prosecution of gas turbine research and development, and was at the service of all British industry. A board of directors was appointed which was independent of sectional and industrial interests, and he had the great privilege of being chairman of that board. It was advised by a Technical Advisory and Co-ordinating Committee which was composed of representatives of those industries actively concerned in gas turbine development, and it would be of particular interest to the Institute to know that the Admiralty was directly re-

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presented on the Committee and that nominations had been received and would shortly be confirmed from the marine industry.

He had been interested to note the authors' remarks at the end of the paper on the value of, and in fact the necessity for, collaboration, because the progress which had been made in recent years had been due largely to the wholehearted collaboration which had taken place between all the firms engaged on the aircraft side and all the associated industries, particularly the metallurgists. There had been what was called a Gas Turbine Collaboration Committee in existence since the end of 1941, and the present plan, under the general direction of the Technical Advisory Committee to which he had referred, was that there should be corresponding committees for land and sea affairs. Those committees were in fact about to start work. Moreover, the State company had for some time past been in contact with the research associations with which members of the Institute would be familiar—the British Shipbuilding Research Association and the more recently formed Parsons and Marine Engineering Turbine Research and Development Association. It was hoped that the contact would become more intimate, and that the State company would be able to be of very great help to the marine industry in the applications of the gas turbine which, stimulated by the present paper, he felt certain that industry must make in the near future.

There was no doubt that they had a very difficult course to steer. They had to face the most severe competition from overseas, and they would need all the enthusiasm and all the initiative which they could put into the work if they were to keep ahead. He would repeat that Power Jets (Research and Development), Limited was a Government organisation at the service of industry generally. The sooner that marine engineers knew all that it was possible to learn from work which so far had been mainly devoted to aircraft applications, the better he and his colleagues would be pleased.

Mr. P. G. Ware said that some of the points which he had intended to raise on behalf of Sir Roy Fedden, who was unable to be present, had already been dealt with by Dr. Roxbee Cox. Sir Roy particularly wished him to emphasise the necessity for getting sufficient energy and drive behind the development of gas turbines in this country.

It was important for engineers in Great Britain to go ahead immediately with some practical developments on the marine side, and see whether they could not lead in this field. Our friends and allies, the Americans, had an advantage over us inasmuch as they had greater industrial resources for such development work, but if they could get going now (and he had had the privilege of working with the President recently on certain projects for marine use which would certainly give scope for the development of the gas turbine), there was no reason why they should not establish, in the realm of these new prime movers, a position equal to that held by them in the steam turbine world.

The authors had referred to the closed cycle turbine, but did not mention one point, though no doubt fully aware of it, namely the need for pumping up and maintaining the whole system at the required working pressure, which would necessarily have to be high if good performance were required without excessive size. On a large plant, leakage from joints and shafts, seals, etc., would be considerable, and this might prove a serious disadvantage. Moreover, the heater itself would probably have to be supercharged to keep its dimensions down, and that would involve further complications.

Mr. W. J. Robinson, B.Sc., A.M.I.Mech.E. (Admiralty Research Department) found it difficult to agree with the authors' definition of a gas turbine. Although the choice of the term "combustion turbine" was a well meaning attempt to cover all types, it presented difficulties when discussing the advantages and disadvantages of the various heat cycles. In his view, gas turbines should for purpose of discussion be divided into *internal* combustion types and *external* combustion types, and internal combustion types should be further subdivided into constant pressure, constant volume, and piston generator types. He thought it was perhaps dangerous to try and put all these types into one class. For example, it was mentioned at the end of the paper that one of the disadvantages of the combustion turbine was the enormous amount of air required. Marine engineers would be very concerned with the problem of getting air in and gas out of a ship. If the quantity of air was "over four times that necessary for an oil or steam installation of equal power", then it might become a very serious matter. This objection did not seem to apply, however, to the external combustion type shown in Fig. 13 (the Escher Wyss type). The scheme suffered, however, from the disadvantage that the lowest temperature of the cycle would be fixed by the temperature of sea water or something higher than that temperature.

Another advantage of the external combustion scheme was that one could put a boiler into the combustion chamber exhaust and get steam without putting any back pressure on the turbine circuit.

He thought that the question of air intake temperature was very important. This was the first attempt he had seen to draw attention to the adverse effect of high intake temperature on performance, and he thought the authors were to be congratulated on their analysis. As regards the temperatures chosen, he could not imagine an intake temperature of the order of 140° F. In gas turbines (at any rate of the open cycle type) enormous quantities of air would be brought into the ship and the maximum temperature to be expected would, he thought, be the ambient air temperature.

On the question of air intake temperature, he wished to stress one point. Dr. Livingston Smith had already given an opinion on the possibility of higher gas temperatures than 1,000° F. abs. There had also been references in the technical press to American projects where temperatures of 1,200° F. to 1,500° F. were targets. When such gas temperatures became feasible, the effects of higher air intake temperatures would be less marked; in other words, raising the maximum gas temperature increased the temperature range of the cycle, and increases of intake temperature would have proportionately less effect. When turbines were designed to operate at higher gas temperatures, the effects on output and efficiency would therefore be less than those shown in Fig. 11. In Table I, as one moved along from Case No. 1 to Case No. 11, the plant tended to become more complex and the pressure drops would go up accordingly. In consequence, departures from the theoretical efficiencies would increase as one moved to the right and the efficiency figures were not therefore strictly comparable. He would like the authors to state whether, in Table I, the cycles were designed for optimum specific output or optimum efficiency.

On page 247, the authors stated "In compounding the compression, the intercooler pressure is the geometric mean of the upper and lower pressures, giving the highest efficiency". He would like the authors to confirm whether this gave highest efficiency of the plant? He wondered whether the condition stated was one of minimum work rather than one of highest overall plant efficiency?

With regard to Fig. 14, on which the authors based most of their discussion, it was stated that the arrangement shown would not work if the intake temperature rose above a certain figure. He had felt a little depressed by Fig. 14 since it implied that to make a gas turbine work it was necessary to fit a reciprocating engine. It was not possible at the meeting to discuss the details of the calculations, but he had found that the use of the reciprocating engine could be avoided by working the compressor at a slightly slower speed and a slightly lower pressure ratio, the gas inlet temperature remaining practically the same.

On page 247, the authors drew attention to different arrangements and said "that a two-shaft arrangement is essential with (c) and offers considerable advantages with (a) and (b)". He would be grateful if the authors would explain what were the operating advantages of putting a variable-pitch propeller with a compound turbine.

Those with more seagoing experience might like to express an opinion on the reversing arrangements and the valving. There was considerable scope for ingenuity in developing the reversing of a gas turbine for marine work and it would be interesting to know whether anyone had any different ideas from those given in the paper.

Dr. N. Booth (Gas Research Board) said that his Director (Dr. J. G. King) had prepared a contribution to the discussion but had unfortunately been unable to attend and had asked him (Dr. Booth) to read this contribution. He added that the gas industry were already in touch with Dr. Roxbee Cox to determine whether his organisation could be of assistance.

Dr. J. G. King (Gas Research Board) expressed the very considerable interest of the research side of the gas industry in the possible development of the combustion turbine.

The authors had discussed both constant pressure and constant volume cycles, although going into greater detail regarding the latter. He would like to ask them whether they could say which form would be more likely to be suitable if the original fuel were coal gas, or alternatively gas of the order of calorific value obtained by underground gasification processes. The calorific value of normal coal gas was about 480 B.Th.U./cu. ft. and from underground gasification was of the order of 100/150 B.Th.U./cu. ft. He understood that the Russians were contemplating the use of underground gasification gas in that way.

In the constant pressure system the compression of the air involved some thermal expenditure, and possibly in order to supply gas to the engine at a sufficiently high rate there would be a further thermal expenditure in the compressing of that gas. He would like to ask the authors, therefore, whether operation at constant volume

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might not be more suitable if gas were the fuel, because of the greater scope for saving compression costs.

The Escher-Wyss system might also be suitable where gas was the fuel. It seemed to hold out considerable promise, but a good deal must depend on the efficiency of heat transfer from the combustion chamber to the working fluid, and also upon the efficient design of a chamber to continue working under what must be fairly severe conditions of heat stress.

The gas industry at present produced a gas at about atmospheric pressure, but there were in progress experiments which were designed to explore the possibility of producing gas at pressures up to 50 atmos. The Lurgi process, which had achieved considerable success in Germany using brown coal as the fuel, would be experimented with in this country using British coals and conditions, and in addition the Gas Research Board were at present working with some success on a process for gasifying coal in hydrogen at pressures up to 50 atmos.

The gas industry was interested, therefore, in three of the classes of development indicated on page 246 of the paper. In class (a), the possibility of an independent power unit of small size operating in special circumstances, such as conditions in which there was an inadequate supply of cooling water. The fuel in that case would probably be gas made during off-peak periods, or possibly coke oven gas. In class (b), there was the possibility of the liquid oxygen—complete gasification project which was attributed to Kapitza, in Russia. He did not know how much information was available about that on which the authors could base a decision. In class (d), there was the possibility of the use of gas under pressure, in which the pressure drop might be from 50 atmos. (the pressure at which the gas would probably be produced) down to 10 atmos. It would be very interesting to hear what the authors thought of the possibility of the application in this country of combustion turbines in the conditions governed by these three classes.

Engineer Rear-Admiral D. J. Hoare (Visitor): While the authors gave a very clear picture of the current position regarding combustion turbines from a thermodynamic point of view, there were certain other aspects of the problem on which the marine engineer would like to be informed.

The principal attractions of the combustion turbine for marine use seemed to lie in the promise of better fuel consumption, the avoidance of the use of feed water and the elimination of main boilers and condensers.

The paper substantiated the first mentioned attraction and the second was obvious, but the extent of the third advantage was not so definite. Heat interchangers could be shown very conveniently on a diagram as inoffensive rectangles, but one would like to be sure in fact that to get the requisite performance as to heat transfer and pressure loss they, together with the compressors and air and exhaust ducts, would be a beneficial exchange for boilers with air intakes and funnels of current dimensions.

The turbines would no doubt be smaller in size than corresponding steam turbines but one must not lose sight of the additional lagging required on practically all hot surfaces and of the starting engines also required.

The positioning further aft of air intakes and funnels would no doubt only require getting used to but the point was important in warships, particularly as the ducts, at any rate in the case of the open cycle plant, were likely to be so much larger than in steam ships.

It was not suggested that these considerations were likely to absorb the balance of weight and space in favour of the combustion turbine; moreover, the authors had shown that the fuel consumption target alone justified serious effort.

With the elimination of the boiler, combustion now took place on the doorstep of the turbine. This constituted the most venturesome part of the scheme. The internal combustion engine had shown that combustion could be tolerated internally but in this case it alternated with draughts of intake air. In the turbine combustion chamber air dilution was continuous but must in the interest of efficiency be restricted to the limit of the turbine's ability to take it.

In prescribing a temperature of 1,040° F., the authors had used discretion, but they showed clearly that to get substantial advantage, higher temperatures than this must be employed. Higher temperatures, as they observed, had been talked about and probably had been made use of in aircraft, but the success attained with small short life turbines gave no ground for expecting early success with longer life turbines of outputs sufficient to compete with steam turbines afloat. Scaling up from aircraft scantlings was going to be a painstaking job for the designer, foundryman, forgemaster and machinist, and one wondered if the practical side were in fact keeping pace with the theoretical. The authors stated that more information concerning the creep strength of materials at high temperatures might

be available in the near future. This seemed to imply that work on test pieces had been in hand for some time, there being no short cut to reliable data on creep. Whether this were so or not it was quite evident that they were a long way astern of those foreign countries which had reached the most important stage—that of running the machinery.

On grounds of fuel consumption, the systems dealt with by the authors offered no immediate challenge to the internal combustion engine. Since the compression ignition engine had proved its amenability to high temperatures it seemed that systems which employed highly supercharged reciprocating engines to supply turbines with gas at temperatures appropriate to their state of development should be capable of quicker development. They should also achieve better fuel consumptions than constant pressure systems. It was certainly not suggested that the advent of either type justified neglect of the other; experience alone could decide the scope of each type.

Mr. F. J. Cowlin (Associate Member) (Chief Engineer, Turbine Department, English Electric Company, Rugby): The general results given in the earlier parts of the paper effectively summarised the known characteristics of the combustion turbine working on the open cycle, including the most convenient methods of improving its overall efficiency, and required little comment. They did, however, serve to bring out the three principal reasons why, in spite of the principles involved having been known for some 40 to 50 years, as mentioned in the authors' historical survey, development and application had been extremely slow.

The first of these reasons was the lack of an efficient compressor. Use of a centrifugal compressor limited to a maximum efficiency of, say, 70 per cent. gave too high a value to the negative work of compression, and it was not until the development of the axial flow compressor enabled efficiencies of 83 per cent. to 86 per cent. to be obtained that the negative work could be reduced to a suitable value. This type of compressor had, however, the disadvantage of a comparatively high surging limit, and because of this many engineers were watching with much interest the development of the displacement type Lysholm compressor. For some purposes, however, the additional weight and cost of this form of compressor might prove prohibitive.

The second reason for slow development was the low turbine efficiencies obtainable, but the steady development of the steam turbine during recent years was now being turned to advantage in the gas field. It might be mentioned in this connection that the combustion turbine, with its low pressure ratio and small parasitic losses, enabled much better efficiencies to be obtained than were possible even with modern high pressure steam turbines.

The third, and probably most effective brake on the development, had been the lack of suitable materials to withstand the comparatively high temperatures necessary to obtain reasonable overall efficiencies. When considering this matter it was essential that the life of the plant be given full consideration, and it was likely that aeronautical developments would seriously diverge from land and marine developments for this reason.

The position at present was that the engineer was definitely awaiting the metallurgist, not only for raw materials to withstand higher temperatures but for reliable data suitable for design purposes on the materials at present in use on steam plants.

One point worthy of note in connection with the advantages of regeneration in improving the cycle efficiency and not mentioned by the authors was that increase in the amount of regeneration was followed by a reduction in the optimum pressure ratio. This, again, reflected on the compressor efficiency and had consequently a dual effect in improving cycle efficiency.

Turning to the marine application of the combustion turbine, perhaps the greatest difficulty to be overcome was the method of transmitting the power from the prime mover to the propeller. Many alternatives were available and because of the poor partial speed characteristics of the combustion turbine there appeared little doubt that the ideal drive would be the variable pitch reversing propeller. Foreign developments in this direction appeared to be in advance of those in this country, and concentration on this development would materially ease the introduction of the combustion turbine for marine purposes. Next in order of suitability from a control and operational standpoint seemed to be an A.C. electric drive, already firmly established on numbers of steam driven vessels. D.C. drive, although suitable from a control aspect, produced difficulties in the construction of the driving motors and considerable increases in the weight of the plant.

The gear drive with reverse turbine, selected by the authors for detailed examination, was one of the least attractive arrangements, and undoubtedly brought out all the disadvantages of the combustion turbine drive. Whilst the two shaft arrangement shown was advantageous to all forms of transmission, the multiplicity of components

Development, Principles and Application of the Combustion Turbine.

introduced in the authors' examples would undoubtedly introduce serious practical difficulties in installation due to the large size of ducts, valves, etc., which were necessary with a combustion turbine, owing to the low pressures and large volumes of gas to be handled. The practical advantage to be obtained from intercooling could soon be completely negated by pressure drop through the intercooler and incidentally destroyed one of the greatest advantages of a gas plant over a steam plant, the fact that the large quantities of cooling water required for a steam plant were eliminated. This point would be appreciated by marine engineers familiar with outages on steam plant due to condenser troubles, and had been recently publicly commented upon by the President of the Institute.

In conclusion, there seemed little doubt that rapid advances might be anticipated in the development of combustion turbines for many purposes during the coming decade, and the crying necessity at the present time appeared to be for plants of various types to be built to test out theoretical predictions and enable operational experience to be obtained.

Mr. B. Pochobradsky (Visitor) (The Fraser & Chalmers Engineering Works—General Electric Company) said his remarks would apply to the single open circuit turbo type machine, since he thought that was the type most suitable for ship propulsion. He agreed with the authors that the principal components—the turbine, compressors, burners and heat exchangers—had proved themselves already in various practical applications. The ship propulsion application had its special problems and these should be examined in addition to the experience already available.

The authors suggested one such problem, the variation of atmospheric temperature. This problem applied to land installation though perhaps not in the same degree and frequency. It applied to the reciprocating oil engine which was affected adversely by higher air temperature. The water temperature entered also into consideration when intercoolers were used, and of course higher cooling temperature affected adversely a steam plant by reduced vacuum.

