

tion, we obtain diagram, Fig. 4-2, *a*. Now if the throttle is partly closed, making the dilution of the charge by the burned gases greater than before, card, Fig. 4-2, *b*, results. Further closure of the throttle makes the combustion still slower, Fig. 4-2, *c*. Similar effects would have been obtained with full throttle if the proportion of air in the fresh charge be seriously increased.

For proper combustion the time of ignition should be so chosen that the combustion line is vertical or nearly so. This means that every different fuel mixture and every different piston speed will have its own proper point of ignition. For that reason the ignition apparatus in every engine should be made adjustable, because the only way to determine the proper time is by trial. For small gasoline engines this adjusting may be done by ear, since the proper point of ignition corresponds nearly with the highest speed, for a given throttle position. For more accurate work the indicator, preferably with a constant-speed drum motion, should be used. It will in general be found when ignition is right that it occurs some time before the piston has reached the dead center, the amount of this lead depending upon the mixture and the piston speed as above stated. What improperly timed ignition results in is shown in a diagram, Fig. 4-3, given by Clerk. With proper ignition the normal diagram is indicated by *a*. As the time of ignition is made later and later, cards *b*, *c* and *d* result, in the last case flame propagation starting so



FIG. 4-2.

late that it barely overtakes the piston before the end of the stroke.

If instead of changing the time of ignition the piston speed had been increased, effects very similar to those of Fig. 4-3 would have been obtained.

Thus the dependence of the shape of the combustion line upon the three factors mentioned at the outset is clear.

The maximum pressure attained during combustion depends upon the heating value of the charge. With all conditions favorable, the maximum pressure should be reached at or before one-tenth stroke. The rich fuel mixtures, those for illuminating gas, natural gas, gasoline, etc., show rapid combustion. They

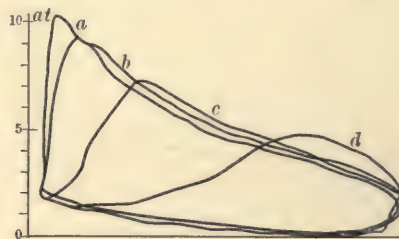


FIG. 4-3.

cannot in general stand high compression, and the ratio of $\frac{\text{maximum pressure}}{\text{compression pressure}}$, simply called the pressure ratio, is usually high, between 3 and 5. The naturally leaner fuels like producer gas and blast furnace gas ignite better when highly compressed, but on account of the generally low heating value of their fuel mixtures the pressure ratio for these gases is usually less, between 1.5 and 3.

The maximum explosion pressure p_x , Fig. 4-1, even assuming complete combustion at constant volume, is in no actual case as high as that computed on theoretical grounds for the heat received by the cycle. There may be several reasons for this. One is undoubtedly the loss of heat to the jacket water during explosion. This amount of heat is a dead loss since it does not even enter the cycle. It is something, however, which cannot be avoided, since cooling is a necessity for other reasons. Another is due to the undoubted fact that the specific heat of the gases increases

with the temperature. We can not, however, as yet mathematically gage this effect, because nothing definite is known of the law of increase.*

Another theory to account for the failure to realize theoretically maximum pressure assumes that combustion is not complete, that is, that not all of the heat of the charge is liberated when the piston starts forward. This results in after-burning, which will be considered later. It is quite likely that in most cases these three things combine to keep the observed pressure below that calculated.

Assume that the combustion takes place at constant volume, and let p_x and V_c , Fig. 4-1, be the pressure and volume at the end of the explosion.

Then
$$p_x = \frac{p_c T_x}{T_c} \text{ and } T_x = \frac{p_x T_c}{p_c}$$

If the combustion line is other than vertical, as indicated by dotted line in Fig. 4-1, let p'_x , T'_x and V'_x be the data for the end of the combustion. The above equations then become

$$p'_x = p_c \frac{T'_x V_c}{T_c V'_x} \text{ and } T'_x = T_c \frac{p'_x V'_x}{p_c V_c}$$

If in the above equations the values of p_x or p'_x are taken from actual diagrams, the equations for T_x or T'_x will give real temperatures. But if p_x should be computed from one of the theoretical diagrams of the previous chapter, then the value of T_x should be multiplied by a factor which expresses how much the *real* value of p_x falls below the *theoretical* value of p_x due to imperfections of combustion. This factor is approximately equal to the ratio

$$\frac{\text{Indicated thermal efficiency}}{\text{cyclic efficiency}}$$

An idea of the maximum pressures p_x or p'_x existing in the cycle may be gained by considering that in most cases the ratio $\frac{p_x}{p_c}$ can be made equal to 3. Turning to the table, page 88, we

* See Chapter X.

see that for $n = 1.35$, and $r = 6$, p_c is equal to about 140 lb. absolute. Thus p_x would be about $3 \times 140 = 420$ lb. The temperature T_x , assuming $T_s = 700$ degrees, which for the same values of n and r , makes $T = 1114$, would then be about

$$T_x = \frac{420 \times 1114}{140} = 3342^\circ \text{ F. absolute.}$$

The following figures give a few of the characteristic diagrams for various fuels.



FIG. 4-4.

FIG. 4-4. From Struthers-Wells hit-and-miss engine. $11\frac{1}{2}'' \times 18''$, 30 H.P., 200 r.p.m., natural gas. Card good throughout, compression 80 lb., max. pressure, 320 lb. Pressure ratio $\frac{320}{95} = 3.5$.

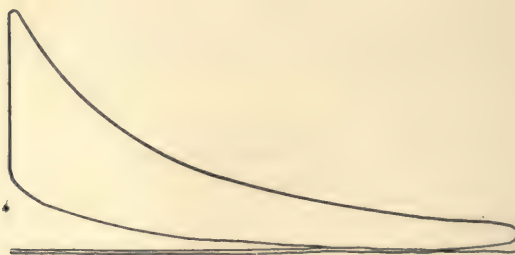


FIG. 4-5.

FIG. 4-5. From Struthers-Wells automatic engine, $16\frac{1}{2}'' \times 22''$, 150 H.P., 200 r.p.m., natural gas, full load. Card good, compression 96 lb., max. pressure 330 lb. Pressure ratio $\frac{345}{111} = 3.1$.

FIG. 4-6. From Stockport engine. Given by Clerk, the Gas and Oil Engine, p. 321. $9\frac{1}{2}'' \times 17''$, 9 h.p., 182 r.p.m. Illuminating gas. Compression 90 lb., max. pressure 270 lb. Pressure ratio = 2.7.

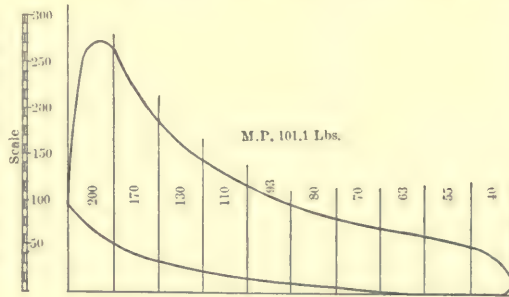


FIG. 4-6.

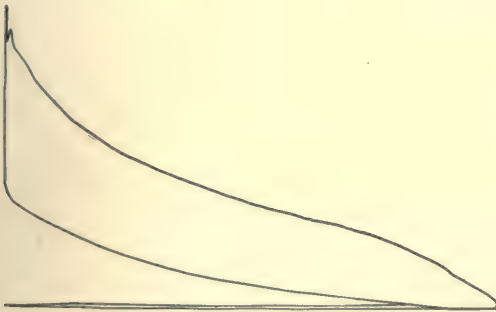


FIG. 4-7.

FIG. 4-7. From Hornsby-Akroyd kerosene engine. 6 H.P., 225 r.p.m., Card at about $\frac{1}{4}$ load. Hot bulb vaporization, hence pressures low. Compression 45 lbs., maximum pressure 116 lb. Pressure ratio = $\frac{116}{45} = 2.2$.

FIG. 4-8. Card from a gasoline engine given by Lucke, Gas Engine Design, p. 71. Vapor prepared outside cylinder. Compression 80 lb., maximum pressure 372 lb. Pressure ratio $\frac{372}{80} = 4.07$.

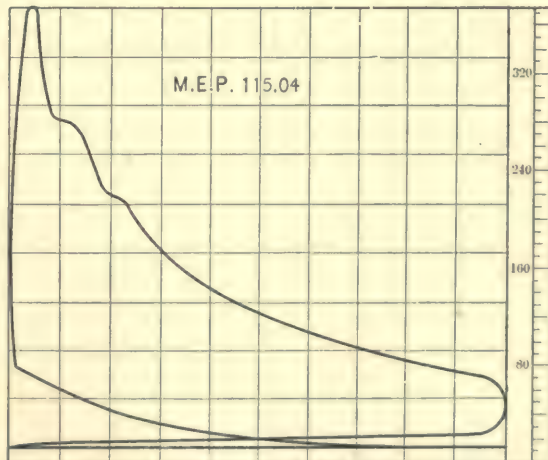


FIG. 4-8.



FIG. 4-9.

FIG. 4-9. Taken from Koerting engine. 700 H.P. Blast furnace gas. Max. pressure 242 lb., compression pressure 127 lb., Pressure ratio = $\frac{257}{142} \div 1.8$.

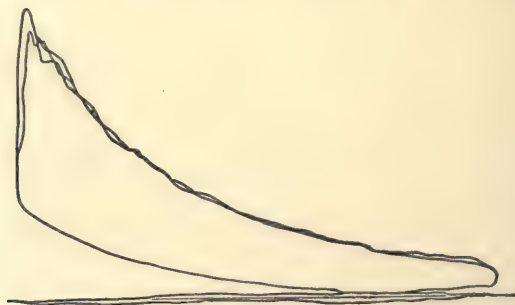


FIG. 4-10.

FIG. 4-10. From American Crossley producer gas engine, given by Langton. $18\frac{1}{2} \times 24"$, 65 K.W., 200 r.p.m. Hit-and-miss governor. Gas rather high in H and low in CO, hence compression low. Compression 83 lb., maximum pressure 248 lb. Pressure ratio $\frac{263}{98} = 2.7$.

The following two diagrams show abnormal conditions:

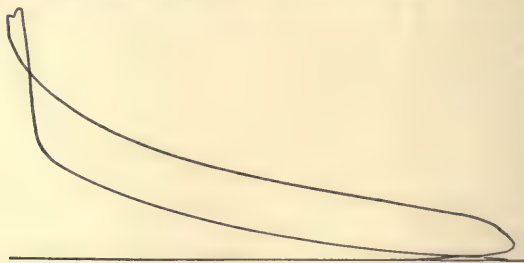


FIG. 4-11.

FIG. 4-11. Pronounced case of pre-ignition in 6 H.P. Hornsby-Akroyd kerosene engine due to too high compression. This was cured by the addition of a little water to the charge.

assumed that this arrangement was disturbed but little during compression. Hence ignition was sure; but as the combustion progressed and reached the leaner layers of mixture, it became more and more slow, was not completed when the piston started forward, and was continued along the expansion line, showing a so-called after-burning. Thus Otto attempted to explain the somewhat slower rise of the combustion line and the absence of the serious shock at the moment of explosion in his gas engines. In the fight against the Otto patent, tests were made by the Deutz Company, by Dewar and by Teichman, all of which seem to support Otto's claim of stratification. Slaby defended it vigorously. It has, however, been pretty clearly shown to-day that, while stratification is not at all impossible, in fact it is not easy to get a uniform mixture, it cannot have any marked effect upon economy or performance; that is, the diagrams would not be very far different. The opinion of the best writers of to-day leans toward the requirements of most uniform mixture and rapid combustion. Clerk has attempted to show that some heat is "suppressed" during the combustion period in every gas engine and released along the expansion line, although to attempt to express these quantities in thermal units would seem superfluous in view of the fact that we know nothing definite of the variation of specific heat at high temperatures. It would seem, therefore, that after-burning, if it exists, is not a peculiarity common to Otto engines only. It is merely evidenced more strongly in the Otto diagrams by the fact of higher piston speeds and proportionately smaller enveloping surface, giving less time and opportunity for heat losses along the expansion line, as compared with gas engines before Otto's time. Hence, as Clerk puts it, the slow dropping of the expansion line is not the cause of the greater economy of the Otto engine, but rather the effect and evidence of it.

The second explanation of the supposed after-burning rests on the so-called *dissociation theory*. It has been shown by Bunsen that a composite gas breaks up into its elements when the temperature exceeds certain limits. Conversely, chemical combinations, such as combustion, can no longer take place when this temperature limit is reached. This theory applied to the gas engine would mean that at the inner piston position, if the temperature rises to the limit, combustion no longer takes place,

but just as soon as the piston starts forward, resulting in a drop of both temperature and pressure, combustion again ensues and is, by the same method, continued along the expansion line until no combustible remains. Clerk strongly leans to this view of the matter. But it has been pretty definitely shown that the temperatures in gas engines rarely exceed 28-3000 degrees Fahrenheit, and that these temperatures are below the dissociation limit. It has also been pointed out by Schöttler that if this theory holds good the expansion line should be an isothermal as long as there is combustion. It may therefore be concluded that dissociation plays but a small part in the combustion phenomena found in gas engines.

Witz, in some tests made with an engine whose piston speeds could be varied, found that in each case after-burning occurred, but that the combustion was the more rapid and the maximum pressure the higher (that is, after-burning the less noticeable), the greater the piston speed and the warmer the jacket walls. Thus he concludes that after-burning largely depends upon the influence of the walls. Slaby, corroborated by E. Meyer, on the basis of other tests, has tried to show that these conclusions of Witz are not generally applicable, but on weighing the evidence, and in the light of later achievements, it must be concluded that they and the principle based upon them, *i.e.*, rapid combustion of the leanest possible mixture at the greatest possible piston speed, at least represent a step in the right direction for gas engine economy.

From the above it is quite evident that none of the theories advanced explain satisfactorily all of the phases of the question of after-burning, the occurrence of which must be held as proven, especially in the light of later tests. Schöttler, indeed, has offered another explanation for the so-called abnormal position of the expansion line by showing that a natural solution of the question may be found in the variation of the specific heat of the expanding gases with temperature. Some figures quoted by him show that this may be the case, but before the idea of after-burning can be dispensed with, a great deal more experimental work is needed and desirable.

The requirements for best efficiency of combustion and expansion have already been briefly explained. In a little greater detail they are as follows:

1. Highest possible compression pressure before ignition. The effect of this is, (a) less admixture of burned gases to the fresh charge, (b) less loss to jacket because smaller volume is involved, (c) greater mean effective pressure, (d) greater ease of ignition of charge.

2. Pure and uniform mixture and rapid combustion to avoid after-burning. The bad effect of after-burning is due to the great jacket loss along the expansion line, and its effect has been compared by Koerting to that of a leaky valve in a steam engine.

3. Avoid external cooling. This of course cannot be entirely eliminated. But since the amount of heat lost by cooling is a function of both time and superficial surface, this requirement calls for high piston speeds and a form of cylinder in which the ratio $\frac{\text{superficial surface}}{\text{volume}}$ is the smallest possible.

To return to the expansion line, at the moment the exhaust valve opens, the gases have expanded to a volume V_y , with a pressure p_y , and a temperature T_y , see Fig. 4-1. We may write

$$p_y = p_x \left(\frac{v_c}{v_y} \right)^n, \text{ and } T_y = T_x \left(\frac{v_c}{v_y} \right)^{n-1}$$

The ratio $\frac{v_y}{v_c}$ is the real ratio of expansion. The expressions for the value of p_y and T_y for the constant-pressure cycle are analogous; care should be taken, however, to use the proper ratio of expansion, which in this case is quite different from the ratio of compression. In the case of the Otto cycle the ratio of expansion is in most instances nearly as great as the ratio of compression r , and we may therefore write, from the above expression for p_y

$$p_y = \frac{p_x}{r^n}$$

It appears from this equation that, with p_x remaining the same, the terminal pressure decreases as the compression increases. In practice, however, the terminal pressure in most cases shows an increase as compression is increased, due, no doubt, to the fact that p_x also increases with compression, unless the mixture is made correspondingly leaner.

(e) THE EXHAUST STROKE. — The velocity of efflux of gas at the instant the exhaust valve opens is very high, approximating 25–3500 ft. per second. The valve should start to open at about

r_0 out stroke, and the opening should be of such size that equalization of pressure is practically established by the time the outer dead center is reached. Too small an opening means increased lost work due to higher back pressure, higher mean cylinder temperatures, and less cylinder capacity. To get some idea of the charging and discharging operations of the four-cycle cylinder it is best to take the so-called loop card with a weak spring, say

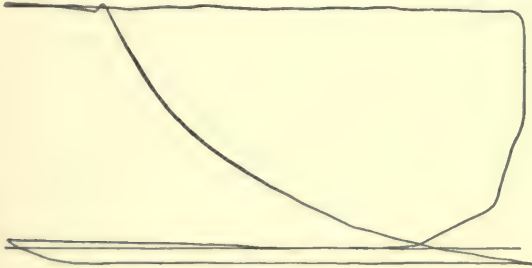


FIG. 4-13.

10-30 lb. scale, with stop attached. Besides giving a measure of the charging work, the so-called fluid friction, this also often reveals defects in the valve mechanism and sometimes curious variations in the exhaust line, which, however, are nearly always due, not to the engine, but to the exhaust piping. The ideal exhaust line should drop quickly nearly to atmosphere and remain so throughout the stroke. Fig. 4-13, from a $16\frac{1}{2} \times 22$

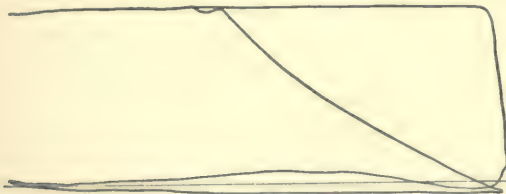


FIG. 4-14.

Struthers-Wells automatic engine on natural gas at full load, shows a nearly ideal exhaust line and in fact a very good loop card. Many loop diagrams, however, show a vacuum at the beginning of the exhaust stroke. This is undoubtedly due to the inertia of the initial gas column once set in motion. In some cases the indication of a vacuum is only very slight, in others it may last for half the stroke. In fact some builders have tried to

utilize the phenomenon to help scavenge the cylinder, but nothing has resulted from it because so many accidental conditions are apt to interfere with the regularity of its occurrence. Fig. 4-14 from a 6 horse-power Hornsby-Akroyd oil engine shows the occurrence of the vacuum very plainly. Owing to some peculiarities in the exhaust pipe, it sometimes happens that the masses of exhaust gas in the pipe are set into vibrations which cause a recurring vacuum in the exhaust line. Fig. 4-15, given by Güldner, shows a diagram from an Otto engine in which this occurred.

The absolute size of the loop, *i.e.*, the fluid friction, varies with the method of regulation, and, in a hit-and-miss engine, with the load on the engine. The effect of the method of regulation will be discussed in a later chapter. In a hit-and-miss engine at low loads the cylinder is cooler than at high loads, and the weight of gases displaced is greater, hence we may expect a greater area of loop than at normal loads. Humphrey, in a test of a 400

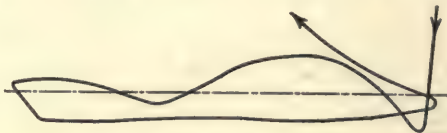


FIG. 4-15.

horse-power Crossley engine, determined a fluid friction loss of 15 horse-power at full load, which is $\frac{15}{400} = 3.8$ per cent. When the engine was taking in air only, this loss was 33 horse-power, but even assuming that there was no increased loss at say half load, the fluid friction would thus have been $\frac{15}{200} = 7.5$ per cent. It is found in general practice that fluid friction in hit-and-miss engines represents from 4-10 per cent of the engine power at full load, depending upon proper design.

3. The two-stroke Otto Cycle. Fundamentally there is no difference between the compression, combustion and expansion lines of the four-cycle and two-cycle types of engines, whether they operate on the constant-volume or constant-pressure combustion principle. The difference between the methods of operation is principally that the exhaust and charging actions of the four-cycle type are done in another way in the two-cycle, and that therefore the crank receives an impulse every revolution instead of every other. Fig. 4-16 shows an idealized two-cycle diagram.

The exhaust opens as before at *a*, the exhaust gases escape, charging commences, and is finished at *b*, where compression commences. That is the ordinary operation. There are some modifications, as for instance fuel is pumped in under high pressure along the compression line, or, as in the Diesel, air alone is compressed and the fuel injected only when the compression is completed; but these are special cases. It will be seen, therefore, from the diagram that in the ordinary case the exhaust and charging actions must be done during the time that the piston moves from *a* to the dead center position *c*, and back again to *b*. This time is short at best and extremely short under high speeds, and therein lies the whole difficulty of two-cycle operation.

