A SCIENTIFIC INVESTIGATION INTO THE POSSIBILITIES OF GAS-TURBINES.

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A prophecy expressed frequently in engineering circles at the present day is that turbines actuated by hot gases, other than steam, will eventually come to the front as prime movers. The idea of employing hot gases (other than steam) to drive turbines is by no means new; but the success of the steam-turbine has recently brought the question into prominence. Although the subject is interesting and important, and although many minds seem to be considering it, there appears to be hardly any literature on the subject, except that which is found in patent records.

There is no doubt that many persons speak of the advantages of gas-turbines without duly considering the difficulties to be encountered. There are probably many others who have valuable ideas on the subject, supported in some cases by experimental data, but who are apt to let their thoughts run in a groove and to consider (rightly or wrongly) that the only possible solution of the gas-turbine problem lies in the particular direction in which they are working.

This Paper is written with the object of expressing and comparing as concisely as possible the advantages and possibilities of gas-turbines worked on different cycles, and the difficulties to be overcome to make these turbines a success. A further and more important object is to draw opinions from other engineers who have studied the question, and especially from those who have conducted experiments. If these objects be attained, even in an imperfect manner, the author believes that a foundation of knowledge will be obtained and placed on record, which will be of considerable use to engineers who may be endeavouring or about to endeavour to produce practical machines.

Carnot's formula for the efficiency of an ideal heat-engine

$$E = \frac{T_1 - T_2}{T_1}$$

is well known, but its real meaning is sometimes forgotten; and it may not be out of place here to put in a reminder that, in Carnot's cycle, *all* the heat is put in at temperature T_1 and *all* the heat withdrawn at temperature T_2. An increase in the range of temperature does not necessarily cause a thermodynamic gain, and it is possible largely to increase the range of temperature (as for example by superheating steam before use in a steam-engine) without thermodynamically increasing the efficiency by more than a small percentage.

The greatest possible efficiency of a gas-engine (reciprocating or turbine) working on Carnot's cycle between the limits of temperature 1,600° C. (2,912° F.) and 17° C., will be found to be:—

$$\frac{(1600 + 273) - (17 + 273)}{1600 + 273} = 0 \cdot 85.$$

If the gas-engine be an explosion motor with compression to 60 lbs. per square inch above atmosphere, combustion at constant volume, and expansion to atmospheric pressure, the greatest possible efficiency between the same limits of temperature is only 0·50; and, if the engine work on the ordinary Otto cycle with the same compression and between the same limits of temperature, the greatest possible efficiency is only 0·37.

Efforts must therefore be made not so much to get the maximum and minimum temperatures respectively as high and as low as possible, but to get the mean temperature at which heat is given to the gas and the mean temperature at which heat is withdrawn from it respectively as high and as low as possible. Of these two temperatures the lower one is usually by far the more important. An ideal gas-engine working on Carnot's cycle between the limits of temperature 2,000° C. (3,632° F.) absolute and 300° C. (572° F.) absolute will lose as much by an increase of 100° C. to the lower temperature as it will by a decrease of 500° C. from the higher temperature.

Coming now to discuss more particularly gas-turbines, there are four cycles on which it seems to the author that these could be worked with the possibility of good results. Two of these are what Mr. Dugald Clerk designates Type 2 and Type 3.* The author will call them respectively Cycle I and Cycle II.

It has not been considered worth while to discuss the Carnot cycle at length, but a few remarks are made about it towards the end of the Paper (page 1102).

A pressure-volume diagram of an engine working on Cycle I is shown in Fig. 1 (page 1064), and an entropy-temperature diagram in Fig. 2.

The working fluid is compressed adiabatically from A to B. Heat is then supplied by combustion at constant pressure from B to C; the gas expands adiabatically from C to D, and the fluid is then cooled at constant pressure from D to A. Reciprocating gas-engines have been worked on this cycle by Brayton and others, but have never come into common use. (The Diesel engine may be considered to belong to this class, although no decided constant pressure line is discernible on indicator diagrams taken from the engine.) One great difficulty that has been experienced in working reciprocating engines on this cycle is that of getting complete combustion during the period B C without the charge occasionally

* "The Gas and Oil Engine," by Dugald Clerk (Longmans & Co.), Chapter III.

firing back. If the air and fuel are brought into contact only on entering the cylinder, it is difficult to get good combustion during the period B C. If, on the other hand, the air and fuel are previously mixed together, it is difficult to prevent occasional firing back. Of course the chamber in which the air and fuel are mixed

FIG. 1.—*Pressure-Volume Diagram.*

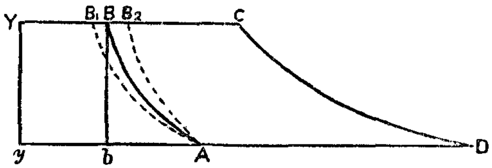
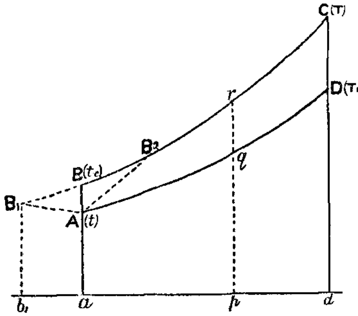


FIG. 2.—*Entropy-Temperature Diagram.*



may be made strong enough to stand explosions; but any back firing upsets the regular working of the engine and is otherwise objectionable.

It has been proposed for gas-turbines to cause air and fuel to unite in a nozzle which thereafter diverges, the idea being that the air and fuel will combine on meeting each other, and the hot products of combustion will then acquire a high velocity in the divergent nozzle with which velocity they will enter the turbine buckets. The results of a trial of such a scheme would be interesting. The author doubts if the combustion would be quick enough to give a good efficiency. If however a combustion chamber of ample size were provided in which the burning gases could rest a short interval before passing to the turbine, better results could, in the author's opinion, be expected. The air and fuel would be separately pumped into the chamber from which the products of combustion would flow continuously and uniformly by one or more passages into the turbine.

At any rate the difficulties should not be as great with turbines working on this cycle as with reciprocating engines, as the latter have to receive the hot gases intermittently, while the turbine receives a continuous flow. This is an important point as regards controlling the flame. With an engine of the Brayton type the fuel has to be ignited in the cylinder for every working stroke, and the supply of gas to the flame has to be cut off for every working stroke. With a turbine the fuel and air could be supplied at a constant velocity to the flame and a steady flame maintained without interruptions. This is important because, if a mixture of air and fuel be always supplied to the flame with a velocity greater than the velocity of propagation of the flame, there can of course be no firing back, and this result can be obtained without the use of a wire-gauze screen. The maintaining of this velocity of supply to the flame above the required minimum when starting and stopping the motor, and when running at low powers, is of course a problem to be considered, and some consideration is given to it later on (pages 1104 and 1105). The strength of the mixture of air and fuel should be kept constant. The power of the turbine can be varied by other means, which will be referred to later (pages 1104–5). It must be noted that if the air and fuel are compressed adiabatically to a sufficient extent, which depends on the nature of the fuel, combustion will occur immediately the two are brought into contact with each other. It is therefore necessary in such cases to keep the air and fuel apart until the instant when combustion is desired. It must also be noted that with a turbine there will be no hot waste-gases mixed with the fresh air and gas to be compressed.

This cycle allows of a fairly high ideal efficiency being obtained with a moderate maximum temperature. Now a moderate maximum temperature is of the utmost importance in the case of a turbine of the Parsons type. A Parsons turbine with steel blades could probably be designed without any great difficulty to stand a temperature of about 700° C. (1,292° F.) without any water jacketing or cooling devices of any sort (except for the bearings). With temperatures above this, the blades would need to be cooled. This would necessitate a radical alteration in design. The question of designing a turbine to stand high temperatures will be considered later on. It is only desired here to point out that great difficulties with a certain class of turbine are avoided by keeping the maximum temperature moderate. The cycle under consideration may therefore have great advantages for turbines.

It had better be stated here that the author has made several assumptions with regard to the working fluid or fluids. These assumptions are as follows:—

(1.) That the specific heats of gases dealt with are constant at all temperatures and pressures, and are as follows:—

Specific heat at constant pressure or $Kp = 0 \cdot 238$.

Specific heat at constant volume or $Kv = 0 \cdot 17$.

(2.) That weight per cubic foot of gases dealt with $= 0 \cdot 0777$ lbs. at a pressure of 15 lbs. per square inch absolute and a temperature of 17° C.

(3.) That $\frac{PV}{T}$ = a constant for all pressures and temperatures.

(4.) That PV = a constant for isothermal expansion and compression at all temperatures and pressures.

(5.) That combustion produces no change of volume except that due to change of temperature.

Some of these assumptions will probably be appreciably inaccurate in certain cases; but it seemed advisable to sacrifice something for simplicity and uniformity. As regards the variability of the specific heats, it has been thought better to assume constancy until more knowledge on the subject has been obtained and a scale of change (if any) has been agreed upon.

Pressures have been reckoned in lbs. per square inch, and temperatures have generally been reckoned on the Centigrade scale, although for convenience the corresponding readings on the

Fahrenheit scale have also been given. The numbers on the diagrams representing pressure and temperature are all representative of absolute pressures in lbs. per square inch, and temperatures on the absolute Centigrade scale.

Referring to Fig. 2 (page 1064), the heat absorbed by the fluid is represented in this figure by the area aBCd, and the heat abstracted or discarded by the area aADd. The heat converted into work is represented by the area ABCD, and consequently, if E represents the ideal efficiency of an engine working on this cycle,

$$\text{E} = \frac{area\ ABCD}{area\ aBCd}.$$

Now, as it can be proved * that $\frac{AB}{aB} = \frac{DC}{dC} = \frac{qr}{pr}$

where pqr is any ordinate cutting the lines ad, AD, and BC, which are all constant-pressure lines,

$$\text{therefore E} = \frac{AB}{aB} = \frac{DC}{dC}. \quad (1)$$

Let t represent the temperature before compression.

Let t_c represent the temperature at the end of compression.

Let T represent the temperature at the end of combustion.

Let T_1 represent the temperature at the end of adiabatic expansion.

Then, from equation (1) and referring to Fig. 2,

$$\text{E} = \frac{t_c - t}{t_c} = \frac{T - T_1}{T}.$$

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* Since all vertical lines represent adiabatic expansion, therefore, by the laws of adiabatic expansion,

$$\frac{\text{temp. at } A}{\text{temp. at } B} = \left[\frac{\text{press. at } A}{\text{press. at } B}\right]^{\frac{\gamma-1}{\gamma}} \qquad \text{where } \gamma = \frac{Kp}{Kv}$$

Similarly $$\frac{\text{temp. at } q}{\text{temp. at } r} = \left[\frac{\text{press. at } q}{\text{press. at } r}\right]^{\frac{\gamma-1}{\gamma}}$$

But press. at A = press. at q, since AqD is a constant-pressure line; and press. at B = press. at r, since BrC is a constant-pressure line,

therefore $$\frac{\text{temp. at } A}{\text{temp. at } B} = \frac{\text{temp. at } q}{\text{temp. at } r}$$

therefore $$\frac{AB}{aB} = \frac{DC}{dC} = \frac{qr}{pr}.$$

This can be proved quite well without an entropy-temperature diagram.* The diagram, however, shows the efficiency better.

It is important to consider the amount of negative work done and the ratio of this to the total or gross work. The negative work is the work of compressing the gas and delivering it in its compressed state. It is true that with some engines there is no work of delivery. In a reciprocating gas-engine in which the gas is compressed in the motor cylinder, the only negative work (ideally) is that of compressing the charge; and, even when a separate cylinder is used for the compression, the work of delivering might be avoided. With a turbine, however, the fluid cannot be compressed in the motor; and, whatever arrangement is adopted, the compressed fluid will have to be delivered after compression. The author has therefore considered it better in all cases to include in the negative work the amount required to deliver the compressed gas. The motor proper of course gets the benefit of this work.

In Fig. 1 (page 1064) the work to compress the gas is represented by the area A*b*B, and the work to deliver it in a compressed state by the area *y*YB*b*. The total negative work is therefore represented by the area *y*YBA. The gross work of the motor is represented by the area *y*YCD, of which the part *y*YB*b* represents the work done before expansion, and the part *b*BCD the work done during expansion. By deducting the negative work from the gross work the net work is obtained; this is represented by the area ABCD. This net work is the same as that represented on the entropy-temperature diagram, Fig. 2 (page 1064) by the area ABCD.

Cycle I, Case 1.

If the gas is required to be used in a Parsons turbine without cooling arrangements the maximum temperature must not exceed 700° C. (1,292° F.). A case with this maximum temperature will now be considered :—

* *See* "The Gas and Oil Engine," by Dugald Clerk, pages 46–48.

In all cases—

Let t and p represent respectively abs. temp. C. and abs. press. (lbs. per sq. in.) before compression.

Let t_c and p represent respectively abs. temp. C. and abs. press. (lbs. per sq. in.) after compression.

Let T and P represent respectively abs. temp. C. and abs. press. (lbs. per sq. in.) after combustion.

Let T_1 and P_1 represent respectively abs. temp. C. and abs. press. (lbs. per sq. in.) after expansion to atmospheric pressure.

Let v represent one cu. ft. of the fluid at temperature t and pressure p.

Let v_c, V and V_1 represent the volume of the same at t_c, p_c; T, P, and T_1, P_1 respectively.

Suppose that in all cases $t = 17°$ C. (290° abs. C.) and the corresponding pressure = 15 lbs. abs. First, by compressing to 42 lbs. abs.: t_c will then be 389° abs. C. This compression is shown by the line AB on the pressure volume diagram Fig. 3 (page 1070), and on the entropy-temperature diagram, Fig. 4.

Let heat be supplied and the gas expand at constant pressure along the line B C till the temperature is 973° abs. C. Let the gas expand adiabatically along the line C D till the pressure falls to 15 lbs. abs. The fluid is then exhausted into atmosphere, and as the new charge is taken at the same pressure and at temperature t, it can be assumed that the discharged gas is cooled at constant pressure and used over again. Both diagrams can therefore be completed by the constant-pressure line D A.

The heat absorbed by the fluid is represented by the area aBCd in Fig. 4, and the heat rejected by the area aADd. The heat converted into work is represented by the area ABCD and

$$E = \frac{area\ ABCD}{area\ aBCd} = \frac{t_c - t}{t_c}$$

$$= \frac{389 - 290}{389}$$

$$= \frac{99}{389}$$

$$= 0{\cdot}25.$$

The negative work is represented in Fig. 3 by the area yYBA, the gross work by the area yYCD, and the net work by the area ABCD,—

therefore $$\frac{\textit{Negative work}}{\textit{gross work}} = \frac{\textit{area } yYBA}{\textit{area } yYCD} = 0 \cdot 4.$$

The expansion line is carried right down to atmosphere. It should be possible in practice without difficulty to do this very

FIG. 3.

Cycle I, *Case* 1. *Pressure-Volume Diagram.*

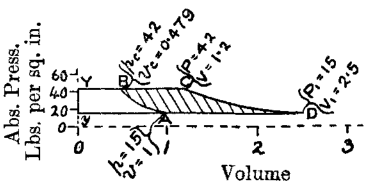
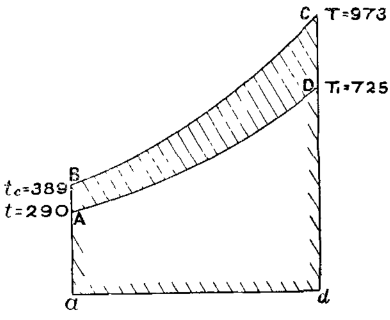


FIG. 4.