The authors showed in a praiseworthy manner the serious effect of increased atmospheric air temperature and they suggested an addition of an oil engine to supply the extra power required by the compressor and still to obtain full output from the propulsion turbine. He proposed a simple and efficient solution of this problem.

When speaking before the Institute a few months previously, he had referred also to the particular machine comprising two turbines arranged in series, somewhat similar to Fig. 14, though with A.C. electric drive in lieu of the astern turbine. The conditions of service were:

(a) normal air temperature 60° F., water temperature 60° F.;
(b) maximum air temperature 100° F., water temperature 86° F.

He considered that 1,200° F. at the turbine inlet would be to-day as safe and probably safer, in view of the progress in heat resisting materials, as 1,040° F. was at the time when machines of this kind were produced commercially for the first time. He assumed 60° F. margin for exceptional service conditions, and so obtained 1,140° F. (1,600° F. abs.) as available temperature at the turbine inlet. The compressor had one intercooler and the water temperature rise from normal to maximum was smaller than the air temperature rise; the compressor part after the intercooler would be affected by both air and water temperature; the input to the compressor with the maximum air and water temperature in that particular service would be about 6.7 per cent. higher; 1,140° F. or 1,600° F. abs. as available gas temperature for maximum air and water temperature would give with the given pressure ratio in the compressor turbine a heat drop 6.7 per cent. higher than with 1,500° F. abs. at the turbine inlet. To make the governing mechanism as simple as possible, it was convenient to design the turbines for 1,500° F. abs. at inlet (or 1,040° F.) which happened to be the same temperature the authors had chosen, with normal air and water temperature of 60° F.

It was necessary to keep in mind that the air delivered by the compressor (excepting leakage losses) passed successively through the compressor turbine and propulsion turbine; the compressor output was correlated to the output of its turbine. The rate of air (gas) flow through the system for a given machine depended on the pressure and temperature before the compressor turbine and on its back pressure, which differed from the inlet pressure of the propulsion turbine only by the pressure loss between the two turbines.

Reduced ratio of flow with unaltered gas temperature meant lower initial pressure and lower pressure before the propulsion turbine; as could be checked by the "law of ellipse", the total heat drop of the two turbines decreased with decreased rate of flow, but the decrease of the heat drop in the compressor turbine was much smaller (especially in percentage) than in the propulsion turbines; actually the compressor turbine would give more output than the compressor required with reduced rate of air flow.

Evidently, then, the compressor turbine was capable at some reduced rate of flow to drive a compressor supplied with warmer air and water; of course, the output of the propulsion turbine would be reduced. For the turbine under discussion with 1,500° F. abs. (1,040° F.) inlet gas temperature, but with air at 100° F. and water at 84° F. (instead of both at 60° F.), the plant would be perfectly balanced at about 75 per cent. of the normal output of the propulsion turbine, with about 88 per cent. of normal air quantity, the fuel consumption being reduced to about 82 per cent. of the full load consumption with normal air and water temperature. Naturally, the propulsion turbine would tend to slow down, but its speed governor, set for a particular speed, would prevent reduction in fuel supply; on the contrary, reduced speed would tend to increase the fuel supply, hence the gas inlet temperature would rise and the available heat drop would increase and that would tend to keep the plant at the set speed. A complete balance and full output were restored, with 100° F. air temperature and 84° F. water temperature, when the gas inlet temperature reached 1,588° F. abs., or slightly less than the available limit. For any intermediate air and water temperature between 60-100° F. and 60-84° F., the speed governor would establish a gas inlet temperature between 1,500 and 1,588° F. abs., the propulsion turbine giving unaltered full output. The fuel consumption at full load with 100° F. air and 84° F. water temperature would be about 1.6 per cent. higher than with normal 60° F. air and water temperature.

The same plant, designed for 60° F. air and water temperature, would still give full load and be perfectly balanced with 110° F. air temperature and 94° F. water temperature, the speed governor supplying enough fuel to raise the gas inlet temperature to 1,618° F. abs. (1,158° F.) which was still acceptable; should it be desired not to exceed 1,600° F., then the propulsion turbine would give a slightly reduced output (about 96 per cent. of the normal).

If, conversely, the air and water temperature decreased below the normal values, then the compressor turbine would give more power than required; it would speed up the compressor which would deliver greater air quantities at higher pressures, the speed of the propulsion turbine would rise, but the speed governor would reduce the fuel supply and so the gas inlet temperature would fall and again at some lower gas inlet temperature (below 1,500° F. abs.) complete balance was restored and the propulsion turbine would give normal output, with substantially unaltered speed.

Apart from this automatic power "balancing" induced by the speed governor, he considered another means to help in achieving the same object which was the provision of a "regulating" first stage of the propulsion turbine, a stage with variable nozzle area. By increasing this area the heat drop of the compressor turbine was increased, that of the propulsion turbine reduced, hence, if for 100° F. air temperature and 84° F. water temperature (instead of normal 60° F.), 1,588° F. abs. gas inlet temperature for full output would be required, by slight increase of the nozzle area we could obtain under the same conditions full output with about 1,550° F. abs. gas inlet temperature. This added means might be useful when exceptionally high air and water temperatures prevailed. Conversely, reducing this nozzle area, we could take advantage of the reduced compressor input with lower air and water temperature and so achieve improved fuel consumption, full output being secured with smaller air quantity and increased heat drop of the propulsion turbine.

The problem of varying air and water temperature could therefore be met by quite simple means.

Partial loads represented another problem common to ship propulsion and land plants. From the explanations in connection with varying air and water temperature, it was clear that the speed governor would reduce the oil supply, the gas inlet temperature would be reduced, the compressor would slow down, deliver less air at lower pressure, the propulsion turbine output being then reduced. If, in addition, his proposal of "nozzle regulation" on the propulsion turbine was applied, higher gas inlet temperatures would result with obvious improvement in fuel consumption at partial loads.

The series arrangement of the two turbines was only one special arrangement; there were other arrangements available, such as parallel arrangement, one turbine driving the compressor and using about two-thirds of the total gas, the propulsion turbine using the remainder of the gas. Some combinations of the two arrangements might be suitable.

The reversing with electric drive was a matter of suitable governing; this problem presented no insuperable difficulties. The well tried A.C. electric drive was quite suitable for gas turbines.

One problem peculiar to ship propulsion was the governing of the plant in heavy seas when the propeller kept on emerging and submerging. In a steam plant the speed governor closed and opened the control steam valves, while the boiler acted as an accumulator. In a gas turbine plant there was no such accumulator. No doubt

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there would be a number of proposals to meet this case and, it was hoped, some of them would be economical. For the first practical application he would suggest that a service should be chosen with a reasonably moderate variation of air and water conditions so as to avoid complications of governing. A form of power transmission should be chosen which allowed complete freedom between the compressor part and the propulsion part, and the propulsion machine should be unidirectional. That was obtainable by electric transmission; he would not advise the use of the variable pitch propeller, which he thought would be costly in the long run because of the necessity for more frequent inspections and less use of the ship; moreover, in many services it did not give any improvement in economy. A.C. electric transmission might show higher first cost, but its long life and reliability ensured more profitable service. In conclusion, he would advise that in the first place a "universal gas turbine" equipped with everything to meet any imaginable condition should not be built; it would be better to choose a service particularly suitable, bearing in mind that no prime mover had yet proved itself supreme in all services and under all conditions. A really successful first gas turbine plant would indicate in the shortest time means and ways for the solution of other more complicated cases.

Mr. E. Markham, Wh.Ex. (Member) (Director of Messrs. Peter Brotherhood, Ltd.), referring to Figs. 1 and 4 of the paper, mentioned that, as well known to engineers, the nearer one got to isothermal for the compression and expansion in a given piston type engine working on the Joule cycle, the greater would be the net power available. He was thus very surprised that so far all speakers, with the exception of one who had talked about town gas, had confined their remarks to air.

As had been pointed out, the drawback on the Joule engine was the small amount of net power obtained from quite a large unit so,

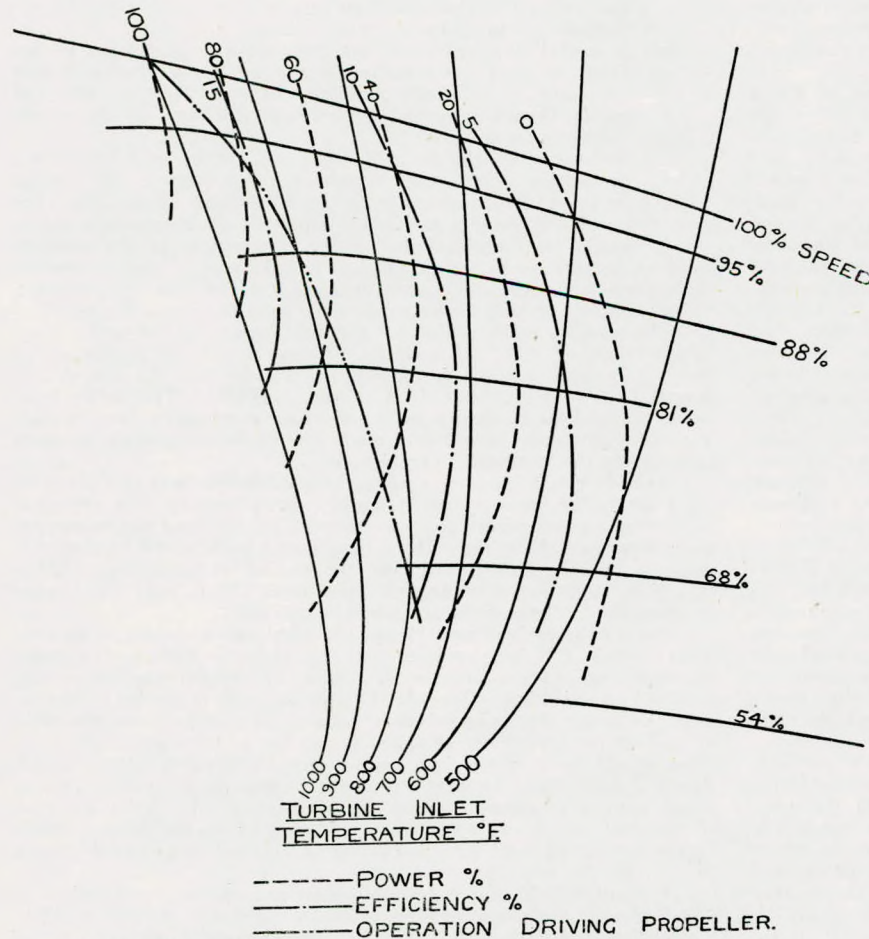
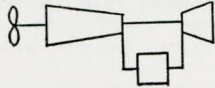


FIG. 18.—Partial load characteristics for simple gas turbine.

when working before the war on this type of engine using air on an open cycle, he had turned his thoughts to employing in a closed cycle some other substance which had a lower ratio of specific heats than air. He finally settled on a Freon (F12 or Dichlorodifluoromethane) which had a ratio of 1.138 as against 1.41 for air.

Thus, when this was compressed the compression curve was much nearer to the isothermal, thereby lowering the work of compression, and when expanded against a piston the expansion curve was nearer to the isothermal so that greater expansion work was obtained, resulting in greatly increased net power.

For a given increase in temperature in the heater more heat was required than was needed for air owing to the greater density and heat capacity of the Freon per cubic foot taken in at the compressor, but his object had been to obtain a greater net power from the engine in question and that was how it had been done.

One should thus be continually searching for other working media which, because of their physical properties, would give better results just as had been done in the case of refrigerating plant where air, working on an open cycle, had been completely superseded by ammonia, CO₂, Freon and other substances operating in a closed system.

Mr. E. C. Sterland, B.Sc., A.M.I.Mech.E. (Turbine Department, The English Electric Co., Ltd., Rugby). With regard to the question of variation of ambient temperature on the performance of a gas turbine plant, it seemed to him when considering this problem that an increase in ambient temperature would result in the following differences in performance of the compressor.

The mass flow would be reduced in direct proportion to the ratio of the absolute temperatures, but the volumetric flow would remain sensibly constant. Thus the gas speeds through the compressor blading would also remain constant, and since the speed of revolution of the compressor also remained constant, one would expect the temperature rise to be the same as under the former conditions. Since, however, the compressor was now being supplied with air at a higher ambient temperature, this temperature rise would correspond to a lower pressure ratio. In this case the actual work done in compressing the air would be less than under the original conditions. This did not, however, affect the main argument of the paper inasmuch as the output of the turbine was also reduced both on account of the lower mass flow and the lower pressure ratio.

When considering the question of partial load characteristics of gas turbines, he had evolved a graphical method for the solution of this problem (which he hoped to be in a position to publish shortly), and found that this method was also applicable to the consideration of variation of ambient temperature. The plant which he had considered was one in which the high pressure turbine drove the compressor and the low pressure turbine was the useful power turbine. There was no regeneration, reheat or intercooling. The plant was designed for an inlet temperature of 1,000° F. to the h.p. turbine, and for an ambient temperature of 70° F. It was decided to estimate the performance when the ambient temperature rose to 120° F. He found that the effect of this increase in the ambient temperature was to lower the speed of the compressor turbine shaft, thus reducing the mass flow and pressure ratio for the power turbine and so reducing its output. However, the power could be increased by increasing the inlet temperature to the high pressure turbine. As a matter of interest, the results obtained were as follows:—

If the inlet temperature to the h.p. turbine was kept constant at 1,000° F., then the compressor turbine shaft ran at 87½ per cent. speed and the power turbine developed 62 per cent. of the designed power. If the charging set were arranged to run at the same speed as formerly, then the inlet temperature must be increased to 1,100° F., the power developed then being 91 per cent. If it were desired to obtain the full 100 per cent. power from the power turbine, then the inlet temperature must be raised to 1,125° F., and in this case the compressor turbine shaft ran at 5 per cent. over the designed speed. It would be noticed that with 1,000° F. inlet, the speed of the compressor turbine shaft was lower than the designed speed, so it seemed reasonable to allow the inlet temperature to exceed slightly the value of 1,000° F. so that the compressor turbine shaft still ran at somewhat less than 100 per

cent. speed. Under these conditions it would be possible to obtain 75 per cent. power output without subjecting the material to any more onerous duty than that for which it was designed.

If, however, the condition of high ambient temperature could be considered as a temporary overload, then the full power could be obtained by allowing the inlet temperature to rise.

The diagram (Fig. 18) showed a typical example of the partial load chart for a simple gas turbine.

Mr. B. Wood (Messrs. Merz and McLellan) suggested that the authors had made the paper somewhat complicated, possibly by covering too wide a field. This might give the impression of offering a multitude of counsels. He thought it should be pointed out that certain of the cycles considered had serious practical difficulties, and, as had been mentioned, nothing was known as to the test results. He considered it was necessary to differentiate between cycles of that type and those which had attained to commercial status and which had achieved or improved upon their guarantees.

Ever since he could first remember, Mr. Holzwarth had been about to produce a more promising gas turbine than his last one but no outstanding success appeared to have been recorded. It might be that his more recent turbines had really achieved something which could not be done better with a steam turbine or a gas engine, but if so it was not clear why the results were not published.

Then there was the Escher-Wyss cycle which offered the great advantage, in theory, of being able to burn solid fuel. It required, however, an "air boiler" which was drawn very small in Fig. 13, but which would, in fact, be as large, if not larger than a corresponding steam boiler. Further, he thought that very serious slagging trouble might be liable to occur when firing coal in the absence of a water cooled combustion chamber. The small coil (5) in Fig. 13 represented the sink which was to dispose of all heat thrown away in the cycle. It served the purpose of the condenser in a steam cycle and would be at least as big as such a condenser. He had heard it suggested that if that cycle had been in use hitherto, some inventor would have come along with a patent claiming an improvement on it by the use of steam instead of air, so cutting out the compressor (4) and all the negative work it entailed, condensing the steam in the condenser (5) and just pumping the water back into the boiler again, showing considerable savings in capital cost, at least as good an efficiency and a nice practical job.