The prime requirement of two-cycle operation is thorough scavenging of the cylinder of burned gases, for upon that depends

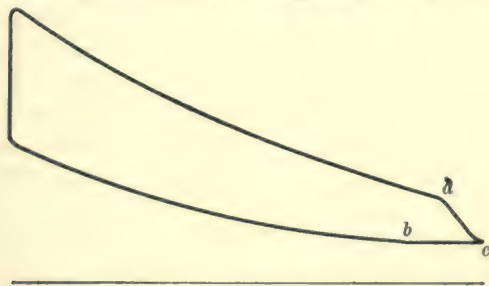


FIG. 4-16.

not only the volume of the fresh mixture that can be taken in, but also the explosibility of the charge. Too great a remainder of such gases not only seriously decreases the capacity of the machine, but it may even go so far as to prevent ignition altogether. Thus, as Güldner aptly says, the two-cycle stands or falls with the perfection or imperfection of the scavenging process. In the light of these facts it is comparatively easy to point out the requirement for good two-cycle operation.

I. The exhaust gases should be at approximately atmospheric pressure by the time the point *c*, Fig. 4-16, is reached. This reduces the volume of the gases remaining in the cylinder and reduces the work displacing them. For that reason the exhaust port should be of ample size, and this explains why the ring of ports, uncovered by the piston, is so much used. An exhaust valve as such is done away with.

II. Scavenging should commence somewhere between *a* and *c*. The scavenging agents used are

(*a*) Air; (*b*) fuel mixture.

The means by which the scavenging agent is furnished are

(*a'*) Separate pumps.

(*b'*) One end of cylinder or cross-head, used as pump.

(*c'*) Crank case used as pump.

Of these combinations, *a-a'* is undoubtedly the best. It is unquestionable that for thorough scavenging some excess of the agent should be employed, and for this only air and not fuel mixture can be used. Now only independent pumps admit of the obtaining of such an excess. It is plain, however, that it would not pay to construct independent pumps for all sizes of machines, hence they are restricted to large or at least medium powers. When independent pumps are not employed there is usually some deficiency of air. The designer, by proper design of combustion chamber and valves, is then compelled to do the best he can with the means at hand. Using the front end of the cylinder or designing the cross-head as a pump is better than using the crank case as such, mainly on account of less leakage, smaller clearance spaces and the possible better arrangement of valves. Fuel mixtures should under no circumstances be used for scavenging, except where cheapness of machine is the primary factor. Hence the small two-cycle machine which uses the fuel mixture as the scavenging agent, compressing it previously in the crank case. Such machines, however, present no true picture of two cycle operation or economy.

III. The scavenging agent should be of low pressure, and if possible of constant pressure. Low pressure is required to prevent the incoming air from piercing through and breaking up the mass of burned gases. The idea is to have the scavenging air shove these gases ahead of itself in a solid column. Constant pressure can be maintained only by the interposition of a reservoir between the pumps and the cylinder. Regarding the pressure of the scavenging air, however, a great deal depends upon valve and port construction, and speed of operation, hence nothing definite can be said to fit all cases.

4. The Constant-pressure cycle. At present this is only carried out as a four-stroke cycle with oil fuel in the Diesel engine. Fundamentally the modification in practice of the suction, compression,

expansion, and exhaust lines do not differ from those already outlined for the Otto cycle. The combustion line needs a little further attention.

The first cycle proposed by Diesel consisted of isothermal and then adiabatic compression, isothermal combustion and adiabatic expansion. This agrees with the Carnot cycle. Difficulties in the way of practical realization, however, lead to modifications of this proposed cycle until the actual cycle of to-day has little in common with it. In the first place, approximately adiabatic compression, such as is used in any gas engine, was substituted for the two compression lines originally proposed. The greatest change, however, is in the combustion line. In a lecture given by Diesel, a translation of which was printed in the *Progressive Age*, 1897, he lays down as the third requirement of his modified cycle "that the fuel must be introduced gradually into the air, which is compressed adiabatically to the combustion temperature, in such a manner that the heat generated by gradual combustion is absorbed in the so-called nascent state in consequence of a corresponding expansion, *i.e.*, by mechanically cooling off the gases, so that the period of combustion is going on constantly isothermally. It is evident that the fuel, in order to fulfil that condition, must be changed in its physical composition to a gaseous, liquid, or powdery form."

"That is to say, that through the combustion and during the same, no, or a relatively small, increase of temperature is caused, an idea which seems to be absurd after having heretofore always effected the increase in temperature by the combustion and during the same."

So far Diesel, even after the experimental stage of his engine had passed. It is quite evident that to approximate this isothermal combustion, it cannot, after ignition, be left to itself, but must be externally regulated to maintain the proper relation between temperature, pressure, and volume, as Diesel himself says. In the Diesel engine as at present constructed no such control is attempted, and it is hence difficult to see how isothermal combustion can be realized. Güldner, from indicator diagrams published by Schröter in 1897, accordingly found upon analysis that there was a decided temperature increase along the combustion line. The air was compressed to 600 degrees Centigrade (1132 degrees

Fahrenheit). At the full cut-off this had increased to about 1500 degrees Centigrade (2732 degrees Fahrenheit), and due to after-burning the maximum temperature was about 150 degrees Centigrade (270 degrees Fahrenheit) higher than this. The mean temperature of the four strokes was about 500 degrees Centigrade (932 degrees Fahrenheit). This in spite of the fact that the combustion line looked isothermal. In this case the maximum temperature was $2\frac{1}{2}$ times that at the end of compression. The temperatures thus realized are *higher* than those found in engines using constant-volume combustion, in spite of claims to the contrary.

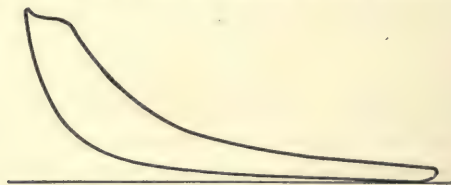


FIG. 4-17.

Fig. 4-17, from a German Diesel engine of late date, shows that the combustion is much nearer that at constant pressure than anything else. This is a step in the right direction, as it can be shown that isothermal combustion is the least favorable to best efficiency. (See previous chapter.)

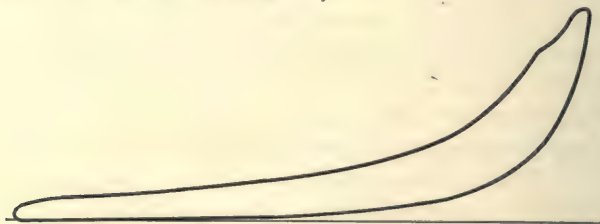


FIG. 4-18.

Fig. 4-18 shows a diagram published by the American Diesel Engine Company as late as 1904 or 1905. In the literature accompanying the diagram, the claim of isothermal combustion is still made. The combustion line certainly has that appearance, but data is unfortunately lacking to analyze the diagram. In view of the fact that the control of combustion in this engine has not changed very materially since 1897, it is safe to assume that the combustion is not isothermal, and it is perhaps to the advantage of the engine that it is not.

CHAPTER V

THE TEMPERATURE-ENTROPY DIAGRAM APPLIED TO THE GAS

ENGINE

THE meaning of the term "Entropy" has already been explained in a previous chapter. It was also shown there what shape various pressure-volume diagrams assume when transformed to a temperature-entropy basis. It is proposed to show here mathematically, and graphically if possible, how this transformation is made.

Just as the p - v diagram, by areas developed, shows the amount of work done during various events of the cycle, the entropy diagram shows heat interchanges for the various parts of the cycle, and it is therefore a valuable aid in giving an insight into the thermal actions of the cylinder. Unfortunately the labor connected with the transposition of the p - v diagram of a gas engine to the entropy diagram is considerable, much greater than is the case for a steam-engine diagram. The reason is that in steam we have a medium whose properties are definite and unchangeable, and which are at once known when a single criterion of the state of the vapor is given. It is possible also to construct entropy tables for them which facilitate the work of transposition very much. On the other hand, a p - v diagram for a gas engine represents factors which hold for that diagram and no other. Such are the composition of the charge, the specific heats of the mixture, the exponents of the compression and expansion lines, etc., and all are important for the accurate determination of the entropy diagram. It may be said that for every different load on a gas engine these factors differ, and they will even change with accidental variation in engine operation at the same load. Hence the construction of entropy tables for gas mixtures, while possible, would be of use only when the conditions of operation

happened to fit the conditions which were assumed in the computations of such tables. That would not often be the case. Even a closely mathematical transformation of the p - v to the entropy diagram is subject to errors, which make many authorities skeptical as to the real value of the diagram. These errors may be grouped under two heads:

1. Errors due to transformation, that is, errors due to incorrectly measuring pressures and volumes from the p - v diagram, and mathematical errors of computation. All these may be kept below the required limit by sufficiently careful work. The original p - v diagram should be carefully enlarged and the measurements taken from this. The larger the scale, the better.

2. Errors due to assuming specific heat constant with changing pressure and temperature. How far this assumption is justified will be shown in a later chapter. We know that there is a variation, but the law is not definitely known for composite gases, as CO_2 , and hence our assumption of constant specific heat results in a distorted entropy diagram. But this assumption is the best we can make with our present knowledge.

It should be clear from the above that a strictly graphical, and therefore time-saving, method of constructing the entropy diagram is out of question. The greatest stumbling block to the general applicability of such a method is apparently the variation of the specific heat with variation in composition of the charge. One of the best graphical methods assumes that the value of $\frac{C_v}{C_p - C_v}$ is always equal to 2.45 for the perfect gases. This law, however, appears to hold only for such gases as H, O, N and CO, whose $\frac{C_p}{C_v}$ is close to 1.41. For gases like superheated steam, or CO_2 the factor is considerably higher. Hence the value of $\frac{C_v}{C_p - C_v}$ at least should be fairly accurately computed even in using this graphical construction, the method for which will be explained later on.

Another point that should be mentioned is the fact that but few gas engine trials are sufficiently elaborate to furnish enough data for the mathematical determination of the entropy diagram. The following data should be known:

1. Composition and weight of fresh charge and burned gases.
2. Heating value of the fuel and of the charge.
3. Temperature at some point of the p - v diagram, preferably at end of suction stroke.
4. Stroke and clearance volume.
5. Index of the expansion and compression lines.

In the following will be given first the mathematical basis for the entropy diagram; this will be followed by the actual construction of such a diagram, and finally it will be shown how the same diagram can be obtained in a way mainly graphical.

The following mathematical exposition is due to Grover.* His demonstration of the general expression for entropy of a gas is especially clear.

Let H = quantity of heat in thermal units added to or subtracted from a mass of gas.

C_v = specific heat at constant volume.

C_p = specific heat at constant pressure.

T = absolute temperature.

p = absolute pressure in pounds per square foot.

v = volume in cubic feet.

J = mechanical equivalent of heat.

ϕ = entropy.

It may be remembered from the statements in Chap. II that

$$\delta\phi = \frac{\delta H}{T} \quad (1)$$

When heat is supplied to a mass of gas the volume, pressure, and temperature of the gas may vary simultaneously. Hence we may write the energy changes occurring under the conditions in the following general terms, following Grover:

$$\text{Addition of heat may produce} \left\{ \begin{array}{l} \text{Additional internal} \\ \text{energy of the gas in-} \\ \text{volving rise of tem-} \\ \text{perature.} \end{array} \right\} + \left\{ \begin{array}{l} \text{External effect of} \\ \text{work done by the} \\ \text{gas expanding be-} \\ \text{tween its contain-} \\ \text{ing walls.} \end{array} \right\}$$

With the notation above given we may write this

$$\delta H = C_v \delta T + \frac{p}{J} \delta V \quad (2)$$

* Grover, Modern Gas and Oil Engines.

but
$$\frac{pv}{T} = R = J(C_p - C_v) \quad (3)$$

hence
$$p = J(C_p - C_v) \frac{T}{v} \quad (4)$$

Substituting (4) for p in equation (2)

$$\delta H = C_v \delta T + (C_p - C_v) T \frac{\delta v}{v} \quad (5)$$

and
$$\delta \phi = \frac{\delta H}{T} = C_v \frac{\delta T}{T} + (C_p - C_v) \frac{\delta v}{v} \quad (6)$$

which is the general equation for entropy of a gas.

Now heat may be added to or subtracted from a body of gas under three conditions: (a) at constant volume; (b) at constant pressure; or (c) pressure and volume may change at the same time.

Case (a). Change of heat at constant volume.

Under this condition

$$(C_p - C_v) \frac{\delta v}{v} = 0 \quad (7)$$

and we shall have from (6) simply

$$\delta \phi = \frac{\delta H}{T} = C_v \frac{\delta T}{T} \quad (8)$$

Case (b). Change of heat at constant pressure.

$$\text{From } pv = RT$$

$$\text{and } p \delta v = R \delta T$$

we may derive

$$RT \frac{\delta v}{v} = R \delta T$$

or

$$\frac{\delta v}{v} = \frac{\delta T}{T} \quad (9)$$

Substituting this value of $\frac{\delta v}{v}$ in equation (6) we have

$$\delta \phi = \frac{\delta H}{T} = C_p \frac{\delta T}{T} \quad (10)$$

Case (c). Simultaneous change of pressure and volume.

For this condition we may write

$$p v^n = p_1 v_1^n = \text{constant} \quad (11)$$

From $p v = R T$, we have

$$p = \frac{R T}{v} \quad (12)$$

Substituting (12) in (11)

$$R T v^{n-1} = \text{constant} \quad (13)$$

Differentiating (13)

$$\delta T v^{n-1} + T (n-1) v^{n-2} \delta v = 0 \quad (14)$$

$$\delta T v^{n-1} = - T (n-1) v^{n-2} \delta v \quad (15)$$

Divide by v^{n-1}

$$\delta T = - T (n-1) \frac{\delta v}{v} \quad (16)$$

or

$$\frac{\delta v}{v} = - \frac{\delta T}{T (n-1)} \quad (17)$$

Substituting (17) in general equation (6)

$$\delta \phi = \frac{\delta H}{T} = C_v \frac{\delta T}{T} - (C_p - C_v) \frac{\delta T}{T (n-1)} \quad (18)$$

To simplify this, write

$$\frac{C_p}{C_v} = \gamma, \text{ or } C_p = \gamma C_v \quad (19)$$

and substitute this value of C_p in (18)

$$\begin{aligned} \delta \phi &= \frac{\delta H}{T} = C_v \frac{\delta T}{T} - (\gamma C_v - C_v) \frac{\delta T}{T (n-1)} \\ &= \frac{\delta T}{T} \left(C_v - \frac{\gamma C_v - C_v}{n-1} \right) \\ &= C_v \frac{\delta T}{T} \left(1 - \frac{\gamma - 1}{n-1} \right) \\ &= C_v \frac{\delta T}{T} \left(\frac{n - \gamma}{n-1} \right) \end{aligned} \quad (20)$$

Restating these results we have

Volume Constant

$$\delta \phi = C_v \frac{\delta T}{T}$$

and entropy change is

$$\phi_2 - \phi_1 = C_v \log_e \frac{T_2}{T_1} \quad (21)$$

Pressure Constant

$$\delta\phi = C_p \frac{\delta T}{T}$$

and

$$\phi_2 - \phi_1 = C_p \log_e \frac{T_2}{T_1} \quad (22)$$

Change of volume and pressure, according to $pv^n = \text{constant}$.

$$\delta\phi = C_v \frac{\delta T}{T} \left(\frac{n - \gamma}{n - 1} \right)$$

and

$$\phi_2 - \phi_1 = C_v \left(\frac{n - \gamma}{n - 1} \right) \log_e \frac{T_2}{T_1} \quad (23)$$

As already stated, it is not easy to obtain tests which give sufficient data to construct the entropy diagram. One of the most complete was a test by Brooks and Steward, made twenty-three years ago at Stevens Institute. The engine tested developed about 7 horse-power on illuminating gas. Although several of the operating conditions of this test, especially the low compression pressure of 43.5 pounds, do not represent modern practice, the complete data furnished makes this test well adapted to the purpose in view.

I. MATHEMATICAL CONSTRUCTION OF THE ENTROPY DIAGRAM.

All pressures and temperatures are absolute.

1. Composition and weight of fresh charge and burned gases.

(a) Composition of illuminating gas by volume.

H	CH ₄	N	C ₃ H ₈	CO	O	H ₂ O
.395	.373	.082	.066	.043	.014	.027

From this we compute the weight of a standard cubic foot of gas at .03882 pounds.

(b) Ratio of air to gas by volume = 6.63, from test.

Ratio of air to gas by weight = 13.78, from test.

(c) Volume and weight of the various gases per stroke.

The figures are given in the report in metric units. They are transposed as follows:

For one charge stroke the engine takes the following volumes:

1.40 liters of gas } at 295° C. = { .0494 cu. ft. of gas } at 533° F.
 9.25 liters of air } { .3265 cu. ft. of air }
 7.94 liters of burned gas at 683° C. = .2805 cu. ft. burned gas at 1261° F.

Now, —

.0494 cu. ft. of gas at 533° F. = .0457 cu. ft. at 493° F.
 which weigh .00178 lb.
 .3265 cu. ft. of air at 533° F. = .3020 cu. ft. at 493° F.
 which weigh .02437 lb.
 .2805 cu. ft. burned gas at 1261° F. = .1091 cu. ft. at 493° F. which weigh .00862 lb.
 Hence total weight of charge = .03477 lb.

(d) Composition of exhaust gases.

From above composition of illuminating gas the theoretical ratio of air to gas by volume is 5.93. Hence excess coefficient is

$$\frac{6.63}{5.93} = 1.118$$

The weights of the products of combustion, therefore, from 1 pound of gas will be

1.940 lb. CO₂ }
 1.740 lb. H₂O } from gas burned.
 .006 lb. N originally in gas.
 9.490 lb. N from air used for combustion.
 1.120 lb. N }
 .340 lb. O } from excess air.

From this the composition per cent by weight of the exhaust gases is

CO ₂	H ₂ O	N	O
13.18	11.82	72.70	2.30

(e) C_p and C_v for the fresh charge and the exhaust gases are found by computation to be as follows:

For fresh charge, $C_p = .265$, $C_v = .191$, $\frac{C_p}{C_v} = \gamma = 1.39$

For burned gases, $C_p = .268$, $C_v = .196$, $\frac{C_p}{C_v} = \gamma = 1.37$

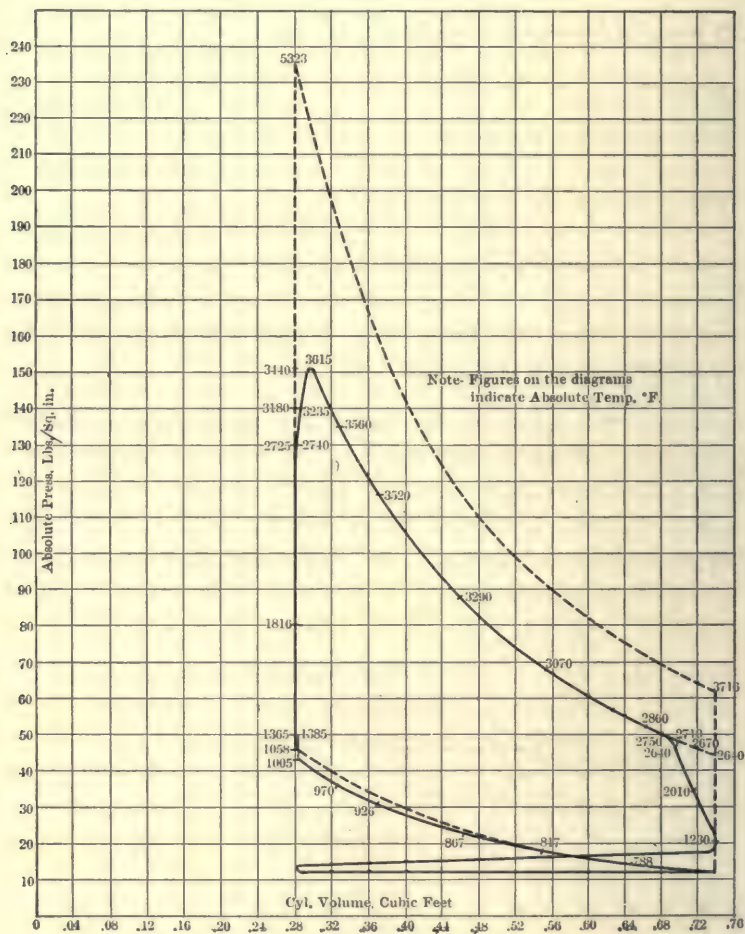


FIG. 5-1.

2. Heating value of the fuel is found by computation to be 617.5 B. T. U. per cubic foot.

Heating value of the charge as computed for the weight of gas in a charge is 29.06 B. T. U.

3. The absolute pressure at the end of the suction stroke, see diagram, Fig. 5-1, was 12.3 lb. Brooks and Stewart calculated the absolute temperature for the same point at 739 degrees Fahrenheit. By using the equation

$$\frac{pv}{T} = \frac{p_1 v_1}{T_1}$$

temperatures were computed for various points around the cycle as indicated on the card. In Fig. 5-1 the full line shows the actual card, enlarged from the original of Brooks and Stewart, while the broken line indicates the ideal cycle which receives the same amount of heat.

4. Stroke volume = .460 cu. ft.
 Clearance volume = .280 cu. ft.
 Total volume = .740 cu. ft.

5. Index for expansion and compression lines.

The index of the ideal card is 1.39 for the compression and 1.37 for the expansion line as computed under (1, e). For the real diagram it is often found that the index is not a constant for the entire line. For that reason each line should be divided into a number of parts and the index determined for each.