Cycle I, *Case* 1. *Entropy-Temperature Diagram.*



nearly in a turbine, although the volume at D is $2\frac{1}{2}$ times the volume at A. In dealing with large volumes and small pressures there is an immense difference between turbines and reciprocating engines. Reciprocating engines require large cylinders. These large cylinders, besides being objectionable on account of bulk and cost, necessitate great frictional losses. The low pressure dealt with is of little import as regards friction, which will be nearly the same whether the pressure is 13 lbs. below atmosphere or 13 lbs. above atmosphere. With a turbine, however, the large volume of the fluid does not necessitate such a bulky machine. Moreover in a turbine the friction depends on the pressure. With high pressures the friction is great, with low pressures very small. (In marine propulsion by steam-turbines it is not considered worth while uncoupling the reversing turbines when the vessel is going ahead. These turbines are allowed to rotate (above their normal speed) in the low pressure which exists at the exhaust ends of the main low-pressure turbines.

CYCLE I, CASE 2.

700° C. (1,292° F.) must not, however, be considered as the limiting temperature for gas-turbines. Much higher temperature can be employed if water cooling or other cooling arrangements be used. Mr. Parsons has circulated steam for heating purposes through passages formed in the rings supporting the fixed blades of his radial-flow steam-turbines.* Water could as easily be circulated, and there should be no great difficulty in passing the water also through the rings supporting the moving blades.

It has been proposed by Mr. Parsons and others to circulate water or other cooling fluid through the actual blades of a turbine, these being formed hollow. It has also been proposed to keep the blades of a single-wheel turbine cool by causing the actuating fluid to act only at one point of the circumference of the wheel, while a cooling fluid is projected on to the blades at another point.

By the employment of cooling devices a turbine might possibly be made to stand a temperature of 1,500° C. (2,732° F.) or even 2,000° C. (3,632° F.). 2,000° C. is a very high temperature, and there would be great difficulty in devising and constructing cooling arrangements which would keep the blades in good working order when acted on by gas at a temperature approaching this. Let it be assumed,

* "The Steam Turbine," by R. M. Neilson (Longmans and Co.), pages 43–45.

however, that 2,000° C. is allowable for the maximum temperature; then, if the same compression is kept as in Case 1, the ideal pressure-volume and entropy-temperature diagrams will be as shown in

Fig. 5.—*Cycle* I, *Case* 2. *Pressure-Volume Diagram.*

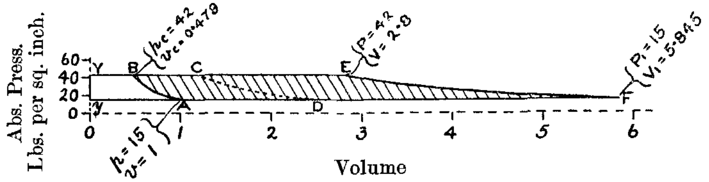
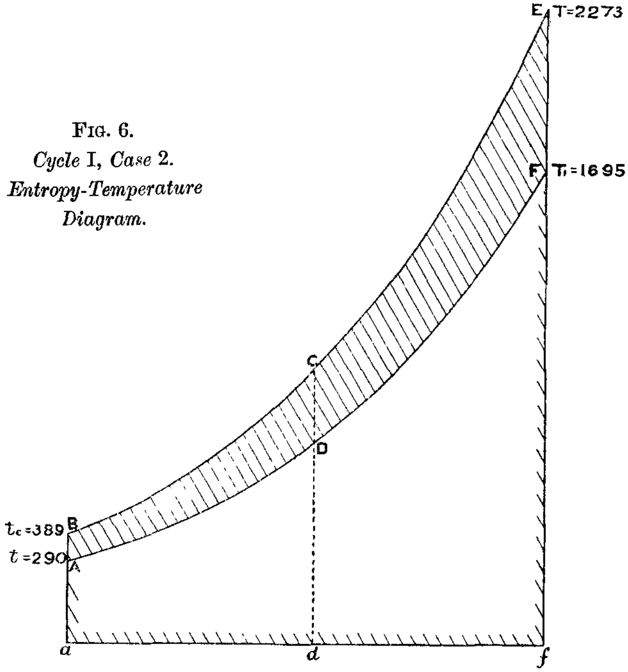


Fig. 6.
Cycle I, *Case* 2.
Entropy-Temperature Diagram.



Figs. 5 and 6. In these Figs. the line C D has been reproduced from Figs. 3 and 4 (page 1070), and is shown in dotted lines in order that the two cases may be readily compared.

Referring to Fig. 6 the heat absorbed by the fluid is represented by the area aBEf, the heat rejected by the area aAFf, and the heat converted into work by the area ABEF. Therefore

$$E = \frac{area\ ABEF}{area\ aBEf} = \frac{t_c - t}{t_c}$$
$$= \frac{389 - 290}{389} = 0 \cdot 25.$$

The increase in the maximum temperature has therefore added nothing to the efficiency, and this will always be the case if the initial temperature and pressure are unchanged and compression is made to the same amount. That is to say, as long as the constant pressure lines are started from the same points, A and B, they can be extended any distance to the right and connected by any adiabatic line; E will remain unchanged. In Fig. 6 the additional area dCEf is divided by the line DF in the same ratio as the original area aBCd is divided by the line A D.

The negative work (in Case 2) is represented in Fig. 5 by the area yYBA; it is the same as in the last case. The gross work is represented by the area yYEF, and the net work by the area ABEF,

$$\text{Therefore } \frac{Neg.\ work}{Gross\ work} = \frac{area\ yYBA}{area\ yYEF} = 0 \cdot 171.$$

The ratio of negative work to gross work has been therefore very considerably diminished.

Cycle I, Case 3.

In Case 1 it was necessary to have a low compression because a high compression with a maximum temperature of only 700° C. (1,292° F.) would have given an impractically high value to the ratio of negative work to gross work. In fact this ratio was high even with the low compression adopted.

With the maximum temperature raised to 2,000° C. (3,632° F.), however, a much higher compression can be adopted. Suppose a compression to 300 lbs. per square inch absolute is adopted. This will make t_c 682·5° abs. C. (1,260·5° F.). The pressure-volume and entropy-temperature diagrams will then be as shown in Figs. 7 and 8 (page 1074).

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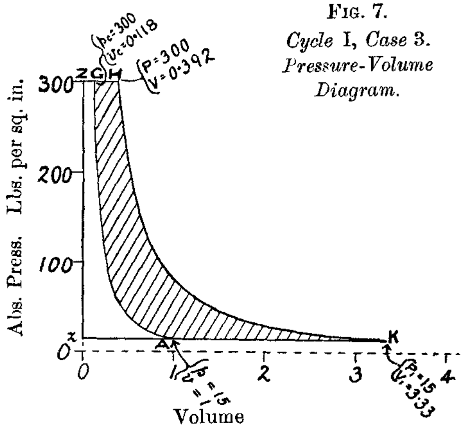
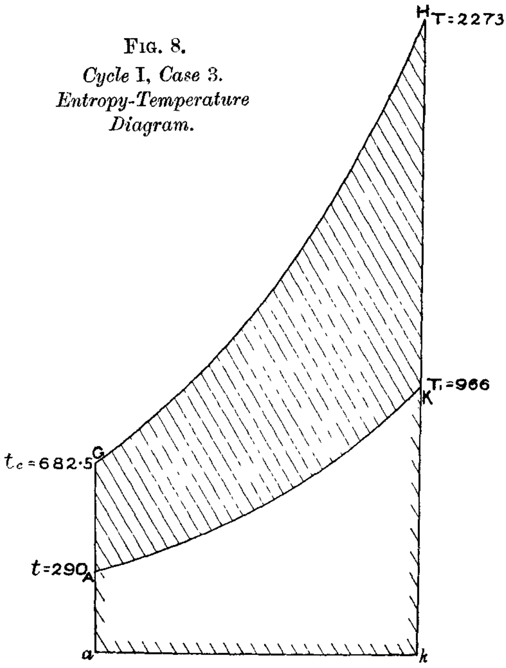


Fig. 7.
Cycle I, *Case* 3.
Pressure-Volume Diagram.

Fig. 8.
Cycle I, *Case* 3.
Entropy-Temperature Diagram.



Referring to Fig. 8 it is seen that

$$E = \frac{area\ AGHK}{area\ aGHk}$$

$$= \frac{t_c - t}{t_c} = \frac{682 \cdot 5 - 290}{682 \cdot 5}$$

$$= 0 \cdot 58$$

which is much better than (more than double) that in Cases 1 and 2. There is however the inconvenience of a high compression, and compared with Case 1 more heat is likely to be lost through radiation owing to the higher average temperature. This question of radiation will be more or less important according to the type of turbine.

The negative work is represented in Fig. 7 by the area *z*ZGA, the gross work by the area *z*ZHK, and the net work by the area AGHK.

$$\text{Therefore } \frac{neg.\ work}{gross\ work} = \frac{area\ zZGA}{zZHK} = 0 \cdot 3.$$

Cycle I, Case 4.

It will be interesting to find what efficiency can be obtained with a maximum temperature of 2,000° C. (3,632° F.) by increasing the compression till the ratio of negative work to gross work is 0·4—the same as in Case 1. This ratio will be attained when $t_c = 909°$ abs. C. (1,668° F.), which corresponds to a pressure of 818 lbs. abs. Then

$$E = \frac{909 - 290}{909} = 0 \cdot 68.$$

The pressure-volume and entropy-temperature diagrams for this case are given in Figs. 9 and 10 (page 1076).

The line BC is shown dotted on Fig. 10 to allow Case 4 to be compared with Case 1.

The sharp corner at M would likely be rounded off in practice. This would reduce the efficiency slightly. It would also however reduce the maximum temperature, and for this reason it might be advantageous in some cases to round off the corner intentionally.

In every case it has been assumed that the compression is adiabatic; it is usually important that it should be at least nearly

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so. If, for example, in Figs. 1 and 2 (page 1064) the compression, instead of being along the adiabatic AB had been along the line AB_1, which is below the adiabatic line, that is, if heat had been

Fig. 9.

Cycle I, *Case* 4.

Pressure-Volume Diagram.

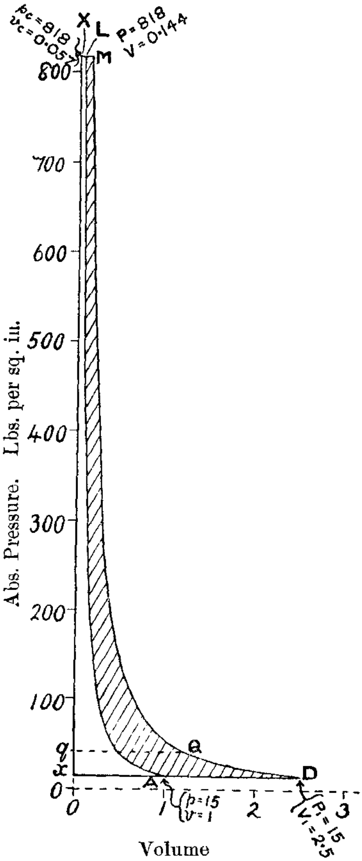
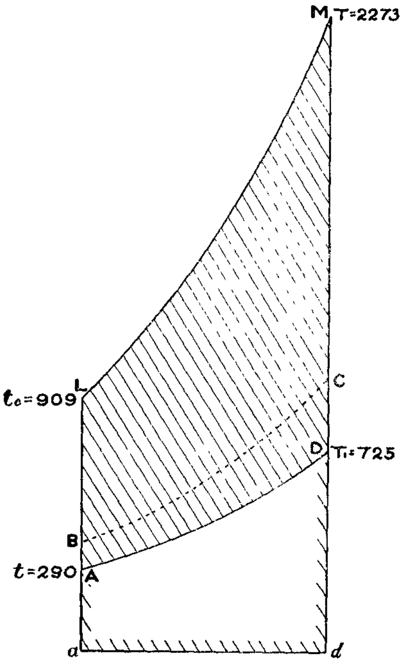


Fig. 10.

Cycle I, *Case* 4.

Entropy-Temperature Diagram.



allowed to escape during the compression, the heat absorbed by the fluid for the same value of T would have been increased by the area b_1B_1Ba in Fig. 2, while the heat converted into work would have been increased only by the relatively small area AB_1B. E would therefore have been reduced.

If on the other hand the compression had been along the line AB_2, which is above the adiabatic, that is, if heat had been put into the fluid during compression, the heat absorbed and the heat converted into work would both have been reduced by the same amount, namely, the area ABB_2. E would therefore obviously be reduced in this case also, assuming that the heat put into the fluid during compression is obtained by the combustion of fuel.

If, however, the heat put into the fluid during compression is obtained for nothing—if, for example, it is heat that would otherwise be radiated away or carried away by convexion—the effect on E is not obvious.

A compression along the line AB_2, Figs. 1 and 2, will give a higher value to E than a compression along the line AB, if the heat absorbed during the compression AB_2 is got for nothing, and if the two cases are otherwise the same; but a compression along the line AB_2 produces a higher ratio of negative work to gross work. This will be clear from Fig. 1. Now with this ratio of negative work to gross work, a still higher efficiency could be obtained by keeping the compression adiabatic and continuing it further. A hot compression, such as along the line AB_2, when the heat is got for nothing, may be advantageous in a few cases, viz., when T_1 is low compared with t_c; but generally such a compression will be harmful.

It is, in general, disadvantageous to heat the air or fuel before compression, no matter what be the source of heat.

If gas is allowed to enter a water-cooled turbine at a high temperature such as 2,000° C. (3,632° F.), there will necessarily be a great amount of heat carried away by the water. In a reciprocating engine the metal surface with which the gas comes into contact is very small compared with that in a multiple expansion turbine; and in a reciprocating engine the bulk of the gas may expand and fall from its maximum temperature to the temperature at exhaust without ever coming near a metal surface. In a multiple expansion turbine, on the other hand, every particle of gas must practically slide along a metal surface immediately it comes to the first ring of blades. With turbines employing gas which enters the turbine casing at such a temperature, the heat lost through the walls and carried away by the water must necessarily be very great indeed. It is true that the metal surface in contact with the gas can be allowed to be at a much higher temperature than the inside of the cylinder walls of a reciprocating engine; but, in spite of this, the heat lost through the walls and carried away by the cooling water (or other cooling medium) will probably be much greater with a turbine actuated by gas entering the turbine casing at about 2,000° C. than in a reciprocating engine in which the maximum temperature is 2,000° C. This loss of heat will cause the actual work done by the engine to be very much below the ideal. This is not only important in itself, but, as will be explained subsequently (pages 1081 and 1082), it prevents useful employment of a high ratio of negative work to gross work. The question of utilising this lost heat will be discussed later on (pages 1088 to 1094).

CYCLE I, CASE 3A.

Instead of employing cooling arrangements for the metal, some or all of the available heat energy of the gas can be converted into kinetic energy before causing it to act on the turbine, so that the latter is not exposed to an unduly high temperature. This can be done by allowing the gas when at the maximum temperature to expand in a divergent nozzle till its temperature falls to a degree that the turbine can stand. More than one nozzle can be employed, but, to reduce the radiation losses, the nozzles should be large and few in number.

Suppose that the gas is compressed adiabatically to 300 lbs. absolute, and then is heated at constant pressure to a temperature of 2,273° abs. C. (4,132° F.), as in Case 3. If now the gas be allowed to expand in a suitable nozzle adiabatic expansion can be obtained; and if this be continued till the pressure falls to 15 lbs. abs. the temperature will be 966 abs. C. (693° C.). This is just below the temperature which was fixed on as a maximum for a turbine without artificial cooling. The entropy-temperature diagram will be the same as in Case 3, Fig. 8 (page 1074), and E will therefore be the same, namely 0·58. The ratio of negative work to gross work will also be the same as in Case 3, namely 0·3.

Referring to the pressure-volume diagram for Cycle I, Case 3, Fig. 7 (page 1074), the area zZHK represents the kinetic energy of the gas leaving the nozzle, which kinetic energy equals 33,840 foot-lbs. This is for a quantity of gas which measures 1 cubic foot at A. The velocity is 5,290 feet per second.