In his view, it was misleading in drawing the curves of Fig. 9 to assume a constant efficiency ratio in the compressor or in the turbine irrespective of compression ratio. As a gas was compressed it inevitably became hotter with the result that of two equally good quality compressors, that designed to work over a wider pressure ratio must show a reduced efficiency ratio even though the stage efficiency and the work per stage remained constant. In the turbine, where the reheat factor worked the favourable way round, the overall efficiency ratio of machines of equal quality increased with pressure ratio. Thus, the overall cycle result obtained by the assumption of constant efficiency ratio irrespective of pressure ratio might be right by accident if the misjudgment of the compression work compensated the under-estimation of the turbine work. He had shown elsewhere a method whereby it was possible to get a better result, i.e. one nearer the truth, and that was to assume a constant polytropic efficiency. That was the efficiency of an infinitely small stage which in practice was not very different from the actual stage efficiency. It was a more fundamental measure of turbine or compressor performance since it was virtually independent of pressure ratio, whereas efficiency ratio was a function of polytropic efficiency and pressure ratio.

BY CORRESPONDENCE.

Dr. W. Davies (The Royal School of Mines): Professor Davies and Dr. Fawzi are, in this paper and in their previous publications, making a notable contribution to the development of the combustion gas turbine by formulating the basic principles in terms that can be immediately applied in practice. Their analyses show clearly the unexpected influence of many factors that might have escaped the attention of the designer. Their work is timely and of the utmost value to the engineering profession in view of imminent developments in this direction.

The maximum temperatures reached at present in the constant pressure turbine are comparatively low and the authors are perfectly justified in ignoring the variation of the specific heats of the gases with changes in temperature in their analysis of this cycle, but in the constant volume type the maximum temperatures pass into the region where the specific heats can no longer be regarded as constant, and the comparison between these cycles on page 243, in which the constant volume cycle is shown to have an efficiency of 0.443 as against 0.329 for the constant pressure cycle, is unduly favourable in the constant volume type. The authors conclude that the relative

efficiency of this type will be lower than that of the constant pressure type for various reasons given in the paper. The increase in the specific heats of the gases at high temperatures might be added as another reason in support of their conclusion.

The closed system shown in Fig. 13 is of exceptional interest in that it permits of the use of any gas as the working fluid. Would the authors consider hydrogen to have any advantage as a working substance apart from the fact that it has a very high thermal conductivity as compared with the gases used in the normal types of internal combustion engines? It may be of interest in this connection to note that hydrogen has been used successfully as a cooling medium in the closed circuit ventilating systems of large alternators. This fact suggests that leakage might not be an insurmountable difficulty.

Finally, would the authors consider it feasible to employ the internal combustion principle in conjunction with the hydraulic jet as a method of propulsion for ships with an arrangement somewhat similar to that of the Humphrey pump?

Air Commodore F. Whittle, C.B.E., R.A.F., M.A., M.I.Mech.E., F.R.Ae.S.: From the point of view of engineers not connected with

aeronautics it must seem a strange thing that the combustion turbine should have reached its greatest degree of practical application so far in the aircraft field, especially as, compared with other applications, the aero engine designer is severely handicapped by the need to keep down weight and frontal area to a minimum. These limitations prevent the use of many devices for improving thermal efficiency which are available in other applications. On the other hand some of the factors operating in favour of the aircraft gas turbine are:

- (1) The required life is only of the order of, say, 300 hours for military purposes and 1,000 hours for civil purposes.
- (2) It has proved possible to get such low specific weight that for many purposes this more than offsets the present lower thermal efficiency as compared with the internal combustion engine.
- (3) The favourable effect of low intake temperatures which has been stressed by the authors is obtained more fully in the aircraft application than in any other, owing to the reduction of temperature with height.
- (4) A useful proportion of the compression process is in the form of ram compression at the intake, a process which can have an adiabatic efficiency of over 90 per cent. and which therefore raises the average efficiency of the whole compression process.
- (5) In the case of the jet propulsion application, only a sufficient proportion of the total expansion takes place in the turbine to drive the compressor, so that the total efficiency of the expansion process is higher than that of the turbine alone.

It should be remembered that the application of the aircraft combustion turbine to jet propulsion is only one of many possible applications and has come first because it is by far the simplest. It has almost certainly come to stay for certain types of aircraft.

The authors make a plea for the substitution of the term "combustion turbine" for "gas turbine". Personally, I am in favour of this. In turn I would ask the authors to abandon the use of °F. and B.Th.U's. in favour of °C. and C.H.U's. The latter have become established practice in the aircraft combustion turbine field in this country and I would very much like to see them made standard throughout the combustion turbine field.

Nobody could class me among the combustion turbine's enemies, yet I am of the opinion that the authors have been far too conservative in their assumptions as to the permissible value of the maximum temperature of the cycle. More than that I cannot say for security reasons, but the firms producing the special materials for blades, etc., have made excellent progress, and their efforts have contributed enormously to the success of aircraft turbines.

One of the main drawbacks of the combustion turbine in general is the serious fall off of efficiency at partial load. This is, of course, the more serious the greater the range of power required in any particular application. Possibly the aircraft case is the least favourable one from this point of view because in certain cases the ratio of maximum power to cruising power is of the order of 5:1. This should be a much less serious consideration in ships, which spend a very large proportion of their time at a steady cruising speed with a maximum power requirement of the order of only 30 per cent. above cruising power. Moreover, in the marine application the use of heat interchangers and other devices can do much to alleviate the position.

A point which cannot be too often stressed, in my opinion, is that in the piston engine the three main processes, namely compression, combustion and expansion, take place in one organ, whereas in the combustion turbine these processes take place in specialised organs,

Discussion.

and when one considers the possible combinations of axial flow, radial flow, and displacement compressors, etc., with axial flow and radial flow turbines, etc., and then to these adds the possible variations in arrangement with intercoolers, heat exchangers, interstage heating, etc., the permutations and combinations are absolutely staggering, and it requires a real effort of will on the part of the designer to make up his mind to go ahead with one particular arrangement and not keep chasing something slightly better just round the corner.

One consequence of this possibility of variation will be that this power plant will have more variants than any other, but each application will be more specialised to its purpose than is usual for other classes of power plant.

Mr. H. J. Wheadon (Member of Council) (Superintendent Engineer, The Royal Mail Lines, Ltd.). The comprehensive survey which Professor Davies and Dr. Fawzi have given us will, I have no doubt, be regarded in the future as heralding a new phase in the development of the combustion turbine—the transition from what the paper shows us to have been a very long period of experiment under great difficulties to the stage of practical commercial application.

The widely varying conditions under which marine machinery operates are such that applications of new engineering developments to ship propulsion invariably follow well in the rear of progress in more favourable fields of application. Although there is no reason to believe there will be an exception in the case of the gas turbine, marine engineers should, nevertheless, keep well informed as to its progress and growing possibilities, for developments may be very rapid.

When, for instance, Squadron Leader Whittle (as he was at the time) expressed the view, during the discussion of a paper on the gas turbine at The Institution of Mechanical Engineers in 1939, that aircraft powering was a most hopeful field for its application, who would have believed that in so short a space of time it would have become an accomplished fact? Even such an enthusiast as Dr. Adolf Meyer, the author of the paper, considered the suggestion rather optimistic.

The authors have pointed out how sensitive the performance of the combustion turbine cycle is to variations in the efficiencies of its component parts—turbines, compressors, regenerators, etc. In this connection the "work ratio", i.e. the ratio of useful work to the turbine output, is important, so much so that I am surprised to find no direct reference to it.

I feel that it would add to the value of the paper if the authors referred to this aspect of design. In this connection, designers of combustion turbines for ship propulsion should not overlook the fact that with the existing types of prime movers, whether they be steam reciprocating, steam turbine or Diesel, are rarely rendered incapable of producing useful power by the partial or complete failure of one or even more components, and owners, underwriters and classification societies will not look kindly upon new types of machinery unless they are equally flexible and reliable in that respect.

Referring to Fig. 14, page 247, where it is shown that the authors propose to direct-couple their astern turbine to the ahead turbine so that one of them must of necessity idle when the installation is operating, they propose to reduce the windage losses by evacuating the idling unit so that it is spinning under conditions of high vacuum, for which purpose a rotary vacuum pump is direct-coupled to the main shaft.

I doubt very much if in practice it would be found that under manoeuvring conditions, when several changes in direction of rotation are sometimes called for in the space of a minute, that the turbine cases would be evacuated rapidly enough with such an arrangement as the present overheating of the idling rotor. In any case, is it really necessary to follow steam practice in this respect? would it not be sufficient to open its inlet and discharge to atmosphere so as to provide a free flow of cold air through it? There would, of course, still be a greater windage loss than would obtain with the idling rotor operating in a vacuum, but I believe this would not be a serious objection, at any rate for astern running.

In view of the above similarity in construction of combustion turbines and axial flow compressors, it occurs to me it might be possible to so arrange them that they exchanged functions, depending upon the direction of rotation. I am aware of some of the objections to this proposal, but its adoption would lead to such a simplification of marine installations as well as the elimination of the windage problem under idling conditions that I venture to seek the authors' views as to its practicality.

The authors have indicated some of the particular difficulties which designers of marine installations will have to overcome, among which the adverse effect of tropical atmospheric temperatures which reach, say, 100° F. is a serious one. There are probably other

peculiarities of marine conditions and requirements upon which members of this Institute, from the experience they have gained in the operation of ships of all types in many waters, could assist in the development of this interesting machine.

I consider, therefore, the suggestion that The Institute of Marine Engineers should form a Marine Combustion Turbine Committee to stimulate cooperation between designers, metallurgists and users an excellent one, and I hope the Council will act upon it and appoint such a committee without delay from among members who are authorities on the subject.

Mr. A. M. Riddell (Member): Would it be possible for the authors to furnish consumption figures, weight, space occupied and, if possible at this early stage, cost based on the e.h.p. calculated as being required for the eight systems of propulsion dealt with in the recent symposium, "The Engining of Post-war Cargo Vessels of Low Power"? Such figures would form a valuable comparative guide.

Mr. J. A. Jaffrey, M.Sc. (Member): The control of temperature is far more difficult than control of pressure, and fundamentally it appears that the use of a combustion turbine on the constant pressure system would entail a considerable number of automatic devices of a rather complex nature.

In view of the salt laden atmosphere inevitable at sea, it would appear that the closed cycle system as shown on Fig. 13 would be preferable, otherwise the erosion which would occur in the compressor and turbine might be unduly high.

The authors have made various proposals regarding the installation of a combustion turbine on board, but might give some indication in regard to the following points:—

- (1) The efficiencies are stated on the designed output, but there is no indication of how the efficiency will change with load, and it might be of assistance if the change of efficiency were indicated over a range of, say, 75 per cent. full load up to full load.
- (2) There is no indication of the lowest speed at which the equipment would work satisfactorily continuously. It appears possible that the characteristics of the compressor might be the limiting feature.
- (3) In a seaway the load may vary from about half load to full load every 5 to 10 seconds and the possible behaviour of the installation is rather complex.
- (4) With a layout as shown in Fig. 14, the stored energy is very high and it might be stated how the installation would compare with the usual marine installations in regard to the time required for reversal.
- (5) As the authors have shown, the efficiency is very susceptible to the ambient air temperature, but their suggestion to employ the Diesel engine for the extra power required at higher temperatures might involve some difficulty. Would it not be preferable to control the temperature of the incoming air?
- (6) Can the authors give any indication of the effect on the behaviour of the plant, resulting from the varying rates of deterioration in the different components over years of service. For example, it is possible that the turbine blading may deteriorate at a more rapid rate than the compressor. It would appear that more frequent inspection of the turbine and compressor than is normal would be required, and it is difficult to see how the necessary adjustments can be made as the result of this inspection, in order to keep the temperatures and efficiencies within satisfactory limits.

From the reliability point of view, the combustion turbine seems to be dependent to a large degree on the metallurgist. The necessity for the "warming up" of a steam turbine is well known, but it does not seem to be generally realised that from a somewhat different point of view a "warming up" period will be necessary for a combustion turbine. In fact, on steam turbines utilising heat resisting steel blades, where the steam is at a high temperature, it might be found more satisfactory if the "warming up" were carried to higher limits of temperature. Generally speaking, steels capable of resisting high temperatures are relatively brittle at low or room temperatures, and as, for example, the exhaust valves of a Diesel engine, or the blades of a supercharger impeller are started up cold in many instances, it is very probable that cracks might be initiated which eventually lead to failure, and which might erroneously be attributed to the high temperature.

Heat resisting steels are very sensitive to surface finish in their resistance to failure at low temperatures. A $\frac{3}{4}$ -inch bar of such a steel was "fine turned", only a small amount of stock being carefully removed. The bar was then divided up into several sections.

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Attempts were then made to bind the bar round a one-inch radius, the load being applied gradually, and it was found that the bar snapped after only a few degrees of bending. Another section was then ground to a "good finish", and a bend of about 20 degrees was obtained before the bar snapped off. Another section was ground and then buffed until no grinding marks were visible, and a bend of 30 degrees was obtained. A Swiss file was pulled once across the surface of a ground bar so that the mark was just visible, and instead of a bend of 20° the bar now broke at 10°. A Swiss file similarly applied to a buffed bar reduced the bend from 30° to 20°. All the experiments were carried out at room temperature and the load applied gradually in all cases. In all cases, also, the fracture was at right angles to the axis of the bar, and there was no gradual indication that the bar was failing.

It is possible to suppose that if a heat resistant steel could be produced, which would retain its initial surface finish at subsequent high temperatures, part of the difficulties associated with "creep" might be overcome.

The author's proposal to provide an astern turbine under vacuum might result in considerable blade failures, and it is possible that it might be necessary to run the turbine with a limited quantity of air present so that the temperature of the blades would be fairly high.

In order to achieve reliability, it would seem necessary to sacrifice efficiency, as the blades of all the turbines might require a greater radius on the edges than is desired from an efficiency point of view.

Mr. B. C. Curling (Member): Reversing is inevitably one of the main problems of the marine combustion turbine designer. In this connection, referring to the authors' proposed arrangement as shown in Fig. 14, a possible simplification would be to drive the vacuum pump, V.P., from the compressor shaft instead of from the power shaft; as the compressor shaft would run unidirectionally, this would eliminate the need for the gear box when reversing the power shaft. This rearrangement would add slightly to the length of the compressor shaft, but it would remove that marine engineers' bugbear, the gear box.

The speed of the compressor shaft is not stated, but as the authors point out that speed changes of the propeller shaft, which affect the power shaft with methods (b) and (c), do not directly affect the compressor shaft, this would seem to be another reason for transferring the vacuum pump to the compressor shaft. This rearrangement would, moreover, avoid the interval in the functioning of the vacuum pump while the power shaft was being stopped and reversed.

Mr. John S. Ashmore, M.A., B.A.I., G.I.Mech.E.: I am certain that the paper will stimulate many who hitherto have shrunk from more advanced and detailed treatment of the subject to renew their efforts and master the problems involved. I therefore would not wish to detract in any way from the general usefulness of the paper by

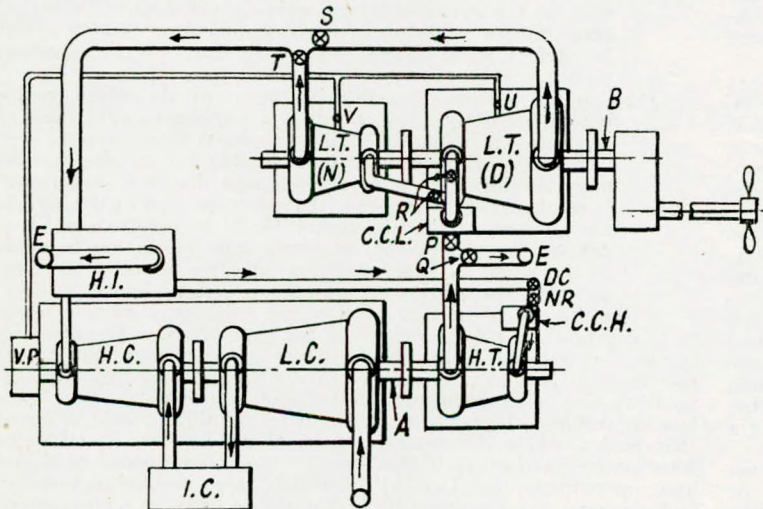


FIG. 19.

The Authors' Reply to the Discussion.

The authors express their gratification that so many engineers have contributed to the discussion.

The President. In reply to the question raised by the President concerning the vacuum necessary to permit these machines to be

suggesting that an overburden of detail should be introduced. I do, however, feel that some information on the starting of the whole system would be beneficial and might most easily be introduced with the details of the installation diagrammatically illustrated in Fig. 14.

In conditions II, III and IV as set out in Table II, the power of the supplementary oil engine would be available to rotate the compressor shaft, thus supplying air to support combustion; resort to the use of glow plugs or ignition elements would have to be made to ignite the fuel.