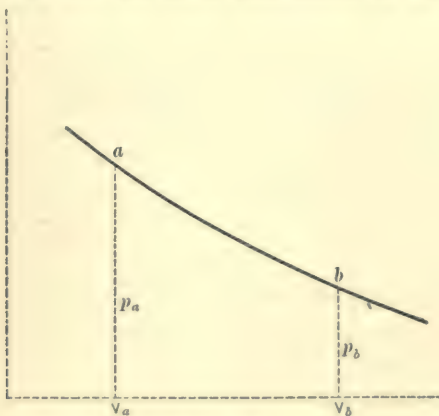


FIG. 5-2.

The method of doing this is as follows: To determine the index between any two points *a* and *b* on the expansion or compression line, Fig. 5-2, find the volumes and absolute pressures

for each point from the diagram. Then the exponent or index in the equation $pv^n = \text{constant}$ is

$$n_{a-b} = \frac{\log p_a - \log p_b}{\log v_b - \log v_a}$$

In the case of the diagram under discussion this method when applied to the expansion and compression lines gave the following results:

Expansion line

Part of line	Index
54.6 to 68.2 lb.	$n = 1.433$
68.2 to 87.6 lb.	$= 1.384$
87.6 to 116.0 lb.	$= 1.281$
116.0 to 135.0 lb.	$= 1.149$

Compression line

Part of line	Index
12.3 to 18.1	$n = 1.390$
18.1 to 23.0	$= 1.390$
23.0 to 30.0	$= 1.351$
31.0 to 43.5	$= 1.272$

We are now ready to make the entropy computations. Since entropy difference from point to point and not absolute value of entropy is the important thing, we may call the entropy at any convenient point in the cycle equal to zero. We therefore assume the entropy at the end of the suction stroke equal to 0, and we will call the entropy difference between any two points $= \phi_1 - \phi_2 = \phi$.

Ideal Card. (See temperatures in Fig. 5-1.)

1. Compression line is adiabatic, hence entropy = 0 also at the end of this line.

2. Combustion line.

$$\phi = \phi_{231.3} - \phi_{46.6} = .196 \log_e \frac{5323}{1058} = .3166$$

3. Expansion line is adiabatic, hence entropy is not changed and $= .3166$ at end of this line.

4. Discharge line.

$$\phi = \phi_{12.3} - \phi_{62.8} = .196 \log_e \frac{739}{3716} = -.3166$$

This locates the end points of the combustion and discharge lines in the entropy diagram. But these lines are curves and hence several intermediate points on each line should be determined by similar computations. The diagram obtained by plotting temperatures and volumes of entropy above computed is shown in broken line in Fig. 5-3.

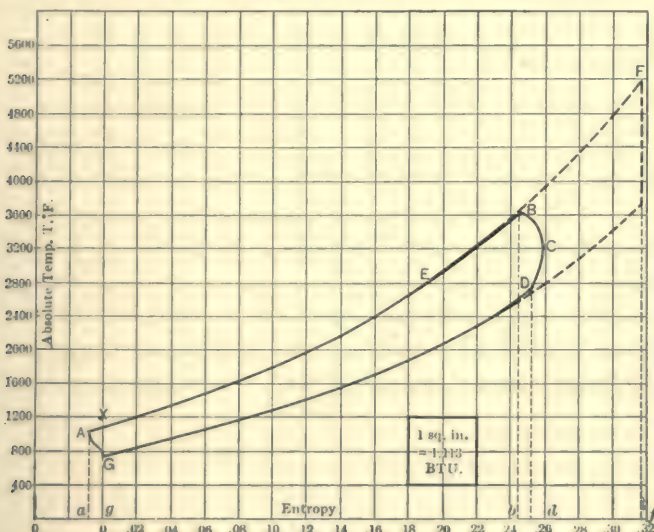


FIG. 5-3.

Actual Card. For the actual card we proceed in an exactly similar manner, using equations (21), (22), or (23) as the case demands. The computations are given below in detail. No further explanation seems necessary, except perhaps for the case where both pressure and volume change but the index of the curve is not known. Then none of the equations given directly apply.

Take the case of the diagram, Fig. 5-4. To determine the entropy at point *a*, find entropy at point *b*, using equation (21), and then add the entropy from *b* to *a* as computed from equation (22). Similarly for the point *c* on the exhaust line. Prolong

the expansion line and determine the entropy at d by equation (23) (index n must be known). Then subtract from this the entropy change due to the drop in temperature from d to c , according to equation (21).

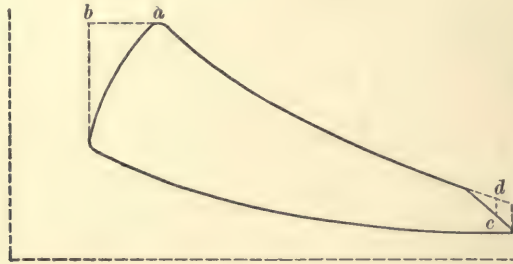


FIG. 5-4.

The following are the computations in detail:

Entropy at		Total Entropy	
Press.	Temp.		
absolute			
12.3	739	assumed = 0	0
18.1	817	adiabatic = 0	0
23.0	867	adiabatic = 0	0
30.0	926	$= .191 \frac{1.351 - 1.390}{.351} \log_e \frac{926}{867} = -.0014$	-.0014
43.5	1005	$= .191 \frac{1.272 - 1.390}{.272} \log_e \frac{1005}{926} = -.0066$	-.0080
60.0	1365	$= .196 \log_e \frac{1365}{1005} + .268 \log_e \frac{1365}{1005} = +.0591$	+.0511
80.0	1816	$= .196 \log_e \frac{1816}{1005} = +.0559$	+.1070
120.0	2740	$= .196 \log_e \frac{2740}{1005} + .268 \log_e \frac{2740}{1005} = +.0807$	+.1877
140.0	3235	$= .196 \log_e \frac{3235}{1005} + .268 \log_e \frac{3235}{1005} = +.1140$ (above 80 lbs.)	+.2210
151.3	3615	$= .196 \log_e \frac{3615}{1005} + .268 \log_e \frac{3615}{1005} = +.1376$ (above 80 lbs.)	+.2446
135.0	3560	$= .196 \log_e \frac{3560}{1005} + .268 \log_e \frac{3560}{1005} = +.1425$ (above 80 lbs.)	+.2495
116.0	3520	$= .196 \frac{1.149 - 1.370}{.149} \log_e \frac{3520}{3560} = +.0029$	+.2524
87.6	3290	$= .196 \frac{1.281 - 1.370}{.281} \log_e \frac{3290}{3520} = +.0033$	+.2557
68.2	3070	$= .196 \frac{1.384 - 1.370}{.384} \log_e \frac{3070}{3290} = -.0005$	+.2552

$$\begin{aligned}
 54.6 \quad 2860 &= .196 \frac{1.433 - 1.370}{.433} \log_e \frac{2860}{3070} = -.0019 & +.2533 \\
 49.7 \quad 2750 &= .196 \frac{1.433 - 1.370}{.433} \log_e \frac{2750}{2860} = -.0007 & +.2526 \\
 47.0 \quad 2640 &= .196 \frac{1.433 - 1.370}{.433} \log_e \frac{2710}{2750} - .196 \log_e \frac{2710}{2640} = -.0053, & +.2463 \\
 34.5 \quad 2010 &= .196 \frac{1.433 - 1.370}{.433} \log_e \frac{2670}{2750} - .196 \log_e \frac{2670}{2010} = -.0566, & +.1960 \\
 20.5 \quad 1230 &= .196 \frac{1.433 - 1.370}{.433} \log_e \frac{2640}{2750} - .196 \log_e \frac{2640}{1230} = -.1550, & +.0976 \\
 12.3 \quad 739 &= .196 \log_e \frac{739}{1230} = -.0978 & -.0002
 \end{aligned}$$

Having thus worked around the cycle, the error seems to be very slight in view of the fact that the exponent, n , for the prolonged expansion line has been assumed = 1.433.

Plotting these values of entropy and temperature finally results in the diagram shown in full line in Fig. 5-3.

INTERPRETATION OF THE ENTROPY DIAGRAM, FIG. 5-3.

In order to evaluate the diagram it is necessary to know the number of heat units per square inch. This is most easily obtained by multiplying one inch of temperature scale by one inch of entropy scale, and then multiplying the result by the charge weight per cycle, since the entropy diagram is drawn for 1 pound of charge weight. In this case we have:

Value of 1 square inch in B. T. U. = $800 \times .04 \times .03477 = 1.113$.

The individual areas of the original diagram were next gone over with a planimeter, and after multiplying each area by the square inch equivalent, the results were as follows:

	B. T. U.	% of Total Heat
1. Heat received during explosion, area $a A B b a$	18.086	62.24
2. Heat received during expansion, area $b B C D d$	1.002	3.44
3. Total heat received, as shown by diagram	19.088	65.68
4. Total heat supplied as calculated (see footnote)	29.316	
5. Difference in heat loss to Jacket and Radiation = area $d D C B E F j d$	10.228	34.89
6. Heat loss to exhaust, area $g G D d g$	13.422	45.78
7. Heat loss during compression, area $a A G g a$	.233	.79
8. Total heat lost per cycle	23.883	81.46
9. Indicated work, area $A B C D G A$	5.431	18.54
	<u>29.314</u>	<u>100.00</u>

The above results do not agree with those of Brooks and Stewart for the same test. Their results are given in the following table:

Heat in indicated work	17.0 per cent.
Heat loss in hot gases	15.5 per cent.
Heat loss in water jacket	52.0 per cent.
Heat loss in radiation	15.5 per cent.
Total	100.0 per cent.

The agreement as regards indicated work is fair. The rest of the figures of Brooks and Stewart are abnormal, in that the radiation loss is as large as the exhaust loss, and in that the jacket water loss is much too large. Brooks and Stewart themselves admit that the jacket water loss was not accurately determined. For that reason, too, the radiation loss cannot be found separately in the entropy analysis above given. If the jacket water loss had been accurately found, item 5 in the above analysis could have been separated into jacket water and radiation loss.

NOTE. — This quantity is greater than the latent heat energy in the gas = 29.06 B. T. U. per cycle, by the heat equivalent of the area $A X g a = .256$ B. T. U.

II. GRAPHICAL CONSTRUCTION OF THE ENTROPY DIAGRAM

The graphical method to be described is due to Prof. H. T. Eddy, and is by him explained in the Transactions of the American Society of Mechanical Engineers, Vol. 21, p. 275. It is based upon the following considerations:

Equation (6), p. 110, after integration may be written:

$$\text{Entropy difference } \phi = \phi_1 - \phi_2 = C_v \log_e \frac{T_1}{T_2} + (C_p - C_v) \log_e \frac{V_1}{V_2} \quad (24)$$

Dividing equation (24) by $(C_p - C_v)$ we have

$$\frac{\phi}{C_p - C_v} = \phi' = \frac{C_v}{C_p - C_v} \log_e \frac{T_1}{T_2} + \log_e \frac{V_1}{V_2} \quad (25)$$

The problem then resolves itself into finding graphical representation for the quantities

$$\frac{C_v}{C_p - C_v} \log_e \frac{T_1}{T_2} \text{ and } \log_e \frac{V_1}{V_2}$$

The construction divides itself into two main parts:

1. The change of the pressure-volume diagram into a temperature-volume diagram, and

2. The change of the temperature-volume diagram so obtained into a temperature-entropy diagram.

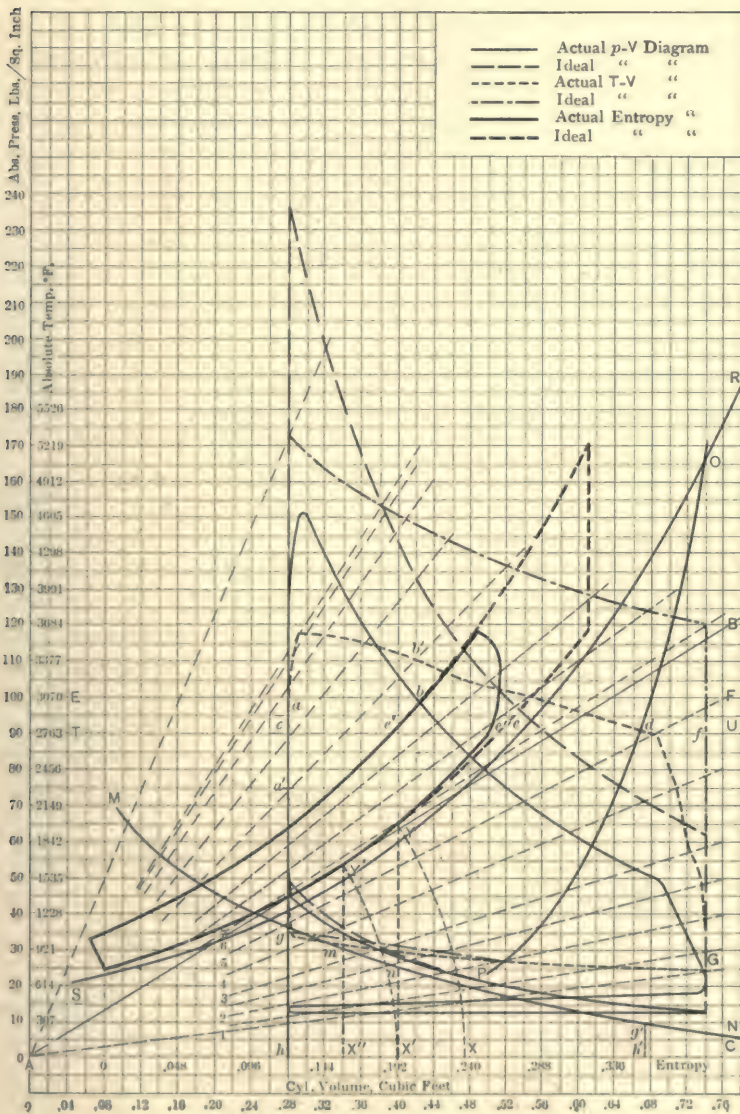


FIG. 5-5.

1. In Fig. 5-5 the actual and ideal p - v diagrams of Fig. 5-1 have been reproduced. Choose any convenient volume ordinate, in this case $V = .38$ cu. ft. has been taken, and from the points of intersection of the various pressure levels with this ordinate draw straight lines through the origin. These straight lines are constant-pressure lines in a temperature-volume field. Thus the line AB in this case is the 60 pounds constant-pressure line. To see the reason for this consider the general equation

$$\frac{pv}{T} = R$$

This may be written

$$\frac{v}{T} = \frac{R}{p}, \text{ or } \frac{T}{v} = \frac{p}{R}$$

But $\frac{T}{v}$ is the tangent of any given angle, BAC , and hence $\frac{p}{R}$ is constant for any point along AB . The same holds for any straight line drawn through any other pressure level.

Next, to construct the temperature-volume diagram, consider any pressure level as $EF = 100$ lb. This cuts the real diagram in the points a and b . At a and b erect perpendiculars until they intersect the constant-pressure line = 100 lb. in the points a' and b' . These will be two points in the temperature-volume diagram. In the same manner the entire diagram may be outlined. The temperature scale, which up to this point has been arbitrary, may next be determined. We know that G , the lowest point on the temperature diagram, must represent the temperature in the cycle at the end of the suction stroke = 739 degrees Fahrenheit. This at once determines the temperature scale.

2. To obtain the graphical representation of the expression $\log_e \frac{V_1}{V_2}$ choose any convenient horizontal line as the zero. In this case the line AC has been taken. With A as a center and any radius AX , draw the arc XY to cut any convenient line, as AB , passing through the origin. From Y draw the perpendicular YX' . Again with A as a center and AX' as radius, draw the arc $X'Y'$ to intersect with AB , and draw the perpendicular $Y'X''$. Where these perpendicular lines YX_1 and Y_1X'' cut any successive pair of equidistant horizontal rulings will be found two

points on the required curve. The horizontal rulings chosen in this case are marked serially on the ordinate $V = .22$ cu. ft. Thus m and n are two points on the curve MN sought. It is evident that the intersections of perpendiculars YX' and $Y'X''$ with any other two successive horizontal rulings might have been chosen as two points on the curve. This would have resulted merely in moving the curve bodily up or down on the field, the shape would have been exactly the same. For the same reason it is immaterial whether the line AC or any other horizontal line is used as the base of construction. The resulting curve is in any case asymptotic to the vertical line $V = 0$. Choosing any other set of equidistant horizontal rulings, nearer together or further apart, has the effect of making the curve rise slower or faster as the case may be. As will be seen later by inspection, this merely changes the position of the entropy diagram in the coordinate field but does not affect the final result.

To prove that the ordinates of the curve MN represent the values of $\log \frac{V_1}{V_2}$, Professor Eddy proceeds as follows:

Let the volume $AX = V_0$, $AX' = V_1$, $AX'' = V_2$, $AX^n = V_n$. It can be shown by plane geometry that, with the construction used,

$$\frac{V_1}{V_0} = \frac{V_2}{V_1} = \frac{V_n}{V_{n-1}} = Z = \text{constant.}$$

or $ZV_0 = V_1$, $ZV_1 = V_2$ $ZV_{n-1} = V_n$

Hence

$$Z^2V_0 = V_2, Z^3V_0 = V_3 \text{ } Z^nV_0 = V_n$$

Now let $Z = e^y$ in which e = Napierian base = 2.7 +, and y = distance between the equidistant horizontal rulings above assumed.

Then

$$e^y = \frac{V_1}{V_0}, e^{2y} = \frac{V_2}{V_0} \text{ } e^{ny} = \frac{V_n}{V_0}$$

and taking the logarithm of both sides of the last term, we have $ny =$ the ordinate at any given volume $V_n = \log_e \frac{V_n}{V_0}$. It is evident that V_0 , the unit of comparison, may be arbitrarily chosen.

The next step is to obtain the graphical representation of the expression $\frac{C_p}{C_p - C_v} \log_e \frac{T_1}{T_2}$. Considering the part $\log_e \frac{T_1}{T_2}$ by itself,

it is evident that the curve representing this may be constructed in the same manner as curve MN . But the coordinates are in this case the line of zero temperatures, AC , and any arbitrarily chosen vertical line, in this case, $V = .74$. The curve must be asymptotic to AC , but the choice of the other line of reference is unrestricted as it merely moves the curve bodily to the left or right. In this case equidistant vertical rulings having the same common distance as those used for curve MN have been employed, hence curve OP is a duplicate of MN .

The ordinates of this curve OP must next be multiplied by the factor $\frac{C_v}{C_p - C_v}$. Professor Eddy, in order to make the construction graphical throughout, assumes that the value of this factor is 2.45 in all cases. The limits of accuracy regarding this assumption have already been pointed out. It is probably *not* sufficiently accurate for most cases, and hence a separate computation of this factor is necessary for every given case. This destroys a great deal of the value of the entire method, but enough is left to make the method much less laborious than the mathematical construction.

In the case under discussion, for the burned gases,

$$C_v = .196, \text{ and } C_p = .268; \text{ hence } \frac{C_v}{C_p - C_v} = \frac{.196}{.072} = 2.72$$

The ordinates of the curve OP are therefore multiplied by 2.72, giving the curve marked RS .

By the aid of the curves MN and RS , the entropy diagram may now be constructed in the following manner:

Take any temperature level as TU . This cuts the temperature-volume diagram of the real cycle in the points c and d , and the curve RS in e . With a pair of dividers determine the ordinate gh of the curve MN , $\left(\log_e \frac{V_1}{V_2}\right)$, corresponding to the volume of point c , and since this ordinate is positive, add it to the ordinate fe $\left(\frac{C_v}{C_p - C_v} \log_e \frac{T_1}{T_2}\right)$ of the curve RS . This gives the point e' as one point of the entropy-temperature diagram. In the same manner, for the second point of intersection, d , of the temperature level TU , determine the ordinate $g'h'$, corresponding to its volume, and add it to the ordinate fe of the curve RS . This

gives e'' as a second point on the entropy diagram. By taking a sufficient number of temperature levels, the entire diagram may be closely outlined, as shown by the full line.

The same thing has been done for the ideal temperature-volume diagram, giving the ideal entropy diagram indicated in broken line.

The last step in the construction is the determination of the entropy scale, if this is desired. Determine by planimeter the area under the combustion line of the ideal diagram down to the line $T = 0$. In the original diagram this was found to be 28.32 square inches. Since the heat applied the cycle was 29.06 B. T. U., the thermal value of each square inch of area = $\frac{29.06}{28.32} = 1.026$ B. T. U. Each inch of ordinate was equal to 614 degrees, hence the entropy scale per inch is $\frac{1.026}{614} = .00167$ for the charge weight of .03477 pounds. This is equivalent to an entropy scale of .048 per inch for one pound of charge weight.

The following table shows how closely the entropy diagrams obtained by the two methods outlined agree:

	Math. Method	Graph. Method
Max. Entropy of Ideal Cycle	.3166	.3158
Max. Entropy of Real Cycle	.2557	.2572

The agreement may be pronounced quite satisfactory. It is quite likely, however, that the lower part of the graphical entropy diagram will show discrepancies. These are in great part due to the fact that toward the lower end of the curve RS , the intersections with the horizontal temperature levels become less definite, impairing the accuracy. Another source of error may lie in the fact that the compression line has been constructed from the curve RS , which was itself constructed from burned gas data, while evidently the data of the fresh charge should have been used.