For the sake of comparison it may be advantageous to mention the velocities of the steam jets employed in De Laval steam-turbines. If saturated steam at 50 lbs. absolute pressure is expanded adiabatically to a pressure of 0·6 lbs. abs., which corresponds to a temperature of 85° F., and its heat energy turned into kinetic energy, the velocity acquired works out at 3,690 feet per second. If saturated steam at 300 lbs. abs. pressure were treated similarly, the velocity would be 4,380 feet per second. The velocities actually obtained in practice must be somewhat less than these figures owing to friction in the nozzles.

To get the best results from a fluid velocity such as 5,290 feet per second would require with a single turbine wheel a vane speed which cannot be obtained at present for want of a sufficiently strong and light material—the stresses produced by centrifugal force are too great. This difficulty is experienced with De Laval turbines. The obvious way out of the difficulty is to employ several wheels in series, the gas passing through the several wheels with diminishing velocity, but with nearly constant pressure. This has been done in steam turbines.

With the same object of reducing the vane speed, a device has been proposed whereby the nozzles are mounted on a wheel which rotates in the opposite direction to the wheel carrying the vanes. If the two wheels rotate at the same speed (in opposite directions) this speed will be half of that of the single wheel if the nozzles were stationary. The centrifugal force is therefore only one-fourth of what it would otherwise be.

The frictional losses in the nozzles of a gas-turbine will probably be less than those in the nozzles of a steam-turbine for the same velocity of exit from the nozzle.

Cycle I, Case 4A.

If one tries to work to the same entropy-temperature diagram as in Case 4, Fig. 10 (page 1076), but employs a divergent nozzle, as in Case 3A, to reduce the maximum temperature to 700° C., so that the gas can be used in a turbine without cooling arrangements, T_1 in this case will be 725° abs. C. (452° C.). It is not necessary therefore to perform all the adiabatic expansion in a divergent nozzle, but a portion of it can be performed in the turbine. If the fluid is expanded in the nozzle only till its temperature falls to 700° C. (1,292° F.), the pressure will then be 42 lbs. abs.; so that 27 lbs. can be dropped in the turbine.

Referring to the pressure volume diagram for Cycle I, Case 4, Fig. 9 (page 1076), the line qQ is drawn to represent the pressure at which the gas leaves the nozzle. The kinetic energy of the gas leaving the nozzle is represented by the area XMQq. It can be ascertained that this amounts to 33,660 foot-lbs. (for one cubic foot of gas measured at A), and the velocity works out at 5,280 feet per second. E will be the same as in Case 4, and so will the ratio of negative work to gross work.

It seems to the author that an engine working on this cycle, according to Case 3A or Case 4A or between these, has good prospects. The ideal efficiency is high—from 0·58 to 0·68. How near one could approach this efficiency in practice would depend of course both on the losses in the motor proper and on the losses in the pump.

The losses in the motor proper may be taken to include the losses in the combustion chamber, if such is employed, and in the nozzles. The motor losses will then consist of:—

(1). Loss of heat by radiation and conduction.

(2). Fluid friction.

(3). Friction in turbine bearings.

(4). Loss due to incomplete expansion.

The first loss will be large, but should be less than in reciprocating engines, owing to the higher velocities employed and to the higher temperatures allowable in the metal.

The second loss will be considerable, but much less than in turbines using saturated steam. It has been found by experiment that hot dry air causes much less friction than wet steam. (The steam is always wet in a De Laval turbine casing, unless it enters the nozzles with a large amount of superheat.)

The third loss will be trifling and the fourth loss should be moderate. The discharge of heat with the exhaust gases is here only considered as a loss in so far as it exceeds that of an ideal engine.

It is difficult to estimate the pump losses. Rotary compressors on the turbine principle seem to have been employed only up to about 80 lbs. pressure. Whether or not they are suitable for high pressures is a point which it is very desirable to ascertain. One would be inclined to believe that the fluid frictional losses with such machines would be very great if attempts were made to obtain high pressures. It by no means follows however that a fairly efficient rotary air-compressor cannot be devised.

A reciprocating compressor always has the disadvantage that the air when drawn in becomes heated by contact with the hot metal surfaces before compression commences. This evil is reduced by compounding. It is an evil which occurs to a serious extent with reciprocating gas-engines working on the Otto cycle.

With a reciprocating compressor it will be difficult to avoid the necessity of jacketing the cylinder if high compressions are employed. This will bring the compression curve below the adiabatic and reduce the efficiency as before explained.

In any case, whatever be the nature of the pump, there is bound to be a certain amount of heat passed through the walls of the pump cylinders or casing. If this loss be made up by friction or impact within the pump, the compression may be along an adiabatic curve, but the loss will still have to be considered.

The ratio of negative work to gross work (in the particular cases here referred to) is somewhat high—0·3 to 0·4. In the case of a turbine one need not fear the increase in the bulk of the engine due to this high ratio; for the bulk of the turbine will probably be very small for the power. Frictional and other losses become, however, of much greater importance when the ratio is high. To show this forcibly, consider an extreme case. Suppose that the ratio of negative work to gross work in an ideal engine is 0·5, or, in simpler language, suppose that the pump requires half the gross power of the machine, there being no friction. If now the machine is not ideal, and if the mechanical efficiency of the pump is only $\frac{2}{3}$ and that of the motor proper only $\frac{3}{4}$, no useful work whatever will be got out of the machine—all the work will be absorbed by friction. For, if the power of the motor proper, including that spent on friction, is 100, the pump will require 50, and as its efficiency is $\frac{2}{3}$, it will take 75. This is exactly what the motor will give out after deducting friction. There will therefore be no power got out of the machine. When there is a high ratio of negative work to gross work, success will therefore be dependent largely on the efficiency of the pump. Unless the pump is at least fairly efficient, success cannot be expected. In the Diesel engine the bulk of the air is compressed to about 500 lbs. per square inch, and the air which carries the oil into the cylinder is compressed from 100 lbs. to 200 lbs. higher.* It would be interesting to know with what efficiency the air is compressed in the Diesel engine.

Otto cycle reciprocating engines having ideal efficiencies of 0·4 to 0·45 have given practical efficiencies of half that amount. By practical efficiency is meant ratio of B.H.P. to thermal units in gas consumed, calculated on the higher calorific value. When the ideal efficiency is increased above 0·45, the ratio of practical efficiency to ideal efficiency usually falls below 0·5—the greater the ideal efficiency, the greater are the losses. With a turbine the losses ought also to increase when the ideal efficiency is increased, but whether to the same extent as with an Otto engine it is difficult to say. When considering high compressions, it is well to note that the Diesel engine with a high compression and an incomplete expansion

* "The Diesel Engine," by H. Ade Clark. Proceedings, 1903, Part 3, page 395.

has given some of the highest practical efficiencies yet attained. The compression should not cause the same trouble in starting a turbine as in starting a reciprocating engine, as with a turbine it should be practicable to arrange that at every instant the gross work is greater than the negative work. With a reciprocating engine having a single cylinder working on the Otto cycle there are of course periods when the negative work exceeds the gross work.

Fig. 11.
Cycle II, *Case* 1.
Pressure-Volume Diagram.

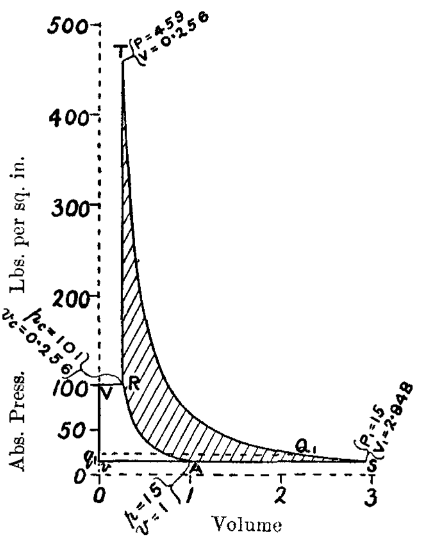
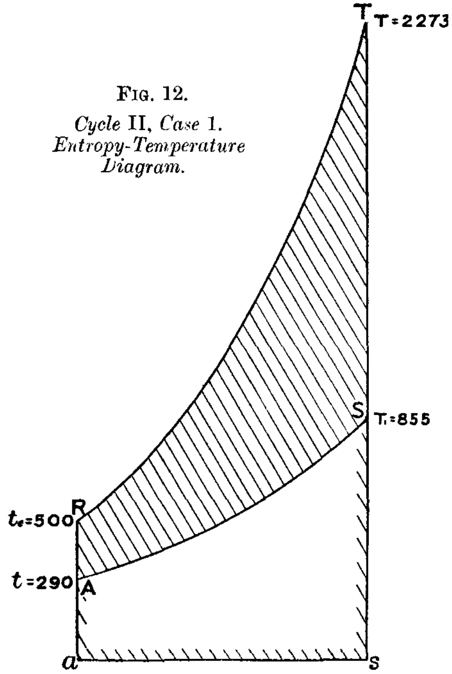


Fig. 12.
Cycle II, *Case* 1.
Entropy-Temperature Diagram.



Cycle II, Case 1.

With regard to explosion turbine engines, suppose that the fluid is compressed adiabatically to, say, 101 lbs. per sq. in. abs., that is to a temperature of 500° abs. C. (932° F.). Let it now be heated at constant volume by explosion, and let there be a mixture of such a strength that the temperature will rise to 2,000° C. (2,273° abs. C.). The pressure will then be 459 lbs. abs. If the gas is now allowed to expand adiabatically till its pressure is atmospheric (when its temperature will be 855° abs. C.), and then cooled at that pressure till it resumes its original state, the pressure-volume and entropy-temperature diagrams will be as shown in Figs. 11 and 12 (page 1083).

In Fig. 12 the heat supplied to the fluid is represented by the area aRTs, the heat rejected by the area aASs, and the heat converted into work by the area ARTS.

$$\text{Therefore E} = \frac{area\ ARTS}{area\ aRTs} = 0 \cdot 55.$$

The negative work can be compared with the gross work in Fig. 11. The ratio of negative work to gross work $= \frac{area\ vVRA}{area\ vVRTS}$ $= 0 \cdot 23$.

Cycle II, Case 1, very nearly resembles common practice to-day with reciprocating explosion engines. The expansion is, however, continued to atmospheric pressure. This as a rule is not desirable in a reciprocating engine, on account of the extra length required to be given to the engine cylinder which not only increases the loss by friction but increases the loss of heat by the expanding gas, and, if the same length of stroke is employed for drawing in the fresh charge, increases the heating of the charge before compression. The case however is very different with turbines; and there seems no good reason why with these the adiabatic expansion should not be carried practically to atmospheric pressure.

In practice the maximum pressure and the average maximum temperature throughout the gas would be less than the values here indicated, owing to radiation losses.

Cycle II, Case 1a.

The gas could not be allowed into an uncooled turbine at the maximum temperature in Cycle II, Case 1; but, if the expansion were performed wholly or nearly wholly in a divergent nozzle, the temperature of exit from the nozzle would be sufficiently low to allow of the gas entering an uncooled turbine.

For example, if the gas at the maximum temperature of 2,273° abs. C. (4,123° F.) and the maximum pressure of 459 lbs. abs. were expanded in a perfect divergent nozzle till the temperature fell to 700° C. (973° abs. C.), which was fixed on as the maximum allowable temperature in an uncooled turbine, the mean pressure on leaving the nozzle would be 23·5 lbs. abs. The kinetic energy of the gas (1 cu. foot at A) on leaving the nozzle would be represented by the area $VRTQ_1q_1$ in Fig. 11, and would amount to 20,500 foot-lbs. The mean velocity (the square root of the mean square) would be 4,120 feet per second.

On comparing Cases 1 and 1A of Cycle II by reference to the Table (page 1102) with Cases 2, 3, 3A, 4 and 4A of Cycle I, which have the same maximum temperature, it will be found that the efficiency is very much greater than Cycle I, Case 2; is nearly as great as Cycle I, Cases 3 and 3A; and is considerably below Cycle I, Cases 4 and 4A. The ratio of negative work to gross work is, however, greater than in Cycle I, Case 2, and less than in Cases 3, 3A, 4 and 4A of Cycle I.

There are two objections to the use for turbines of a cycle such as Cycle II, and these objections must be set against the advantage which turbines would possess over reciprocating explosion motors, in being able to make better use of the tail end of the pressure-volume diagram.

One objection is that explosions at constant volume have to take place intermittently, while a turbine desires a continuous supply of fluid. If the supply is not continuous the power of the turbine is less than it would otherwise be for a given size of machine; and the initial cost, the bulk and—most important—the loss by friction are greater in proportion to the power developed than they would otherwise be.

The other objection is that the fluid must leave the explosion chamber at varying pressure. This necessitates, unless special means are provided to prevent it, the fluid entering the turbine casing either at varying pressure or at varying velocity, which is of course objectionable, as the speed of rotation of the turbine cannot during the period of a cycle be made to vary correspondingly.

The second objection might be met by employing in a parallel flow turbine of the De Laval type long radial blades, and causing the nozzles to be altered in position according to the pressure, so as to direct the gas on to the outer ends of the blades at high pressures and on to the inner ends of the blades at low pressures. The difficulty could also be met by an arrangement of reciprocating engine combined with a turbine, the gas being first expanded in the reciprocating engine to a certain pressure and then passed on to the turbine to complete its expansion. If several reciprocating cylinders were employed, the first objection also would be got over, but it is true that with such a combination some of the most important advantages of the turbine would be lost. The idea is, however, in the author's opinion, worthy of consideration. Reciprocating steam-engines have been successfully combined with steam-turbines in this manner.*

Cycle II, Case 2.

An explosion-engine, in which a very high compression pressure is employed, will now be considered. If compression be carried to 818 lbs. abs. as in Cycle I, Case 4, one obtains with a maximum temperature of 2,000° C. (3,632° F.) a maximum pressure of 2,045 lbs. abs. and a very high ratio of negative work to gross work. If a much lower compression—namely 417 lbs. abs.—is adopted, this will give a temperature of compression of 750° abs. C. (1,382° F.). Working on the same cycle as in the last case and arranging the explosive mixture to give a maximum temperature of 2,000° C. (2,273° abs. C.), a maximum pressure of 1,265 lbs. abs. is obtained, and the pressure-volume and entropy-temperature diagrams will be as shown in Figs. 13 and 14 (page 1087).

Referring to Fig. 14,

$$\mathrm{E} = \frac{area\ AUWW_1}{area\ aUWw} = 0{\cdot}68.$$

* *See* Paper by Professor Rateau read before the North of England Institute of Mining and Mechanical Engineers at Newcastle-on-Tyne, 13 Dec. 1902; or Paper by the same author read at the Chicago Meeting of the Institution of Mechanical Engineers, Proceedings 1904, Part 3 (page 737).

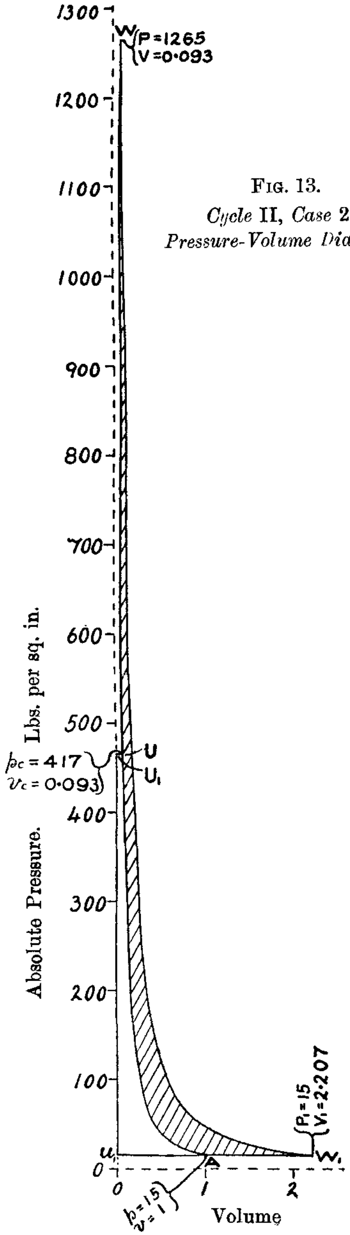


FIG. 13.
Cycle II, *Case* 2.
Pressure-Volume Diagram.