Case I is an entirely different proposition, there being no supplementary power available. This case therefore deserves special consideration with regard to starting. Dr. Meyer, in his paper on "The First Gas Turbine Locomotive", to which reference is also made in the text, described in detail the starting of a turbo generator unit, the power required for starting being supplied by an auxiliary Diesel engine driven generator which operated the main generator as a motor for four minutes prior to ignition of the fuel.

C. C. Pounder in the Fifteenth Thomas Lowe Gray Lecture referred to the potential challenge of the combustion turbine to normal marine prime movers, and mentioned that starting was effected by an electric motor which enabled the compressor to deliver sufficient air to the combustion chamber when lighting up.

In each of the foregoing cases, starting is effected by first rotating the compressor, which supplies sufficient air to support combustion. I therefore wonder whether, in cases where no auxiliary power is available to rotate the compressor shaft, the starting process could not be somewhat varied in order to make use of compressed air to support combustion. If compressed air were admitted to the combustion chamber and the fuel ignited by electrical means, the products of combustion would pass through the turbine, thus starting the system.

Incidentally, it was assumed that a non-return valve would in any case be fitted between the compressor and the combustion chamber to prevent the danger of blow-back. A decompression valve could be fitted on the compressor side of this valve and arranged to operate in conjunction with the admission of compressed air; hence, while compressed air was being admitted, the compressor would deliver against only atmospheric pressure. This arrangement would permit the use of compressed air at any desirable pressure.

A marine installation with auxiliary Diesel generators would already have a supply of compressed air for starting purposes at 350lb./sq. in., and this, delivered through a suitable reducing valve, would be available for starting the turbine also. The practicability of the suggestion as a whole depends upon the quantity of compressed air required to initiate the cycle.

While it is realised that Fig. 14 is purely diagrammatic, nevertheless the path of the gases from H.T. to C.C.L. appears to be unnecessarily tortuous compared with the path H.T. to L.T.(N). Considering arrangement No. 10, Case (C), involving transmission by reduction gears with a fixed blade propeller, I would suggest the following alterations to Fig. 14:—

- (1) Place L.T.(D) in the position occupied by L.T.(N);
- (2) reduce L.T.(N) in size so that with a combustion chamber fitted it develops only 60 per cent. of the torque of the ahead turbine;
- (3) turn this reduced L.T.(N) so that its inlet end is next to, instead of remote from, the inlet end of L.T.(D);
- (4) place valve P in the position occupied by valve R;
- (5) between P and L.T.(D) fit C.C.L.;
- (6) between C.C.L. and L.T.(D) fit an additional valve P₁;
- (7) between C.C.L. and P₁ fit R.

By these alterations the inertia of the astern turbine would be reduced, the one combustion chamber would operate for both ahead and astern running, the risk of oil accumulating in an unused combustion chamber during manœuvring would be eliminated, and the system would give more positive manœuvrability. Disadvantages would be that the valves P₁ and R and the astern turbine would have to withstand higher temperatures.

Since the rotation of shaft A was unidirectional, whilst the rotation of shaft B was not, it would appear more feasible to fit V.P. to shaft A than to shaft B, and thus eliminate G.B. This would necessitate a slight increase in the output required from H.T.

driven reversed, the authors stated that the suggestion to use a vacuum pump to exhaust the casing of the idling turbine was inspired by a test witnessed by one of them some years ago. The pump used was a Northey rotary machine, which works on the displacement principle; when used as an air compressor it has a nominal

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displacement of 40 cu. ft. of free air per minute at 1,500 r.p.m. In the test it was used to evacuate a closed vessel of 4.5 cu. ft. capacity from an atmospheric pressure of 29.12 in. of mercury. During the test the speed of the pump was 1,590 r.p.m. The following observations were made:—

Vacuum, in. of mercury	0	10	15	20	22	24	26	27	28
Time, seconds ...	0	2.2	4.2	7.1	8.3	10.5	15.3	19.1	30.6

The final absolute pressure was under 0.5 in. of mercury. In view of these results, there would appear to be no difficulty in reducing the resistances in combustion turbines under idling conditions below those of comparable steam turbines.

Dr. Livingston Smith. The authors are glad to have the support of Dr. Smith in their view of the importance of the present subject to the marine engineering branch. Dr. Smith is under a misapprehension concerning the change of specific heat of the gases; while the simplified comparison of Figs. 7 and 8 is based on assumed constant values, in all other cases, including those of Tables I and II, the actual properties of the gases are used in the calculations. Dr. Smith regards the value 1,500° F. abs. for T_B as too conservative and requests the figures for $T_B=1,600$ and $1,700^\circ$ F. abs. The authors have calculated these for installations Nos. 1, 2 and 10 and now give, in Table III, the results for all three values of T_B , tabulated for comparison. These results show, as has been pointed out, that raising the practicable value of T_B improves the picture considerably.

as the installations become more complex, it must be remembered that the pressure ratios also increase, so that the relative effect will not vary greatly. The values of pressure ratio taken in the various cases are those giving a compromise between maximum efficiency and maximum specific output. Thus, although the values of efficiency and specific output in Table I will be reduced by hydraulic losses, the relative picture given remains a true one.

In the final version of their paper the words "of compression" have been added on page 247 in connection with the mean intercooler pressure. It corresponds, of course, to minimum work of compression.

The authors recognise that the installation of Fig. 14 would work if output were sacrificed and speed reduced, but do not regard this as a desirable condition. Mr. Robinson's suggestion also involves a reduction of efficiency, as is indicated by case I in Table II.

Mr. Robinson asks concerning the operating advantages of combining a variable pitch reversible propeller with a compound turbine. These are: (1) lower inertia of the propeller shaft, improving flexibility; (2) less variation of speed of the compressor and the high pressure turbine, giving higher efficiencies at partial loads.

Dr. J. G. King raised some interesting matters affecting the gas industry. He asked whether, with a gaseous fuel, the constant volume cycle did not offer advantages over the constant pressure cycle, from the greater scope for reduction of the work of compression of both gas and air. The test data available from the constant volume type

TABLE III.

Type of Installation.	Simple.			Simple with heat interchanger.			Combined compounding with heat interchanger.					
	1,500	1,600	1,700	1,500	1,600	1,700	50% intercooling.			100% intercooling.		
Max. temperature of the cycle °F. abs.	1,500	1,600	1,700	1,500	1,600	1,700	1,500	1,600	1,700	1,500	1,600	1,700
Efficiency ...	0.177	0.201	0.222	0.248	0.268	0.303	0.289	0.288	0.319	0.313	0.346	0.371
Sp. output B.Th.U./lb. of air	29.6	36.3	42.8	29.8	36.3	44.1	42.3	56.01	66.61	45.9	67.4	77.2
Fuel consumption lb./h.p./hr. ...	0.787	0.693	0.627	0.562	0.520	0.460	0.482	0.484	0.436	0.440	0.402	0.375
Air consumption lb./h.p./hr. ...	86.00	70.1	59.4	85.4	70.1	57.7	60.1	45.3	38.2	55.5	37.7	32.9
Air consumption cu. ft./h.p./hr. (at 60° F. and 14.7 lb./sq. in. abs.) ...	1,103.8	899.4	762.0	1,096.1	899.4	740.2	771.9	581.2	490.1	712.2	483.7	422.1
Relative efficiency (simple case=100)	100	100	100	140	133	136	163	143	143.6	177	172	167.1
Relative output (simple case=100)	100	100	100	100.6	100	103	143	154	155.6	155	185	180.4

It is regretted that it has been impossible to carry out the comprehensive calculations necessary to estimate the power/weight ratio for the installation of Fig. 14, asked for by Dr. Smith.

Dr. Roxbee Cox. The valuable and interesting contribution of Dr. Roxbee Cox does not require reply, beyond an expression of their appreciation, by the authors.

Sir Roy Fedden. The authors are glad to have the support of Sir Roy Fedden, whose views were put forward by Mr. P. G. Ware, on the importance of the present subject. As regards the closed system adopted by Escher Wyss, the basic pressure appears to be 9 atm., which, with a pressure ratio of about 5, gives a maximum pressure of 50 atm., not an excessive value. High rates of heat interchange are claimed. The authors, however, await further test results, and regret that they are not able to give fuller information on this interesting development.

Mr. W. J. Robinson gave some good reasons for adhering to the term "gas" turbine and pointed out the advantage of the Escher Wyss installation, which he called the "external combustion" type. Mr. Robinson regarded 140° F. as excessive for the air temperatures, but this value was only employed in the general case of Fig. 11; for the marine installations of Table II, the maximum value taken was 110° F., probably the highest possible value of ambient air temperature to be expected on a ship. Mr. Robinson made a good point when he affirmed that, with increase of T_B , the effect of air temperature would be relatively less; the authors have confirmed this by calculations. Concerning the effect of pressure drops in the cases of Table I, while it is true that pressure drops will be greater

are very limited and the installation is clearly more complicated. With the information at present available, the authors would prefer the constant pressure type, even for gases of low calorific value. The volume ratio of gas to air and the change of volume on combustion are two matters which must be considered in the calculations.

As regards Dr. King's questions concerning the applications on page 246, the authors see no special difficulties in connection with class (a); in class (b), further data is awaited concerning Kapitza's installation, especially concerning the radial inward-flow turbine employed; in class (d), assuming the gas to undergo a fall of pressure from 50 to 10 atm., power generation by turbines would easily be possible within reasonable ranges of temperature.

Mr. F. J. Cowlin, in his very interesting contribution, added important considerations to those dealt with in the paper. He finally commented on the installation in Fig. 14, selected by the authors as an example for discussion, and expressed the view that the multiplicity of components in this case would lead to difficulties. He opposed, in particular, intercooling the compression. This is again a matter for compromise; either greater simplicity or a higher efficiency. But the authors do not agree that an intercooler on a compressor is so liable to breakdown as a steam condenser; neither is the necessary water quantity so large in the former case. These questions are, however, very largely speculative and the authors associate themselves wholeheartedly with Mr. Cowlin's view that the crying need is for plants to be built and tested in order to check the validity of various assumptions.

Eng'r Rear-Admiral D. J. Hoare raised a number of important practical considerations. The authors agree that the heat inter-

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changers and the various ducts demand special attention in design. The answers to these and the other questions posed by Admiral Hoare can, however, only be given on the basis of actual test results. On page 246 the authors pointed out that marine combustion turbine installations "are in the stage of study and preliminary experiment". The results from one well designed experimental installation, equipped with the means of alternative combinations, and put into service in a ship, would be of enormous value to marine engine designers in this country. The authors would also welcome more test data from the combined compression-ignition and turbine installations referred to finally by Admiral Hoare. This type and those working respectively on the open and closed cycles, constitute three competitive combustion turbines worthy of examination for marine purposes.

Mr. B. Pochobradsky, in his contribution, added considerably to the information the authors had given, and gave a number of interesting combinations for the main and auxiliary shafts. The authors took the view that, given a certain practicable value of the maximum temperature, T_B , this should, in order to obtain the best performance, always be reached, and that the propeller speed should be constant. Mr. Pochobradsky departed from both these desiderata, and put up a good case for doing so. It is, of course, for the designer to decide which combination best meets the desired conditions of running. Mr. Pochobradsky's appeal for a successful first combustion turbine plant accords with the view expressed above by the authors.

Mr. E. Markham made an important suggestion that a working substance with a low value of the ratio of the specific heats should be sought for, and referred to his tests with Freon. This could, of course, only be used in a closed cycle turbine, but the authors agree with Mr. Markham that, in this type of installation, all the relevant properties of possible working substances should be examined.

Mr. E. G. Sterland's contribution provided a further combination for the main and auxiliary shafts, but one which allowed a reduction of full load output and increased maximum temperature. The authors welcomed these further suggested arrangements, since it is evident that the range of possibility is wide. Mr. Sterland's chart for the simple case is interesting, but the authors would emphasise that it is dangerous to apply the conclusions from one type of installation in Table I to another, since each type has its own narrow characteristics.

Mr. B. Wood suggested that too wide a field had been covered in the paper. In the present state of the science, the authors felt that a fairly comprehensive treatment had greater value than one more limited, bearing in mind the relatively small body of engineers who have studied the subject; especially was it desirable to plan the development in proper perspective, both generally and nationally. Having considered the course of development of other prime movers, the authors saw the position of combustion turbines as follows: side by side with practical achievements such as the Holzwarth, Escher Wyss and Brown Boveri, on which the available test data of the last named is greatest, there are, on the one hand, combinations of reciprocating engines with combustion turbines and, on the other, paper proposals of various kinds; all of these matters have arisen in a period of rapid technical advance especially in such matters as metallurgy and aerodynamics. The authors would hesitate to prophesy the position as it would be even five years hence and they consider it essential to take a broad view of a rapidly developing subject. Reliable test data provide the criterion for the engineer, but the absence of this in relation, for example, to the Escher Wyss installation, does not justify the authors in failing to bring this type to the notice of the members of the Institute.

Mr. Wood wished the authors, in expressing compressor and turbine efficiencies, to vary these with pressure ratio. In their view, Table I is already sufficiently complex and it is only by making simplifying assumptions that such comparisons of types can readily be made. Provided the reader is not misled by the assumption, the gain in clarity is well justified.

Air Commodore F. Whittle made some valuable comments on the application of the combustion turbine to the propulsion of aircraft, and pointed out certain favourable conditions which do not apply to marine practice. The authors would like to use °C., but, in view of the established practice in marine and general engineering circles, both here and in the U.S.A., °F. had been used. Air Commodore Whittle's views on maximum temperatures reinforce the hopes of the authors for an increase beyond 1,500° F. abs. in the near future. Finally, he emphasises that the "permutations and

combinations" of possible arrangements are very numerous, as the present discussion has shown.

Dr. W. Davies pointed out the importance of taking variable specific heat into account in the comparison of Figs. 7 and 8, and justifiably pointed out that neglect of the changes favoured the constant volume type. The main point of this comparison, however, was to emphasise the need for the enormous cooling of the gases in the constant volume case, and the heat to be removed, given the same maximum temperature, is obviously still greater when variable specific heats are taken. He questions the suitability of hydrogen, from its high thermal conductivity, as the working substance for the closed cycle engine. The authors feel, however, that other properties, in addition to that of thermal conductivity, should be taken into account, and especially the value of the adiabatic index. Finally, on the evidence before them, the authors would hesitate to express a view on the use of the hydraulic jet for ship propulsion.

Mr. H. J. Wheadon, in the course of his interesting contribution, referred to work ratio. The authors saw no special reason to emphasise this matter; it is, for example, never used in connection with reciprocating internal combustion engines although compression in these engines similarly involves negative work; more important, in their view, is the fact that, in combustion turbines, all parts, at constant output, are subjected to steady pressures and temperatures. Mr. Wheadon rightly stressed the importance of flexibility and reliability in new types of machinery. He also referred to the proposals to evacuate the casing of the idle turbine and doubted the possibility of this under manoeuvring conditions. In their reply to the President the authors have given further particulars of this which they hope will convince Mr. Wheadon of the rapidity with which pressure may be reduced. Incidentally, the windage losses vary as the square of the speed of revolution, so that at low speeds they may be negligible, even with casing pressures at or near atmospheric pressure. Mr. Wheadon's interesting proposal to interchange the functions of compressors and turbines could only be carried out at the expense of efficiency. Finally, the authors are glad to have Mr. Wheadon's support for their suggested Marine Combustion Turbine Committee.

Mr. A. M. Riddell asked for comparative data to enable the combustion turbine to be compared with the other systems of propulsion given in the recent symposium. The authors regret that the provision of this information is beyond present possibilities; the work involved would be at least equal to that of preparing the present paper.

Mr. J. A. Jaffrey, in his valuable contribution, raised a number of important matters, most of which, however, go beyond the point to which the subject has been taken at present. The authors, however, had dealt with the points he raised as well as they could; they agree with his warning against erosion but think that corrosion might require to be even more carefully guarded against.

The efficiency at reduced loads depends on the method of fuel and air control. So long as T_B is kept at 1,500° F. abs., by reducing the fuel and air proportionately, the efficiency is constant for the range he mentions; if T_B falls, then efficiency falls off rapidly. Mr. Jaffrey is correct in stating that slow running depends on the compressor characteristics, but the authors have not specified a particular type. With the arrangement in Fig. 14, the low pressure turbine will always have a supply of pressure gases, and the installation will not differ from a steam turbine one. To answer this point definitely, the authors would, of course, have to make a complete design. It is not easy efficiently to control the temperature of the ambient air and the arrangement proposed has other compensating advantages. Since no running experience is yet available, the authors cannot answer his sixth point. This matter and all the interesting ones in the remainder of his contribution can only be settled on the basis of practical tests under service conditions.