CHAPTER VI

COMBUSTION

1. The Perfect Gases. — The perfect gases are those which follow the general law:

$$\frac{pv}{T} = R \quad (1)$$

where

p = pressure expressed in pounds per square foot.

v = volume in cubic feet.

T = absolute temperature.

R is the amount of work done by 1 pound of gas when heated 1 degree Fahrenheit, the pressure remaining constant at p pounds per square foot. R is thus a constant for any one gas, but differs for different gases.

The data for the gases of most use in gas-engine practice are those of atomic and molecular weight, density and weight per cubic foot. The following table gives these figures for some of the more important gases:

Gas	Atomic Weight	Molecular Formula	Molecular Weight	Density Air = 1	Weight per cu. ft. at 29.92" Hg and 32°F
Hydrogen	1	H ₂	2	.0692	.00559
Oxygen	16	O ₂	32	1.106	.08921
Nitrogen	14	N ₂	28	.971	.07831
Carbon Monoxide .	14	CO	28	.967	.07807
Carbon Dioxide ...	14.6	CO ₂	44	1.529	1.267
Dry air	—	—	29*	1.00	.08072
Water vapor	6	H ₂ O	18	.623	.05020
Acetylene.....	6.5	C ₂ H ₂	26	.915	.07251
Methane	3.2	CH ₄	16	.554	.04464
Ethylene	4.7	C ₂ H ₄	28	.974	.07809
Benzol	5.75	C ₆ H ₆	78	2.695	.21758
Alcohol.....	6.50	C ₂ H ₆ O	46	1.601	.12958

*Only apparent value.

The weight of a cubic foot of any of the above gases under standard conditions may be found with sufficient accuracy by

dividing the molecular weight of the gas by the constant 359.* The weight of a cubic foot of CO_2 under standard conditions, for instance, is $\frac{44}{359} = 0.1225$ pounds, as computed by slide rule.

This is sufficiently accurate for all practical purposes.

To find the weight of a cubic foot of gas under other than standard conditions, the formula

$$\frac{p_0 v_0}{T_0} = \frac{p_1 v_1}{T_1} \quad (2)$$

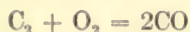
may be employed for a change of either pressure and temperature separately or a simultaneous change of both. The weight of a cubic foot is inversely proportional to the volume.

2. Combining Weights and Volumes, Combustion, Heating Value, Air Required.

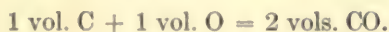
COMBINING WEIGHTS AND VOLUMES. — All elements, when they do combine, unite only in certain fixed proportions, although there may be several proportions for any given pair of elements. Thus carbon forms two combinations with oxygen, CO and CO_2 .

The gases also combine in definite volume proportions. The resulting volume is either the sum of the original volumes or is in a definite ratio less than this sum. It is necessary to remember merely that anything that can be said of molecules according to Avogadro's Law, applies with equal force to combining volumes.

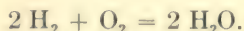
Thus take the combination of C and O to CO. Always remembering to use the combining weights of the various elements we can write



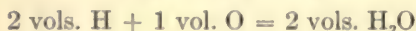
that is



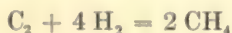
Similarly,



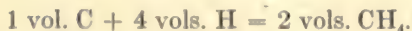
that is,



and again



or



The elements mostly concerned in combustion phenomena are carbon and hydrogen, together with the compounds carbon monoxide and the various hydro-carbons. The following table

* Derived, in connection with Avogadro's Law, from the fact that the kilogramme-volume of perfect gases is equal to 22.33 cubic meters under standard conditions.

shows the combustion formula, also oxygen and air required, for the first three of these combustibles:

Combustible	Combustion Formula	Combining Weights	Combining Volumes	Air required		
				Pounds per Pound	Cu. ft.*	Cu. ft. per* Cu. ft.*
H	$H_2 + O = H_2O$	1 lb. H + 8 lb. O = 9 lb. H_2O	2 vol. H + 1 vol. O = 2 vol. H_2O	34.78	.194	430.9
C to CO	$C + O = CO$	1 lb. C + 1.33 lb. O = 2.33 lb. CO	1 vol. C + 1 vol. O = 2 vol. CO	5.78	—	71.6
C to CO_2	$C + O_2 = C_2O$	1 lb. C + 2.66 lb. O = 3.66 lb. CO_2	1 vol. C + 2 vol. O = 2 vol. CO_2	11.57	—	143.4
CO to CO_2	$CO + O = C_2O$	1 lb. CO + .57 lb. O = 1.57 lb. CO_2	2 vol. CO + 1 vol. O = 2 vol. CO_2	2.48	.194	30.6
						2.388

* Standard conditions, *i.e.*, 29.92 inches barometer and 32° F.

If the combustible be a hydrocarbon, the combustion of the carbon and the hydrogen in its composition can be treated separately and the results combined. Thus 1 lb. of CH_4 may be considered to consist of $\frac{3}{4}$ lb. of C and $\frac{1}{4}$ lb. of H.

HEATING VALUE. — Every chemical change is accompanied by a thermal change, either positive or negative, *i.e.*, either heat is given out or it is absorbed during the change. In the case of the combustibles, the union with oxygen is accompanied by a very decided development of heat. The heat given off when one pound of any combustible is completely burned is known as its *heating value*, or *calorific power*. It is to be noted that the heating value of any combustible is constant, whether the combustible

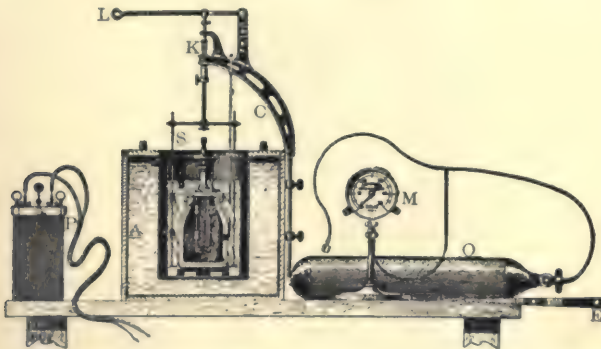


FIG. 6-1. — Mahler Bomb Calorimeter.

is burned in oxygen, theoretical amount of air, or an excess of air. The resulting temperature, the *calorific intensity* so called, is, however, different for each of these cases, as will be explained later.

The heating values of the simple combustibles like hydrogen and carbon can only be determined by means of the calorimeter; that of a complex fuel, like hydrocarbons, the various coals, etc., can be found either by the calorimeter or it may be computed with fair accuracy from its chemical composition.

For solid and liquid fuels the calorimeters used are mostly of the type of the Mahler bomb, or the Carpenter calorimeter; for gaseous fuels Junker's gas calorimeter holds the first place. The principle of any of these calorimeters is to transmit to water

the heat evolved by burning the fuel in oxygen or air, and from the temperature rise of the water to compute the heating value.

The Mahler Bomb Calorimeter, Fig. 6-1, is a strong steel or bronze vessel into which the finely powdered fuel is introduced, held in a small cup or crucible. Through the cover of this vessel or bomb two wires pass which at their lower ends are cross-connected by a fine iron wire which in turn dips into the powdered fuel. The bomb is charged with oxygen to a pressure of about 150 pounds. The oxygen can now be obtained commercially prepared in steel tubes, under pressure. The charged bomb is then placed in a vessel containing a known quantity of water; the two wires above mentioned are connected to a source

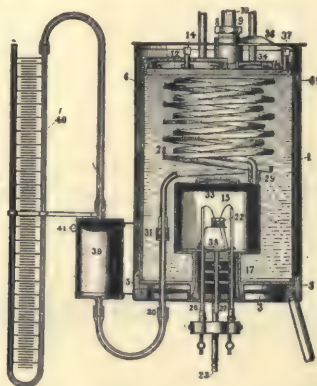


FIG. 6-2. — Carpenter Coal Calorimeter.

of electrical energy, P , which fuses the iron wire connecting their ends and fires the fuel. The water is kept thoroughly stirred by means of the apparatus LKS , and the temperature rise is carefully determined. The whole apparatus is carefully protected against radiation by an outer vessel, A . From the observed temperature rise, the weight of water, and such corrections as are necessary for radiation, heat of fusion of wire, etc., the heating value of the sample is easily computed. The drawbacks of the instrument are the labor of charging, and the fact that if the wire fails to ignite the coal all labor of weighing and charging is lost. The joint at the top, which must be tight against considerable pressure, also often gives trouble. The Mahler is applicable to both solid and liquid fuels.

The Carpenter calorimeter, Fig. 6-2, is of a different type. The fuel is powdered and held in the cup, 22, just as in the Mahler. The coal, however, is fired by the heat of a platinum wire, 15, which does not fuse, and is extinguished on the instant the coal fires. No correction is therefore necessary for the heat of the wire. The combustion chamber, 33, is kept charged with a steady stream of oxygen under a pressure not exceeding one-half pound. The hot gases of combustion pass up and down the spiral tube, 28, and finally escape at 41, practically at the tem-

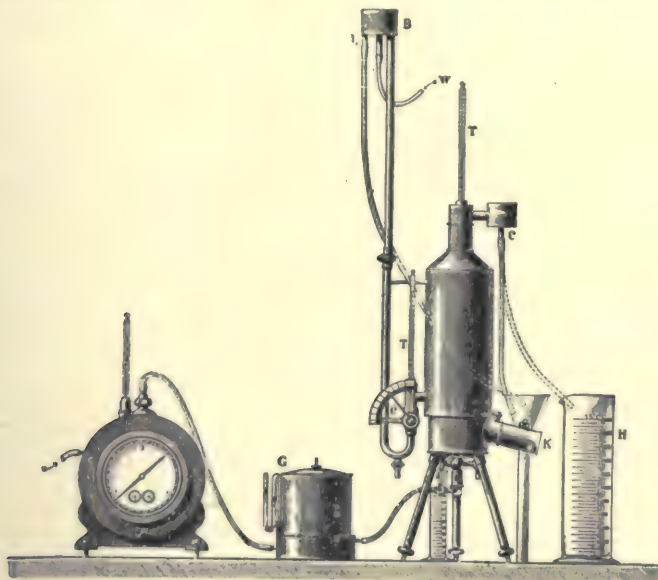


FIG. 6-3. — Junker's Gas Calorimeter.

perature of the surrounding air, the nozzle at 41 regulating the outflow velocity to produce this result. The calorimeter is completely filled to some height on the glass tube, 10, with water or kerosene oil so that the gases in their passage heat the fluid, expanding it up the tube 10. The amount of expansion gives a relative measure of the heating value of the fuel. It is only necessary to calibrate the instrument, *i.e.*, to obtain the amount of heat required to expand the fluid up the tube say 1 inch. This is quickly and accurately done by using carbon obtained by

burning and coking sugar. The calorimeter is subject to one correction only, that for radiation. It is encased in a bright nickel case to reduce radiation as far as possible, but this can not be entirely done away with. The correction is made by determining the distance the liquid level in the tube will fall, after the fuel sample is burned out, for the same length of time that the fuel burned. Care must be taken to have the temperature of the calorimeter slightly above the temperature of the room to start with, and if the temperature of the room has not changed during the time of burning and that of the determination for radiation, the fall in the water level is a true measure of the heat loss due to radiation.

For gas fuels, Junker's calorimeter, Fig. 6-3, is usually preferred. The gas to be tested is led first through the gas meter where its volume and temperature are determined, and second through the regulator *g*, which maintains any desired gas pressure during the determination. It is burned in air, with a burner of the Bunsen type. The heat evolved is transmitted to water, entering through the vessel *b*, and discharged through the vessel *c*, the gases escaping at or near the room temperature at *K*. The vessel *b* maintains a constant head of water on the calorimeter. The amount of water flowing is regulated at *e*. The water is measured in the graduate *h*. The water of condensation formed during the combustion is caught and measured at *d*. Temperatures of cold and hot water are measured at *t* and *t'* respectively. The apparatus is very simple, easily set up in any desired place, and the results obtained by its use are consistent and reliable. Its operation may be made continuous by furnishing suitable means for measuring the water flowing.

The heating value of carbon when burned to CO_2 is now definitely determined as 14647 B. T. U. per pound (8080 calories per kilo), that of carbon to CO at 4429 B. T. U. In actual practice the figures 14500 B. T. U. for C to CO_2 , and 4500 B. T. U. for C to CO are easier to remember and accurate enough for most purposes. The combustion of C to CO is usually known as the incomplete combustion of carbon. The heating value of the combustion of CO to CO_2 has been found to be 4380 B. T. U.

Hydrogen gives for the *higher* heating value 62100 B. T. U., and for the lower value 52230 B. T. U. Hydrogen may burn

either to water vapor, or, if condensation occurs, to water. In the former case the heat necessary to maintain the water as vapor is lost, and hence we obtain the lower heating value 52230 B. T. U., the temperature being 212 degrees Fahrenheit before and after combustion. If condensation occurs, the heat in the water vapor is recovered and the higher value 62100 B. T. U. results.

In gas-engine practice the water resulting from the combustion of hydrogen almost always escapes as water vapor, and hence it is usual to employ the *lower* heating value in computations.

For the same reason we distinguish a lower and higher heating value in all complex fuels containing hydrogen to any considerable extent, and always use the *lower* value in computations.

If no direct determination of the heating value of a complex fuel can be made by means of a calorimeter, the heating value can in most cases be found with fair accuracy by computation, provided the chemical analysis of the fuel is known.

In the case of hydrocarbons the statement of its chemical formula gives at the same time the weight proportions of its chemical composition. The computation of the heating values of the hydrocarbon groups C_nH_n and C_nH_{2n} may be based upon the following considerations:

C_nH_n contains n atoms of C and n atoms of H. Hence the gram-molecule weighs $12n + n = 13n$ grams. Now 1 gram of C develops 32.29 B. T. U., and 1 gram of H 115.15 B. T. U. lower heating value. The heating value of C_nH_n is therefore

$$\frac{(12 \times 32.29) n + (1 \times 115.15) n}{(12 + 1) n} = 38.67 \text{ B. T. U. per gram}$$

This reduces to:

Lower heating value of $C_nH_n = 17540$ B. T. U. per lb.

C_nH_{2n} contains n atoms of C and $2n$ atoms of H, one gram-molecule therefore weighs $14n$ grams. The lower heating value of this family of hydrocarbons is therefore generally

$$\begin{aligned} \frac{(12 \times 32.29) n + (1 \times 115.15) 2n}{(12 + 2) n} &= 44.13 \text{ B. T. U. per gram} \\ &= 20016 \text{ B. T. U. per lb.} \end{aligned}$$

It should be remembered, however, that these formulæ give approximate results only.

Slaby, quoted by Güldner,* in his Calorimetric Investigations, gives a formula based upon experience for heavy hydrocarbons, according to which:

Lower heating value = $1000 + 10500 y$ calories per cubic meter, where y = weight of a cubic meter of the gas in $K g s$.

In English units this formula would be:

Lower heating value =

$$[112 + 18880 y] \text{ B. T. U. per cubic foot,} \quad (3)$$

the cubic foot being under standard conditions, 29.92 inches Hg barometer and 32 degrees Fahrenheit, and y = weight of a cubic foot.

The following table of heating value of hydrocarbons is taken from Güldner, the figures being transposed to English units. The last two columns show for purposes of comparison the values as computed by Slaby's formula and the approximate formulæ first given:

HEATING VALUES OF HYDROCARBONS

	Weight of cu. ft. in lbs. Standard	Density air=1	Higher Heating Value per lb. B. T. U.	Lower Heating Value		Lower Heating Value per cu. ft Slaby's formula B. T. U.	Lower Heating Value Approximate Formula per cu. ft. B. T. U.
				per lb. B. T. U.	per cu ft. B. T. U.		
CH ₄ ..	.04464	.554	23842	21385	952	952	
C ₂ H ₂ ..	.07251	.915	21429	20673	1499	1479	1272
C ₂ H ₄ ..	.07809	.974	21429	20025	1564	1584	1562
C ₂ H ₆ ..	.08329	1.0367	22399	20434	1700	1682	
C ₃ H ₄ ..	.11157	1.3819	20992	20009	2232	2238	
C ₃ H ₆ ..	.11699	1.4512	21224	19820	2317	2318	2341
C ₃ H ₈ ..	.12256	1.5204	21825	20039	2455	2424	
C ₄ H ₈ ..	.15599	1.9349	20912	19508	3032	3055	3120

In the case of the liquid hydrocarbons, the crude oils and their distillates, it is usual to compute their heating value directly from the chemical composition. Thus the lower heating value

* Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, 2d ed., p. 581.

of a Pennsylvania crude oil containing C 84.9 per cent, H 13.7 per cent, O 1.04 per cent, would be

$$\begin{aligned}
 &= 14500 C + 52230 \left(H - \frac{O}{8} \right) \quad (4) \\
 &= [14500 \times .849] + \left[52230 \left(.137 - \frac{.0104}{8} \right) \right] \\
 &= 12310 \quad + \quad 7102 \\
 &= 19412 \text{ B. T. U.}
 \end{aligned}$$

The determination of the heating value of this oil by means of the calorimeter gave 19210 B. T. U., a difference of 1.05 per cent between this and the computed value.

In many instances, however, the agreement is not as close, and it should therefore be made a general rule to obtain actual calorimeter determinations whenever possible, and to resort to computation only when unavoidable.

Solid fuels, as coal, coke, wood, etc., may be treated in a similar manner. For hard coals the computation gives results which agree fairly closely with calorimeter determinations. As the amount of volatile matter in the coal increases, *i.e.*, as the hydrocarbons increase, there is less certainty of fair agreement, although the computed result is usually within 5 per cent of the true value. The general formula for a solid fuel may be stated as follows:

Heating value =

$$14500 C + 52230 \left[H - \frac{O}{8} \right] + 4000 S - 1000 H_2O \quad (5)$$

where

C = percentage of fixed and volatile carbon.

H = percentage of hydrogen.

O = percentage of oxygen.

S = percentage of sulphur.

H_2O = percentage of water in coal as received, C, H and S being determined on the dry fuel, and then recomputed to the basis of fuel as received.

The following table shows the analyses of two coals and the heating values as found by calorimeter and by above formula. Both analyses are taken from Poole's "The Calorific Power of Fuels:"

Name	Kind	C	H	O	N	S	H ₂ O	Ash	Heating Value as received	
									By Cal.	Computed
Trearton..	Anthracite	90.66	1.73	.78	.001	—	.84	6.83	14029	13988
Carnegie ...	Bituminous	77.20	5.10	7.22	1.68	1.42	1.45	5.93	13842	13367

It will be noted that the agreement in the results for the hard coal is good, in the case of the soft coal the computed result is 3.5 per cent smaller than that determined by calorimeter.

Attempts have been made from time to time to adapt formulæ to the approximate analyses of coal, *i.e.*, those in which only fixed carbon, volatile matter, water and ash are determined. All these attempts have not given satisfactory results owing to the varied composition of the volatile matter in various coals.

AIR REQUIRED FOR COMBUSTION AND THE PRODUCTS OF COMBUSTION. — As already pointed out, the combustion of 1 lb. of C to CO₂ requires $\frac{2 \times 16}{12} = 2.66$ lb. O, and that of 1 lb. of H to H₂O requires $\frac{.5 \times 16}{1.0} = 8$ lb. O. Since air contains in each pound only .23 lb. of oxygen, the air required for the two cases will be $\frac{2.66}{.23} = 11.57$ lb. and $\frac{8}{.23} = 34.78$ lb. respectively.

The products of combustion are, in the case of carbon, 3.66 pounds of CO₂ and, in the case of hydrogen, 9 pounds of H₂O, if the theoretical amount of oxygen has been used in each case.

With the aid of these fundamental figures, the air required and the resulting products of combustion may be computed for any given fuel, solid, liquid or gaseous, if the composition of the fuel is known.

In general, any fuel which contains per pound say C lb. carbon, H lb. hydrogen and O lb. oxygen, requires

$$\frac{2.66 C + 8 H - O}{.23} \text{ pounds of air for its complete combustion.}$$

In gas-engine practice, however, the case of gas fuels is of much more importance, and for this we can derive the following general formulæ:

(a) FOR ONE POUND OF GAS. — Let the gas consist of

N_1 lb. CO + N_2 lb. H + N_3 lb. CH_4 + N_4 lb. C_2H_4 + N_5 lb. C_2H_2
+ N_6 lb. O + N_7 lb. N + N_8 lb. CO_2 + N_9 lb. H_2O = 1
pound.

Air theoretically required per pound of gas

$$= \frac{.57 N_1 + 8 N_2 + 3.99 N_3 + 3.43 N_4 + 3.07 N_5 - N_6}{.23} \text{ pounds} \quad (6)$$

Products of combustion per pound of gas

$$CO_2 = [1.57 N_1 + 2.74 N_3 + 3.14 N_4 + 3.38 N_5 + N_8] \text{ pounds} \quad (7)$$

and

$$H_2O = [9 N_2 + 2.25 N_3 + 1.29 N_4 + .69 N_5 + N_9] \text{ pounds} \quad (8)$$

Besides the above amounts of CO_2 and H_2O , the products of combustion will also contain any excess oxygen that may have been used together with the nitrogen brought in by the oxygen.