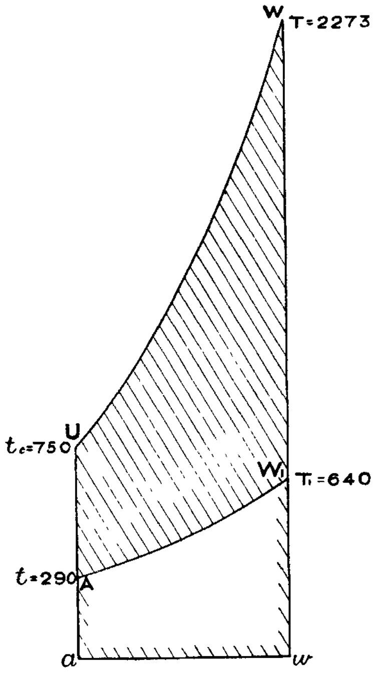


FIG. 14.
Cycle II, *Case* 2.
Entropy-Temperature Diagram.

Referring to Fig. 13,

$$\frac{Neg.\ work}{gross\ work} = \frac{area\ u_1 U_1 U A}{area\ u_1 U_1 U W W_1} = 0 \cdot 38.$$

E is the same as in Cycle I, Case 4, and the ratio of negative work to gross work is also the same. The compression is lower than in Cycle I, Case 4, but the maximum pressure is very much higher.

The excessively high maximum pressure is an objection to this case.

CYCLE II, CASE 2A.

If the expansion took place in an ideal divergent nozzle as before till the temperature fell to 700° C. (973° abs. C.), the gas would still have a pressure of 70 lbs. abs., while the mean velocity of exit from the nozzle would be 4,300 feet per second. If the gas were expanded in the nozzle down to 25 lbs. abs., the temperature would then be 741° abs. C. (1,366° F.), and the mean velocity of the gas leaving the nozzle would be 4,830 feet per second.

CYCLE III, CASE 1.

It has been proposed, when a water-jacket is employed, to utilize the heat passed into the jacket water by causing this heat to generate steam from the water. This steam could then receive further heat from the products of combustion, which would therefore be reduced in temperature, while the steam would be superheated. The steam and products of combustion could then expand adiabatically, doing work in the same or in separate turbines. The carrying out of this idea would affect the efficiency in the several cases considered of Cycle I. Cooling arrangements are not required in Cycle I, Case 1, so this case need not be further considered. In Cycle I, Case 2, let it be supposed that the combustion chamber is jacketed and that the jacket water is heated and converted into steam by heat taken from the products of combustion, which have their temperature thus lowered from

2000° C. to 700° C., that is, to the temperature at which they can safely be allowed into an uncooled turbine, the steam being superheated up to 700° C. Let this be called Cycle III, Case 1.

Referring to Fig. 6 (page 1072), the heat in the products of combustion which is converted into work is now represented by the area ABCD instead of by the area ABEF. The heat represented by the area *d*CE*f* has, however, been employed in heating water and generating and superheating steam. The fraction of this heat which is converted into work will not now be as great as in the original scheme of working. That is to say, the net work got out of the heat put into the water and steam will be less than the area DCEF. By transferring heat to the water and steam from the gas, E is therefore reduced. There must, however, in any case, as already mentioned (page 1063), be lost in practice a large amount of heat from the products of combustion when these products of combustion enter the turbine casing at a temperature such as 2,000° C., and, by adopting this combined steam and gas scheme, a much higher practical efficiency may possibly be obtained than would otherwise be possible. As the net work ideally is less than in Cycle I, Case 2, and as the negative work is not less (and may be greater by the amount of work required to pump the water into the jacket if under pressure), the ratio of negative work to gross work is increased. In Case 2 of Cycle I, the ratio of negative work to gross work is low, and it will be therefore allowable to increase this ratio.

CYCLE III, CASE 2.

Case 3 of Cycle I could be modified in the same way by reducing the temperature of the products of combustion from 2,000° C. to 700° C., and by employing the heat so given up in heating water and generating and superheating steam. The steam could be generated at 300 lbs. pressure abs. (the same pressure as the products of combustion) and superheated to 700° C. at this pressure. The steam and gas could then be expanded adiabatically in the same or in separate turbines. As in the previous case, E would be reduced, and the ratio of negative work to gross work increased. As in the previous case, the practical efficiency might also be largely increased.

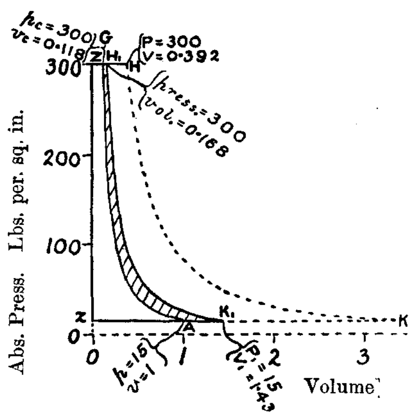
The pressure-volume and entropy-temperature diagrams for the gas in this case (called Cycle III, Case 2) are shown in Figs. 15 and 16 respectively (pages 1091–2). The gas is compressed along the line AG as in Cycle I, Case 3, till its pressure is 300 lbs. abs., and its temperature is 409·5° C. (682·5° abs. C.). It is then heated by combustion at constant pressure along the line G H as in Cycle I, Case 3, till its temperature is 2,000° C. (2,273° abs. C.). Heat is now withdrawn from the gas at constant pressure and transferred to the water and steam, the temperature of the gas falling along the line H H_1 to 700° C. (973° abs. C.) at H_1. The heat transferred from the gas to the water is represented, Fig. 16, by the area $k_1 H_1 H k$. The gas now expands adiabatically along the line H_1 K_1 till the pressure is 15 lbs. abs., when the temperature will be 140° C. (413° abs. C.) The contraction of the gas at constant pressure along the line K_1 A completes the cycle. Dotted lines have been placed on Figs. 15 and 16 to illustrate Cycle I, Case 3, where this differs from the present cycle. The two cycles can thus be compared.

Pressure-volume and entropy-temperature diagrams for the water are shown in Figs. 17 and 18 (page 1092). Referring to Fig. 18, the water is heated at a constant pressure of 300 lbs. per square inch abs. along the line *fc* from 100·6° C. (373·6° abs. C.) to 214° C. (487° abs. C.), which is the boiling point at this pressure. The water is now converted into steam, this process being represented by the line *c g*; and the steam is superheated at constant pressure, as represented by the line *g d*, till its temperature is 700° C. (973° abs. C.). The steam is then expanded adiabatically along the line *d e* till it falls to 15 lbs. absolute pressure, its temperature then being 184° C. (457° abs. C.). The steam is now exhausted and cools along the line *e h*. At *h* it is saturated, its temperature being 100·6° C. (373·6° abs. C.), and thereafter it condenses along the line *h f* and is compressed to its initial state.

Fig. 17 shows the work done by the steam in its generation, superheating and adiabatic expansion. The work done in forcing the water into the chamber at 300 lbs. pressure is not shown in Fig. 17, and is negligible in the present investigation.

The heat required to raise the water from 373·6° abs. C. to 487° abs. C., is represented in Fig. 18 by the area f_1fcc_1. The area c_1cgg_1 represents the latent heat of steam at a pressure of 300 lbs. abs. (the temperature being 487° abs. C.), and the area g_1gde_1 represents the heat required to superheat the steam from 487° abs. C. to 973° abs. C. The area f_1fhh_1 represents the latent heat of steam at a pressure of 15 lbs. abs., and the area h_1hee_1 represents the heat required to superheat this steam from 373·6° abs. C. to 457° abs. C.

Fig. 15.—*Cycle* III, *Case* 2. *Pressure-Volume Diagram.—Gas.*



Comparing this case with Case 3, of Cycle I, it is found that the total heat absorbed is the same in both cases, being represented by the area *a*GH*k* in Fig. 16. The portion of this heat which is converted into work in Case 3, Cycle I, is represented by the area AGHK, while the corresponding portion in the present case is represented by the sum of the areas AGH_1K_1, Fig. 16, and *fcgdeh*, Fig. 18. This sum is less than the area AGHK, and E in this case is only 0·33 as compared with 0·58 in Case 3 of Cycle I. The fall in the value of E is due to the relatively low efficiency of the steam portion which has an ideal efficiency of only 0·28. (For a steam-engine this is really not low.)

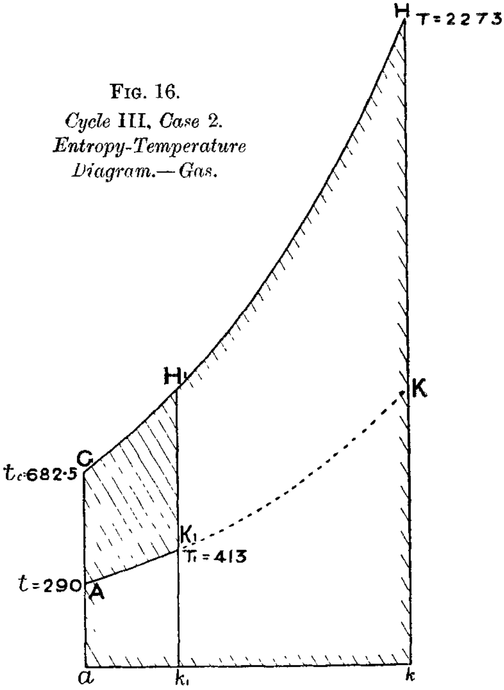


Fig. 16.
Cycle III, *Case* 2.
Entropy-Temperature Diagram.—Gas.

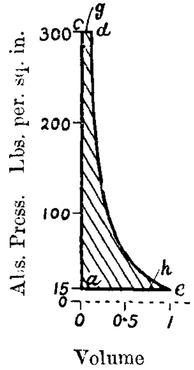
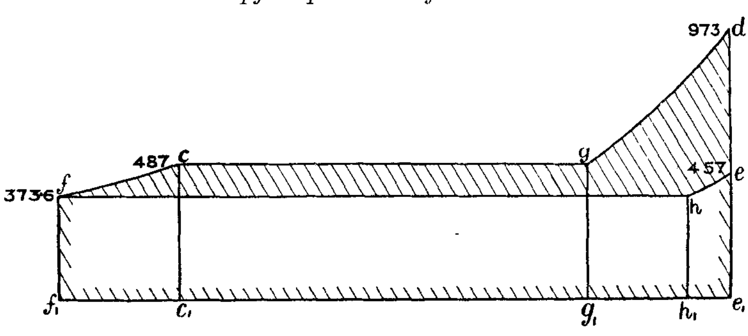


Fig. 17.
Cycle III, *Case* 2.
Pressure-Volume Diagram.—Steam.

Fig. 18.—*Cycle* III, *Case* 2.
Entropy-Temperature Diagram.—Steam.



The feed-water has been taken at a temperature corresponding to atmospheric boiling-point. It has been assumed that the steam is exhausted into the atmosphere, and is not condensed for use over again. It would therefore be necessary, in order to follow the cycle, to heat the feed-water to 100° C. It should not be difficult to approximately accomplish this by utilising the heat of the exhausting gases. By heating the feed-water still more, the efficiency could be improved; but the improvement would be slight (less than in an ordinary steam-engine) and the feed-water would have to be under pressure. As, moreover, any increase of exhaust or back pressure is a serious matter with a turbine, and as feed-water heaters must to a certain extent affect this back pressure, any prospect of gain by heating the feed-water beyond 100° C. need not be considered.

The gross work in the present case is represented, Figs. 15 and 17, by the area z Z H_1 K_1 + the area a c d e. This is less than the gross work in Cycle I, Case 3, which is represented by the area z Z H K. The negative work in Cycle I, Case 3, was represented by the area z Z G A. In the present case it is also represented by this area, neglecting the work of pumping the water into the jacket. The ratio of negative work to gross work in the present case is 0·41 as compared with 0·3 in Cycle I, Case 3. This ratio (0·41) is rather high. It will, however, probably not be so objectionable in the present case as the ratio 0·40 in Case 4 of Cycle III, as the real efficiency in practice will come nearer to the ideal in this case than in Case 4 of Cycle III. In the present case the ratio could be reduced by lowering the compression. This would reduce E.

As the mass of the water employed is not the same as the mass of the air and fuel, the scale for entropy in Fig. 18 has been made different from that in the other entropy-temperature diagrams, so that in all these diagrams, areas represent quantities of heat to the same scale. In all the pressure-volume diagrams the scales are the same except in Fig. 19 (page 1094), which will be referred to hereafter. It might be mentioned here that all the numerical results given in this Paper have been obtained by calculation and not by scaling the diagrams.

It will be seen that it has been assumed that the gas and steam expand adiabatically separate from each other. The adiabatic curve of the one is different from that of the other, as the specific heats are different; and, while the gas falls to a temperature of 140° C. (413° abs. C.), the steam falls only to 184° C. (457° abs. C.). This will be correct if the steam and gas are not mixed. It is much simpler to consider this case than to consider the case where the gases are intimately mixed. In this latter case the diagram Fig. 16 would be altered, and it could not so easily be seen where the loss of efficiency came in. In practice, however, it will probably be found convenient to mix the gases. This will alter the diagrams and the efficiency somewhat; but what has been

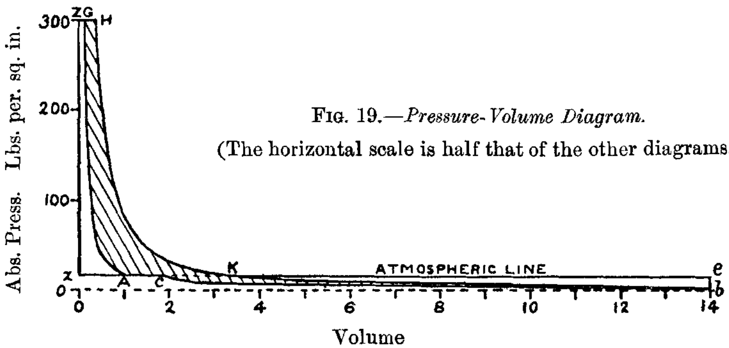


Fig. 19.—*Pressure-Volume Diagram.*
(The horizontal scale is half that of the other diagrams.)

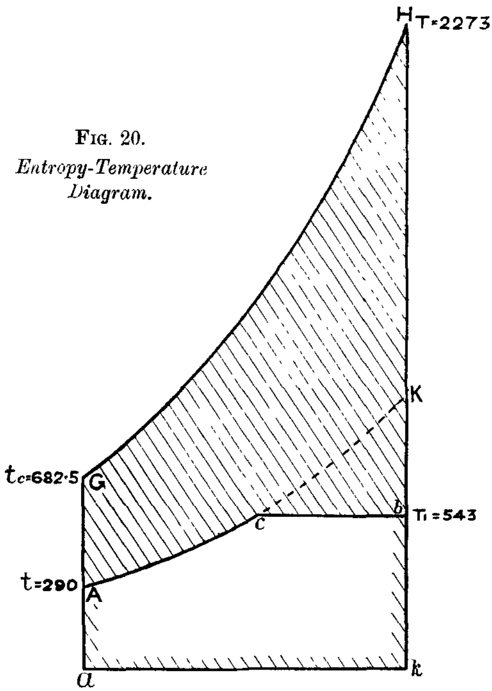
considered gives a good idea of the general effect of the employment of steam in conjunction with gas. If the steam and gas are not mixed, a condenser could be employed for the former. The steam could then be expanded to a much lower temperature and pressure, and the efficiency would be considerably raised.