Mr. B. C. Curling makes the valuable suggestion that the vacuum pump should be driven from the compressor shaft; it is agreed that this would be an improvement. The change would have all the advantages Mr. Curling claims for it.

Mr. J. S. Ashmore, among other useful suggestions, also proposes moving the vacuum pump to the compressor shaft. The authors did not think it worth while to go into the starting question since this would be bound up with the whole question of the auxiliaries necessary for other purposes on board. Mr. Ashmore's proposed arrangements for starting strike the authors as very good, while his layout is, in several respects, an improvement on that in Fig. 14.

The discussion has emphasised the wide range of problems to be solved in adapting the combustion turbine to marine purposes. Their

solution will provide the bases for many future papers and discussions. The contributions to the discussion have given the authors several new lines of study and have more than justified the labour

of preparing the paper. Finally, the authors express their gratitude to all those who contributed to the discussion; they appreciate very much the kindly compliments paid them.

CORRESPONDENCE—*The Engining of Post-war Cargo Vessels of Low Power.

The following communication, received from Götaverken A/B, Gothenburg, has been referred to Mr. P. L. Jones, Dr. J.-B. O. Sneed and Mr. J. Calderwood for their comments, which are also published below:—

In his summary of the different proposals for post-war cargo ship machinery, given at the June meeting of The Institute of Marine Engineers, Mr. James Calderwood states, with regard to the Götaverken system: "It would appear that for given cylinder sizes the indicated horsepower is 30 per cent. lower than with the Bauer-Wach proposal". This statement is not correct.

According to the paper read by Mr. P. L. Jones at the above mentioned meeting, a triple expansion engine, with the Bauer-Wach exhaust steam turbine cut out, develops 80 per cent. of the power obtained when the exhaust steam turbine is at work. The increase in power with the Bauer-Wach turbine is thus 25 per cent. In the paper concerning the Götaverken system, read at the same meeting by Dr. J.-B. O. Sneed, the indicator cards reproduced from s.s. "Harpasa" prove that the power of a triple expansion engine increases from 1,533 to 1,904 i.h.p., i.e., 24 per cent., when the turbo-compressor is engaged. From what is said above it is obvious that there is no essential difference in the rise in power if a triple expansion engine is combined with a Bauer-Wach exhaust steam turbine or if it is combined with a Götaverken turbo compressor. For equal horse-power and r.p.m. the size of the engines thus becomes the same for both systems.

With the steam engine referred to by Mr. Jones, with cylinder dimensions 24in., 40in., and 66in. x 45in. stroke, the power developed is stated to be 3,600 i.h.p. at 90 r.p.m. and a boiler pressure of 225lb./sq. in. For this engine the indicated mean pressure, referred to the h.p. cylinder, becomes 389lb./sq. in. or 1.73 times the boiler pressure. This "referred" mean pressure is very high and a comparison with existing installations proves that many installations with triple expansion engines and Bauer-Wach exhaust steam turbines normally at sea develop a mean indicated pressure, referred to the h.p. cylinder, of 1.35 times the boiler pressure. It therefore seems that the horse-power given by Mr. Jones is the full power and not the "seagoing horse-power". Mr. Calderwood compares this steam engine, combined with a Bauer-Wach exhaust steam turbine and designed for a full power of 3,600 i.h.p. with a steam engine, combined with the Götaverken turbo compressor, intended to develop a normal power at sea of 3,600 i.h.p., and this is the reason why the comparison between the two engines is not correct.

In the summary, referred to above, it is also stated, with reference to the Götaverken system: "The size and weight are apparently greater than with the Bauer-Wach system". With regard to the size of the reciprocating engine we have already shown that there is no essential difference between Götaverken's system and that of Bauer-Wach, and with regard to the space needed for both the engines the following can be said:—

In the Götaverken system the turbo compressor is mounted on top of the condenser and needs, in most cases, no extra space. In the Bauer-Wach system, coupling, gears and in many cases also the turbine are fitted aft of the reciprocating engine. The Bauer-Wach system therefore needs a larger engine room than is necessary with the Götaverken system.

In regard to the weight of the two systems the following comparison can be made:—

Mr. Jones states in his paper, "As a matter of interest it is found that in nearly all cases the total weight of a Bauer-Wach exhaust turbine installation is almost exactly the same as that of normal triple expansion machinery of the same power and r.p.m.". The weight of the turbo-compressor of the Götaverken system is much less than the weight of the turbine, couplings and gears of the Bauer-Wach system. The total weight of the complete machinery with the Götaverken turbo compressor is only 90 per cent. of the weight for a normal triple expansion engine plant with the same output and r.p.m. Consequently, machinery with the Götaverken turbo compressor is not heavier than the Bauer-Wach system, as was stated in Mr. Calderwood's summary. On the contrary, it is 10 per cent. lighter than a corresponding engine plant with Bauer-Wach exhaust steam turbine.

Another advantage of the Götaverken system is that the engine

can be reversed without disconnecting the turbo compressor. The manœuvring therefore becomes as quick as with an engine without exhaust steam turbine. Full power is also obtained when running in reverse, which is useful when navigating in ice.

Mr. Calderwood, in his summary, asks for data relating to the Götaverken system more directly comparable to the proposal under consideration. In addition to what has been said above, the following information about the fuel economy can be given.

Several new Swedish ships of 3,000 tons d.w. have 1,500 i.h.p. triple expansion engines with Götaverken turbo compressor. The steam pressure is 220lb./sq. in., the steam temperature about 575° F., the feed-water temperature about 285° F. and the temperature of the combustion air about 290° F. The fuel consumption in these ships is 0.95lb./i.h.p./hr., when burning coal with an effective calorific value of 13,500 B.Th.U's./lb.

Some ships under construction will be provided with steam driven generators and electrically driven auxiliaries, steering engine, windlass and winches. For these ships the fuel consumption is expected to be somewhat less at sea and much less in harbour than for the ships mentioned above.

Mr. P. L. Jones, M.C., B.Sc., Wh.Ex., M.I.Mar.E.:

It is true that I stated that, if the turbine is cut out of operation, the reciprocating engine in the Bauer-Wach system can then develop 80 per cent. of the full power for which the installation is designed. This, of course, presupposes that the boilers are supplying the same amount of steam as when the turbine is in operation. The mere act of cutting the turbine out of operation would reduce the revolutions by more than 10 per cent. and the power by about 30 per cent. The power of the engine working alone would be brought up to the 80 per cent. mentioned by increasing the engine cut-off. In other words, the engine would be adjusted to develop the maximum power for which the boilers can supply the steam.

This is not the same thing as saying that the power of the triple-expansion engine is increased by 25 per cent. when the Bauer-Wach turbine is engaged. This statement should be qualified by adding that in each case the steam consumption is the same. Now, in the case of the "Harpasa", it is true that the trials show that the power was increased from 1,533 i.h.p. with the turbo compressor cut out to 1,904 i.h.p. with the compressor engaged. This is certainly an increase of 24 per cent., but, as the h.p. cut-off was not altered, it is accompanied by an increase in the steam consumption pro rata to the increase in revolutions, i.e., from 65 to 70.5 r.p.m. or 8.5 per cent., so that the actual increase in power for the same steam consumption is 17.5 per cent. only; this compares with 25 per cent. obtained with Bauer-Wach.

It is noted that the mean indicated pressure with the Bauer-Wach machinery described in my paper works out to 389lb./sq. in. referred to the high pressure cylinder, or 1.73 times the boiler pressure. These figures are correct, but I do not agree that they are abnormally high. Messrs. Götaverken say that a comparison with existing Bauer-Wach installations shows that many of them develop a mean indicated pressure referred to the high pressure cylinder of only 1.35 times the boiler pressure. This is a novel way of comparing steam engines and I do not consider it a satisfactory criterion, but, since the matter has been put in this form, I have looked into the question and elucidated the following facts:—

On the last fifty Bauer-Wach installations for which my firm have been responsible, the m.i.p. referred to the high pressure cylinder varies from 307 to 423, the true average being 363, and the ratio of m.i.p. to boiler pressure varies from 1.52 to 1.89, with a true average of 1.63. These figures all refer to the i.h.p. developed under *continuous service operation*, and it was made quite clear in my paper that the installation described therein was capable of developing 3,600 i.h.p. at about 90 r.p.m. *continuously in service*. A large number of similar installations have been developing full power continuously at sea for many years with highly satisfactory results.

As regards size and weight, I gave complete figures in my paper for a modern Bauer-Wach installation, and it would be interesting if Götaverken would supply exactly similar information for machinery designed for the same power, revolutions and steam conditions.

Dr. J.-B. O. Sneed, B.Sc., Ph.D., F.R.S.E., M.I.Mech.E.:

In the third paragraph of their comments Messrs. Götaverken have offered an additional explanation for the difference in cylinder

*Trans.I.Mar.E., Vol. LVI, Nos. 6 and 7, July and August, 1944.

Obituary.

he brought a wealth of knowledge and experience unrivalled by any of his predecessors. His service included about 10 years at sea, another 10 years teaching at Greenwich and Keyham Colleges, and about 15 years at the Admiralty. His ships included the "Cumberland", the "Britannia", the "Delhi", the "Emperor of India", the "Rodney", and the "Lion", and he was "senior" engineer of the last when as flagship of Admiral Beatty she was nearly lost through gunfire at the Dogger Bank in January, 1915. Preece's recollection of those days was that from Sunday to Wednesday, January 24-27, it was just one continuous struggle to utilize or improvise pumping facilities to rid the ship of thousands of tons of water. By the time the "Lion" reached Queensferry she was nearly upright and Preece was rather indignant that the Academy picture of her "Home Coming" gave her a considerable list.

Preece's years at the Admiralty, first as assistant, then as deputy, and finally as engineer-in-chief was a very strenuous period. The Engineering Branch of the Navy, both ashore and afloat, does much of its work under "forced draught" in more ways than one, but its doings are seldom in the news. Speaking on the Naval Estimates in February, 1940, Mr. Churchill referred to the wonderful steaming

done by his Majesty's ships and asked the House of Commons to join him in a tribute "so that these many thousands of faithful, untiring engineers may learn, as they will learn, that we here in London understand what they have done and are doing and that we admire their work, and thank them for it". The machinery of those hundreds of ships and the thousands of officers Mr. Churchill referred to were under the supervision of Admiral Preece and his staff.

When Preece retired, to facilitate the promotion of his juniors, it was but to take up other work, for he was almost immediately chosen to succeed Lord Mottistone as President of the Institute of Marine Engineers. To the Institute during the last two years he has given unstintingly of his time and energy. Not only did he preside over its gatherings with dignity and humour, but he contributed to its proceedings by observations on his own experiences, and was especially interested in the education and status of the sea-going engineers of the Merchant Navy. It is thus that his death at the comparatively early age of 60 will be regretted, not only by those who had the privilege of enjoying his friendship, but by a large number of officers in both navies.

Mr. JAMES McKIE DEWAR

The Institute has suffered another great loss by the death, which occurred suddenly at Liverpool on December 29th, 1944, of Mr. James McKie Dewar, a Past Chairman of Council and Vice-President for the London area.

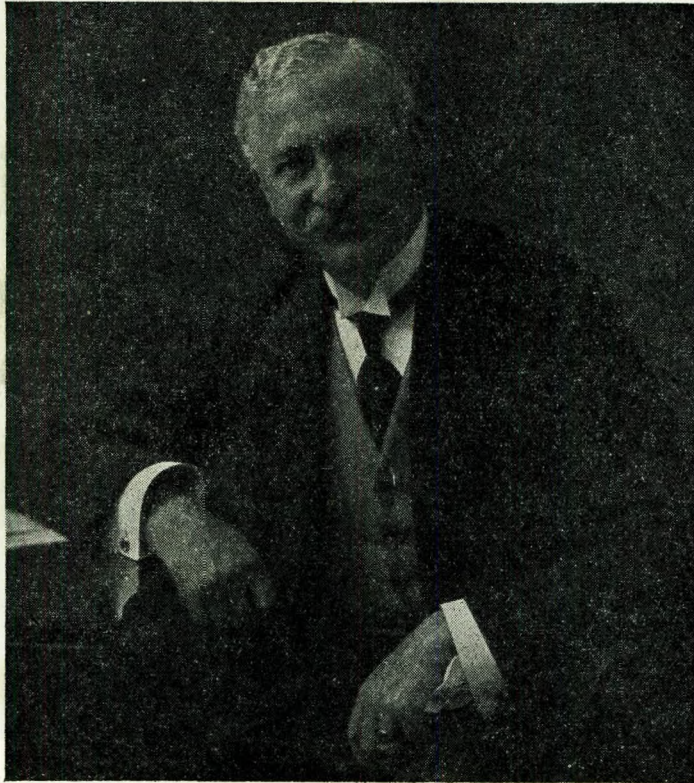
Mr. Dewar, who was 78 years of age, was born at Ayr, a son of the late Robert Dewar, J.P., and was educated at Ayr Academy, and later at the Royal Technical College, Glasgow. He was apprenticed first to the firm of J. and A. Taylor, crane builders and general engineers, of Ayr, and subsequently to Messrs. Kincaid and Company, Greenock, marine engine builders. After a period as a draughtsman with the latter firm he went to sea in the service of the British India Steam Navigation Company, obtaining a first class Board of Trade certificate. After a short time in the employ of the Glasgow Patents Company, he came to London in 1892 as a departmental manager in the works of Messrs. Maudslay, Sons and Field, Limited. In this capacity he was closely associated with the introduction of the Belleville water-tube boiler into the Royal Navy. He remained with Messrs. Maudslay until the liquidation of the firm in 1900. He then became general manager with Messrs. Bow, McLachlan and Company, Paisley, relinquishing the appointment in 1903 to return to London with the intention of setting up in private practice. Instead, he joined the London staff of Messrs. Cammell, Laird and Company, eventually becoming a director and remaining with them for 20 years, during which period he gained a high reputation at home and abroad as a leading marine engineer. On the outbreak of war in 1914 he was called upon by the Director of Dockyards to advise on and organise the repair work in the Royal Dockyards at Chatham, Portsmouth and Devonport. He was also associated with the maintenance of vessels, under Government control, engaged in the inland water transport system of this country. For some years he was a director of Messrs. Gwynnes, Limited, and

was concerned with the firm's wartime construction of Clerget and Bentley rotary aero-engines.

At the end of 1923 Mr. Dewar retired from the position of director of Cammell, Laird and Company, Limited, to start the consulting practice in London in which he was mainly occupied throughout the remainder of his life, in association with his son, Mr. Norman M. Dewar, M.I.N.A., M.I.Mar.E. Among the vessels with which Mr. Dewar was associated in the design and construction were the Italian Cosulich Line motor-ships "Saturnia" and "Vulcania". He was elected a Member of the Institute in 1913; later he held office as a Member of Council, Chairman of Council for 1920/21, and a Vice-President from 1921 until his death.

Mr. Dewar was a member of the Institution of Naval Architects, the Institution of Mechanical Engineers, and of the Institute of Metals, a Fellow of the Royal Society of Arts, a liveryman of the Worshipful Company of Shipwrights, and a member of the committee of the Maudslay Society. For some years he had been a director of Messrs. Chadburn's (Ship) Telegraph Company, Limited, and chairman since the death of his predecessor, Mr. Chadburn Bamford, three years ago. At the company's annual general meeting on December 29th, at which he presided, Mr. Dewar agreed to withdraw his resignation, which he had tendered on account of ill-health. A few hours later he collapsed and died. He was held in affectionate regard by his countless friends in the marine engineering and shipbuilding world, and he will be sadly missed by his colleagues on the Council and by his fellow members of the Institute. The cremation took place at Landican Crematorium on January 3rd, at which the Institute was represented by Mr. Sterry B. Freeman, who is a director of Messrs. Chadburns and a Vice-President of the Institute for Liverpool.

Mr. Dewar is survived by his wife, a son and a daughter.



Abstracts of the Technical Press

Neither The Institute of Marine Engineers nor The Institution of Naval Architects is responsible for the statements made or the opinions expressed in the following abstracts.

The New Harland-B. and W. Double-acting Engine.