If the analysis of the exhaust gas shows free oxygen, the excess coefficient for the air used, provided the fuel gas itself carries no nitrogen, may be computed from the formula

$$U = \frac{N}{N - 3.76 O}$$

where N = per cent of nitrogen, O = per cent of free oxygen in the exhaust gas, by volume and 3.76 is the volume ratio of N to O in air. This formula, together with the two given above for CO_2 and H_2O , allow of a complete determination of the products of combustion.

In case the fuel gas itself carries N, the determination of the excess coefficient for the air used is not so simple and should be made according to the method outlined on p. 144.

(b) FOR ONE CUBIC FOOT OF GAS. — Assume that the gas has the following analysis:

N_1 per cent by volume of CO
 N_2 per cent by volume of H_2
 N_3 per cent by volume of CH_4
 N_4 per cent by volume of C_2H_4
 N_5 per cent by volume of C_2H_2

N_6 per cent by volume of O_2
 N_7 per cent by volume of N_2
 N_8 per cent by volume of CO_2
 N_9 per cent by volume of H_2O

Air required per cubic foot, theoretically,

$$= \frac{\frac{N_1 + N_2}{2} + 2 N_3 + 3 N_4 + 2.5 N_5 - N_6}{.21} \text{ cubic feet} \quad (9)$$

Products of combustion per cubic foot of gas

$$CO_2 = [N_1 + N_3 + 2N_4 + 2N_5 + N_8] \text{ cubic feet} \quad (10)$$

$$H_2O = [N_2 + 2N_3 + 2N_4 + N_5 + N_9] \text{ cubic feet} \quad (11)$$

Besides these amounts of CO_2 and H_2O there will in most cases be additional amounts of free oxygen and of nitrogen. The volumes of these can be determined from the exhaust gas analysis as before.

The table, page 139, gives the main constants for the principal gases met in gas engine practice.

The constant, R , can be computed from

$$\frac{R}{J} = (C_p - C_v) = \frac{2}{m}$$

where,

J = the mechanical equivalent of heat = 778

and

m = molecular weight of the gas.

SAMPLE COMPUTATION. — Given the following chemical analysis of a producer gas, p. 140, to determine its heating value, its molecular weight m , C_p and C_v , R and $\frac{C_p}{C_v}$. Also these quantities for various mixtures of this gas with air, and the amount of and the constants for the burned gases after combustion of these fuel mixtures.

Gas	Lower Heating Value per		Weight per cu. ft., lbs., at 28.92 inches 32° F.	Theoretical Amount of			Products of Combustion		Specific Heat * per pound		Constant $\frac{p_v}{T}$ in equation
	Lb.	Cu. ft.		Oxygen for 1 lb., lbs.	Air for 1 lb., lbs.	1 cu. ft., cu. ft.	CO ₂ lbs.	H ₂ O lbs.	C _p	C _v	
Hydrogen	52230	297	.00559	8.00	34.78	2.40	—	9	3.409	2.412	775.6
Carbon Monoxide ..	4380	342	.07807	.57	2.48	2.38	1.57	—	.245	.174	55.23
Methane	21385	952	.04464	3.99	17.30	9.56	2.74	2.25	.593	.468	97.25
Ethylene	20023	1564	.07809	3.43	14.95	14.58	3.14	1.29	.404	.333	55.23
Acetylene	20673	1499	.07251	3.07	13.35	11.99	3.38	.69	.346	.270	59.13
Butylene	19512	3040	.15590	3.43	14.95	28.86	3.14	1.29	.404	.333	55.23
Nitrogen	—	—	.07831	—	—	—	—	—	.244	.173	49.79
Oxygen	—	—	.08921	—	—	—	—	—	.217	.153	53.75
Air	—	—	.08072	—	—	—	—	—	.2375	.1684	85.58
Water Vapor	—	—	.05016	—	—	—	—	—	.48	.37	35.01
Carbon dioxide	—	—	.12267	—	—	—	—	—	.20	.155	—

* From Güldner, p. 578.

SAMPLE OF DOWSON GAS, CLERK, p. 363

Compo- nents of Gas	Composition by		Density Wt. per Cu. ft. Stand- ard lbs.	Constants for Constituent Gases					V^m	$W \cdot C_p$	$W \cdot C_v$	$W \cdot R$	Heating Value of Com- ponent Gases B. T. U.
	Volume	Weight		Lower Heat Val- ue per lb. B. T. U.	Mole- cular Weight m	C_v	C_p	R					
H	.1873	.0153	.00559	52230	2	2.412	3.409	775.6	.3746	.0522	.0369	11.87	799.1
CO	.2507	.2902	.07807	4380	28	.174	.245	55.23	7.0196	.0711	.0505	16.03	1271.1
CH ₄	.0031	.0020	.04464	21385	16	.468	.593	97.25	.0496	.0012	.0009	.19	42.8
C ₂ H ₄	.0031	.0035	.07809	20025	28	.333	.404	55.23	.0868	.0014	.0011	.19	70.1
N	.4898	.5689	.07831	—	28	.173	.244	55.23	13.7144	.1388	.0984	31.42	—
O	.0003	.0005	.08921	—	32	.153	.217	49.79	.0096	.0001	.0001	.02	—
CO ₂	.0657	.1196	.12267	—	44	.155	.200	35.01	2.8888	.0239	.0185	4.19	—
									24.14	.2887	.2064	63.91	2183.1
								=	m	C_p	C_v	R	B.T.U. per lb. of gas

Weight of gas per standard cubic ft. $\ast = \frac{m}{359} = \frac{24.14}{359} = .0673$ lbs.

Heating value per cu. ft. of gas = $2183.1 \times .0673 = 146.9$ B. T. U.

A briefer way of determining R would have been $R = J (c_p - c_v) = 778 (.2887 - .2064) = 64.0$

$\ast$ See page 127.

According to equation (9), the theoretical amount of air required by this gas per cubic foot

$$\frac{.2507 + .1873}{2} + (2 \times .0031) + (3 \times .0031) - .0003$$

$$= \frac{.2342}{.21} = 1.12 \text{ cubic feet.}$$

The following table gives in the first column the constants for the theoretical air-gas mixture. The second and third columns assume that an excess of air is used, in the first case equal to $1.5 - 1.12 = .38$ cubic feet, in the last case equal to $2.0 - 1.12 = .88$ cubic feet:

CONSTANTS FOR VARIOUS MIXTURES OF ABOVE DOWSON GAS
WITH AIR

Ratio air to gas:			
By volume, V = ...	1.12	1.5	2
By weight, W = ...	1.35	1.79	2.39
Weight of standard cu. ft. of mixture, lbs. = $\frac{.0673 + .08072 V}{1 + V}$	.0744	.0753	.0772
Heating value of standard cu. ft. of mixture, B. T. U. = $\frac{146.9}{1 + V}$	69.4	58.8	49.0
$R = \frac{64 + 53.7 W}{W + 1}$	58.2	57.6	56.9
$C_p = \frac{.2887 + .238 W}{W + 1}$	.2595	.2560	.2529
$C_v = \frac{.2064 + .169 W}{W + 1}$	.1849	.1824	.1800
$\frac{C_p}{C_v} =$	1.403	1.404	1.405

The next table gives the corresponding constants for the burned gases resulting from the combustion of the fuel mixtures assumed above.

The changes occurring during combustion cause a change in the values of R , C_p , C_v , $\frac{C_p}{C_v}$ etc. For the theoretical case, i.e., with 1.12 cubic feet of air to 1 cubic foot of gas, the products of combustion will be according to equations (10) and (11).

$$\text{CO}_2 = (.2507 + .0031 + .0062 + .0657) = .3257 \text{ cubic feet.}$$
$$\text{H}_2\text{O} = (.1873 + .0062 + .0062) = .1997 \text{ cubic feet.}$$

Since 1.12 cubic feet of air were used, there must also necessarily be nitrogen to the volume of

$$[(1.12 \times .79) + .4898] = 1.3746 \text{ cubic feet,}$$

.4898 cubic feet being due to the nitrogen in the fuel gas itself.

The volumes of the exhaust gases in the second column of the table are found as follows. The volumes of CO_2 and of H_2O are of course the same as before, since 1 cubic foot of gas is burned in every case. But since only 1.12 cubic feet of air are required and 1.5 cubic feet have been used, the excess air is $1.5 - 1.12 = .38$ cubic feet. This consists of $.38 \times .79 = .3002$ cubic feet of N and .0798 of O. Hence the excess O appearing will be .0798 cubic feet, while the N now is $1.3746 + .3002 = 1.6748$ cubic feet.

CONSTANTS FOR THE BURNED GASES

Ratio air to gas by vol.	1.12	1.5	2.0
Vol. of exhaust gases, cu. ft. CO ₂	.3257	.3257	.3257
H ₂ O	.1997	.1997	.1997
O	.0000	.0798	.1848
N	1.3746	1.6748	2.0698
Vol. of exhaust gases to 1 cu. ft. of Dowson gas, cu. ft. V ₂	1.9000	2.2800	2.7800
Vol. of mixture before Combustion	V ₁ 2.12	2.50	3.00
Ratio	V ₂ .896	.912	.926
	V ₁		
% contraction	10.4	8.8	7.4
Principal Burned Gas Constants $\left\{ \begin{array}{l} R_e = R \frac{V_2}{V_1} \\ C_p \\ C_v \end{array} \right\}$ per pound	52.1	52.5	52.7
.....	.2478	.2462	.2446
.....	.1809	.1787	.1769
$\frac{C_p}{C_v}$	1.369	1.379	1.383

The values of C_p and C_v in the foregoing table are found as follows. Consider the mixture with ratio = 1.5:

Burned Gas	By Volume	By Weight
CO ₂	.3257	.0399
H ₂ O	.1997	.0100
O	.0798	.0071
N	1.6748	.1311
	2.2800 cu. ft.	.1881 pounds

$$\text{For CO}_2 \quad C_p = .0399 \times .20 = .0080$$

$$\text{H}_2\text{O} \quad C_p = .0100 \times .48 = .0048$$

$$\text{O} \quad C_p = .0071 \times .217 = .0015$$

$$\text{N} \quad C_p = .1311 \times .244 = .0320$$

$$\text{For } .1881 \text{ lbs. } \Sigma C_p = .0463$$

$$C_p = \frac{.0463}{.1881} = .2462$$

$$\text{Since } C_p - C_v = \frac{R}{J}, \quad C_v = C_p - \frac{R}{J}$$

$$C_v = .2462 - \frac{52.5}{778} \\ = .1787$$

$$\text{and } \frac{C_p}{C_v} = 1.379$$

Attention should be called to the fact that although considerable contraction of volume occurs, in the case of this gas, during combustion, still the values of R , C_p and C_v are not greatly different from the corresponding values before combustion. In some other gases, as illuminating gas for instance, the change is even less. So that in most ordinary cases it is sufficiently accurate to assume that these gas constants are the same before and after combustion. Only in cases where extreme accuracy is desired is this assumption not permissible.

3. Computation of the amount of air used in excess of theoretical requirements from the exhaust gas analysis.

In actual practice the exhaust gases are analyzed for CO₂, O, and N. By the ordinary method of collecting these gases, the water vapor originally present is thrown down and does not appear in the analysis. As mentioned before, if the fuel gas itself carries

no nitrogen, the excess coefficient for the air actually used may be computed from the formula given on page 137.

To show an example of the method of computation when the fuel gas carries N, we will take the case of the Dowson gas above given, and assume that the exhaust gas analysis gives the following results: CO_2 — 14.36%, O — 5.31%, and N — 80.33% by volume. We proceed as follows:

Products of Combustion for theoretical ratio per cubic foot of gas, are:

CO_2	.3257	cu. ft.	
H_2O	.1997	"	
O	.0000	"	
N	.4898	"	due to gas itself
N	.8848	"	due to air used
} = 1.3746 cu. ft. of N,			
of which { 35.63% is due to gas.			
{ 64.37% is due to air.			

Total 1.9000 cu. ft.

On the basis of the above exhaust gas analysis, we now have:

Total N	80.33
Of this amount, N due to excess air will be 3.76×5.31	= 19.96
Leaves N due to the gas itself and to air actually burned	= 60.37
Of this remainder, as above shown, 35.63% is due to the fuel gas	= 21.49
Leaves N due to the air actually burned	= 38.88
Hence the excess coefficient	

$$U = \frac{38.88 + 19.96}{38.88} = \frac{58.84}{38.88} = 1.5$$

and the real ratio of air to gas for the original fuel mixture was $1.5 \times 1.12 = 1.68$.

4. Calorific Intensity. — By calorific intensity is meant the temperature that can be realized theoretically when a unit weight of any fuel is completely burned under stated conditions of oxygen or air supply. If H represents the heating value of the fuel in B. T. U., A , B , C , etc., the weights of the various resulting products of combustion, and C_{pA} , C_{pB} , C_{pC} , etc., the specific heat at constant pressure of these products, the general statement for calorific intensity, supposing the pressure to remain constant, is

$$\text{Theoretical Temperature Rise} = \frac{H}{AC_{pA} + BC_{pB} + CC_{pC} + \dots}^{\circ F}.$$

Thus the calorific intensity of hydrogen with theoretical air would be, the products of combustion being water vapor and nitrogen,

$$\frac{52230}{[9 \times .48] + [26.8 \times .244]} = 4800^{\circ} F.$$

That of C to CO₂ with theoretical air would similarly be

$$\frac{14500}{[3.66 \times .20] + [8.91 \times .244]} = 5000^{\circ} F.$$

Such high temperatures, however, are practically never realized, due probably to two causes. On the one hand it is claimed that the specific heat of gases is not constant at all temperatures, but that it rises with the temperatures; on the other, dissociation is supposed to set in before such temperatures are reached. These matters will be taken up somewhat more in detail in a later chapter.

CHAPTER VII

GAS-ENGINE FUELS; THE SOLID FUELS; GAS PRODUCERS

THE general requirement for a gas-engine fuel is that it must mix readily with air to form a combustible gas or vapor. Further, it should burn with little or no residue. This latter requirement is not met by the solid fuels, as coal dust for instance, and while isolated attempts at using powdered coal directly have been made, they have so far not been successful, owing to the fact that the resulting ash soon seriously interferes with operation.

The gas-engine fuels may be classed under three heads:

1. The solid fuels.
2. The liquid fuels.
3. The gas fuels.

It is the rule that the working medium in all internal combustion engines is either a combustible gas or a combustible vapor, no matter what the fuel may have been from which it was derived. This implies gasification of the solid, and vaporization of the liquid, fuels.

As already pointed out above, the solid fuels cannot be employed in their natural state. From coal we derive by distillation illuminating gas, and from coal and sometimes other materials, as wood, refuse, etc., by gasification, the various classes of producer or power gas.

Illuminating gas will be further considered under the head of gas fuels.

1. The Conversion of the Solid Fuels to Gas: Producer Gases.

— Gasification of solid fuel differs from distillation in the fact that the process is carried one step further, *i.e.*, not only are the gases, if any, driven off from the fuel, but the carbon itself is gasified, leaving behind nothing but ash.

The fundamental principle of all producer-gas processes is

therefore, first, dry distillation of the fuel, and, second, the conversion of the solid carbon into a combustible gas, which can only be carbon monoxide. If the producer gas is found to contain other gases than those mentioned, it can only be due to changes in the process, unavoidable or otherwise.

Producer practice may be carried on in the following ways:

1. No steam or water introduced with the air, resulting in air gas.

2. Producer blown up with air for one period, then blown with steam alone. Product during first stage is air gas, during the second, water gas.

3. Producer furnished with air carrying a certain quantity of water vapor. Product is ordinary producer gas, Dowson gas, etc.

The first of these is seldom employed on account of limitations pointed out below. The second and third introduce modifications into the simple air-gas process owing to the presence of water or steam.

AIR GAS. — Considering the case of the gasification of carbon alone, resulting in the production of the so-called air gas, assume the combustion of C to CO complete; we then have

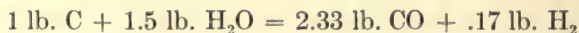
$$1 \text{ lb. of C} + 1.33 \text{ lb. O} = 2.33 \text{ lb. CO}$$

If C had been burned completely to CO₂, the calorific power would have been 14647 B. T. U.; burning only to CO, however, we obtain only 4429 B. T. U., so that the remainder or 10218 B. T. U. is carried out of the producer by the gas made, and thus represents its heat energy. 4429 B. T. U. appears as sensible heat in the gas, and if the gas is cooled before entering the engine cylinder, as it usually is for good reasons, the greatest possible efficiency which can be realized from the gasification of 1 pound of carbon in this way is $\frac{10218}{14647} = 69.6$ per cent. It will be shown below that this is by no means the maximum possible producer efficiency.

WATER GAS. — When water vapor is led through or over incandescent carbon the following reactions take place:



I occurs at temperatures less than 1250 degrees Fahrenheit, while II alone occurs at temperatures exceeding 1800 degrees Fahrenheit; both may occur between these temperature limits, but the higher the temperature the greater the formation of CO. The maximum amount of CO is of course the end in view, and assuming that no CO₂ is formed, *i.e.*, temperature at or above 1800 degrees Fahrenheit, we have the following quantitative statement:



from which 1 pound of water gas must contain

$$\frac{2.33}{2.33 + .17} = .932 \text{ lb. Carbon monoxide}$$

and

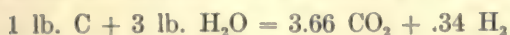
$$\frac{.17}{2.33 + .17} = .068 \text{ lb. Hydrogen}$$

The gasification of 1 pound of carbon, therefore, in the presence of water vapor results in products which, on complete combustion, develop the following amount of heat:

2.33 lbs. CO	×	4380	=	10205 B. T. U.
.17 lbs. H ₂	×	62100	=	10557 B. T. U.
Total,				<u>20762 B. T. U.</u>
Heating value of C to CO ₂ ,				<u>14647 B. T. U.</u>
Excess,				<u>6115 B. T. U.</u>

The excess of 6115 B. T. U. can only be due to heat rendered latent during the process. Water vapor on coming in contact with incandescent carbon dissociates into H₂ and O. The latter unites with C to form CO₂, but as the temperature of the producer is at or over 1800 degrees Fahrenheit, CO₂ is dissociated to CO. The heat thus rendered latent accounts for the excess above shown. Now it is evident that this heat can only come from the stock of heat present in the producer when the blowing with steam first starts. Hence there must be a continual cooling of the producer contents during the period of water-gas making. This results finally in a serious production of CO₂ according to reaction I, when the steam must be shut off and the contents of the producer brought back to incandescence by blowing with air.

The most unfavorable operation of the producer occurs when the reactions are according to equation I. Under this condition



so that 1 pound of water gas then contains

$$\frac{3.66}{3.66 + .34} = .915 \text{ lb. CO}_2$$

and

$$\frac{.34}{3.66 + .34} = .085 \text{ lb. Hydrogen}$$

The production of water gas reckoned on the basis of coal or carbon is not at all efficient, since the heat in the poor gas made during the blowing-up period is very often wasted, and only in rare instances of utility. Other losses are, of course, those through incomplete combustion, radiation, etc.; but these are inherent in all producers to a greater or lesser extent.

PRODUCER GAS. — Midway between air gas and water gas we find the great class of power gases for the production of which the producer is blown continuously with a mixture of air and water vapor.

To get a fair insight into the working of a power gas producer and of the efficiencies that may be realized, we will assume the following definitions and quantities.*

1. The heat supplied to a producer consists of the heat furnished to it in the fuel plus the heat contained in steam and air above a certain fixed temperature, say 32 degrees Fahrenheit.

2. The heat leaving the producer in the gas is made up of the latent heat of the gas plus the sensible heat. What the quantity of heat considered as the useful effect in efficiency should be depends upon circumstances. In furnace work, where it may be of advantage to employ the hot gas, the useful effect would be the sum of the latent and sensible heats of the gas. In gas-engine practice, on the contrary, the opposite is the case, and the useful effect would be the latent heat only. In the first case we speak of the *hot-gas efficiency*, in the second of the *cold-gas efficiency*.

* Adapted from the discussion of E. Meyer, *Zeitschrift des Vereins deutscher Ingenieure*, 1895, p. 1523.

3. Inside of the producer the following reactions take place, some endothermic, others exothermic.

Of every pound of carbon the larger part burns to CO, the remainder to CO₂. The heat generated from these two combustions is utilized in the following ways: Part of it dissociates the steam present, forming H, and CO or CO₂, or both. How this action varies with the temperature of the producer has already been pointed out. A second part of the heat serves to bring the fresh fuel up to the temperature of the producer, a third is lost by radiation from the exterior producer walls, and the remainder appears as sensible heat in the gas made.