Cycle II could be modified by combining steam with the gas, in the same way as Cycle I was modified. A case of this nature has not been worked out; but Case 1 of Cycle II could probably be modified in this way. Case 2 of Cycle II could not be so treated on account of the high ratio of negative work to gross work that would occur.

One might try to improve on all these cycles, by extending the adiabatic expansion line of the gas below atmosphere, instead of stopping it at atmospheric pressure. It would, of course, be necessary to compress the fluid back again to atmospheric pressure; but, if this compression were isothermal or between the isothermal and adiabatic, there would be an increase in efficiency. Carnot's cycle is in fact being approached in the lower part of the diagram.

Figs. 19 and 20 are respectively pressure-volume and entropy-temperature diagrams of Cycle I, Case 3, modified by continuing the adiabatic expansion to a pressure of 2 lbs. per square inch abs.

Fig. 20.
Entropy-Temperature Diagram.



The scale for volumes in Fig. 19 has, for convenience, been made half that of the other diagrams. K *b* represents the addition to the adiabatic line of expansion, and *b c* represents isothermal compression of the gas from 2 lbs. abs. at *b* to 15 lbs. abs. at *c*.

There should be no difficulty in a turbine in extending the expansion from K to *b*. There may be difficulty however in getting isothermal compression from b to atmospheric pressure at c. As the volume at b is 14 times the initial volume, it will be desirable to get the fluid discharged as quickly as possible. A rotary compressor will probably be best for this purpose. A compression, sufficiently near to the isothermal and sufficiently remote from the adiabatic to raise the efficiency appreciably, should be obtainable.

The temperature at b is 270° C. (543° abs. C.), and if the compression were isothermal, this would of course be the temperature all along the line $b\,c$. The gases could be passed through or around water-cooled tubes to keep down the temperature during compression. With the gas at a temperature of 543° abs. C. it would not do to spray water into it, unless sufficient water were sprayed to cool the gas below the boiling-point of the water, which is 326° abs. C. at this pressure.

If compression takes place along the isothermal line $b\,c$, a net amount of work will be gained, represented by the area Kbc. The gas will be discharged into the atmosphere at c, the volume at discharge being 1·874 of the original volume (at A). Even if the compression is not isothermal, an amount of work may be gained which will wipe out the extra losses in the machine, provide for pumping out the cooling water, and perhaps leave a margin of net gain.

In Fig. 20 the heat absorbed by the fluid is represented by the area aGHk, the heat rejected by the area aAcbk, and the heat converted into work by the area AGHbc. As the heat absorbed remains unchanged, while the heat converted into work is increased by the area Kbc, E is of course increased.

$$E = \frac{\textit{area } AGHbc}{\textit{area } AGHk}.$$

This enlarging of the diagram of course affects the ratio of negative work to gross work. Referring to Fig. 19 (page 1094),

gross work = area zZHKebc
negative work = area zZGA + area Keb
net work = area AGHbc

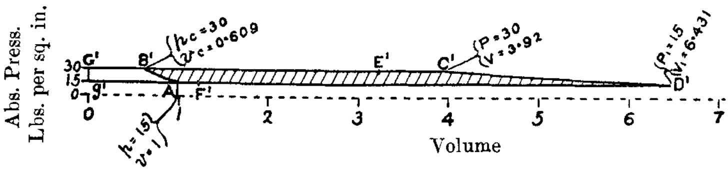
$$\frac{\textit{neg. work}}{\textit{gross work}} = \frac{\textit{area } zZGA + \textit{area } Keb}{\textit{area } zZHKebc}.$$

In the free piston explosion engines, which were at one time in fairly common use, the best known of which is the Otto and Langen, the expansion was carried to a pressure considerably below the atmosphere. The compression to atmospheric pressure which followed this must have been between the isothermal and the adiabatic.

If this continuation of the adiabatic expansion below atmospheric pressure is not found to be advisable to the extent that has just been described, it may be found advisable to a less extent. If it is found advisable in any case, it is more likely to be so in a case in which the high pressure of the gases after combustion is reduced to a low pressure in divergent nozzles, before the gas is allowed into the turbine casing, than in a case in which the whole fall of pressure takes place in the turbine casing. In the former case very high vane speeds are necessary, and the friction between the rotating parts and the fluid in the casing is an extremely important matter. The reduction of the pressure within the turbine casing from atmospheric pressure (or above that) to one-quarter or one-eighth of that amount may therefore very much reduce the frictional losses. It is true that the rotary pump, if such is employed for completing the cycle, has to deliver at atmospheric pressure, but the rotating parts of the pump can revolve at a much lower speed, and the friction will therefore be of much less consequence.

With such high-speed turbines there is another question to be considered. It has been stated in discussing Cases 3A and 4A of Cycle I, and 1A and 2A of Cycle II, that the velocity of the gases escaping from the divergent nozzles would be over 4,000 feet per second, if the heat energy converted into kinetic energy was as mentioned. The author is not however aware of any results of experiments having been published in which velocities of these amounts were obtained, when the pressure of the medium into which the divergent nozzle discharged was atmospheric. It is supposed by some that there is a maximum limit to the velocity of a gas leaving a divergent nozzle and escaping into a given medium which is at a given pressure, etc., and that this limit velocity is dependent on the pressure in the medium into which the nozzle discharges, and is less when the pressure in this medium is greater, and *vice versâ*. That is to say, it is supposed by some that, after a certain velocity of discharge has been attained, no increase in the initial temperature or

Fig. 21.—*Cycle* IV. *Pressure-Volume Diagram.*



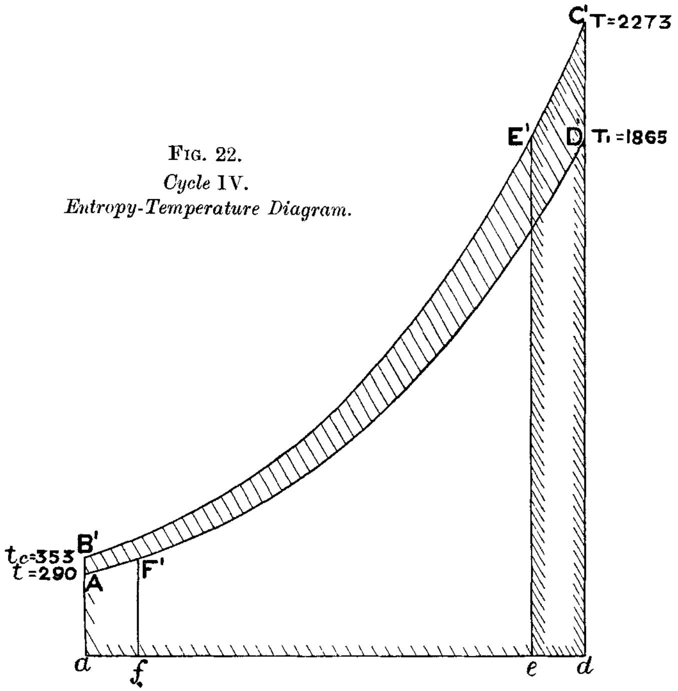


Fig. 22.
Cycle IV.
Entropy-Temperature Diagram.

pressure will increase this velocity; but a reduction of the pressure in the medium may do so. The author does not express any opinion himself on this point, but if it should be found that the reduction of the pressure inside a turbine casing below atmospheric pressure enables the heat energy of the gas to be more effectively converted into kinetic energy, this will be a further argument in favour of so reducing the pressure. Whether or not there is an advantage to be gained remains to be proved, but there is at any rate a possibility of gain by thus extending the expansion, and it is a possibility which, in the author's opinion, should not be ignored. In dealing with large volumes and small pressures there is, as already mentioned, an immense difference between turbines and reciprocating engines.

Cycle IV.

The fourth cycle which will be considered in this Paper is one in which a high ideal efficiency can be obtained with a low compression, and without having an abnormally high ratio of negative work to gross work.

Figs. 21 and 22 are, respectively, pressure-volume and entropy-temperature diagrams for an engine working on this cycle. In explaining the cycle it is best to start at E^1. At this point the temperature of the fluid is 1,592° C. (1,865° abs. C.), and the pressure is 30 lbs. abs.

Let the fluid be heated by combustion at constant pressure along the line E^1C^1 till the temperature reaches 2,000° C. (2,273° abs. C.). Now let the gas expand adiabatically from C^1 to D^1 till the pressure is atmospheric. The temperature will then be 1,592° C. (1,865° abs. C.). Now let the gas pass through a regenerating chamber and be cooled at a constant pressure from D^1 to F^1 till the temperature is 80° C. (353° abs. C.). The gas escapes at F^1 into atmosphere, and thereafter cools at constant pressure to 17° C. (290° abs. C.) at A. A new charge is taken at A and compressed adiabatically to B^1 where the pressure is 30 lbs. abs. and the temperature 80° C. (353° abs. C.). The fluid is now passed through the regenerating chamber, and is heated at constant pressure along the line B^1E^1, taking back the heat given up by the last charge. This will raise its temperature to 1,592° C. (1,865° abs. C.) and place the fluid in the condition it was at the start.

Referring to Fig. 21, the gross work is represented by the area $g^1G^1C^1D^1$, the negative work by the area $g^1G^1B^1A$, and the net work by the area $AB^1C^1D^1$.

$$\text{Therefore } \frac{\textit{negative work}}{\textit{gross work}} = \frac{\textit{area } g^1G^1B^1A}{\textit{area } g^1G^1C^1D^1} = 0 \cdot 16 \ (0 \cdot 1553).$$

The heat absorbed by the fluid (other than that obtained in the regenerator from a previous charge) is represented in Fig. 22 by the area eE^1C^1d. The heat rejected (other than that given to the regenerator) is represented by the area aAF^1f. The heat converted into work is represented by the difference of these two areas.

$$\begin{aligned} \text{Now area } AB^1C^1D^1 &= \text{area } aB^1C^1d - \text{area } fF^1D^1d - \text{area } aAF^1f \\ &= \text{area } aB^1C^1d - \text{area } aB^1E^1e - \text{area } aAF^1f \\ &= \text{area } eE^1C^1d - \text{area } aAF^1f. \end{aligned}$$

Therefore the area $AB^1C^1D^1$ represents the heat converted into work.

$$\text{Therefore E} = \frac{\textit{area } AB^1C^1D^1}{\textit{area } eE^1C^1d} = 0 \cdot 84$$

The ideal efficiency is high; but the highest actual efficiency which could practically be obtained would be very much below this. Besides the losses in the motor proper and in the pump, there would be a very great loss in the regenerator. It would not be practicable to reduce the temperature of the exhausting gases in the regenerator to 80° C., or to raise the temperature of the fresh gases in the regenerator to 1,592° C.

If the losses in the regenerator and in the passages leading to it and from it amounted to 50 per cent. of the heat which is ideally given to or taken from the regenerator, these losses would have to be made up by extra heat given to the fluid by combustion, and the efficiency would fall to 0·3. This does not take into account the losses in the motor proper and in the pump. The heat losses in the motor would probably be very great. This cycle may, however, when used with turbines, give results sufficiently good to justify its use. It certainly seems to promise better results with turbines than with reciprocating engines, on account of the lower frictional losses that might be expected with turbines. With reciprocating engines the large volumes and the low pressure of the fluid would cause extremely high percentage losses in friction.

The cycle has the disadvantage that gas at a very high temperature has to be conveyed from the regenerator to the turbine. This practically makes it absolutely necessary to have the turbine quite close to the regenerator. It would seem to be expedient to build the regenerator of brickwork, and to erect the turbine in this brickwork. This would very much limit the usefulness of the cycle, as it would not be quite feasible in many cases either to have a regenerator of the nature required at the place where power is wanted, or to transmit the power from a place suitable for holding the regenerator. Nevertheless there will be cases in which it will be quite practicable to build a regenerator beside the turbine, and this cycle therefore seems to be worthy of consideration. A rotary pump driven from the turbine spindle could easily be used for compressing the gases, thus simplifying the mechanical moving parts.

The several cases can be compared in the Table (page 1102). It will be seen that a high ideal efficiency is, as a rule, accompanied by a high ratio of negative work to gross work. Cycle 4 is however an exception to the rule. The cycle has the highest ideal efficiency and the lowest ratio of negative work to gross work. As has been already pointed out, however, the efficiency which could be actually looked for with this cycle would be very much below the ideal, and the cycle has other objections, as already stated.

Cycle I, Case 1, has a high ratio of negative work to gross work, although the efficiency is the lowest. This is because all the heat is supplied to the gas at a comparatively low temperature.

Engineers interested in any particular cycle can work out other cases for themselves if they consider it necessary, but it is suggested that after a careful perusal of this Paper the effect of any change can be guessed at with fair accuracy. It might be possible to use the exhaust gases from a turbine working according to Cycle I, Cases 1 and 3, or Cycle II, Case 1, to heat the fluid after compression and so to save fuel. Consider Fig. 4 (page 1070). The heat supplied to the fluid is represented by the area aBCd. Part of this heat might be obtained from the hot exhaust gases and the efficiency of the cycle thus raised, as will be clear from the description just given of Cycle IV. With Cycle I, Cases 1 and 3, and Cycle II, Case 1 there should be no necessity to use a regenerator of brick or such like refractory material. The exhaust gases could be passed through tubes and the fresh air passed over the outside surfaces of the tubes or some equivalent construction might be employed.

Many other cycles or modifications of cycles might have been

TABLE

Comparing the several Cycles and Cases.

Cycle.	Case.	Compression. Temp. abs. C.°	Compression. Press. Lbs. per sq. in. abs.	Max. Temp. abs. C.°	Max. Press. Lbs. per sq. in. abs.	Ideal Efficiency. (E)	Ratio of negative work to gross work.
I.	1	389	42	973	42	0·25	0·40
I.	2	389	42	2273	42	0·25	0·17
I.	3 & 3A	682·5	300	2273	300	0·58	0·30
I.	4 & 4A	909	818	2273	818	0·68	0·40
II.	1 & 1A	500	101	2273	459	0·55	0·23
II.	2 & 2A	750	417	2273	1265	0·68	0·38
III.	2	682·5	300	2273	300	0·33	0·41
IV.	—	353	30	2273	30	0·84	0·16

investigated; but the author has considered it inadvisable to burden the Paper with them.

As regards the Carnot cycle, an engine working on this cycle would have the same value for E as one working on Cycle I for the same values of t and t_c; and, if the isothermal expansion were carried far enough, it would (under ideal conditions) do the same work per cycle for the same amount of fluid, and have the same ratio of negative work to gross work.

The maximum volume of the fluid would, however, be very much greater, and although this is not such a serious matter with a turbine as with a reciprocating engine it is nevertheless not a condition to be accepted without due recompense.

If an engine working on Cycle I has $T_1 = t_c$, then an engine working on the Carnot cycle with the same values of t and t_c would require, in order to do the same work, to commence its adiabatic expansion at the point where the engine working on Cycle I leaves off. If p and P_1 on the Cycle I engine are atmospheric pressure, then, on the Carnot cycle engine, the whole of the adiabatic expansion would take place below atmospheric pressure. It is interesting to compare the Carnot cycle with other cycles, but it hardly seems useful in the present investigation to devote any more space to this comparison.

Although the practical efficiency of an engine is usually of great importance, there are many occasions on which a very poor efficiency will be tolerated if other conditions are satisfactory. Small gas-engines using lighting gas, costing 2*s*. to 3*s*. per 1000 cubic feet, are employed in great numbers, although the fuel cost per B.H.P.-hour is very high. Small electro-motors are extensively used consuming current which costs over 2*d*. per Board-of-Trade Unit. The fuel cost and the energy cost, respectively, in the two cases are high; but the user prefers to put up with this, rather than employ a power plant which has a higher initial cost or which requires more attention or is generally more inconvenient.