Since the Harland-B. and W. double-acting 2-stroke marine Diesel engine was first placed in service some 12 or 13 years ago, machinery of this type totalling well over 1,250,000 b.h.p. has been built. The main features of this design included the adoption of through scavenging, so that the scavenge air passed direct from the centre of each cylinder to the ends, the exhaust gases being discharged through piston-type exhaust valves at the top and bottom respectively. The time of admission of the scavenging air is controlled by the main piston and the discharge of the exhaust gases by the exhaust pistons. This arrangement makes it possible to utilise a higher mean effective pressure than when loop scavenging is used, with the result that the engine is of lighter weight and more compact. As originally designed and built, the Harland-B. and W. d.a. engine had exhaust pistons of smaller diameter than the main pistons, but among the modifications embodied in the improved design developed by Burmeister and Wain at Copenhagen, is an increase in the size of the exhaust pistons to that of the main pistons, and the elimination of the cylinder covers. At the same time, prefabricated welded construction was adopted for the bedplate, the A-frames and other parts of the engine, thereby reducing its weight. The first of these new coverless engines to be constructed by Harland and Wolff is a 6,000-b.h.p. 6-cylr. unit installed in a Lamport and Holt cargo liner which recently completed successful sea trials. The new engine weighs only 370 tons, compared with 490 tons for an engine of equal output, based on the earlier design. This corresponds to a specific weight of under 140lb./b.h.p. at a conservative rating of 115-120 r.p.m. The cylinders have a diameter of 550 mm. and the stroke of the main pistons is 1,200 m.m., that of the exhaust pistons being 400 mm. The m.i.p. for an output of 1,000 b.h.p. per cylinder at 122 r.p.m. is about 94lb./in.² for the top and 85.3lb./in.² for the bottom, the average being 89.6lb./in.². As the mechanical efficiency of the engine is 84.5 per cent., the corresponding b.m.e.p. is 75.7lb./in.². The fuel consumption between about three-quarter output and 10 per cent. overload is 0.36-0.37lb./b.h.p.-hr. The top and bottom exhaust pistons, which reciprocate together, are driven by eccentrics through a bottom yoke to which the lower exhaust piston is bolted, this yoke being attached to a corresponding upper yoke by four bolts, instead of two as in the earlier design. The eccentric sheaves and balance weights are integral with the cast-steel crank webs. The angle of advance of the eccentrics is such that there is a lag between the opening of the exhaust ports and the scavenge ports respectively, with the result that the pressure in the cylinder drops to about atmospheric at the end of the expansion. As the main and exhaust pistons are of the same diameter, there are no unbalanced pressures, and longitudinal stay bolts are, therefore, unnecessary. There are two fuel-injection valves per cylinder supplied from one fuel pump at a pressure of 5,000lb./in.² through a distribution box and connecting pipes. The ratio of compression is 13.04 to 1 at the top of the cylinder and 13.4 to 1 at the bottom, at a compression pressure of just under 500lb./in.². The vertical fuel pumps are located at the level of the entablature and are operated by a horizontal pump shaft which is chain-driven from the crankshaft. The cylinders and jackets are cooled with fresh water, whilst the main and exhaust pistons are cooled by lubricating oil which is delivered to each piston head through an annular passage between the C.I. sleeves surrounding the piston rod and the piston rod itself. The return flow takes place through a longitudinal hole in the rod. All the pistons have chrome molybdenum cast-steel crowns screwed into a C.I. body piece. Six piston rings and one scraper ring are fitted at either end of each main piston, whilst the exhaust pistons have six rings each, the piston-rod stuffing-box being provided with four twin rings and three double sealing rings. The flanged end of the piston rod is sandwiched between the top and bottom C.I. body pieces of the piston. Each complete cylinder comprises a centre section bolted to the scavenging belt and forming the entablature and a top and bottom liner, the lower end of the latter being detachable to facilitate the examination of the rings of the exhaust piston. As in the earlier engine, the scavenging arrangements comprise a two-unit blower with a capacity

of 20,000 cu. ft./min. driven by duplex chains and spur gearing from the crankshaft at 3.5 times the speed of the latter. Change-over valves in the inlet and outlet passages come into operation automatically when the engine is reversed. The scavenging air pressure at full output is in the neighbourhood of 3 to 3½ in. of mercury, and the corresponding exhaust temperature at the top is about 560° F. and at the bottom about 510° F. The various auxiliary pumps driven from the crankshaft are of the rotary type and a separate motor-driven fan is provided to remove oil vapour from the engine crankcase.—*The Motor Ship*, Vol. XXV, No. 297, October, 1944, pp. 206-211.

Some Suggestions for Reducing Cylinder Wear.

This paper is primarily concerned with cylinder wear in Diesel engines of the industrial type having cylinder diameters of between 8 in. and 22 in., piston speeds of or below 1,300 ft./min. and normal running speeds of up to 700 r.p.m. The whole subject is dealt with under seven separate headings, the first of which is engine design. The author suggests that a "well designed" engine might be considered as: "An engine which, when running under the normal conditions for which it was designed, and intelligently operated, shall be capable of a service period of 5,000 running hours without the necessity for withdrawing pistons; and that when the pistons are withdrawn, all the piston rings shall be operatively free". The second section is devoted to materials and finish, with particular reference to machine finishes and special finishes of cylinders or cylinder liners, pistons, piston-ring grooves, piston rings and other castings. The third section concerns the lay-out for a new installation and covers the arrangement of the induction and exhaust systems, the circulation of cooling water and the lubricating-oil cooling system. The fourth section of the paper deals with fuel combustion and includes references to the quality of the fuel oil, idling and light-load running of the engine, the inevitable slight dilution of the lubricating oil with fuel oil, fuel-injection equipment and defective inlet or outlet ducts. The fifth section is devoted to the question of lubrication and deals with the choice of lubricant, additives, oil cleaning, priming and distribution. The sixth section covers various aspects of engine operation, such as cold starting, idle and light-load running, overloading, distribution of load, general cleanliness and the keeping of accurate daily and weekly logs for record purposes. The seventh and last section of the paper deals with routine maintenance of the engine, separate sub-sections being devoted to fuel pumps and fuel injectors, as well as to the fitting of piston rings. Some suggestions for routine examinations are also put forward.—*Paper by H. Ferguson, read at a meeting of the Diesel Engine Users' Association on the 12th October, 1944.*

American Minesweeper with Diesel-electric Propelling Machinery.

The U.S.S. "Ardent" is one of a series of twin-screw minesweepers of 1,100 tons equipped with Diesel-electric propelling machinery of between 3,500 and 4,000 s.h.p. She is 222 ft. in o.a. length, with a beam of 32 ft. and a moulded depth of 17 ft. The Diesel-electric power plant and propulsion motors are installed in two separate and self-contained engine rooms, arranged in tandem and separated by cold-storage rooms and fresh-water tanks, etc. Each engine room contains two 8-cylr. four-stroke Diesel engines of the trunk-piston type, each directly coupled to a 500/720-kW. 250-volt Westinghouse generator of the d.c. type. The engines are of the De la Vergne design which enjoyed considerable popularity in the U.S. some years ago. Each engine has a continuous rating of 1,000 b.h.p. at 600 r.p.m., but the power of the ship is based on an output of 720 b.h.p. per engine developed at 560 r.p.m. At the after end of each engine room are two 882-h.p. propulsion motors which drive the propeller shaft through 2.5 to 1 reduction gearing. These twin motors take current at 250 volts d.c. and can, if necessary, take the maximum power output of the generator sets at any speed between 700 and 900 r.p.m. One of the engine rooms also contains two auxiliary Diesel generators. Each of these consists of a 150-b.h.p. 3-cylr. General Motors engine running at 1,200 r.p.m. and directly coupled to a 100-kW. Westinghouse generator of the a.c.

type, since alternating current is used for auxiliary purposes. The entire lay-out is very compact and the total length of each engine room is only 32ft. The width is considerably less than the ship's beam, as fuel-oil tanks are arranged along the sides of both engine rooms as well as in the D.B. compartments below them. The motor-driven deck machinery includes a minesweeping winch, an anchor windlass and boat hoists. The ship has two squat oval funnels, the spacing of which corresponds to the positions of the main Diesel generators, a straight stem and a stern of the usual minesweeper type. There is a single tall mast abaft the conning tower and bridge.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 36,395, 5th October, 1944, p. 3.*

Liner Wear in Opposed-piston Engines.

The following is a practical record of the working life of a twin set of cylinder liners of the two 4-cylr. Doxford engines installed in the Prince Line motorship "Siamese Prince", which was placed in commission in 1929, since which time she has been employed solely on the Far Eastern services. The cylinders of these engines have a bore of 600 mm. and the combined stroke of the opposed pistons is 1,840 mm. The i.h.p. per cylinder is 850 at the service rating of 3,000 b.h.p. of each engine, the maximum firing pressure is 570-580lb./in.² and the exhaust temperature is approximately 580° F. The exhaust manifolds are water-cooled and the engine frames are of cast-iron (not welded) construction. Throughout her service, the ship maintained an average speed of 14.4 knots on a daily fuel consumption of some 24 tons for all purposes. The liner wear was measured at fairly regular intervals over the whole period and plotted in the form of yearly graph readings. The liners of these engines are of the engine-builders' standard material and the

AVERAGE WEAR OF LINERS OF FOUR-CYLINDER DOXFORD ENGINES.

Average wear port engine per 1,000 hrs.	00404in.
Average wear starboard engine per 1,000 hrs.	00432in.
Maximum wear port one cylinder per 1,000 hrs.	00491in.
Maximum wear starboard one cylinder per 1,000 hrs.	00530in.
Lowest maximum wear port one cylinder per 1,000 hrs.	00252in.
Lowest maximum wear starboard one cyl'r per 1,000 hrs.	00233in.
Average wear port engine (top) six years	2.81 mm.
Average wear starboard engine six years	3.06 mm.
Average wear port engine (bottom) six years	3.22 mm.
Average wear starboard engine six years	3.67 mm.

LINER WEAR IN DOXFORD ENGINES.

Cyl.	Top max. wear.			Bot. max. wear.			Least max. wear.			Ovality.		Com. chr.		Average wear top.	Average wear bot.	Average engine top and bot.	Engine average.	Per 1,000 hours.	Per 1,000,000 ft. piston travel.	R.p.m. av. 6 yrs.	Piston speed.	
	F. and A.	P. and S.	F. and A.	P. and S.	F. and A.	P. and S.	Top.	Bot.	F. and A.	P. and S.	Top.	Bot.										
No. 1	3.14	2.14	3.63	3.23	3.03	2.14	1.0	.2	-.65	.35	2.64	3.13	2.88	—	.00327 in.	.00011 in top	—	—	—	—	—	—
No. 2	3.7	2.2	3.68	4.01	3.68	2.2	1.5	.33	-.4	.15	2.85	3.84	3.34	3.16 mm.	.00431 in.	—	—	110.7	561.6 ft. p.m.	767.8 ft. p.m.	—	
No. 3	3.82	2.3	3.07	3.53	3.07	2.3	1.5	.46	-.15	.05	3.06	3.3	3.18	—	.00411 in.	.00096 in bot.	—	—	—	—	—	
*No. 4	2.5	1.75	2.57	2.67	2.5	1.75	.75	.1	-.8	.29	2.12	2.62	2.36	—	—	—	—	—	—	—	—	
No. 1	3.92	3.27	3.85	3.24	3.85	3.24	.65	.61	-.23	-.22	3.59	3.54	3.565	—	.00460 in.	.000118 in top	—	—	—	—	—	
No. 2	3.42	2.05	3.56	4.1	3.42	2.05	1.37	.45	.02	.1	2.73	3.83	3.28	3.87 mm.	.00420 in.	—	—	110.8	561.6 ft. p.m.	767.8 ft. p.m.	—	
No. 3	3.2	2.48	2.9	3.9	2.93	2.48	.72	.97	-.43	-.35	2.84	3.41	3.12	—	.00400 in.	.000103 in bot.	—	—	—	—	—	
No. 4	3.8	2.4	3.74	4.1	3.74	2.4	1.4	.36	-.15	-.18	3.10	3.92	3.51	—	.00451 in.	—	—	—	—	—	—	

*Liner renewed when 4 years 10 months old due to cracks.
Total S.T. hours, 30,895. Distance, miles, 421,441.

pistons are each fitted with five C.I. rings, those in use during the life of the liners having been supplied by a firm specialising in piston rings. They appeared to be of a softer iron than the liners and their average life was in the region of 85,000 miles. The average length of each round voyage was about 30,000 sea miles, and it was found necessary to fit a complete new set of piston rings during the third voyage. The maximum wear of the upper halves of the cylinder liners has invariably taken place in a fore-and-aft direction, although it is more usual for the greatest wear to occur in the athwartship line; this fact is probably accounted for by the method of guiding the upper pistons. In the lower halves of the liners, however, the wear is greatest in the athwartship line, probably because the attachment of the piston to the crosshead is the more usual rigid type. The ovality reached in the upper halves of the liners is much greater than in the lower halves, despite the fact that the total maximum wear in the latter is the greater. In more recent types of Doxford engines a different method of attaching the pistons to their respective crossheads has been adopted; they are left free to adjust themselves to the cylinder-bore alignment, and this reduces the wear on the liner. A certain amount of distortion is invariably present in the centre or combustion-chamber portion of the liners examined. This part of the liner tends to decrease in diameter, and though the distortion occurs in the same horizontal plane in each case, its actual location does not remain constant, but creeps to vary-

ing positions in that plane. It amounts to a maximum of 0.6 mm. in some cases. The maximum wear in the last liner to be changed had only reached 5 mm. after some eight years and four months of service, although the engine-builders consider that 6 mm. wear is the approximate life of a liner. The best of the eight liners shows an estimated life of 18 years and the worst nine years, which represents a satisfactory performance. The renewals of the liners were not carried out because of wear, but owing to the development of cracks at some of the valve apertures, this being a fairly common occurrence with this type of engine. The cause of the cracks developing was fully understood and occurred early in the life of the respective engines. Most of the liners were changed by the ship's staff as occasion required, and this usually took 22 working hours. The water spaces, when examined on removal, proved to be quite clean and free from scale, the surfaces being covered only by an easily removable brownish deposit. These liners were renewed in periods of 6½ to 8½ years, and during their lifetime had only been cleaned with a weak solution of grease-removing compound in fresh water. No liner was removed at any time for cleaning. The cooling medium employed was the usual one of distilled water with just enough bichromate of potassium to give it a distinct yellow colour. Special care was taken to prevent contamination by sea water, as the corrosive action of the mixture would thereby have become far greater than that of salt water alone. With this type of engine fuel-valve timing is of great importance, the fuel pressures being as high as 6,000-6,500lb./in.² and the lift of the valves 2.8 mm. The supply of cylinder lubricating oil for these engines was gradually reduced until it was found that about 9 gall. per 24 hours gave the most satisfactory results. This included the amount necessary for the respective scavenge pumps. The rate of liner wear was proof that sufficient oil was being used. The cylinder lubricating oil was fed to the cylinders in the ratio of 2 to 1 approximately, the upper pistons receiving the larger amount. The results of over-lubricating are readily discernible when opening out the cylinders, pistons and scavenge trunks for overhaul; in combination with, say, late valve timing, liners will suffer from much heavier wear, carbon deposits and overheating. It is possible with this type of 4-cylr. engine to obtain perfectly smooth running at as low as 20 to 25 r.p.m. over long periods by taking care to see that the valve timing is correct and that the lubrication is efficient. The balance of the engines was so satisfactory that it was found unnecessary at any time to harden

up any holding-down bolts during the whole nine years, an unusual state of affairs even with most steam engines. Some particulars of the average and general wear of the liners are given in the accompanying tables.—A. A. Righton, *"The Motor Ship", Vol. XXV, No. 297, October, 1944, pp. 226-228.*

Roller Bearings on Board Ship.

Some of the U.S. Maritime Commission's ships equipped with geared Diesel engines have been fitted with taper roller bearings to support the pinions of the mechanical gearing. Bearings of this type have also been employed for transmitting the propeller thrust on board a variety of U.S. naval vessels ranging from high-speed motor-boats to cruisers, and their behaviour is said to compare most favourably with that of the normal thrust bearing, test data showing that the coefficient of friction is only about 0.003. The design most commonly adopted comprises a steep-angle bearing having a double-cone inner race, the housing being of the spherical self-aligning type in order to allow for deflections of the hull. As the bearing can take the radial load as well as the thrust load, the supporting bearings can be dispensed with, thus making the unit more compact. Further, while oil can be supplied under pressure if desired, it is claimed that satisfactory lubrication can be effected by maintaining a suitable oil level in the sump.—*"Shipbuilding and Shipping Record", Vol. LXVI, No. 15, 12th October, 1944, p. 339.*

Heavy-oil Engine Progress.