The formulæ to be derived will be based upon one pound of C rather than upon one pound of coal, for the reason that coals vary greatly in composition, and it is in every case quite easy to change from this basis to that of coal if the qualities of the coal be known. Let

14500 B. T. U. = heat of combustion of 1 lb. of C to 3.66 lb. CO₂.

4400 B. T. U. = heat of combustion of 1 lb. C to 2.33 lbs. CO.

10100 B. T. U. = heat of combustion of 2.33 lb. CO to 3.66 lb. CO₂

6900 B. T. U. = $\frac{62100}{9}$ = heat required to dissociate 1 lb. of

water vapor under producer conditions.

x = part of 1 lb. of C burning to CO₂.

$(1 - x)$ = part of 1 lb. of C burning to CO.

y = pounds of steam introduced per lb. of C.

A = heat furnished in steam.

B = heat furnished in air.

C = heat required to bring fresh fuel to temperature of producer.

R = heat lost by radiation.

S = sensible heat of the gas.

All of the above heat quantities are per pound of carbon gasified, and above a temperature of say 32 degrees Fahrenheit.

With this notation the general heat equation for the producer may be stated as follows, based on 1 lb. of carbon:

$$4400 (1 - x) + 14500 x + A + B = 6900 y + S + R + C$$

The heat that will be generated by the combustion of the volume of gas formed comes from CO and H.

Heat generated in gas per pound of C gasified

$$= [10100 (1 - x) + 6900 y.] \text{ B. T. U.}$$

6900 y for the heat generated by H is obtained by considering that we must receive as much heat from the combustion of the H in the gas as was rendered latent during dissociation of the amount of H_2O required to furnish it.

Heat supplied per pound of C gasified = $14500 + A + B$.

Hence

$$\text{Cold-gas efficiency} = \frac{10100 (1 - x) + 6900 y}{14500 + A + B}$$

$$\text{Hot-gas efficiency} = \frac{10100 (1 - x) + 6900 y + S}{14500 + A + B}$$

In the latter case, no part of the sensible heat of the gas is abstracted before the gas is used. In practice there is always some loss of temperature between the producer and the place where the gas is used, hence S is never fully obtained.

If in the above general heat equation the values of x , A , B , C , R and S are known, the value of y , i.e., the pounds of steam to be used per pound of carbon, may be computed. The composition of the resulting producer gas may be computed as follows, using the above notation:

COMPOSITION BY WEIGHT PER POUND OF CARBON GASIFIED.

$$\text{CO} = (1 - x) \frac{28}{12} = 2.33 (1 - x) \text{ pounds}$$

$$\text{CO}_2 = \frac{44}{12} x = 3.66 x \text{ pounds}$$

$$\text{H} = \frac{2}{18} y = \frac{y}{9} \text{ pounds}$$

The amount of N is found as follows:

$$\text{Oxygen required for CO} = (1 - x) \frac{16}{12} = 1.33 (1 - x) \text{ pounds}$$

$$\text{Oxygen required for CO}_2 = \frac{32}{12} x = 2.66 x \text{ pounds}$$

Oxygen produced by dissociation of H_2O .

$$= \frac{16}{18} y = \frac{8}{9} y \text{ pounds}$$

$\therefore$ O required from air blast

$$= 1.33 (1 - x) + 2.66 x - \frac{8}{9} y \text{ pounds}$$

$$= 1.33 (1 + x) - \frac{8}{9} y \text{ pounds}$$

$$\text{Hence air required} = \left[1.33 (1 + x) - \frac{8}{9} y \right] \frac{100}{23.5} \text{ pounds}$$

Hence also N brought in by air blast is

$$N = \left[1.33 (1 + x) - \frac{8}{9} y \right] \frac{76.5}{23.5} \text{ pounds}$$

The total weight of gas produced by one pound of carbon therefore is

$$\begin{aligned} &= 2.33 (1 - x) + 3.66 x + \frac{y}{9} + \left[1.33 (1 + x) - \frac{8}{9} y \right] \frac{76.5}{23.5} \text{ pounds} \\ &= (6.67 + 5.67 x - 2.77 y) \text{ pounds of producer gas.} \end{aligned}$$

COMPOSITION BY VOLUME PER POUND OF CARBON GASIFIED.

Volumes at 32 degrees and 14.7 lb. pressure.

Weight per cubic foot of the various gas under standard conditions.

$$\text{CO} = .07807 \text{ lb.}, \text{CO}_2 = .12267 \text{ lb.}, \text{H} = .00559, \text{N} = .07831.$$

Hence, from the above weight computations, we may write directly.

$$\text{Volume of CO} = \frac{2.33 x}{.07807} (1 - x) = 29.84 (1 - x) \text{ cu. ft.}$$

$$\text{Volume of CO}_2 = \frac{3.66}{.12267} x = 29.02 x \text{ cu. ft.}$$

$$\text{Volume of H} = \frac{y}{9 \times .00559} = 19.87 y \text{ cu. ft.}$$

$$\begin{aligned} \text{Volume of N} &= \left[1.33 (1 - x) - \frac{8}{9} y \right] \frac{76.5}{23.5 \times .07831} \\ &= (55.51 + 41.74 x - 37.10 y) \text{ cubic feet.} \end{aligned}$$

From this

Total volume of gas per pound of C gasified

$$= [29.84(1 - x) + 29.02 x + 19.87 y + 55.51 + 41.74 x - 37.10 y] \text{ cubic feet.}$$

$$= (85.35 + 40.92 x - 17.23 y) \text{ cubic feet.}$$

2. Theoretical Yield of Producer. — If it is supposed that no CO_2 is formed, that all the sensible heat produced in the generator is recovered in making steam and preheating air and fresh fuel, and that no radiation has taken place, we shall have

$$x \text{ and } R = 0, \text{ and } S = A + B.$$

Under these theoretical conditions, the general heat equation (p. 151) then becomes

$$4400 = 6900 y + C.$$

The value of C is approximately $.2 \times 1800 = 360$ B. T. U. Hence

$$4140 = 6900 y,$$

from which $y = \frac{4140}{6900} = .600$ pounds of steam per pound of C gasified.

The theoretical yield of gas per pound of carbon under these conditions will be

$$29.84 \text{ cubic feet of CO.}$$

$$19.87 \times .600 = 11.92 \text{ cubic feet of H}$$

and

$$55.51 - 37.10 \times .600 = 33.25 \text{ cubic feet of N.}$$

Total volume of yield = 75.01 cubic feet per pound of C .

Composition of gas, per cent by volume,

$$39.8 \text{ per cent CO, } 15.9 \text{ per cent H, } 44.3 \text{ per cent N.}$$

Taking the higher heating value of H at 346 B. T. U. per cubic foot, and the heating value of CO at 342 B. T. U. per cubic foot, the gas yield from 1 pound of C will develop

$$(29.84 \times 342) + (11.92 \times 346) = 14249 \text{ B. T. U.}$$

The higher heating value per cubic foot of this theoretical gas is therefore

$$\frac{14249}{75.01} = 187.2 \text{ B. T. U. per cu. ft.}$$

In actual practice, however, x and R cannot = 0, neither is $S = A + B$; i.e., not all of the heat appearing as sensible heat in the gas leaving the producer is ever recovered. To

make clear what happens under these circumstances, the following table is constructed. In this table it is assumed that the heat furnished in steam and air per pound of carbon equals 1000 B. T. U. = $(A + B)$ in all cases, and that the sum of the heat losses due to radiation, R , and sensible heat of the gas, S , = 1640 B. T. U. per pound of carbon.

The heat furnished the producer per pound of C will then in all cases be $14500 + 1000 = 15500$ B. T. U., while the heat accounted for will be $15500 - (1640 + 360) = 13500$ B. T. U., so that the generator efficiency on hot gas in all cases = $\frac{13500}{15500} = 87.2$ per cent. It is further assumed for illustration that only the amount of carbon burned to CO_2 per pound of carbon gasified varies.

% of Carbon Burned to CO ₂	Steam used lbs.	Cubic feet of gas per pound of C (Gasified)					Composition, % by volume				Lower Heating Value per cu. ft. at 32° and atmo. press.
		CO ₂	CO	H	N	Total cu. ft. lbs.	CO ₂	CO	H	N	
0	.493	0	29.84	9.80	37.22	76.86	0	38.9	12.6	48.5	170.0
10	.639	2.90	26.85	12.70	35.94	78.39	3.7	34.2	16.1	46.0	164.7
20	.783	5.80	23.90	15.55	34.84	80.10	7.2	29.9	19.4	43.5	159.7
30	.932	8.70	20.90	18.50	33.51	81.60	10.7	25.6	22.7	41.0	155.0
40	1.079	11.60	17.90	21.45	32.18	83.13	13.6	21.5	25.8	39.1	150.2

In commenting upon the above table, Meyer points out that as the percentage of CO_2 in the gas increases, the heating value of the gas decreases, but since at the same time the percentage of H and the volume of gas per pound of C increase, it is not always right to conclude from analysis alone that the efficiency of the generator is less with a fairly high than with a low percentage of CO_2 in the gas. In general terms it can be stated, however, that the lower the temperature of the generator, the greater the formation of CO_2 , and the greater also the loss of heat in sensible heat of the resulting gas.

Referring to the gas engine itself, a high percentage of CO_2 in the gas means a low engine capacity, since this high percentage is usually also accompanied by an increased amount of the other indifferent gases.

3. Gas Producers in Practice. — Turning now to actual generator practice, we find the following main points of difference: The fuel is not pure carbon, but some impure form of it, as coal, coke, lignite, peat, or wood. This in itself merely results in a lower yield of gas per pound of fuel fired than that above computed, and this decrease is further emphasized by the fact that some of the unburned carbon in the fuel is always lost in the ash. A complex fuel being used containing gases which are distilled off during the first part of the process, the resulting producer gas will have a somewhat different composition than that above computed. The main difference is due to the addition of hydrocarbons, and this difference is therefore greater with bituminous coals than with any of the other fuels. The use of any of the above-mentioned fuels results also in other complications more or less difficult, depending upon the fuel used. Such are the formation of tar, dust carried by the gas, etc., all of which make a cleaning of the gas imperative before it can be used.

The primary consideration in the operation of the gas producers is perhaps the kind of fuel used. The points to be considered in this connection are: percentage of water carried by fuel, amount and kind of ash, tar-forming ingredients of the fuel, size of fuel, and whether it cokes or not.

A high percentage of water has a direct effect in lowering the temperature of the producer, besides lowering the heating value

of the gas per cubic foot as made. A large amount of ash makes more frequent cleaning out necessary, or it is likely to result in a partial stoppage of the air supply. If the ash should be easily fusible the case is much more complicated, as this results in bad clinkering. The size of the fuel should be a happy medium. Small fuel, *i.e.*, screenings, etc., clog up easily and in any case require a higher blast pressure. Large fuel, on the other hand, offers too little surface for gasification and is apt to let much water and CO_2 escape unreduced. A coking coal nearly always gives trouble from this cause, and it necessitates constant breaking up of the charge.

The formation of tar, which results especially when bituminous coals are gasified, makes a cleaning of the gas for engine purposes indispensable. Tar results when some of the hydrocarbon gases are condensed through cooling in the gas mains and pipes. If these gases reach the cylinder their combustion is likely to result in a strong deposit of soot. In either case the operation of the engine will soon be seriously interfered with. Tar can be almost entirely removed from the gas by washing it, but this process requires constructions fully as costly as the producer itself and hence other methods have been employed.

The tar-forming gases are always those which are formed from the dry distillation of the coal, hence most trouble is encountered with bituminous coal, less with lignite and still less with anthracite. For this reason anthracite and coke producers have been most successful, although producers using brown coals and lignites are in operation, as are also those using bituminous coal, but with less success. This does not apply to steel works where bituminous coal is used extensively for gasification. But there the gas is used mostly hot and less trouble from tar is experienced.

The tar gases can be "fixed," *i.e.*, changed to permanent gases when the producer gas containing them is led through an incandescent bed of fuel before entering the gas mains. In this case the tarry hydrocarbons are changed either to H_2O and CO_2 or split up into CO and H . In some producers only the gases resulting from the dry distillation are handled in this way. In either case the tarry hydrocarbons are fixed, and no elaborate cleaning apparatus for the gas is required. The necessity for

treating bituminous producer gas in this way has resulted in various constructions of producer, a few of which are given below.

Gas producer installations may be divided into three classes:

a. Pressure Producers.— In these air and steam are furnished to the fuel bed by a blower or fan. The ash pit of the producer must be enclosed, making the removal of ash complicated or the action of the producer intermittent, unless the water-bottom type is used. Steam for blowing is usually furnished by a separate boiler. Since the rate of production of gas is usually not regulated according to the demand for gas directly, a gas holder is usually necessary for this type.

b. Suction Producers.— In this class the air and steam are drawn through the producer by the suction of the engine cylinder. The production of gas is thus directly regulated by the demand. The ash pit remains open, and steam enough can usually be generated by the sensible heat of the gas.

Suction gas producers have nearly replaced pressure producers for gas-engine purposes. Some of the obvious advantages, as open ash pit, absence of separate boiler and of gas holder, have been pointed out above. The dangers at first supposed to be inherent in this system have failed to materialize. Leaks in a pressure system may lead to a poisoning of the atmospheric air by the color- and odorless CO, positively dangerous to attendants. Leaks in a suction system only result in an in-leakage of air. That this can never happen to such an extent as to form an explosive mixture, except through a combination of extraordinary circumstances, is at once evident when we consider that the ratio of air to producer gas for such a mixture would have to be at least 1 to 1.

In spite of such advantages the suction system is by no means perfect. The regulation of the water supply to control the amount of H in the gas, the vaporizer for the water, and the cleaning apparatus are still points which admit of improvement even in the most recent form.

c. Combination Producers.— The air and steam mixture is drawn through the producer by a fan and the resulting gas forced by the same fan to the engine. The producer in this system is thus of the suction type.

PRESSURE PRODUCERS.—*Taylor.* Fig. 7-1 shows the Taylor producer made by R. D. Wood & Co. of Philadelphia. In their publications on the producer this company lays down the following requirements for a successful pressure producer. Most of these, however, apply to producers in general.

1. A continuous and automatic feed; the former for regularity and uniformity of gas production with improved quality, the latter for eliminating negligence of attendants.

2. A deep fuel bed carried on a deep bed of ashes; the first to make good gas, and the second to prevent waste of fuel.

3. Blast carried by conduit through the ashes to the incandescent fuel.

4. Visibility of the ashes, and accessibility of the apertures for their removal, arranged so that operator can see what he is doing.

5. Level, grateless support for the burden, insuring uniform depth of fuel at all points, and consequent uniformity in the production of gas.

These points are well covered in the design of the producer. The fuel is admitted through a distributing hopper which keeps the layer of fuel level over the cross-section. The bed of ashes is kept at about 6 inches over the top of the air pipe, thus protecting it from direct heat. The entire charge in the producer is supported by a plate whose diameter is somewhat greater than that at the bosh. As necessity requires, this plate can be revolved and the ashes are scraped, or they fall off, into the closed ash pit, which is under blast pressure. The grinding action ensuing when the plate is revolved settles the contents of the producer and thus closes up any free air channels that may have been formed. Once a day the pit must be opened for the removal of the ash. Blast is supplied generally by a steam jet.

Morgan. Somewhat similar in design is the Morgan producer, Fig. 7-2. The main point of difference is in the removal of the ash. The fuel here is also admitted through a continuous automatic feeding device. The blast is controlled by a steam injector so designed as to maintain a proper proportion between



FIG. 7-1. — Taylor Producer.

air and steam. The make of gas can be completely controlled by the adjustment of a $\frac{3}{4}$ -inch steam valve.



FIG. 7-2. — Morgan Producer.

This producer is of the water-bottom type, *i.e.*, the ashes fall into a water seal at the bottom, and may there be removed without stopping the operation of the producer. This is not easily done when a grate or similar device is used in a pressure producer. For certain fuels, especially those highly bituminous or those where ash is apt to clinker, the water-bottom producers possess some advantage over the others. About three feet above the water level in the ash pan, and some inches above the top of the blast distributing pipe, a number of sight holes are arranged around the circumference of the producer. Through these the zone of combustion may be watched. The top of the producer is covered with a shallow water pan. Poke holes through the top with water-sealed covers are also provided.

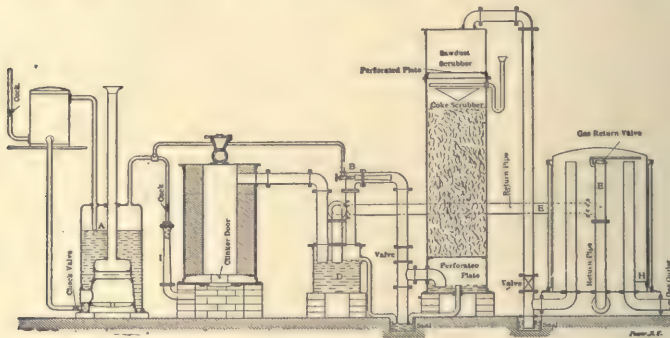


FIG. 7-3. — Wile Producer Installation.

Wile. A complete Wile pressure gas plant is shown in Fig. 7-3.* Steam under about 40 pounds pressure is generated in the boiler, A, and enters the generator through the injector, I, mixed with air. The gas made passes through the seal box, D, and the

* J. I. Wile in Power.

scrubber to the gas holder. This is the ordinary arrangement, but it is open to objection when the load is extremely variable. In such a case it is usual to arrange the gas holder so that it shuts off steam at *I* when the holder is full. The contents of the producer then cool, and a further cooling results when the steam is next turned on. The temperature ranges in the producer are therefore apt to be high under such conditions. To meet this difficulty the design can be changed by placing the steam injector at *B*, above the seal box *D*. The gas holder is connected with the seal box by a return pipe *E*. When the gas holder is up, catch *H* in the gas holder opens the return valve, and the injector *B* merely draws on the gas holder, placing the generator temporarily out of commission. When the gas holder falls, the return valve closes, the injector draws on the generator, and gas is again made. In this design the arrangement at *I* is then merely a saturator, and the plant is really a combination plant, the generator being under suction.

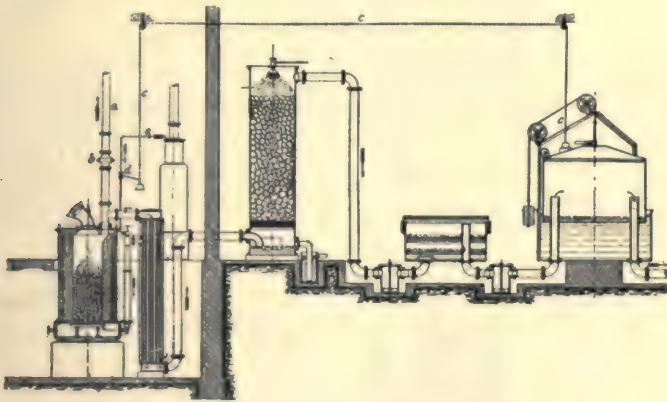


FIG. 7-4. — Koerting Pressure Producer.

*Koerting, Hannover.** Fig. 7-4 shows a Koerting pressure plant. The gases made during the firing-up period escape through the pipe *a*. Should the natural draft not be strong enough, it may be increased by means of a small steam blower in pipe *a*. After the flame at the try cock shows dark red, and not blue, the valve *b* is closed, and the gas made sent through the pre-heater, scrub-

* Guldner, *Entwerfen und Berechnen der Verbrennungsmotoren*, p. 384.

ber, and sawdust purifier to the gas holder. The make of gas is controlled by the gas holder through the chain *C*, which acts upon the blast through a throttle valve at *d*. According to published figures, the average analysis of the gas made is, by volume,

H, 18 per cent; CO, 26 per cent; $C_n H_{2n}$, 2 per cent.

CO₂, 7 per cent; N, 47 per cent; Efficiency, 80–82 per cent.

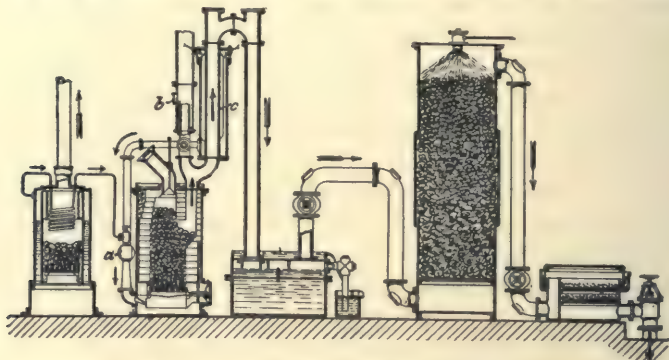


FIG. 7-5. — Deutz Pressure Producer.

*Deutz.** The steam used for the blower *a*, in the Deutz plant, Fig. 7-5, is superheated in a coiled pipe above the fuel in the boiler.

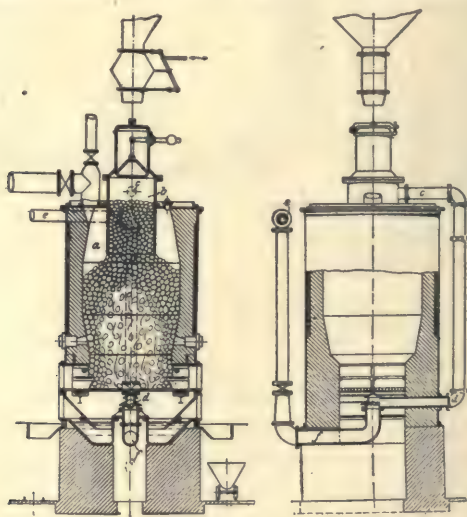


FIG. 7-6. — Poetter Producer.

* Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 384.

The air is pre-heated by the heat of gases in the heater *C*. The opening *b* is used to try the gas during firing up. Arrangements of scrubber and purifier are similar to those already described.

Poetter. The Poetter producer,* Fig. 7-6, is especially designed for bituminous coal. The fuel charged is distilled while still in the hopper, the gases formed are drawn off by special steam blower and are led through the pipe *Cd* under the grate. Air for blast is provided through *e f*. The gas made escapes at *a*, and is first used to raise the steam required in a boiler. It is then led through cooler, scrubber, coke- and sawdust purifier to the gas holder. Schöttler, in describing a plant of Poetter producers at Johannesburg, surmises that their operation might cause trouble, although the kind of coal used at Johannesburg is not stated.

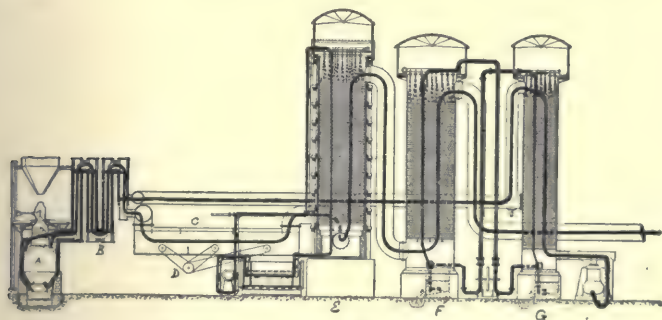


FIG. 7-7. — Mond Producer Plant.

Mond. When a producer is blown with a large excess of steam, a great deal of it will go through undecomposed. At the same time, however, the quality of the gas made undergoes some radical changes. The quantity of H in the gas will be high, sometimes up to 25 per cent, while on account of the low producer temperature, a great deal more of the C is burned to CO_2 than is ordinarily the case. A further, and important, change is that a great deal of the N is changed to ammonia, a reaction which does not take place in producers run under the ordinarily higher temperatures. To recover this ammonia is an important consideration. This is the process of Mond.

A Mond gas plant consists essentially of two parts, the producer and the condensing and recovery plant, Fig. 7-7.

* Rev. Mec., 1904, p. 484.

The details of the producer *A* do not differ much from those already described. It is of the water-bottom type. The gas made passes through the regenerators *B*. These consist of double wrought-iron tubes united alternately top and bottom. The hot gases pass through the inner tubes, heating the mixture of steam and air, used for blowing the producer, which flows through the outer tubes in the opposite direction. The gas next passes through the washer *C*. This is a large chamber partly filled with water. By means of dashers, driven by *D*, the chamber is kept filled with water spray. The gas is cooled considerably in passing through *C*, the water vapor in the gas being condensed to a great extent. Up to this point all of the apparatus is necessary even if no ammonia recovery is attempted.

For a gasification capacity of less than 30 tons of coal in twenty-four hours it is not usual to install a recovery plant on account of the high cost of installation. Beyond this capacity the installation is justified. A recovery plant consists of the acid tower *E*, the gas cooling tower *F*, and the air heating and saturating tower *G* (see Fig. 7-7).

The gas after leaving the condenser *C* enters the tower *E* at the bottom and flows upward through firebrick checker work. In doing so it meets a descending rain of sulfuric acid liquor containing about 4 per cent of free acid, and by this the free ammonia in the ascending gas is fixed, being changed to sulfate. The acid liquor is circulated by a special pump, and kept at the proper strength by drawing off the sulfate liquor and adding a corresponding amount of fresh acid solution from time to time.

The gas then enters the cooling tower *F*, at the bottom, and in its ascent is cooled by descending cold water. The gas gives up its burden of steam, which in turn heats the descending cold water. The gas is then conveyed to the gas mains.

The hot water leaving the gas-cooling tower is pumped to the top of the air-saturating tower. Here it flows downward through checker work, and in its descent saturates and heats the air, which is driven upward through the tower by blowers. Leaving this tower, the air-water vapor mixture then goes to the regenerator *B*, where it is further pre-heated before entering the producer.

Sexton* states that the amount of steam used is about 2½ tons per ton of fuel, and estimates that about two tons of this go through undecomposed.

The method has the advantage that slack coal may be gasified with success. The recovery of ammonia amounts to about 1 ton for 23 tons of coal gasified, or adding the fuel required for making steam, about 1 ton for 28.5 tons of fuel.

The average analysis of Mond gas is, by volume,

11 per cent CO, 17.1 per cent CO₂, 1.8 per cent CH₄, .4 per cent C_nH_{2n}, 27 per cent H, and 42.5 per cent N.

This gas is free from tar and excess of moisture, and burns with a non-luminous flame.

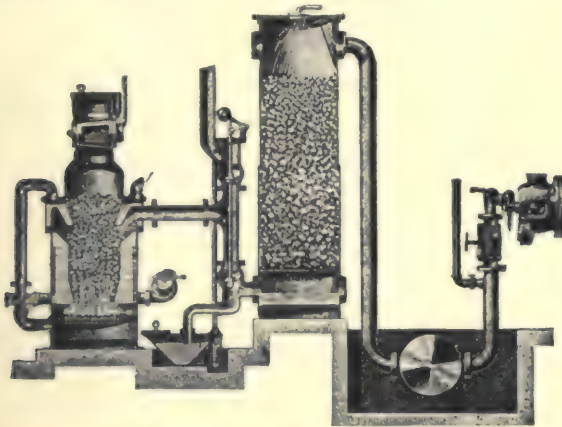


FIG. 7-8. — Deutz Suction Producer.

SUCTION PRODUCERS. — *Deutz.*† In the Deutz suction producer, Fig. 7-8, the vaporizer is arranged around the top of the generator. During the suction stroke of the engine, fresh air enters the vaporizer, is here mixed with water vapor, and then, flowing through the connecting pipe at the left side of the generator, reaches the under side of the grate. The gas made leaves the generator through the pipe at the right and enters the wet scrubber at the bottom. From here it passes to the engine through a

* Sexton, *Producer Gas*, p. 90.

† Güldner, p. 386.

regulator. Just before entering the engine the gas passes a vessel containing a wire brush for a final cleaning of tar. The larger Deutz suction installations show a somewhat different construction.

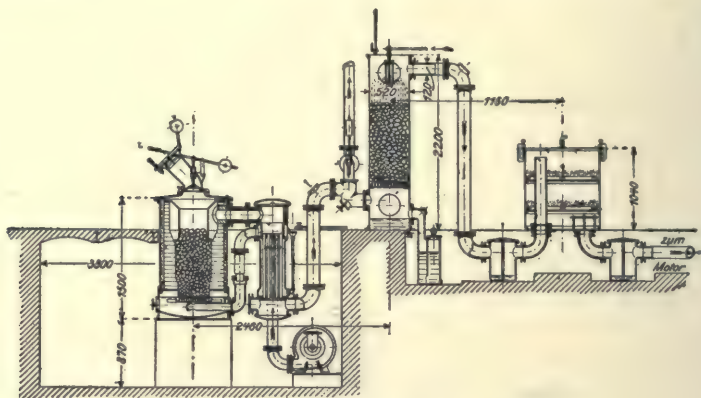


FIG. 7-9. — Koerting Suction Producer.

*Koerting.** Fig. 7-9 shows a Koerting suction plant designed for from 30-35 horse-power. In this case the vaporizer is independent of the generator. The latter is heated by the hot gases on their way to scrubber and purifier. During the heating-up period, the air is furnished by the blower shown. The smoke escapes through a pipe which branches off beyond the vaporizer, so that this will be heated during the blowing-up period.

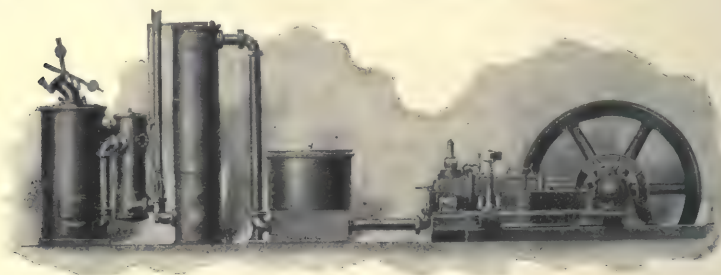


FIG. 7-10. — Koerting Installation.

A complete view of a Koerting plant is shown in Fig. 7-10.

* Güldner, p. 386.

American Crossley. The American Crossley suction gas plants in this country are built by the Power & Mining Machinery Company. Fig. 7-11 shows a complete plant. The producer is of conventional design. The fuel bed rests on a shaking grate. The fuel is admitted as shown. The waste heat boiler or saturator surrounds the feed tube. The water level in this saturator is automatically regulated. During the firing-up period the producer is blown by means of the fan shown, the gases escaping by the purge pipe. A new idea seems to be the saturating of only part of the air supply. The makers point out the difficulty of

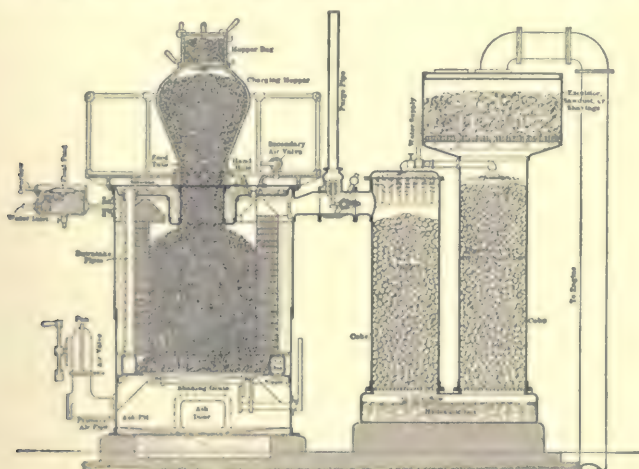


FIG. 7-11. — American Crossley Producer.

making uniform gas at all, especially at low, loads, owing to the difficulty of controlling the amount of steam admitted. To remedy this, part of the air only passes through the saturator, while the rest is directly admitted to the ash pit. Valves on these two air inlets can be set so as to give satisfactory gas over wide ranges of load.

Another arrangement in which this producer plant differs from most others is that of the cleaning apparatus. This consists of a wet scrubber, an hydraulic box, and a combination wet and dry scrubber. The gas enters the first and passes downward, passing upward in the combination scrubber. The operation is shown plainly in the cut.

Fairbanks-Morse. Fig. 7-12 shows a Fairbanks-Morse installation for 21 horse-power, the producer itself being shown in Fig. 7-13, in greater detail. The general design of the producer is very

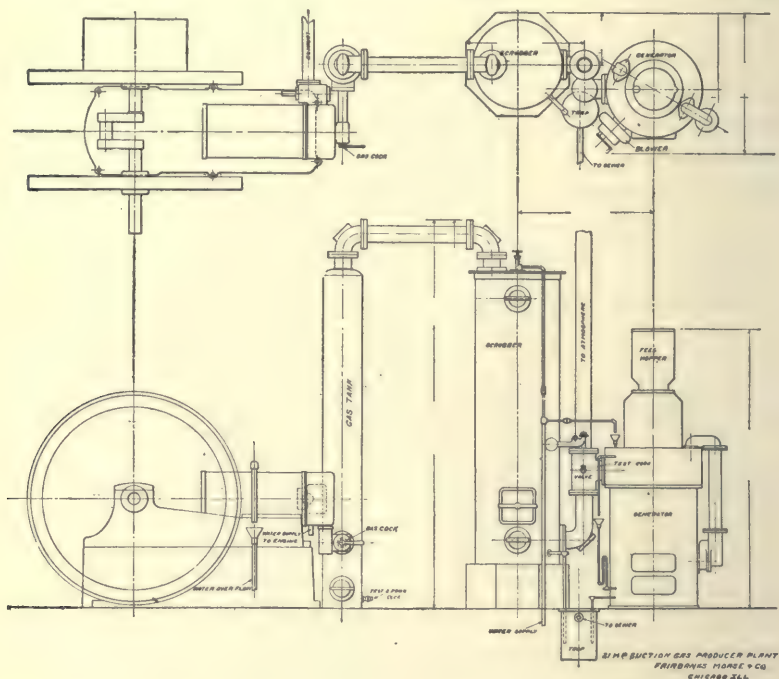


FIG. 7-12. — Fairbanks-Morse Installation.

similar to that of the American-Crossley, except that all of the air passes through the vaporizer. Apparently only a wet scrubber is provided, it being evidently intended to use anthracite only. A gas tank of considerable volume is interposed between scrubber and engine.

The suction producers so far described have been for anthracite, coke, or charcoal. The desire to utilize soft coal, lignites, etc., has led to special designs, of which the following are a few examples. In some of these all of the gas made from the fuel is passed through other highly incandescent but gas-free fuel to fix the tar-forming gases, in others only the gases of distillation are so treated.

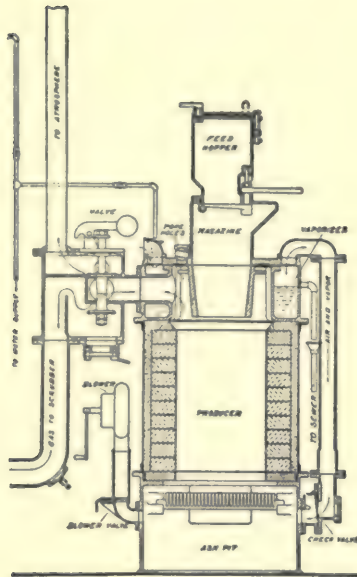


FIG. 7-13. — Fairbanks' Producer.

Riche. * Riche's producer, Fig. 7-14, is especially designed for wood, but it may also be used for hard coal and coke. The

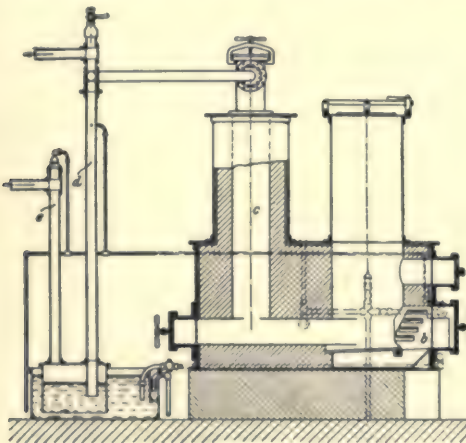


FIG. 7-14. — Riche's Producer.

* Génie Civil, 1901-2, p. 398.

column at the right serves as a magazine for the fuel bed at *b*. The gases formed are sucked up through *C*, which is kept filled with incandescent charcoal. The fixed gas passes through pipes *d* and *e*, being washed by means of sprayed water in its passage.

Lencauchez.^{*} Instead of using two separate stacks, the two fuels may be put together in one furnace, as is done by *Lencauchez*, Fig. 7-15. Long-flaming bituminous coal is charged through *a*, while through *b* coke from coke ovens or gas works is charged. Water is evaporated in the ash pit. The openings *c* and *d* show only black smoke, while *e* shows a colorless gas. The gases from the soft coal are fixed in passing through the incandescent coke layer. *Schöttler* mentions that this gas is not used as power

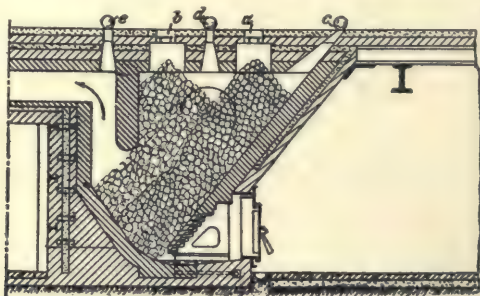


FIG. 7-15. — *Lencauchez Producer*.

gas but for heating purposes. The proportions of the charge are about 20-25 per cent coke to 80-75 per cent soft coal.

Lencauchez.[†] Another way of reaching the same end, that is, fixing the tarry gases, is by the use of so-called double generators. In these generators the charge burns both top and bottom, the gases being drawn off at about the middle of the producer. Fig. 7-16 shows a generator of this type designed by *Lencauchez*. It consists essentially of three conical parts. The upper one serves as a magazine. In the middle one the fire burns on top, in the bottom one at the bottom. Air enters at *D* and *B*. Water is introduced at *A* and evaporated in the ash pit. The air and gas currents are shown by the arrows. The gases of distillation formed in the magazine are drawn downward through the

^{*} *Zeitschrift des Vereins deutscher Ingenieure*, 1905, p. 1903.

[†] *Zeitschrift des Vereins deutscher Ingenieure*, 1905, p. 1905

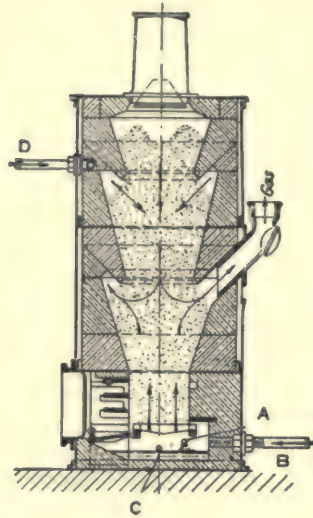


FIG. 7-16. — Lencauchez Double Zone Generator.

incandescent fuel in the middle section and are so fixed. Any unburned coked fuel then passes on toward the grate and is gasified on reaching the incandescent zone in the lower section.

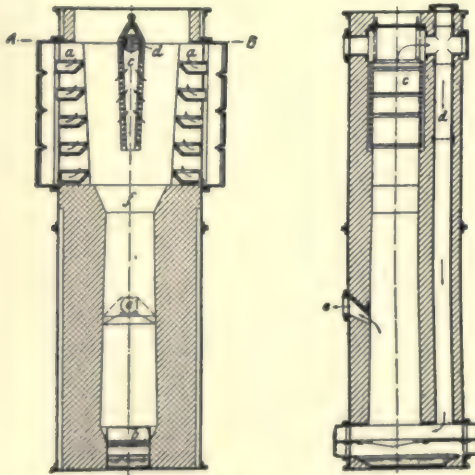


FIG. 7-17. — Koerting Producer for Peat.

Koerting. * Koerting's producer for peat, Fig. 7-17, is built along lines similar to the above. The producer consists of two parts, of which the upper and larger one is fitted with step grates, *a a*, the lower one with an ordinary grate *b*. When in operation, the charge of peat in the upper parts burns only near the grates *a a*, while the inner core of peat is only coked. The gases so formed pass into the perforated pipe *c* and are led through *d* under the grate *b*. The coked peat formed in the upper part passes downward and is gasified over the grate *b*. The gases from *d* pass through the incandescent layer above *b*, are fixed, and together with the gases formed over *b* pass out at *e*. One would naturally assume at first sight that the gases of distillation would pass directly downward and out at *e*, instead of entering *c* and passing through *d*. The direction of the movement is regulated by creating a slight vacuum under the grate *b*, and by contracting the producer section at *f*. This contraction produces an extra resistance to the passage of gases downward, and they consequently take the more convenient way through *c* and *d*. Schöttler gives the analysis of a peat gas so made as

14 per cent CO_2 , 4 per cent CH_4 , 15 per cent CO , 10 per cent H ,
and 57 per cent N ,
producer efficiency being about 75 per cent.

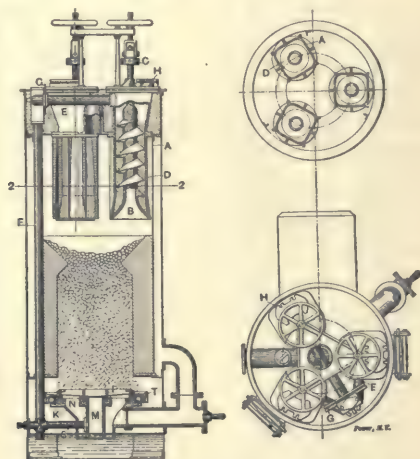


FIG. 7-18. — Crossley Producer.

* Zeitschrift des Vereins deutscher Ingenieure, 1905, p. 1909.

COMBINATION SYSTEMS. — *Crossley*. The Crossley producer, Fig. 7-18, is suitable for bituminous coal, or other fuel high in volatile matter. The fuel is first charged into the retorts *A*, and is there distilled by the sensible heat of the gases formed below. When charging, the valve *B* is drawn tightly against its seat, and the fuel is projected downward by turning the spiral *D* by means of the capstan. When the distillation is complete the casting *B* is lowered and the coked fuel is thrown into the main part of the producer by turning *D*. At the same time it is broken up by the spirals. The gases of distillation are drawn out of the retorts through the pipe *F*, and pass under the grate *T*, at *M*. The suction is produced by a fan blower, which at the same time draws air and steam to the main bed of fuel, and discharges the resulting producer gas into a holder at suitable pressure.