If, therefore, small gas-turbines could be sold at a low price, and if they required little attention and did not readily get out of order, they might be in great demand, even although the gas-consumption per B.H.P.-hour was high. With the average user of a small engine, producing say 100 B.H.P.-hours per week, a reduction of £10 in the initial cost is of more consequence than a reduction of 2 cubic feet per B.H.P.-hour in gas-consumption. The same user would no doubt be quite willing to allow an additional 5 cubic feet of gas per B.H.P.-hour if he were saved trouble and anxiety and small expenses in the working of the engine.

To produce a gas-turbine cheaply it seems necessary to entirely avoid reciprocating parts and to be content with a low compression. Cycle I, Case 1, appears to lend itself to cheapness of construction and simplicity, but it might be advisable to reduce the compression at the expense of efficiency. A rotary pump could undertake the compression.

In many cases the vibration of a reciprocating engine is extremely objectionable; and a motor that ran with practically no vibration would be popular, even if its initial cost were greater and it were more extravagant with fuel. In motor cars, for example, oil or spirit explosion engines are used for their lightness and compactness; but the vibration they cause is objectionable. If a satisfactory turbine were obtainable, there is no doubt that motor-car builders would eagerly buy it and instal it on their cars, even if the cost were greater and the efficiency less than the many arrangements of explosion reciprocating engines now in use.

One might mention many other uses to which gas-turbines could advantageously be put, if they were obtainable as fairly efficient and reliable machines. In many factories and engineering works, electric motors fed by current from a central station are used to drive individual machines or groups of machines, in order to save the losses and inconveniences produced by driving by belts or ropes. Gas-engines (reciprocating) have been used to a limited extent for the same purpose; but the foundations required for these and the vibration caused by them have prevented their extensive use. If a gas-turbine were obtainable which could be set down anywhere like an electric motor, it would serve splendidly for this purpose, and, in order to displace electric driving, it would only require to possess an efficiency greater than the efficiency of the central station engine multiplied by the efficiencies of dynamo, mains, and motor.

Suction gas-producers are coming into extensive use, and by their means gas-engines can take the place of steam-engines where they otherwise could not. Whatever objections there may be to the supply of gas to gas-engines from these suction producers, these objections should be less rather than greater with turbines than with reciprocating engines.

The power of a gas-turbine could be effectively varied with an insignificant variation of speed, by cutting out one or more of the nozzles or admission ports which admit the fluid to the turbine buckets or first set of buckets when several are employed in series, the fluid being passed through the acting ports or nozzles at a uniform pressure and with a uniform velocity. To enable the acting nozzles or admission ports, no matter how many may be in use, to deliver the fluid always in the same uniform manner, it will be necessary, if one flame supplies all the nozzles or admission ports, to control the fuel and air supplies to the flame in conjunction with the mechanism which cuts off the nozzles or admission ports. It will however be usually advisable to supply the air and fuel to the flame at a constant velocity which, although it could be done when one flame supplied a varying number of nozzles, would involve complications; and it may therefore be found expedient to have a separate flame and a separate combustion chamber for each nozzle. This would involve the necessity of igniting or extinguishing a flame for each change of power, and, although this could be done automatically, it is objectionable. Much careful consideration will be required to determine which is the least evil to put up with and which course had best be taken.

The author hopes that this comparison of the several cycles treated will be of some use in showing what may be expected from each, and which will be best suited for a motor which is required to work under given conditions and which it is important should have given characteristics. One cannot, however, compare the several cycles and estimate the actual efficiencies, &c., which might be expected in practice in anything like so satisfactory a manner as could be desired, without having more information obtained by experiment on the following three points:—

1. Losses in pneumatic compression to high pressures.
   (*a*) With reciprocating compressors.
   (*b*) With rotary compressors.
   (*c*) With a combination of reciprocating and rotary compressors.
2. Expansion of hot gases in divergent nozzles.
3. Radiation losses and transference of heat from gases to metals at high temperatures.

It would immensely aid the solving of the gas-turbine problem if a thorough set of experiments on these three points were made and the results published. This would naturally cost a considerable amount of money; but the information obtained by the engineering world would be very good value at the price. Money has been spent and is being spent by engineering and scientific societies on investigations which, while no doubt interesting and instructive, are not of so far-reaching importance as experiments which would materially aid in the production of a successful gas-turbine.

The Paper is illustrated by 22 Figs. in the letterpress.

Discussion.

The PRESIDENT said that the author of the Paper had been for a long time associated with Mr. Dugald Clerk. It was rather an exception for the Council of the Institution to accept a Paper based upon theoretical considerations alone, the rule being to reject Papers unless they were founded upon something that had actually been accomplished in a concrete form. But he thought the large audience before him that evening justified the Council in having admitted the present Paper for theoretical discussion. With the American Society of Mechanical Engineers no formal vote of thanks was passed for a Paper. Thus time was saved for the technical discussion. The procedure of this Institution had latterly followed the same course, and the appreciation of the present Paper might now be indicated by a round of applause.

Mr. NEILSON said there were many methods of working and devices suggested in the Paper which had been previously proposed, some of them over and over again. It was extremely difficult in many cases to ascertain to whom the credit was really due for those devices or methods of working, and therefore he had in most cases refrained from giving acknowledgment to anyone. He mentioned that fact in case anyone should feel aggrieved at not being given the credit of something stated in the Paper. He wished to emphasise what he had said at the beginning of the Paper—that its most important object was to draw opinions from other engineers who had studied the question, and especially from those who had conducted experiments.

Mr. Henry Davey, Member of Council, said he had made some experiments, but they had been on a small scale, and, with a view to starting the discussion, he would just state what his experience had been. First of all he would like to tender his thanks to the author for bringing the Paper before the Institution. It dealt with an important subject. When a Paper was brought before engineers,

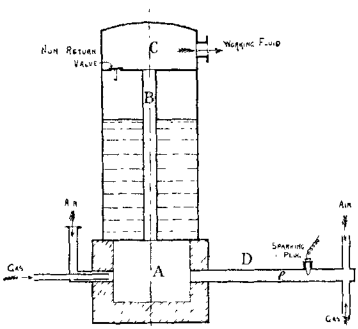


Fig. 23.
Experimental Working-Fluid Producer.

and the theory of the subject was put before them so lucidly as it had been put by the author, it encouraged them to look more minutely into the matter, and to see if some practical good might not come out of it. The Paper was valuable as indicating the direction in which the possibilities of the problem lay. The first part of the problem was that of producing in a practical way the working fluid; then the temperature difficulties followed, and they must be succeeded by the mode of application to the turbine.

It was from the point of view of producing the working fluid that he had made two or three experiments, on quite a small scale. They were undertaken with an apparatus, shown in Fig. 23, based on Cycle III, Case 2 (page 1091). A was a combustion chamber

(Mr. Henry Davey.)
lined with fire-brick, and above it was a small steel boiler consisting of a shell containing the water, having fire-tubes B extending to the smoke-box or vessel C. A non-return valve opened into the smoke-box. When steam was raised above the pressure in the furnace, it passed through the valve and mixed with the products of combustion. The object of the apparatus was to produce a mixture of steam and products of combustion, the steam being highly superheated. The furnace was fed by means of gas and air pumps, *of the relative capacities for delivering* a burning mixture. The gas and air were delivered in separate pipes and came together just inside the furnace. To start the apparatus it was necessary to get the fire-brick lining into a red-hot state, so that it might maintain combustion; then the air and gas pumps were put to work. He experimented with this apparatus on a small scale, but he found many practical difficulties. He commenced with low pressures, intending to go on step by step to higher and higher pressures; but what happened was that, if the gases ceased to burn in the furnaces from any accidental cause during one or two strokes of the pump, then a big explosion frequently ensued. It seemed to him that the system promised well, if the practical difficulties of keeping the combustion constant could be overcome. If *the divergent nozzle* was as efficient as one might imagine, it might be that a turbine worked with a fluid produced from an apparatus of that kind, at moderately low pressure, would give a good efficiency; but the loss of efficiency in the air and gas pumps would be considerable. His pumps were reciprocating; and the loss would be probably much more considerable if they were rotary ones. That would tell very much against the scheme, as it was all a question of what useful work could be got out of a given quantity of gas. Instead of bringing air and gas from the furnace into the combustion chamber to be burnt with a constant flame, he altered the apparatus as shown at D. The gas and air united together in the pipe *e* and pushed forward the products of combustion of the last charge, forming a kind of cartridge in the pipe. Then the supply was cut off, and a sparking plug at once fired the cartridge. With a moderate sized brick-lined chamber A, a fairly constant pressure

could be kept up. The cartridge might be small and fired very frequently, or three or four of them might be used and fired intermittently, to avoid intermittent impulses on the turbine. He had not got beyond what might be termed laboratory experiments, but had succeeded in getting the thing to work fairly at a low pressure. He had not used the working fluids to propel a turbine, but he had taken them, in the particular case he mentioned, into a reciprocating engine. It was desirable to know what were the losses due to compression both with reciprocating and rotary pumps, and then came the very great difficulties of using high temperatures in the turbine itself. With regard to the divergent nozzle, there was, as far as he knew, scarcely any practical information available. It was one of those subjects which required to be thrashed out experimentally, and one of the most important points in connection with the development of the gas-turbine.

Professor F. W. Burstall said it was a source of great gratification to him to be able to attend and—he would not say discuss the Paper, because it was not a Paper which admitted of a great deal of discussion—speak on a subject which was one of immense possibilities and potentialities. It might not be for the present generation to inherit the gas-turbine, but it was probable that it would come to subsequent generations. If he might say one word of criticism in regard to the Paper, he thought perhaps the author was a little optimistic on the subject. He knew very well the enormous difficulties which stood in the way of a commercial gas-turbine. From the days of Watt it had taken nearly 120 years to develop a comparatively simple thing like the steam-turbine, and, when it was considered that the gas-turbine had not got to turn the energy of the fluid into kinetic energy on the shaft, but had to compress the fluid, to ignite the fluid, and deal with a fluid which was infinitely more troublesome to deal with than steam, it would be seen what difficulties had to be overcome. He did not for a moment say that a gas-turbine could not be made; perhaps he would plead guilty to having schemed a good many himself. There was no difficulty in making one; the

(Professor F. W. Burstall.)
difficulty was to make a turbine which would produce any useful work at the engine shaft, and for a reason which he would very soon make clear. In the Otto cycle engine the compression was produced almost entirely in the motor cylinder itself, and therefore any heat which escaped from the charge of air and gas into the walls of the cylinder was only there for a short time, if at all. When the charge exploded, the cylinder walls were heated, and cooled on exhaust, but at any rate they were probably very much above the temperature of the compression charge. It was clear, he thought, that no turbine could ever be made to work on that principle. The compression of the air and gas must take place outside the turbine casing. Therefore one had to contemplate the possibility of economically producing that compression.

The author had alluded to the rotary compressor. The rotary compressor, as far as he knew it—and that was only up to about 7 lbs. or 8 lbs. per square inch—was singularly inefficient, and he felt convinced that any rotary compressor at present known would take the whole power of the engine to compress the air and gas to begin with, because its efficiency was probably not more than 0·4; and, when it was considered that the negative work was 0·4, it would be found there was about 30 per cent. of the work of the engine absorbed in compression. Referring to the experiment of Diesel, by taking a portion of the air during compression and compressing it in an independent cylinder, a very much higher efficiency had been obtained. On that point he could answer the question with regard to the efficiency which Diesel got in his compression cylinder. The pressures were not correctly stated in the Paper, though perhaps they were the pressures found by Mr. Ade Clark. In his own Diesel engine the compression in the cylinder was 35 atmospheres, and the compression on the blast used for spraying the oil was 57 atmospheres. The efficiency in the small compressor was not high, but then that small compressor took such a very small charge that it did not affect the general efficiency of the Diesel engine, or hardly appreciably.

When dealing with the steam-engine, one was dealing with a fluid that could be carried over very large distances without very much loss of heat. A 5 per cent. liquefaction in a pound of dry steam would liberate quite a considerable quantity of heat. In the gas-engine, however, those conditions were reversed. Any fall of temperature of the gas came entirely from the internal energy of the gas, and therefore an explosion charge could not be trusted to remain in contact with the metallic surface for any length of time without an excessive loss of heat. From that, he inferred that any form of Parsons turbine was not the most desirable form for working with gases. The cooling in passing through the comparatively small passages would be so immensely rapid as to prevent anything like a reasonable efficiency being obtained. In the reciprocating engine there was a rather different set of conditions. There was a large volume of gas in a cylindrical vessel, and only the outer portion of it became cooled at all, the centre core remaining always hot, there being not sufficient time to pass its heat to the outer walls. Hence the losses from cooling in a reciprocating engine were not serious.

With regard to the possibility—and he was afraid it was the only possibility in the last instance—of using the divergent nozzle, the author had clearly indicated what were the limiting conditions for that divergent nozzle, namely, that a velocity was required of about 5,200 feet per second in order to turn the heat energy of the charge into the kinetic energy of the turbine vanes. Probably, at present, that was not possible, but he saw no reason whatever to suppose that, with the advances in metallurgical science, engineers would not be in possession of a material which would stand the high stresses and which might probably be lighter. It seemed to him by no means impossible to get that, and, granting the fact of a material, then the turbine was materialised almost at once. It was a matter of mechanical difficulties. As he had stated before, there was no reason to suppose that it would be anything like economical, owing to the very high ratio of the negative work, which did not occur with the reciprocating engine.

The questions which the author raised with regard to data were of course extremely important, particularly the questions relating to the reciprocating compressor and the expansion of hot gases in the divergent nozzle. The author would probably know very well that

(Professor F. W. Burstall.)
any one of those experiments was a most serious matter to undertake, and to get results which were in any way commensurate with practical work meant conducting the experiments on a large scale. When experiments such as those were conducted on a large scale the expenses were apt to be enormous, and therefore one could hardly expect any private individual to give his information for the good of the world at large. If the experiments were made, no doubt they would be made by some individual who, naturally, was seeking his own advancement and who wished to produce a commercial article. Whether such a thing was possible he did not know, but he felt very strongly there was no possibility of advance in that direction until more efficient compressors were obtained. That was the real gist of the matter. In the last five years he had had on the average from six to eight patents for gas-turbines put before him, and in every case he had discovered that they were perfectly vague on the subject of the negative work. If the author only succeeded in persuading engineers that that was the real rock on which they split in devising a gas-turbine, then he thought the object of the Paper would be thoroughly accomplished. He thanked the author for having brought such a subject before the Institution, even if it were only in the form of a scientific investigation.

Mr. James Atkinson said the author had given a most excellent Paper on the theory of gas-turbines, and as a matter of fact the gas-turbine simply existed now in theory. Whether it would remain so or not was a question that might take a good deal of deciding. With regard to the temperature which the author assumed could be put into a turbine of the Parsons type, the author had taken it as 700° C. (1,292° F.). Professor Burstall had spoken about the probabilities of a metal which would stand those temperatures and the scouring action which must take place in a turbine, but he, Mr. Atkinson, thought such a metal was a long way from being discovered. Any metal that had iron in its composition commenced to oxidise at about 400° C. (752° F.), or 500° less than the temperature given by the author. If that temperature existed in the presence of free oxygen, metal would begin to oxidise, and the scouring action which took place in the turbine would very soon wash away the blades. He thought the author might have made a slight mistake in talking about superheated steam in steam-engines and the efficiency being only a small percentage. As a matter of fact, he thought superheating in the steam-engine was more efficient than the actual steam working in the steam-engine. In a turbine, superheating the steam increased the efficiency very largely, and it would increase the efficiency of a steam-engine equally as well, if it were not for the fact that superheated steam went a very small way through a steam-engine, and in a triple-expansion engine it very rarely went beyond the high-pressure cylinder, if as far as that. He expressed his admiration for the care, attention, and trouble the author had taken in producing his calculations, which would be of very great use to engineers who had to deal with the question. The Paper practically did for the gas-turbine what Mr. Dugald Clerk did for the gas-engine many years ago.