The paper constitutes a brief survey of the progress made during the past 50 years in the design and construction of heavy-oil engines, with special reference to the development of the combustion chamber and fuel injection system. Three main stages of development are considered, viz.: (I) the low compression engine, (II) the medium compression engine, and (III) the high compression engine of the present day. The author then proceeds to discuss and illustrate the various developments which took place in the design and construction of the combustion chamber and fuel injection system during each of these stages.—Paper by J. W. Armstrong, "Transactions of the Institute of Marine Engineers", Vol. LVI, No. 9, pp. 185-192.

Manganese Bronze Co.'s Propeller Shaping Machine.

A machine for shaping the backs, and in some cases the faces, of ships' propellers is illustrated in Fig. 2. The result is obtained

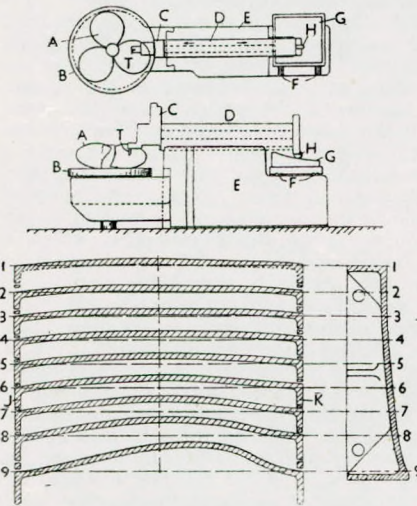


FIG. 2.

automatically and without the aid of a master propeller. Referring to the upper diagrams on the left, the propeller (A) is mounted on a horizontal rotary table (B) which can be raised or lowered intermittently by gearing. The horizontally reciprocating part (D) of the head operates in slides carried by a frame (E). The frame also embodies cross slides (F) in which a former (G) moves in a transverse direction. The intermittent movements of the former are arranged to be in step with those of the rotary table. The former (G) comprises a rectangular plate with predetermined contours. There is a roller (H) on the planing head (D) maintained in contact with the plate. When the head is reciprocated, the tool (T) performs a stroke compounded of two rectilinear motions at right angles. Both these motions take place in the vertical plane containing the axis of the table (B), the tool (T), the planing head (C, D) and the roller (H). The part (C) is connected by link mechanism, whereby vertical motion is imparted to the head as the roller rises and falls in contact with the former (G). The proposed diameter of the roller is 3in., the width 1.5in., and it is rounded at the periphery to a radius of 0.75in. A former for a three-bladed propeller is shown in the lower left-hand diagram. The propeller would have blades 20in. long, the sections (1-9) being taken at radii of 20.5in. 18in., 16in., 14in., 12in., 10in., 8in., 6in. and 3in. The edge (I) of the former corresponds with the leading edge of the blade, while the edge (K) corresponds with the trailing edge. In the right-hand diagram are shown sections of a blade (1-6) taken at 14in., 12in., 10in., 8in., 6in. and 3in.—"The Motor Ship", Vol. XXV, No. 299, December, 1944, p. 302.

Design and Construction of High-pressure Boilers.

The design and construction of marine boilers for the generation of steam at pressures up to 1,500lb./in.² and temperatures of 900°-1,000° F. involve fewer problems than had been anticipated. Since the ratio of the density of water to saturated steam at 150lb./in.² pressure is only one-fifth of that at 400lb./in.², it is sometimes thought that the design of a marine boiler to operate with natural circulation at a steam pressure of 1,500lb./in.² is a matter of some difficulty, where it has been proved both by calculations and practical experience that these conditions do not involve the use of an abnormal number of downcomers. In designing the superheaters for the higher temperatures the velocity of steam must be kept high enough to maintain the tubes at a temperature at which no undue "creep" of the tubes will take place. The higher the velocity, the greater the pressure gradient, so that in order to avoid any excessive drop

of pressure in the superheater, it becomes necessary, when the rate of heat absorption is high, to use alloy steels such as "18-8", which can be employed at a higher temperature than low carbon or low alloy steels. The higher the pressure of the steam the lower its specific volume, which means that for the same pressure gradient in a tube, a higher mass flow can be carried with high-pressure than with low-pressure steam. Either radiant of convection superheaters, or a combination of both may be used for high-pressure boilers. Since the intensity of heat absorption is greater with a radiant superheater than with a convection superheater, the mass flow of the steam is usually greater, which increases the pressure gradient; but since the total surface of the radiant superheater is less than that of the convection superheater, the overall pressure drop need not be relatively large. The radiant superheater is particularly suitable for use with a boiler having a separately fired superheater or reheater. The author's firm (the Foster Wheeler Corporation) have built and operated a marine type boiler delivering steam at a pressure of 825lb./in.² and a total temperature of 850° F., and are completing the construction of others for a working pressure of 1,500lb./in.² and total temperature of 1,000° F.—Abstract of paper by J. Blizard presented at the 1944 annual meeting of the Society of Naval Architects and Marine Engineers, "Marine Engineering and Shipping Review", Vol. XLIX, No. 8, August, 1944, pp. 204 and 206.

Pescara Supercharged Gas Generator and Turbine Propulsion System.

In Fig. 3 are shown two lay-outs of free-piston oil-engined gas generators and turbines for a propulsion plant. The parts in the upper diagram comprise a rotary blower (A) driven by a gas turbine (B) which is supplied by generators (C). The main gas generators (D) supply a turbine (H). This turbine drives the propeller (34) through reduction gearing. It is stated that the plant can maintain a high efficiency over a wide range of load. At low loads, the clutch (35) is engaged, while the clutch (38) is disengaged and the valve (30) is closed. The gas generators (D) are stopped, together with the blower (A), while the generators (C) draw air from the atmosphere through a valve (29). The turbine (B) alone drives the propeller through the clutch (35) and the gearing. At the intermediate loads both clutches are engaged. The valve (30) is closed and the gas generators (D) are stopped. The blower (A) delivers supercharged air to the generators (C) and the turbine (B) drives the blower (A) as well as the propeller. At high loads the clutch (35) is disengaged. The valve (30) is opened and the gas generators (D) are brought into action. The air-compressing section (A, B, C) now serves wholly to feed compressed air, the blower (A) supercharging its own gas generators (C) and the generators (D) which supply motive gas to the turbine (H) for driving the propeller. A method of controlling the adjustment rods of the gas-generator fuel pumps is included in the mechanism shown in the lower diagram. The turbo blower (B, A) of the auxiliary group drives a centrifugal governor (44), while the turbine (E) drives a second governor (45). The governors actuate parallel rods (46, 47) sliding in a frame (48). There is also a third rod (49) and all the rods are connected by a lever (50) which slides in joints in the rods (46, 47). The rod (49) is connected to the injection-pump control rods (12) of the auxiliary group (C). The position of the lever (50), shown in full lines, corresponds to a reduced load, the full load position being indicated by a dotted line.—"The Motor Ship", Vol. XXV, No. 299, December, 1944, p. 302.

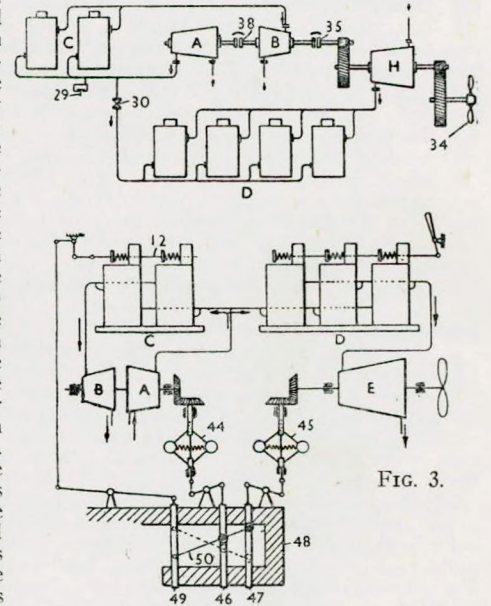
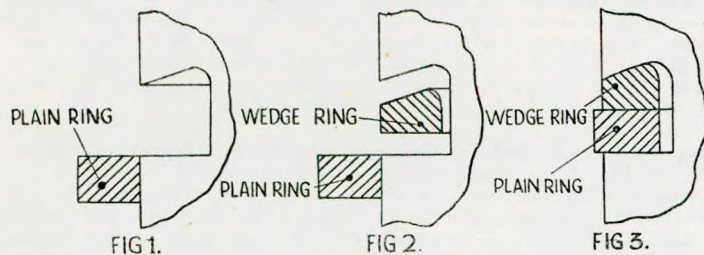


FIG. 3.

The M.E.P. Piston Ring.

An improved form of oil-engine piston ring, known as the M.E.P. piston ring, has recently been put on the market in this country. It is actually made up of two separate components, viz., a

plain ring and a wedge ring, the object of the latter being to eliminate any vertical play of the plain ring in the piston groove. Referring to the accompanying sectional diagrams, the wedge ring tends to spring outwards and therefore nips the plain ring and prevents any play of the latter in the piston groove. This represents a partial restraint of the outward spring of the plain ring, owing to its tightness in the piston groove. When there is pressure behind the rings due to gas leakage, especially in the top, or firing, rings, the wedge ring exerts sufficient pressure on the plain ring to hold it firmly in the piston groove and prevent any undue outward spring. This theory has been borne out in practice by the fact that where M.E.P. piston rings have been used, the cylinder bore does not become tapered at the firing position, as it frequently does when plain rings are fitted. When a piston fitted with M.E.P. rings is withdrawn for examination, it is usually found that the wedge ring and the plain ring in the top grooves are apparently jammed, but a slight tap in their circumference will show that this is merely due to the contraction of



the grooves in a cold condition. However, this fact demonstrates the impossibility of fitting a single solid ring in a cold piston groove and expecting it to work without vertical clearance in that groove after several hours of high-temperature running. The illustrations show how the rings are inserted. The plain ring is first placed on the land just below the groove into which it is to be put (Fig. 1), and the wedge ring is then inserted in the groove (Fig. 2). One end of the plain ring is then inserted by holding the wedge ring against the bevelled edge of the groove. The wedge ring is kept in that position while the piston is worked round until both rings are inserted in the groove in their correct position (Fig. 3). When the wedge ring is first inserted in the groove it should be placed with the opening 180° offset from that of the plain ring.—*"The Motor Ship"*, Vol. XXV, No. 298, November, 1944, p. 263.

Superheaters for Watertube Boilers.

In a paper on the above subject read before a recent meeting of the Institute of Fuel by L. C. Southcott, B.Sc. and Dr. D. W. Rudorff, it was pointed out how progress has increased the amount of heat imparted to the steam by the superheater from about 10 per cent. of the total for working pressures of 200lb./in.² to over 26 per cent. when the working pressure is 1,400lb./in.². This, and the increasing temperatures (600° to 950° F. for the two pressures just cited), has caused the superheater to be brought nearer to the furnace as an alternative to an unwieldy increase in its size. An early development of this form of superheater was the Babcock and Wilcox multi-pass interdeck superheater with headers situated in the gas path. The introduction of water-cooled furnace walls reacted on the superheater for two reasons, the first being the higher percentage of CO₂ which could be carried by the employment of a water-cooled furnace and the consequent reduction in the weight of the hot gases available for superheating. At the same time, there has been some reluctance on the part of the boiler designers to reduce the amount of the convection heating surface between the furnace and the superheater, and this has made it necessary to place some of the superheater tubes between the front rows of steam generating tubes in order to obtain a location possessing a sufficiently high gas temperature and maintain a reasonably uniform steam temperature at various loads. The employment of higher steam temperatures brought up the problem of control. With temperatures around 850° F., turbine makers could not accept the possibility of rises above this figure for any length of time, so that superheater manufacturers had to include in their guarantee clauses to safeguard users against damage resulting from an excessive rise of temperature. Although control of superheat by the use of both contact and non-contact de-superheaters, as well as by the regulation of gas flow by dampers or by means of separate furnaces, has been practised for many years, it was not until 1932 that the first order was placed in this country for an inter-controlled superheater in which primary and secondary stages were used with a non-contact de-superheater regulating the degree of inter-stage cooling. Regulation by control of gas flow has also been employed in this country, sometimes by by-passing a proportion of the gases and sometimes, as in the Babcock-Johnson and Yarrow boilers, by

providing two gas outlets. In some cases, de-superheating by direct injection of feed water between two superheating stages or—more usually—after full superheating, has been adopted; but this method is open to the objection that dissolved salts may be brought in with the water. Whatever the system employed, adequate superheater surface must be provided to meet the specified requirements under the conditions most adverse to the superheater, so that at all other times the superheater surface will be in excess of requirements. It is possible, however, to utilise a minimum superheater surface if the control of the superheater is effected by means of separate specially-located burners. Where steam temperatures of more than 850° F. are likely to be experienced for extended periods, welded joints are generally used in the superheater itself. The headers are manufactured with welded-on stub tube ends and the elements are welded to these stubs when the boiler is erected. It is usual to cut out up to about 3 per cent. of the welded joints for inspection. The main causes of trouble with superheater installations are, in general: (1) ashing of the external surfaces causing a reduction of superheating, the load in some cases being restricted because of the reduction of draught through the superheater; (2) failure of elements due to bursts caused by stoppages from salts carried over in the steam; and (3) failure by overheating during the period when the steam pressure is rising. As regards ash, reasonable freedom from this can sometimes be secured by soot blowers, but this is dependent upon the type of ash and the facilities afforded to the soot-blower designer. Washing the superheater with water or steam has also been adopted in the last few years. Undoubtedly much of the ash trouble has been due to the very close pitching of the superheater tubes, which is inevitable if a high degree of superheat is to be got from a convection superheater; but the design may be such that open pitching is possible for a portion of the superheater, and promising results have been obtained by giving an open pitch to the first two or three rows of superheater tubes. Such pitching does not necessarily prevent deposits of ash on the front tubes, but the greater space between those tubes lengthens the time before bridging occurs, and this makes mechanical cleaning easier. By the time the gases reach the closely pitched tubes they are cooler and the tendency for ashing is consequently reduced. From the carrying over of solids, failures can be extensive. In one installation where frequent trouble was experienced a reduction in the degree of water concentration coupled with the tightening up of the separating baffles in the steam drum led to a marked reduction in the number of element failures. Overheating of the superheater elements while the steam pressure is being increased is by no means uncommon, although it has been proved by actual tests that steam can be raised in a cold boiler to a pressure of 1,400lb./in.² in 2½ to 3 hours without permitting the metal temperatures in the superheater to exceed a safe figure. As to materials, mild steel has been found to be quite suitable and safe for pressures up to 700lb./in.² and steam temperatures of 850° F. Where somewhat higher temperatures are to be dealt with, molybdenum steel tubing is to be recommended, and for still higher temperatures low chrome-molybdenum tubing is employed. All these steels, however, have poor resistance to scaling; therefore for steam temperatures above 900° F. it is desirable to maintain high rates of flow to keep the metal temperature as close as possible to the steam temperature. Moreover, these high steam velocities entail an adequate pressure drop, and so make for uniform steam distribution if the header design is reasonable. Rectangular headers were used in some of the early superheaters, but increasing pressures and temperatures made it necessary to adopt the circular form. For high pressures solid forgings, suitably machined, are general, with drawn tubes for lower pressures. With forged headers, the flanges and branches can be integral. A number of 0.5 per cent. molybdenum headers are in use for temperatures of 900° to 960° F.—*"The Siren"*, Vol. CXCII, No. 2,510, 4th October, 1944, pp. 15 and 25.

Boiler Control.

The authors state that the purpose of their paper is to discuss the control of marine boilers, and particularly the use of various devices for improving the responsiveness and efficiency of steam generating units. In advocating the use of meters and automatic controls for regulating the operation of ships' boilers, the authors make the following claims in regard to the advantages to be derived from their adoption: (1) simplified operation and more efficient use of personnel; (2) fuel economy; (3) increased life of furnaces and auxiliaries; (4) improved engine efficiency through closer regulation of steam pressure and steam temperature conditions; and (5) less smoke. The above claims are briefly discussed and the authors then proceed to describe the various types of metering and control devices available for marine work, these being illustrated by diagrams. The concluding portion of the paper is devoted to an explanation of the safety precautions to be taken to prevent furnace explosions in marine boilers which are being operated by means of automatic controls.