Lt.-Colonel R. E. B. Crompton, C.B., thought that the two great difficulties which stood in the way of producing a gas-turbine had been fully stated by Mr. Davey and Professor Burstall, the latter of whom had rounded up the subject so completely that it left little for others to say. There was one point however which might perhaps be mentioned, although he was uncertain whether it could be fairly discussed under the question of gas-turbines. He had recently been present at the Electrical Congress at St. Louis, and had taken part in the discussions which followed two interesting Papers, read by Mr. Emmett, of the General Electric Co., of Schenectady, on the Curtis form of steam-turbine, and by Mr. Hodgkinson, of the Westinghouse Co., of Pittsburg, on the Parsons form of steam-turbine. Several speakers appeared to think that one method of obtaining the maximum efficiency in electrical energy from the energy in the coal would be by using the gas-producer and gas-engine to deal with the higher temperatures, and by utilizing the energy remaining in the jacket-cooling water and in the exhaust of the gas-engine by raising steam in a suitable boiler and utilizing this steam in a low-pressure

(Lt.-Colonel R. E. B. Crompton, C.B.)
steam-turbine. In this way it seemed possible to increase the present highest practical efficiency obtainable with the gas-engine from the figure of about 28 per cent. up to possibly 38 per cent. He thought that there were no practical difficulties in the way of the mechanical engineer in carrying out this development. What would then remain to be considered would be whether the economical advantage in increased efficiency would not be more than counterbalanced by the extra first cost and afterwards by the maintenance cost of the extra plant that would have to be added to obtain this increased duty from the fuel. It appeared to him that, in many cases where fuel was dear, it was quite probable that the answer would be favourable, and that therefore such a proposition ought to be seriously taken in hand by those interested in the question. Speaking as a power engineer, he thought that this combined use of gas-engines and steam-turbines was likely to be of more immediate practical importance than the remote possibility that the mechanical difficulties connected with the gas-turbine would be successfully surmounted.

Mr. H. M. Martin said that Professor Burstall had mentioned, in connection with rotary air-compressors, 0·4 as the highest efficiency he had come across. In a turbine compressor built for the Farnley Ironworks, in which the rotary air-compressor was driven by a steam-turbine, the combined efficiency as determined by Professor Goodman was 61 per cent. That was to say, the "air horse-power" was 61 per cent. of that theoretically due from the steam. This meant that the efficiency of the compressor *per se* was something like 80 per cent.

With respect to experiments on divergent nozzles, it might be stated that some had been made by Professor Stodola, who found that with a pressure drop of from $10\frac{1}{2}$ to $\frac{1}{10}$ atmospheres, the loss was 20 per cent. The experiments were not entirely satisfactory, as the measurements were made by means of a small tube passing centrally down the nozzle, which acted as a sort of Pitot gauge. In fact Professor Stodola concluded that in the absence of the resistance caused by this tube the loss would have been about 15 per cent.*

There was another possibility with respect to gas-turbines which he did not know whether it would prove practicable to work out. Consider the well-known water-ejector, such as indicated in Fig. 24. High-pressure water entering through a suitable nozzle into a combining cone drew in and carried with it low-pressure water to form a combined jet. The efficiency claimed for the appliance was over 90 per cent. He did not know whether it would be possible to work an air-ejector on the same lines. The method would be to let a jet of products of combustion at high-pressure develop its full

Fig 24.
Water-Ejector.

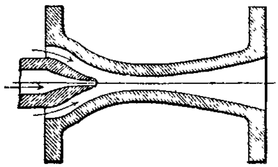
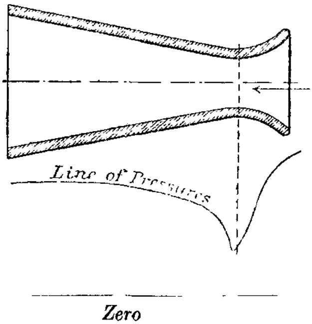


Fig. 25.
Divergent Nozzle, showing Varying Pressure.



kinetic energy in a divergent nozzle. This would then be utilized to draw in, say, four or five times its weight of air by an ejector, and the combined jet having a correspondingly reduced velocity and temperature would be utilized to drive the turbine. Some experiments by Professor Stodola went to show that the necessary

* The use of a thermo-couple would seem to afford a means of measuring the loss in a divergent nozzle with the least possible disturbance of the flow. With such an instrument the distribution of temperature throughout the nozzle could be determined, and this known, together with the weight discharged per second, the efficiency of the nozzle could be readily calculated by elementary thermodynamics.

(Mr. H. M. Martin.)

suction could be obtained with an air-jet. Thus he found that if a divergent nozzle designed for a high drop of pressure was used for a smaller drop, the distribution of pressures in the nozzle was such as indicated in Fig. 25 (page 1115). Obviously, if holes were put through the walls of the nozzle in the region of low-pressure, air would be drawn in, and Professor Stodola had attempted to construct an ejector somewhat on these lines, but so far had failed to obtain an efficient apparatus.

Professor Robert H. Smith said that much useful and interesting information, especially upon the outflow of high-pressure fluids through varying shapes of nozzles, was to be found in the book by Professor Stodola on "Steam Turbines." * As the author in his Paper referred to a theory that more than a certain critical velocity could not be obtained in such nozzles, he might mention that Professor Stodola's experiments showed velocities in the nozzles which were considerably higher than the critical velocity calculated on the assumption which the author referred to; but, when the critical velocity was passed, extremely interesting phenomena occurred, which had been suggested by the diagram shown by Mr. Martin, Fig. 25 (page 1115). The pressure oscillated, not in time, but in distance along the axis of the nozzle. Great waves, steady waves, of pressure existed. The most interesting practical point resulting from these experiments was that such waves of pressure did not serve any useful purpose, and that the nozzle should be designed so that they did not arise. The nozzle could be designed for each discharge pressure so that the waves could be smoothed down and made to disappear. That, he thought, was the most useful result of Professor Stodola's very long and exceedingly interesting series of experiments last year.

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* "Die Dampfturbinen," by Dr. A. Stodola, of Zurich, 1903, published by J. Springer, Berlin. *See* pages 17–37.

Other recently published books on the subject are "Le Turbine a Vapore ed a Gaz," by Ing. Guiseppe Belluzzo, published by U. Hoepli, Milan, 1905; and "Dampfturbinen, Entwickelung, Systeme, Bau u. Verwendung," by Wilhelm Gentsch, published by Williams and Norgate, London, 1905.

The difficulty to which Mr. Atkinson had alluded seemed to be the greatest difficulty to be feared in gas-turbines, because the fluid resulting from combustion must be diluted very greatly in order to keep down pressure and temperature within practical limits, and the only cheap way of diluting it was with an excess of air. There would be, therefore, a very large excess of extremely hot air containing large quantities of free oxygen passing through the narrow nozzles and wheel buckets, and he fancied these would be rapidly burned away. He was surprised to hear the statement that air-compressing could not be done at a greater efficiency than 40 per cent. He knew many reciprocating air-compressors in which it was possible easily to attain over 60 per cent. efficiency. Mr. Henry Davey had sketched an apparatus he had made upon a small scale, Fig. 23 (page 1107). He, Professor Smith, had spent two years in experimenting upon a similar apparatus on a larger scale, which was designed for 600 lbs. working pressure and 45 indicated horse-power, and he had worked it at 130 lbs.; but it was not for the combustion of gas, but for that of oil. The difficulty had been, so far, one similar to that which Mr. Davey referred to, namely, the occasional sudden extinction of the flame and the difficulty of controlling the flame, leading, in his apparatus, to an inconvenient rush of the water over into the fire-tubes.

In the Paper there was a very interesting and important remark (page 1062) relating to Carnot's formula for the efficiency of an ideal heat-engine. This formula was not intended for practical application to working plant, but was framed only for the purpose of proving a theoretical proposition, a proposition of immense value in the science of thermodynamics. It was not a measure of efficiency applicable to any heat-engine which it was possible to construct and work. The upper and lower limits of temperature might be greatly changed without to any extent changing the thermodynamic efficiency. It was an historical fact that every engine that had been made and worked successfully had cut down its upper temperature limit before it had obtained practical success.

He was struck by the remark (page 1103) that "With the average user of a small engine, producing, say, 100 B.H.P.-hours per

(Professor Robert H. Smith.)
week, a reduction of £10 in the initial cost is of more consequence than a reduction of 2 cubic feet per B.H.P.-hour in gas-consumption." It occurred to him that that was rather an extravagant estimate of the value of low initial cost. It was usually his fate to argue in favour of greater consideration of initial cost. Considering that 2 cubic feet per B.H.P. at 100 B.H.P.-hours per week was 200 cubic feet of gas per week, and taking gas at 2*s.* 6*d.* per thousand, that meant 26*s.* a year; 26*s.* was 13 per cent. upon £10, and 13 per cent. was rather a high percentage to take in the comparison. Perhaps a saving of 3 cubic feet of gas per B.H.P.-hour in an engine of this small size working with so bad a load factor might be more fairly comparable with the advantage of £10 decrease in capital cost. On page 1104 the author suggested the use of gas-turbines upon motors for certain reasons which he gave, but surely the difficulty of using any sort of gas-engine on the motor was not in the engine itself but in the weight and bulk of the vessels that had to be carried to hold the gas. Some years ago he worked that out, and he found it was practically impossible, even with the high pressures that were used in weldless steel gas-cylinders, to carry any reasonable bulk of gas in a motor car. Lower down on the same page the author made a very useful remark, pointing out that in order to displace electric driving the turbine would only require to possess an efficiency greater than the efficiency of the central station engine multiplied by the efficiencies of dynamo, mains, and motor. If those four efficiencies were multiplied together a very low compound efficiency would be obtained in the end.

Mr. NEILSON, in reply to the discussion, stated that Professor Burstall's remarks on rotary compressors and their efficiencies had been already replied to by Mr. Martin (page 1114). It by no means followed that an efficient rotary compressor could not be used, although the turbine compressors were not successful at high pressures. Turbine compressors had been used up to about 80 lbs. pressure or so, that was, compressors arranged like a converse steam-turbine. It was quite reasonable to suppose that such compressors would not be efficient at high pressures, as they had a great number of stages, and the air was somewhat roughly handled at each stage. He thought there was great room for investigation in the matter of trying to find an efficient rotary compressor on another principle. Professor Burstall had stated that a gaseous charge could not be allowed to remain in contact with the metal for long without a great transference of heat, and that gas was very different from steam in that respect. In steam-turbines with superheated steam there was practically a gas at the high-pressure end of the turbine. Superheating had very much improved the efficiency of steam-turbines, and it would be reasonable to suppose that no great loss of heat occurred at the high-pressure end of the turbine. He believed experiments had been made at the Manchester Technical School with a Parsons turbine, and the temperatures taken at different stages of the expansion, but he had not seen the figures. Those experiments were with steam slightly superheated.*

Professor Burstall had referred to the possibility of getting a material some day that would stand the high stresses caused by high speeds. During the last few years there had been very great improvement in that respect. He did not think that a De Laval turbine, such as the 300-H.P. turbine now built, could have been built 20 years ago, as there was no material then to stand the enormous stresses. It had to be remembered that the stresses varied pretty nearly as the square of the velocity, so that, if the velocity was doubled there was four times the stress. The experiments mentioned at the end of the Paper certainly would cost a great deal of money. Those with divergent nozzles would perhaps entail least expense, as such experiments would not require a great amount of apparatus. Professor Stodola's experiments were exceedingly interesting, but it was impossible to take the results as being very exact. Professor Stodola admitted that his measuring apparatus was of such a nature as would be apt to distort the results. For example, in his nozzle he had a measuring tube which was very

* The author subsequently saw the figures through the courtesy of Mr. Mellanby; but the superheat was so slight that he considered the figures, although interesting, were of no value in the present instance.

(Mr. Neilson.)
small, but still of a diameter that was appreciable considering the minimum diameter of the nozzle. Experiments on the effect of wind pressure on different bodies had been made and they were described before the Institution of Civil Engineers * by Dr. T. E. Stanton. The experimenters discovered that in order to get reliable results the models they put in the air duct had to be very small compared with the dimensions of the duct. That was with a comparatively small velocity of air. In Professor Stodola's experiments the velocity of the fluid was enormous, and the effect of any body in the centre of the jet would presumably make a much greater difference. Therefore,

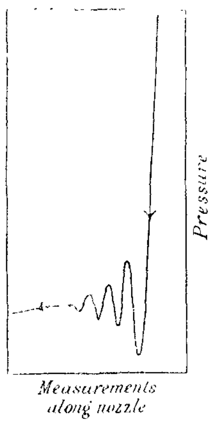

Fig. 26.
Fluctuating Pressure of Steam passing through Nozzle.

while Professor Stodola's experiments were interesting, further investigations were required in connection with the matter.

With regard to the fluctuation of pressure mentioned by Mr. Martin and Professor Smith as being found by Professor Stodola, the steam in passing through certain nozzles fell in pressure more than it should do, and then fluctuated up and down in regular beats before it remained steady at the final pressure. Fig. 26 illustrated the oscillations of pressure which were like those of a spring. In the diagram, heights represented pressure, and horizontal distances represented measurements along the nozzle. He did not think

* Proceedings Institution of Civil Engineers, 1903, Vol. clvi, page 78.

Professor Stodola was the first to discover that. It illustrated what he had said earlier, that it was very difficult in many cases to tell to whom credit was due. The fact was mentioned in a Paper* which Mr. Hodgkinson read in America some few years ago, but even he did not seem to be the first to discover the phenomenon.

Mr. Atkinson had referred to the statement about superheating (page 1062). Of course, superheating did increase the efficiency, *not owing to thermodynamic reasons*, but because of the reduction in friction and for other reasons which he need not go further into. Mr. Martin alluded to the proposal to draw in an extra quantity of air to mix with the products of combustion in order to cool down the temperature of the explosion and get a more reasonable temperature, and he mentioned the proportion as 4 lbs. of air to 1 lb. of products of combustion. The lower temperature suited the turbine, but it had to be remembered that in a practical turbine the extraction of the available energy of the fluid was never complete, and that with this air-dilution scheme, instead of discharging 1 lb. of fluid with a certain amount of unrecovered energy, it would be necessary to discharge 5 lbs., thus throwing away five times the available energy that would be otherwise thrown away. If water instead of air were used to dilute the products of combustion, the mass of fluid would not be increased to the same extent. A somewhat similar proposal had been made as regards steam-turbines, namely, to dilute the steam from a divergent nozzle with water or other fluid in order to bring down the velocity so as to enable the velocity to be utilized conveniently in the turbine wheel. This would allow the wheel to run at a lower speed. Instead, however, of discharging 1 lb. of fluid with a certain final velocity, the scheme would involve the discharging of, say, 5 lbs. or 6 lbs.

Professor Smith had referred to the question of the relative advantage of low initial cost and low working expenses (page 1118). The case mentioned by Professor Smith he had worked out, but he forgot what the saving amounted to as a percentage of the extra capital. Taking it as 13 per cent. (as given by Professor Smith), he

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* Proceedings, Engineers Society of Western Pennsylvania, Nov. 1900.

(Mr. Neilson.)

thought that with a small turbine the reduction in initial cost would be of more importance than the 13 per cent. saving. It was not like putting money into a building. No one could say how long the engine was going to be up-to-date, and no one knew how long it would be before another engine was required. Moreover, small users generally wanted money at the time. He thanked the Institution for the vote of thanks that had been accorded to him.

Communications.

Mr. Edward Butler wrote that perhaps the greatest obstacle in the way of the production of an efficient gas-turbine—namely, "the very low reactionary value and destructive property of a highly heated gas, as compared with steam of the same pressure"—could be removed by a combination of apparatus in which gas and air would be supplied to a mixing chamber communicating with a series of long explosion chambers with siphon formations for receiving water. The *modus operandi* being for the series of explosion chambers to be charged with explosive mixture and fired in rotation, the effect of the explosions would force out the water contained in the well or siphon formation of each tube in succession and this water could be directed on to the wheel of a water-turbine. Succeeding charges of gas mixture would enter automatically from the mixing chamber through non-return valves immediately after each explosion. The only operating mechanism required would be means for quickly filling the siphons with water and for igniting the mixture in the series of five or six explosion tubes, by which an almost continuous stream of water would be directed on to the turbine wheel at high velocity, thus avoiding all the trouble that would follow from any method of employing a turbine wheel actuated direct by the hot rarefied gases of combustion. He offered this solution of a gas-power turbine, after having had some experience in the working of a steam-turbine worked somewhat on these lines, namely, with water accelerated by a steam injector and boiler.

Mr. Dugald Clerk wrote that he had read the Paper with much interest, and he quite sympathised with the author in his effort to clear up in a preliminary way the many abstract points which required consideration before the practical problem of the gas-turbine could be attacked with any chance of success. In view of the wonderful results obtained by the Hon. C. A. Parsons and his imitators with the steam-turbine, it was very natural that the attention of engineers should be called to the problem of the gas-turbine. Belonging to the older school of engineer inventors and having become somewhat mentally fatigued by the numerous difficulties experienced with cylinder and piston gas-engines in every stage of their progress, it might be that the writer was less likely to take a hopeful view of the gas-turbine problem than a younger engineer, whose life and practical engineering difficulties were still before him. Whether this were so or not, he must confess that his view of the future of the gas-turbine was not so hopeful as the author's. The difficulties appeared to be even greater than the author had apprehended. Mr. Neilson rightly laid a certain stress on the relatively low efficiencies possible in turbine compressing plant; but he did not think he had laid sufficient stress upon the similarly low efficiency of expansion in steam-turbines of existing types. No doubt the author had clearly in his mind the fact that there were special advantages in the steam-turbine which counterbalanced the disadvantages of relatively inefficient expansion. The steam-turbine had, to begin with, the great advantage of practical freedom from losses due to initial condensation. It had the further great advantage of expansion to a much lower pressure point than could ever be effectively attained with a reciprocating steam-engine. These two advantages, in the case of steam, undoubtedly enabled the Parsons turbine to reach figures of economical steam consumption which were not touched at all by ordinary reciprocating steam-engines, and were very rarely reached even in special reciprocating steam-engines using high superheats. These advantages of no initial condensation and largely extended expansion were advantages special to steam; they were not advantages which would be found in using a turbine construction even for the

(Mr. Dugald Clerk.)
purpose of expanding high pressure cool gases, such as compressed atmospheric air. In an ordinary gas-engine cylinder, the efficiency of compression and expansion of the gaseous contents of the cylinder (without combustion at all) was so high that practically no difference could be detected between the compression and expansion lines of the air within the cylinder during successive compression and expansion strokes. Practically, what one might call the efficiency of gaseous compression and expansion in an ordinary gas-engine cylinder was certainly not less than 99 per cent. This, undoubtedly, enabled engines to be operated economically with large proportionate values of negative work. If, however, the efficiency of compression had been, say, 70 per cent.—a figure higher than any turbine air-compressor could give at present—and the efficiency of expansion 80 per cent.—a figure higher than any existing turbine could give operated by expanded compressed air—even then the united efficiencies would only amount to 56 per cent. That is, assuming the gases to be heated for the expansion period, the loss to be made good to bring the diagram up to unity would be 44 per cent.; that is, if the volume of the gas were increased by heating from 56 to 100, then the expansion would only be sufficient to produce a diagram which would keep the engine running if it were quite frictionless. In the writer's view, therefore, the problem to be faced required not only the invention of a turbine air-compressor compressing, say, to 200 lbs. per square inch, with an efficiency of something nearly 90 per cent., but it also required the invention of a turbine motor-engine which would give a like efficiency of expansion of the gases so compressed. High efficiencies of compression and expansion of gases, such as air, were undoubtedly theoretically possible; but he knew of no turbine yet in existence which would give any efficiencies such as he had indicated. Until these efficiencies were obtained, it seemed to the writer impossible to design any gas-turbine having a chance of success.

The author had very accurately made clear the point that, in some types of turbine, low temperature, and therefore large negative work, was necessary for working conditions. He himself could not help thinking that, although the high temperatures he proposed—

2,000° C. (3,632° F.) and so forth—certainly diminished negative work, they yet introduced for the inventor a much worse condition from the points of view of durability of blades, and heat losses if the blades were kept cool. There seemed to be no possibility whatever of working a turbine at temperatures approaching 2,000° C. The De Laval type was proposed by the author to enable temperatures to be reduced while the energy was maintained in the shape of velocity of the moving gases. He could not help thinking that any scheme which allowed of such reduction would also cool the expanding gases by the very conditions required in the constructions of the expanding nozzle. He did not see any immediate future for a gas-turbine, except in possibly utilizing the exhaust gases from reciprocating engines, which at present were liberated under considerable pressures. He feared that the attack of the problem of an efficient turbine air expander was more within the province of the practical inventor than the scientific investigator. He would be much interested to hear any ideas Mr. Neilson might have in the direction of producing efficient turbine compressors and expanders.

He hoped that the author would not be discouraged at the somewhat pessimistic tone of these remarks. He had often thought over this subject, and had not as yet seen any hope of getting as good results, either as to economy or durability with gas-turbines as were obtained with reciprocating engines.

Mr. W. J. A. London wrote that, in studying the Paper, one could not help feeling impressed by the thoroughness with which the author had dealt with the various possible cycles to which engineers were to look in the future gas-turbine. The most interesting point, however, in the writer's opinion, was the way in which the author's investigations showed the difficulties to be met with in the successful designing of such machines. It was a pity that he had not attempted to consider from a more practical standpoint several of the cycles set forth; had he done this, the writer could not help feeling confident that he would have omitted several of the cycles referred to.

(Mr. W. J. A. London.)

On page 1065 a reference was made to a combustion chamber where the burning gases could rest before entering the turbine. He thought this would be a great source of loss, because this chamber would have to be cooled, and, to allow the gases to remain in a chamber with cooled walls, a great amount of efficiency would naturally be lost. Further, on the same page, the author referred to a mixture of air and fuel being always supplied to the flame with a velocity greater than that of the propagation of the flame. This, in the writer's opinion, would be a very difficult thing to do, especially at the high outgoing velocities attending the expanding gas. On page 1066 the author referred to the Parsons turbine with steel blades for working at about 700° C. (1,292° F.). This temperature was very high, and it was doubtful whether any blades with sharp edges could be made to withstand it for any length of time.

In Cycle I, Case 2 (page 1071), the author proposed to circulate water in the revolving part. If he would consider the difficulties to be met with in circulating water in the revolving chamber, the writer thought he would find that it would be a more difficult problem than he considered. The water would be thrown against the outer walls, which was the point to be desired, but whether the circulation could be arranged without seriously interfering with the running balance of the machine was another question. Further, even if the rotating drum were cool, this would not effectively cool the blades, and to do this as the author suggested, by making the blades hollow, would require very large blades with internal pipe arrangements for ensuring circulation. With his device he claimed that 2,000° C. was allowable; this corresponded to 3,632° F., or above the melting-point of steel, so that it was hardly likely that any sharp edges could be expected to remain on the blades. For mechanical reasons he did not think one should look for the gas-turbine on the Parsons principle, but more on the lines of the De Laval, as set forth in Cycle I, Case 3A (page 1078), where the gas was expanded in nozzles, and the temperature reduced before the working fluid came into contact with the moving blades.

The author referred (page 1086) to a combination of reciprocating engines and turbines working with gas on the principle now being adopted in connection with steam-engines and turbines. It would be interesting to learn what gain this system had over reciprocating engines. The idea in the steam combined system was that the reciprocating engines were perhaps more economical at lower speeds than the normal. The steam-turbine working non-condensing had great difficulty in competing with the reciprocating engine, but when working condensing the conditions were different, and the turbine was more capable of taking care of what might be called the tail end of the expansion curve. These conditions when working condensing did not come into account when working non-condensing, as in the case of a gas-turbine, so that it was doubtful whether the combined system would be even as economical as a gas-engine of the ordinary reciprocating type.

The system which the writer thought offered the greatest possibilities was on the principle of Cycle III, Case 1 (page 1088), where the steam-jet was inserted into the gas, thus utilizing the expansion of the superheated steam and reducing the temperature of the working fluid to within practical limits. The ideal efficiency of 0·33 and the ratio of negative work to gross work of 0·41 did not look as satisfactory as some of the other cases set forth, but it was undoubtedly more practical, and it seemed more than likely that the solution of the difficulty would be found in this direction. Referring to the author's remarks (page 1097) on the velocity of gases issuing from diverging nozzles, there must be a limiting velocity for the flow of gas through an orifice, this velocity being at a point where the weight discharged multiplied by the velocity due to difference in pressure was a maximum. He thought this point had been experimentally investigated, and it had been found that the point of maximum discharge was when the ratio of internal to external pressure varied between 0·5 and 0·6 according to the nature of the gas. This pressure of 0·5 to 0·6 initial pressure would be at the throat, or, in other words, at the smallest part of the nozzle, and the velocity had been found to be somewhere about 1,500 feet per second. However much the difference of pressure was increased, this

(Mr. W. J. A. London.)
law remained good, and no more gas would pass through the orifice, nor would the velocity at the throat increase. The terminal velocity, however, would be greater in proportion to the drop in pressure.

Mr. E. Kilburn Scott wrote that he thought the author struck the keynote of the present position when he suggested (page 1106) that a thorough set of experiments should be made and the results published. The information so obtained by the engineering world would be extremely valuable, whatever the price, as it would save much time and overlapping if the several fundamental considerations involved could be investigated straight away. It was essentially a matter for some public body, and he suggested that the Institution of Mechanical Engineers should undertake the work. The tendency of the times was well shown by the working arrangement between the General Electric Co. of America and the Allgemeine Elektricitäts Gesellschaft of Berlin, whereby experimental data, drawings, etc., were now the common property of both firms. This arrangement was come to with the idea of cutting down expenses, and at the same time facilitating the progress of experimental research and of new designs.

The high temperatures and high velocities involved in working turbines with gas, and the fact that the hot gases came so intimately into contact with the metal parts, called for an investigation as to the most suitable metal to employ.

Separate blades, caulked in as with the Parsons turbine, would hardly do for gas-turbines, as this construction was not even satisfactory at the temperature of highly superheated steam. New turbines were, however, coming forward which had not so delicate a constitution. The Bucholz,* for example, Fig. 27, had a series of metal discs with holes drilled through them at an angle, and against which the working medium impinged; and the ingenious Patschke turbine, Fig. 28, had the rotating part simply in the form of a drum with a number of holes drilled in the rim. With such solid construction, exceedingly high velocities could be attained, and

* "Electrical Review," 8th January 1904, page 66.

temperature had very little effect, whilst these two designs had the further advantage of being reversible.

The question of water-supply to the heated metal was mainly a matter of mechanical design, and he thought it was less difficult to arrange water-circulation in the interior of the rotating part of a turbine than it was to get it into the pistons and exhaust valves of a large slow-running power-gas engine.

Principles of Two German Arrangements of Turbines.

FIG. 27.
Principle of Bucholz Turbine.

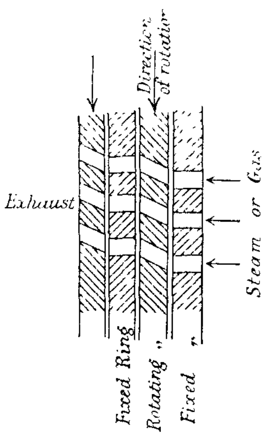
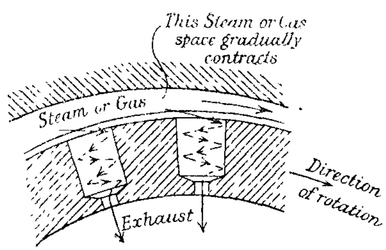


FIG. 28
Principle of Patschke Turbine.



In comparing gas-turbine with electric-motor driving, it must be remembered that practically all turbines ran at speeds which were unsuitable for driving shafting and machine tools, and that the latter were *par excellence* the very things for which the electric motor was most suitable, by reason of the ease with which the motor could be started and stopped or its speed varied, and also moved about for use with portable tools. A pipe containing gas could never compete with an electric cable for such work. As a matter of fact, the author need not worry as to uses for gas-turbines; only let it be clearly demonstrated that they were commercial, and they would become the favourite prime-mover for driving electric

(Mr. E. Kilburn Scott.)
generators, and in that direction alone there was scope enough. Anyone prophesying two years ago that the Curtis turbine would attain its present position in the United States would have been thought a dreamer, yet the writer had recently seen five stations equipped with 8,000 horse-power Curtis sets running without a hitch. So far as power-station work in the United States was concerned, steam-turbines were preferred to reciprocating steam-engines, and he believed that one day gas-turbines would come to the front just as rapidly.

Mr. George A. Wigley wrote that he thought it was remarkable that during the discussion no direct reference was made to oil-turbines or engines, with the exception of a few remarks on the Diesel engine. One of the first problems to be considered with regard to a gas-turbine was the production of the gas, and if it were necessary to supply a gas-producer even of the modern suction type, a pump for compressing this gas and a mixing chamber, it was useless to compare such a plant with an electric motor, irrespective of maintenance costs, but if a mineral oil were substituted for the gas the problem immediately assumed a different aspect.

In Cycle III, Case 2 (page 1090), the author clearly pointed out the advantages to be obtained by the use of water or steam with the gas, and this would of course hold good with oil as fuel, the ratio of negative work to gross work being decreased, but it was this question of negative work which, in the writer's opinion, was most important. In oil (mineral oil) they had what was practically a gas compressed to liquefaction all ready to hand in a very convenient form, and in such a condition that, when intimately mixed with the proper amount of oxygen, it formed a gas which was ready to give up a great proportion of the work which would be or had been necessary to compress it to liquefaction. So that they had what was to all intents and purposes a gas already compressed; and as this gas had had work done upon it far greater in amount than was necessary for the required immediate object, it followed that the oil, when fired, gave up this work, and not only reduced the negative work given out by the turbine to the extent of that required to compress the gas to the necessary pressure, but also reduced the excess work required to liquefy it, less that corresponding to the latent heat of vaporization. The result was that the negative work calculated for gas was greatly reduced if oil were used, the extra work being given in the convenient form of heat and kinetic energy of the particles.

Mr. Neilson, in reply to the written communications, wrote that Mr. London had raised the question as to whether, if the burning gases were allowed to rest a short interval in a combustion chamber before being taken to the turbine, there would not be a great loss of heat. He (the author) agreed that this question deserved consideration, but he did not think that there need be anything like as great a loss of heat to the walls of this combustion chamber as to the walls of the combustion chamber of a reciprocating engine, which required to be kept much cooler than would be necessary in the case of a turbine. As regards the last remarks made by Mr. London, it must be noted that the highest velocity obtainable when a gas issued from a simple orifice was not necessarily the highest velocity that could be obtained when a divergent nozzle was placed on the delivery side of the orifice.

With regard to supplying the air and gas at a velocity higher than the velocity of the propagation of the flame, mentioned by Mr. London, he did not think that this was really a difficult problem with the gas-turbine. With a reciprocating engine it was complicated, but with a gas-turbine having a continuous flow of gas it should not be difficult. With a mixture of a given gas and air in constant proportion, the flame would have a given rate of propagation; and, if care were taken to supply the air and gas to the combustion chamber at a greater velocity, there should be no firing back. Mr. London had referred to the proposal to circulate water in the turbine. It was said in the Paper that this had been proposed, but he (the author) did not propose it himself, and he had not much faith in it.

He agreed with Mr. Kilburn Scott as to the benefit to be derived from experiments made by a public body.