The authors express the opinion that a more extensive use of meters and automatic control will accompany the increasing trend towards the employment of high pressures and temperatures in marine boilers. —Paper by E. G. Bailey and P. S. Dickey, "Transactions of the Institute of Marine Engineers", Vol. LVI, No. 8, September, 1944, pp. 173-181.

English Electric Co.'s Engine Drive with Gearing and Electro-magnetic Slip Couplings.

A diagrammatic arrangement of ship's propelling plant with gearing and electro-magnetic slip couplings is illustrated in Fig. 1. The engines (1, 4) are coupled to the shaft (7) having pinions (29, 30). One pinion (29) meshes with an idler pinion (31) and the

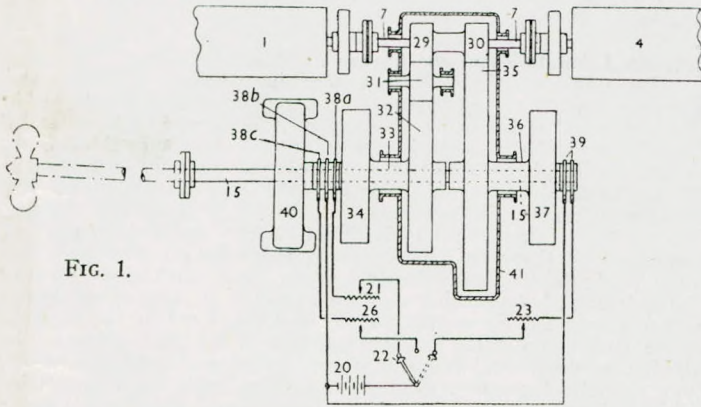


FIG. 1.

corresponding gear-wheel (32) is secured to a hollow shaft (33). This shaft is mounted freely on the propeller shaft (15). The forward pinion (30) meshes with the gear-wheel (35) on another hollow shaft (36), which is also free on the propeller shaft. The input member of the electro-magnetic slip coupling (34) is secured to the shaft (33) and the output member drives the propeller shaft (15). At the forward end, the input member of the slip coupling (37) is secured to the shaft (36), the output member also being attached to the propeller shaft, as in the case of the after unit. An electro-magnetic brake (40) retards the speed of the propeller shaft when the ship is required to go astern. The pinion (29) drives the propeller shaft in one direction through the gear-wheel (32) and the coupling (34), while the pinion (30) drives the shaft in the reverse direction through the gear-wheel (35) and the coupling (37). The field windings of the coupling (34) are connected between slip-rings (38a, 38b), while those of the brake (40) are connected between slip-rings (38b, 38c). With the coupling (37), the field windings are connected across a pair of slip-rings (39). The switch (22) and the rheostatic regulators (21, 23, 26) are wired to a supply unit (20). The ahead and astern gearing is enclosed in a casing (41), the hollow shafts (33, 36) projecting from opposite sides, with the propeller shaft (15) extending right through both of the hollow shafts. The electro-magnetic slip couplings are on the slow speed side of the gear, which is a suitable arrangement for an installation employing high-speed engines.—"The Motor Ship", Vol. XXV, No. 299, December, 1944, p. 302.

Blowing Down Boilers.

The Ministry of Fuel and Power have recently issued a *Bulletin* for the guidance of boiler attendants and users of steam boilers dealing with the blowing down of boilers. It is suggested that there is a general impression that boilers are blown down simply to blow out sediment and mud from the bottom of the shell or drums, and that scale is lessened by blowing down. Little is known about the safe limits of concentration of the salts in the boiler water in relation to priming, whilst a hydrometer is frequently regarded as a scientific instrument only to be handled by a chemist. After explaining why it is necessary to blow down a boiler, the *Bulletin* gives a useful guide as to the best time to blow down. The routine control of blowing down is then described and the factors which influence the amount of blowing down necessary to prevent priming are examined. The value and use of the hydrometer in testing samples of boiler water to ascertain the amount of dissolved solids it contains is dealt with at length. The need for careful supervision where continuous blowing-down connections are fitted, in order to prevent unnecessary loss of heat (and fuel), is stressed. Then follows a section on heat recovery from continuous blowing down. Much has yet to be done in this direction as regards fuel saving, and many operators fail to realise that they have in their boiler blow-down much useful heat which might be put to work in order to reduce the amount of heat

to be provided by the boilers. Clear directions are also given for calculating the amount of blowing down required. The *Bulletin* is available free of charge from the Ministry of Fuel and Power, London, or from Regional Offices of the Ministry.—"Metallurgia", Vol. 30, No. 180, October, 1944, p. 328.

Richardsons, Westgarth Engine Piston Construction.

The accompanying sectional diagrams show the construction of an improved design of piston for Diesel engines recently developed and patented by Richardsons, Westgarth & Co., Ltd. The top plan is a section at the line A of the elevation; the centre half-plan is taken at the line B; and the bottom half-plan is through the top part of the piston at the line C. The assembly comprises four main parts, viz., the piston crown (1), the bottom crown (4), the conical

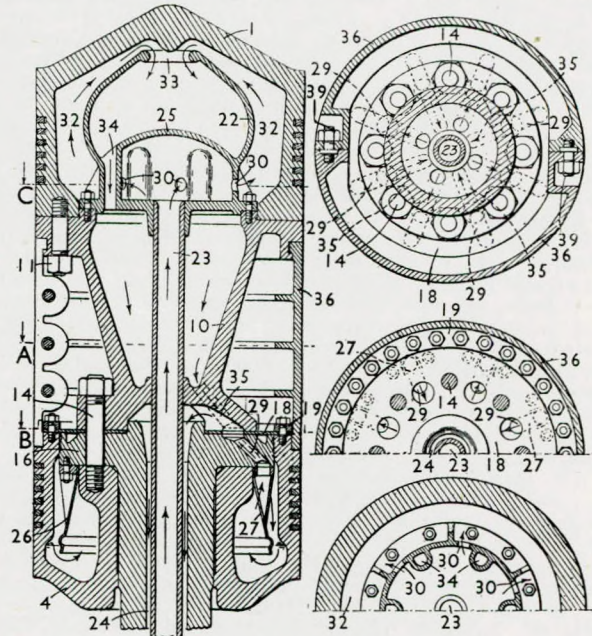


FIG. 1.

distance piece (10), and the piston-rod flange (16), which fits between the distance piece and the bottom crown. The top crown is secured to the bottom crown by studs (11) and the bottom crown is attached to the piston-rod flange (16) by studs (14) which pass through the latter. As some differential expansion may take place between the inner and outer walls of the bottom crown, a flexible disc (18) is fitted between it and the piston-rod flange, being secured to the latter by a ring (19). A globular deflector (22) is located in the top crown for directing the flow of the cooling fluid, which is delivered through a pipe (23) in the bore of the piston rod. A conical deflector (26) is fixed in the bottom crown. The cooling fluid passes up the pipe (23) and is deflected by a diaphragm (25) through holes (30) in the base of the globular deflector into the space (32) inside the piston crown. From this annular space the cooling fluid passes down through a hole (33) at the top of the globular deflector to the passages (34) and holes (35) into the bottom crown. The return flow is by way of openings (27) in the wall of the conical deflector (26) which communicate with corresponding holes (29) in the piston-rod flange and enable the fluid to flow into the annular space (24) between the central pipe and the bore through the piston rod. In order that the complete piston may be cylindrical and flush from end to end, a split casing (36) is clamped round the distance piece (10). The halves of this casing are held together by nuts and studs (39), which pass through the vertical flanges of the casing in a manner shown in the top plan view.—"The Motor Ship", Vol. XXV, No. 298, November, 1944, p. 268.

British Merchant Shipbuilding in 1943.

Some particulars have now been released of representative types of ships ordered for Government or private account, and delivered by British shipyards in 1943. It would appear that, on the whole, the 10,000-ton d.w. tramp steamer constituted the backbone of the output during that year. A typical design of a vessel of this class is the "B" type closed shelter-decker having a deadweight of about 10,400 tons and gross tonnage of about 7,000. The midship portion of these vessels has been fabricated by constructional engineering works, but the ends, which are not adapted for pre-fabrication, are

built in the usual way in the shipyards. This class of ship has six holds, one of which is arranged as a reserve coal bunker, with steel grain divisions clear of the hatches, and a special feature of the equipment is the full provision for handling cargo. In many of the vessels a 40-ton and a 50-ton derrick are fitted. Compared with pre-war practice the ballast capacity of these ships is large, and some 1,000 tons more water ballast than was formerly considered necessary can be carried in deep tanks abreast the machinery spaces, in the wings between Nos. 1 and 2 holds, and at the sides of the tunnel aft—a disposition which gives the ship a good trim and a proper immersion of the propeller, when light, without imposing any undue stresses on the hull structure. The ship's officers and petty officers are accommodated in deck-houses amidships, while the seamen and firemen are berthed aft. A typical machinery installation is a triple-expansion engine and three coal-fired boilers working at 220lb./in.² with forced draught. Cargo vessels of this general class have been built by Armstrong Whitworth, Burntisland, Connell, Harland and Wolff, Readhead, J. L. Thompson, Lithgow, Bartram, Short, Gray and Pickersgill. A series of motorships with similar characteristics to the steam tramps have been built by Doxford, Barclay Curle and other shipbuilders. In addition to what might be termed standard cargo tramps, vessels of special design have been constructed for the M.o.W.T. These include a number of 10,000-ton cargo ships with three 120-ton derricks for handling heavy and bulky items of cargo; partially refrigerated steam tramps with an insulated-cargo capacity of 250,000 cu. ft., as constructed by Short, Lithgow and Armstrong Whitworth; fully-refrigerated motorships of 10,000 tons d.w. with a refrigerated capacity of about 450,000 cu. ft.; and smaller, general-purpose tramp steamers of some 4,700 tons d.w., having two 80-ton derricks and suitable for use as crane ships. Apart from these cargo vessels, the 1943 deliveries included a number of steam and motor tankers of from 11,000 to 15,000 tons d.w. and a variety of smaller vessels, such as Diesel-engined coasters of 350 tons d.w., oil-burning steam tugs, and steam puffers with all-welded hulls carrying about 140 tons of cargo. Mention must also be made of a number of fast cargo liners, of which the "Empire Chieftain", 11,950 tons d.w. built by the Furness Co., may be taken as an example. This vessel is equipped with geared turbines and has two Foster-Wheeler type watertube boilers. The cargo-handling equipment includes an 80-ton, a 50-ton and a 30-ton derrick, and there is accommodation for 12 passengers. The ships ordered by private owners include a much wider diversity of types and vary in size from about 8,000 to 11,200 tons d.w. Most of the fast cargo vessels have twin screws driven by oil engines or geared turbines.—*Fairplay*, Vol. CLXIII, No. 3, 198, 24th August, 1944, p. 296.

English Electric Co.'s Oil-cooled Bearing.

An improved design of oil-cooled bearing has recently been patented by the English Electric Co., Ltd., and is illustrated in Fig. 2. Some of the oil supplied to this bearing passes into the interior of

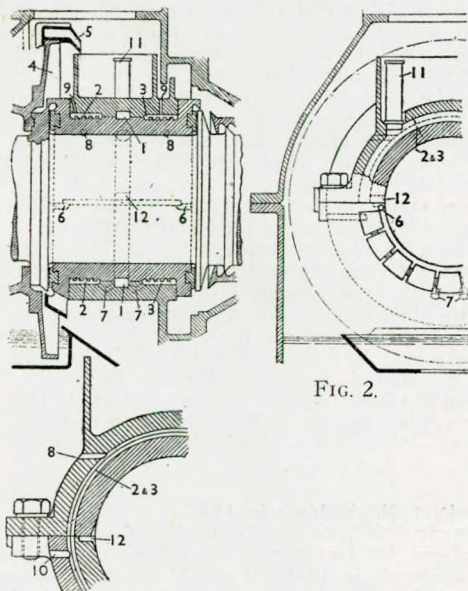


FIG. 2.

the bush, whilst the remainder is discharged through annular chambers around the outside, thus giving a cooling effect. The bush is of the usual split type and has a channel (1) on the outside, communicating with further channels (2, 3) by means of holes (7) at the bottom of the bearing. The shaft carries a disc (4) which picks

up oil and discharges it through a spout (5) into a collecting trough. The oil in this trough reaches the central channel (1) through filters (11) and grooves (12), and there are holes (6) for lubricating the thrust surfaces at the ends of the bush. In order to increase the cooling area, the annular chambers (2, 3) are provided with fins. The level of oil in these chambers rises, due to the head of oil in the trough at the top. Escape holes (8) are provided for the oil to overflow and run into the sump below, while air locks are prevented by the provision of holes (9) at the top of the bearing. Additional escape holes (10) are arranged in the vicinity of the split in the bush. These holes do not allow the oil to escape as rapidly as it enters from the trough, so that there is always enough oil to reach the upper part of the bush and thus pass away through the upper escape holes (8).—*The Motor Ship*, Vol. XXV, No. 298, November, 1944, p. 268.

Research Laboratories of the Bureau of Ships.

The functions of the Bureau of Ships of the U.S. Navy Department include the administration of a number of laboratories in which technical and scientific research work is carried on by specially trained naval officers and civilian scientists. These establishments include the Naval Research Laboratory of Anacostia, D.C.; the David W. Taylor Model Basin at Carderock, Md.; the Naval Engineering Experiment Station at Annapolis, Md.; the Naval Boiler and Turbine Laboratory at the Philadelphia Navy Yard; the Industrial Test Laboratory at that yard; the Material Laboratory at the Brooklyn Navy Yard; the Radio and Sound Laboratory at San Diego, Cal.; the two Industrial Laboratories at Mare Island, Cal., and at the Puget Sound Navy Yard, Bremerton, Wash.; and the two Test Laboratories at the Charleston, S.C., and Portsmouth, N.H., Navy Yards. In addition to these establishments, there are several others in which more specialised research work is carried on, these being the Paint Laboratories at the Mare Island and Portsmouth, Va., Navy Yards; the Rubber Research Station at Mare Island; the Metals Laboratory at Munhall, Pa.; the Rope Laboratory at Boston, Mass.; the Petroleum Laboratories at Houston, Tex., and at Norfolk, Va.; and the Experimental Diving Unit at the Washington, D.C. Navy Yard. Among the research work which is being carried out in some of these establishments at the present time is the development of heat-resisting lubricating oils; research into the life of electrical storage batteries; the development of suitable materials for salt-water pipe systems in ships, the study of boiler mountings, O.F. burners and similar appliances; the development of special pipe and hull insulating materials; the testing of electric cables and electrical appliances; the testing and development of substitutes for manila ropes; and the development of fire-resisting paints, which minimise the fire hazard aboard ship. The Training School for Water Tenders and Firemen at the Philadelphia Navy Yard, where these E.R. ratings undergo their initial training, works in close touch with the Boiler and Turbine Laboratory of that yard.—*Marine Engineering and Shipping Review*, Vol. XLIX, No. 8, August, 1944, p. 182.

Sufficient Oil for 100 Years.

Not long ago it was said that the world's oil reserves would only suffice for a period of 14 years, this estimate being based on the belief that the proved reserves of petroleum amounted to some 3,000 million tons, which corresponds to about 14 times the world's annual consumption. The position has now been analysed by the Standard Oil Company, of New Jersey, whose pronouncement on this matter represents the official views of the American oil industry. It is pointed out that "proved reserves represent a conservative estimate of the readily recoverable oil in *actually developed fields only*". They reflect no part of the petroleum we expect to find to-morrow". In the U.S. some 1½ million square miles or 960 million acres are favourable for the accumulation of petroleum, and on the basis of past experience it is anticipated that at the very least 15,000 square miles will prove to be petroleum-producing land. An average yield of 10,000 barrels per acre is certain, thus providing a total of 96,000 million barrels, of which 28,000 million barrels have already been produced. This leaves approximately 70,000 million barrels to be recovered, but the increased efficiency of present-day recovery processes makes this amount equivalent to 100,000 million barrels or 15,000 million tons. It is considered that oil fields outside the U.S. will produce a further 15,000 million tons, giving a prospective supply of at least 30,000 million tons. This should be more than enough to last for over a century at the present rate of consumption. The analysis of the Standard Oil Co. concludes with the following encouraging comment: "Whenever it becomes possible to permit our system, which has been so successful in the past, once more to function freely (after the removal of war restrictions) we may confidently expect our oil industry to provide the base of cheap fuel on which this country has built its high living standards".—*The Motor Ship*, Vol. XXV, No. 296, September, 1944, p. 173.

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