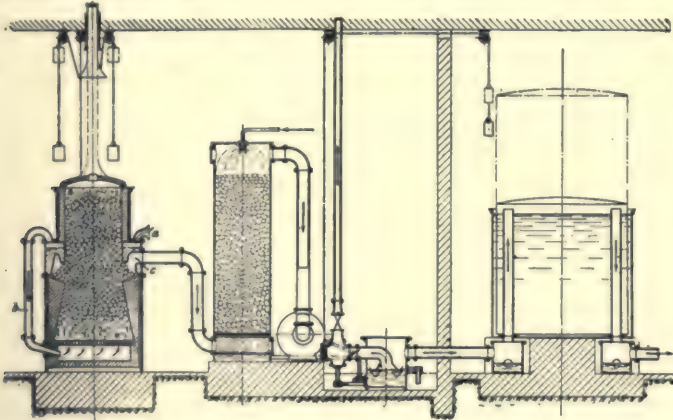


FIG. 7-19. — Deutz Double Zone Producer.

Deutz. The Deutz plant for bituminous coal is shown in Fig. 7-19. The producer is of the double-generator type. The air enters the vaporizer at *a*, and reaches the ash pit through pipe *b*. The fire burns top and bottom. Green coal is charged at the top. It is distilled, the gases formed pass downward through incandescent fuel and are so fixed, passing out at *c*. The greater part of the fuel coked near the top passes downward and is finally gasified in the lower part of the producer, the gases also passing out at *c*. Suction is maintained by a fan connected beyond the coke scrubber. Owing to the fixing of the tarry gases it is found

that a coke scrubber is all that is required. The fan, which is almost a necessary part of such a plant, owing to the increased friction entailed by the design of the producer, delivers the gas to a holder. The holder regulates the make of gas by controlling the fan. During the firing-up period, a purge pipe is lowered over the top of the generator, which has no special charging bell. The same fan is then used as a blower to furnish air to the producer, by-passing the scrubber.

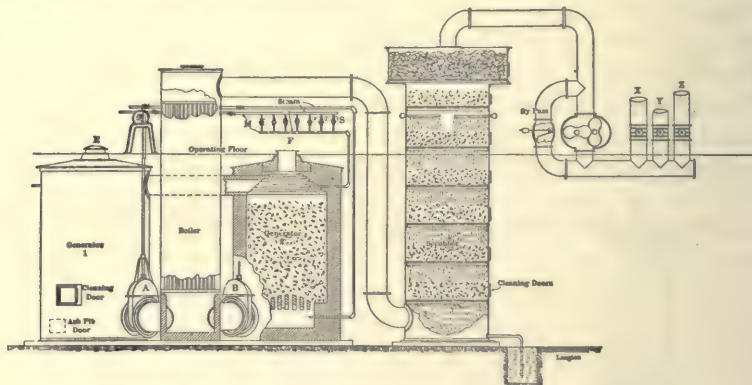


FIG. 7-20. — Loomis-Pettibone Producer.

*Loomis-Pettibone.** The Loomis-Pettibone gas apparatus, Fig. 7-20, may be used for the making of producer gas alone, or for alternate manufacture of water gas and lean power gas. Each plant consists of two producers, a boiler, a combination wet and dry scrubber, and an exhauster which delivers the gas made to the proper holder or the purge pipe, depending upon the positions of the valves *x*, *y* and *z*. The producers are charged through the top. Through the same doors the air also enters. Thus the gases pass downward through incandescent fuel, so that any kind of fuel may be employed. Leaving the producers at the bottom, the gases pass through the boiler, where they are cooled, and then through the scrubber and purifier. The make of gas is controlled by the speed of the exhauster, the suction pressure being ordinarily from 12 to 24 inches, the delivery pressure about 6 inches of water.

Suppose it is desired to make water gas. The feed doors *E*

* The Power Plant of the Montezuma Copper Co., John Langton.

and *F*, and the valve *B*, are closed. Steam is then admitted under the grate of producer No. 2, the right-hand producer in Fig. 7-20. The water gas formed in No. 2 is passed through the top connection to No. 1, passes down No. 1, out at *A*, and thus up the boiler and through the scrubber to gas holder. The next time water gas is run, the directions are reversed, or this may be done when one half of a water-gas run is completed. To make water gas, the aim is to blow the producers up with air as hot as possible, and then pass steam only through them until a great deal of it goes through undecomposed. Then air alone is again used to get the fires hot. This procedure means a wide range of temperature in the producer, and while this does not affect the quality of the water gas seriously, it does affect the producer gas made during the blowing-up periods, as a great deal of CO_2 is formed when the producers are comparatively cool. In a large plant where several producers are at one time delivering to the same gas holder, this irregularity is not much felt at the mixing valve of the engine where the two gases are mixed in proper proportion. For a single set plant this feature of varying producer gas make is more serious, and for this case the producers may be operated in a different manner.

To this end the steam connections, *M*, are used so that steam may be introduced along with the air, thus making water gas and lean gas concurrently, maintaining the producer temperature fairly constant for a long time; in other words, true producer gas is made. To get around the too fine adjustment of air and steam for constant temperature, somewhat less top steam than could be used is employed, thus allowing the temperature to slowly rise. A short water-gas run brings it down again to the proper point. The water gas made is slowly fed from its holder to the main gas holder through a controlled throttle valve so that its rate of feeding approximately equals its rate of production.

4. Some Producer Details. — Mathot, in *The Engineering Magazine*, May, 1905, gives the following data for suction producers:

To get the greatest production of H and the most effective reduction of CO_2 , make the cross-section of the base of the producer from 0.6 to 0.9 of the piston area of the engine. This is for a single cylinder 4-cycle engine at from 600-800 feet piston

speed. The depth of fuel bed should be from 3 to 5 times the diameter at the base, for $\frac{1}{2}$ -inch to $\frac{3}{4}$ -inch lump coal. Amount of water dissociated is from 0.8 to 1.2 times the weight of coal consumed.

More than twenty analyses give the following average figures: CO, 24 per cent; CO₂, 5 per cent; H, 17 per cent; N, 54 per cent. Calorific power from 135 to 150 B. T. U. per cubic foot. Average coal used shows 89 per cent C, 2 per cent H, 4 per cent O + N, and 5 per cent ash. Best size coal is from $\frac{1}{2}$ -inch to $1\frac{1}{4}$ -inch lump with 8 to 10 per cent of ash, and not more than 8 to 10 per cent volatile matter.

Water in the ash pan is satisfactory for steam production if the air is pre-heated. But preference is now given to internally heated vaporizers. Tubular vaporizers produce sufficient steam after 10 to 13 minutes. With well-designed apparatus, any suction plant should be in operation 25 minutes after lighting up.

The volume of the scrubber should be from 6 to 8 times that of the producer. Its height should be from 3 to 4 times its diameter. The filling is coke in pieces 3 to 4 inches in diameter, the coarser pieces near the bottom, the smaller near the top. Amount of water used for washing is from 3 to 5 gallons per horse-power per hour for anthracite gas.

According to the DeLavernne Company, the impurities to be removed from one ton of fuel are, for anthracite, from 1 to 2 pounds of ammonia, traces of sulfur, and from 5 to 10 pounds of tar; for bituminous coal, from 4 to 5 pounds of ammonia, sulfur from traces to 5 per cent, and from 10 to 12 gallons of tar.

The Morgan Construction Company point out the effect of sulfur on the formation of clinker. A coal with a large per cent of ash may work satisfactorily in a producer provided the sulfur does not exceed 1 per cent. Above 3 per cent the effect of sulfur in forming clinker is badly felt unless special facilities are at hand to break up such formations.

The same company makes the following statements regarding pressure-producer capacities. The best rate of combustion seems to be about 10 pounds of bituminous coal per square foot cross-section per hour, the coal carrying 10 per cent of ash and $1\frac{1}{2}$ per cent of sulfur. With coals of lower ash content the rate

may be increased to 12 pounds, and in some special cases to 15 pounds. For poorer coals, however, the rate may sink to 6 or 8 pounds. Regarding the size of gas flues and pipes, the statement is made that each square foot of gas flue will take care of a gasification of 200 pounds of coal per hour, and will serve a gas-making area of 16 square feet in the producer.

CHAPTER VIII

THE GAS-ENGINE FUELS: LIQUID FUELS, CARBURETER AND VAPORIZERS

1. The Crude Oils and their Distillates.—The crude petroleum oils are practically mechanical mixtures of various hydrocarbon families. It is therefore to be expected that crude oils from various fields show compositions considerably different. From these crude oils we obtain by distillation all of the mineral oils now so widely used for a great variety of purposes. Of these oils the various gasolines, kerosenes, and the so-called "distillates," besides crude oil itself, are used for gas-engine fuels.

CRUDE OIL.—The following table* gives a few analyses of crude oils from different parts of the world. These oils are usually of a dark green color, the specific gravity is between 0.80 to 0.90 at 32 degrees, and the flash point between 76 and 93 degrees Fahrenheit.

	Sp. Gravity at 32°	C	H	O and Impuri- ties	Lower Heating Value per pound
Heavy crude W. Virginia	.873	83.5	13.3	3.2	18324
Light crude W. Virginia	.841	84.3	14.1	1.6	18400
Heavy crude Pa.	.886	84.9	13.7	1.4	19210
Light crude Pa.	.826	82.0	14.8	3.2	17930
Heavy crude Ohio	.886	84.9	13.7	1.4	19210
Salt Creek, Wyoming	—	—	—	—	19463
Bothwell, Canada	.857	84.3	13.4	2.3	20410
Lima, Ohio	—	80.2	17.1	2.7	21600
Schwabwiller, Lower Rhine . .	.861	86.2	13.3	.5	18781
Gallicia, East	.872	82.2	12.1	5.7	18010
Gallicia, East	.885	85.3	12.6	2.1	18416
Light Crude, Baku	.884	86.3	13.6	.1	19440
Heavy Crude, Baku	.956	86.6	12.3	1.1	19496
Keudong, Java	—	85.0	11.2	2.8	18330

* Poole, the Calorific Power of Fuels.

GASOLINE. — After the distillation of the very light products from crude oil, we obtain a series of gasolines of varying flash points and gravity. The first of these is 86 degree gasoline, as measured on the Beaumé scale. This forms a mixture with air so readily even at ordinary temperature that it is somewhat dangerous and not often used. The next gasoline, the original liquid gas-engine fuel, is 74-degree gasoline. Owing to the much greater consumption of gasoline, due to the introduction of the automobile mainly, the gravity of gasoline and also the flash point have slowly gone up, until to-day 69-degree gasoline is quite common.

Gasoline vaporizes readily at ordinary room temperatures, and it is therefore necessary to keep it covered, not only to prevent loss but also accidents due to explosions. Insurance companies usually specify that any quantity of it must be kept in an underground tank outside of the building, and that is undoubtedly the best way.

Data on the heating value of gasoline is not at all plentiful. A sample the writer had the opportunity of testing recently gave the following figures:

Specific gravity, Beaumé 69.5 at 60 degrees Fahrenheit.

Composition, 84.76 per cent C, 15.24 per cent H.

Lower heating value as computed from analysis, 20411 B. T. U.

Heating value as determined by Junker's calorimeter, higher value 19606, lower value 18482 B. T. U.

It should be noted that this is one of the instances where the heating value as computed is much higher than the actual value, nearly 2000 B. T. U. = 10 per cent in this case. It shows the necessity of calorimetric determination for accurate work.

KEROSENE. — Kerosene, the next heavier distillate beyond gasolines, is not as extensively used as gasoline in this country for gas engines. It will not form an explosive mixture with air at ordinary temperatures, and therefore requires more elaborate apparatus for the formation of such a mixture. The following data is given by Mr. S. A. Moss for ordinary American kerosene:

Sp. Gr. at 60°	Flash point	C	H	Lower Heating Value
	open	by weight		per lb. B. T. U.
.80	100°	.85	.15	18520

Further reliable data on kerosene is given in a lecture by Diesel in 1897, in connection with Schöttler's test of the Diesel engine.

Specific gravity at 24°C (75.2° F.) .789.

Average composition, C 85.13 per cent.

H 14.21 per cent.

O .66 per cent.

Lower heating value computed from analysis, 19874 B. T. U.

Lower heating value by Junker's calorimeter,
average of five tests, 18242 B. T. U.

DISTILLATES. — A distillation product resembling kerosene in its general properties is sometimes used as fuel. These so-called distillates are not as well refined as kerosene, but are handled the same when used for engine work.

The following table, due to Hofer and transposed from Güldner, shows a good method of presenting the various distillation products from crude oil:

(a) *Volatile Oils.*

SP. GR., .65-.75, FLASH POINT BELOW 70° F., BOILING-POINT
BELOW 300°

	Sp. Gr.	Boiling-Point
Petroleum Ether	.65-.66	95°-122° F.
Benzine	.66-.68	112°-158°
Various Gasolenes	.67-.74	149°-300°

(b) *Illuminating Oils.*

SP. GR., .78-.86, FLASH POINT 70°-158°, BOILING-POINT
ABOVE 300° F.

	Sp. Gr.
American Kerosene	.78 -.81
Russian Kerosene	.82 -.825
Standard White	.808-.812
Prime White	.80 -.806
Astralin	.85 -.86

SP. GR., .86-.96, FLASH POINT 374-482° F.

	Sp. Gr.
Solar Oil	.86-.88
Lubricating and Cylinder Oils.	.88-.90, etc., can be used as lubricating oils only.

Ethyl alcohol,* whose chemical formula is C_2H_6O , may be made in various ways, but the commercial alcohol of to-day is the result of fermentation, generally of grape sugar, in the final stage. The raw materials are various. Thus, according to Sand † they may be divided into three classes:

1. Those containing starch — potatoes, with 15 to 24 per cent starch.
rye, with 50 to 56 per cent starch,
corn, with 60 per cent starch.
2. Those containing sugar — sugar beet, with 8 to 18 per cent sugar,
sugar cane, with 12 to 16 per cent sugar.
3. Those containing alcohol — wine with 9 to 16 per cent alcohol.

† Sand, *Zeitschrift des Vereines deutscher Ingenieure*, 1894, p. 933.

The method of manufacture, of course, varies with the raw material, but need not be described in detail here. Theoretically, 100 pounds of grape sugar should yield 51 pounds of pure alcohol; in reality the yield is from $\frac{1}{3}$ to $\frac{1}{4}$ less than this amount.

A second method of producing alcohol, notably mentioned by Witz in his "Moteurs à Gaz et à Pétrole," is to start with calcium carbide as a raw material. This, by a somewhat complicated process, can be changed from CaC_2 through the stages of C_2H_2 and C_2H_4 to alcohol, $\text{C}_2\text{H}_6\text{O}$. Barium carbide or strontium carbide can be used in the same way. Witz states that from 1 kilogram (2.2 pounds) of calcium carbide 0.8 liter (1.69 pints) of alcohol can be obtained; this is equivalent to 0.096 United States gallons of alcohol from 1 pound of carbide. Estimating the price of carbide at 3 cents per pound, which is even now somewhat below the market price, 1 gallon of alcohol would therefore cost, in raw material alone, 31.2 cents, to say nothing of the cost of the chemical operations. The by-products in the case of the calcium carbide do not amount to much. There is consequently little likelihood that the so-called synthetic or mineral alcohol will ever seriously compete with gasoline or kerosene for power.

The heating value of alcohol cannot be accurately computed from its chemical composition, because nothing definite is known of the arrangement of the atoms entering the composition. We therefore have to depend upon the calorimeter. The figures determined for absolute alcohol by various experimenters are as follows:

	Higher Heating Value per pound	Lower Heating Value per pound
Thomson.....	13310 B. T. U.	12036 B. T. U.
Favre & Silberman	12913 B. T. U.	11664 B. T. U.

The value 11664 B. T. U. is the one most generally used. Absolute alcohol has a specific gravity of 0.7946 at 15 degrees Centigrade (59 degrees Fahrenheit), so that one gallon of pure alcohol weighs 6.625 pounds, and has a lower heating value of 77274 B. T. U.

One pound of C_2H_6O contains 0.522 pound carbon, 0.130 pound hydrogen, and 0.348 pound oxygen.

According to this there will be required for the combustion of one pound of absolute or 100 per cent alcohol,

$$\frac{(0.522 \times 2.66) + (0.130 \times 8) - 0.348}{0.23} = 9 \text{ pounds of air.}$$

This is the equivalent of 111.5 cubic feet of air at 62 degrees Fahrenheit, per pound of C_2H_6O . Commercial alcohol, however, is never pure, but nearly always contains a certain quantity of water, the admixture being measured according to volume per cent. Thus, 90 per cent alcohol means that the mixture carries 10 per cent by volume of water. The heating value of such alcohol is of course correspondingly reduced from that of 100 per cent alcohol according to the following table, due to Schöttler:

Absolute alcohol volume—per cent	Specific gravity	Absolute alcohol weight—per cent	Lower heating value per pound B. T. U.
95	0.805	93.8	10880
90	0.815	87.7	10080
85	0.826	81.8	9360
80	0.836	76.1	8630
75	0.846	70.5	7920
70	0.856	65.0	7200
70	0.856	65.0	7200

It is required by law, in countries where alcohol is now used in the industries, to so fix the fuel that it is rendered undrinkable. This process is called "denaturizing" the alcohol. The bill passed by Congress at the last session provides for the same thing. The materials used for this purpose differ in the various European countries. Some of them try to keep the process a secret, hence some of the information given in the following table * is based upon analyses.

* Zeitschrift des Vereines deutscher Ingenieure, June, 1905.

MATERIALS USED TO DENATURIZE ETHYL ALCOHOL

Country	Sp. Gr. of Denaturized Alcohol at 15° C	Methylene (wood alcohol) and its impurities per cent	Pyridine or Pyridine Bases per cent	Acetone per cent	Benzol per cent	Benzine per cent
France	0.832	7.5	—	2.5	—	0.5
Germany						
Denat. Alcohol ..	0.819	1.5	0.5	0.5	—	—
Motor Alcohol ..	0.825	0.75	0.25	0.25	2.0	—
Austria						
Denat. Alcohol ..	0.835	3.75	0.5	1.25	—	—
Motor Alcohol ..	0.826	0.5	trace	trace	2.5	—
Russia	0.836	10.0	0.5	5.0	—	—
Italy						
Motor Alcohol ..	0.835	6.5	0.65	2.0	1.0	—
Switzerland	0.837	5.0	0.32	2.2	—	—

It will be noted from the table that the material most used for denaturizing ethyl alcohol is wood alcohol. The heating value of the fuel is by the addition of the denaturizing liquid changed but little in most cases.

Benzol, C_6H_6 , besides being used for denaturizing, is sometimes used in larger quantities than indicated in the above table, for the purpose of increasing the heating value of the fuel mixture per pound. Benzol has a specific gravity of 0.866 and a heating value of 17190 B. T. U. per pound. A mixture of x per cent by weight of absolute alcohol with y per cent of benzol, will therefore have a heating value of

$$[11664 x + 17190 y] \text{ B. T. U. per lb.}$$

If the alcohol is not absolute, its proper heating value should be substituted from the table above given. In this way from 10 to 40 per cent of benzol is sometimes employed, thus raising the heating value of the fuel, and at the same time decreasing the specific heat cost, *i.e.*, the cost per heat unit.

There is a second reason why benzol is employed. Under certain circumstances there will be formed acetic acid in the products of combustion of alcohol. This causes rusting of the engine parts. On examination it will be found that this is due to a combustion

with insufficient air supply, and the surest way to prevent rusting, therefore, is to use a good excess of air, and to have a perfect mixture. Under such conditions there will be no danger of corrosion. It is also found that a good addition of benzol acts as an additional safeguard. It should not be forgotten in this connection, however, that the great advantage possessed by alcohol in its odorless exhaust is sacrificed to some extent by the use of benzol.

3. Mixing Devices for Liquid Fuels. — As has been already pointed out, the fuel mixture of a gas engine is always a gas or vapor mixed with air. Hence in the case of liquid fuels special devices are required to convert these fuels into gases or vapor. Such devices are indiscriminately known as carbureters, vaporizers, mixers, or mixing valves. In the strict sense, a carbureter is a device in which the mixture is formed by passing air over or through the liquid fuel. When no special heating of the fuel is done, these devices are applicable to the more volatile fuels only, as gasoline. The name carbureter, however, is also used when in this device gasoline is mechanically atomized or sprayed into the current of incoming air. The term vaporizer is usually employed when considerable heat for gasification or vaporization is required, as is the case for kerosene, crude oil, and alcohol.

MIXING DEVICES FOR GASOLINE: CARBURETERS. — Carbureters in which the air is passed over or through gasoline have the drawback that when the carbureter chamber is filled with fresh gasoline the lighter constituents of the liquid distil off first. The parts harder to vaporize remain behind, so that the fuel mixture is anything but constant, growing leaner as the vaporization progresses.

Atomizing or spraying carbureters are not open to this objection, a definite amount of gasoline being injected each time. Hence they are much used, especially in automobile work.

Figure 8-1 shows a carbureter which is of what has been very aptly termed the bubbling type. The suction stroke of the engine establishes a partial vacuum in *a* through the check valve *c*, and the chamber *d*, interposed for reasons of safety. This causes air to enter through the screen *b*, rising through *c* in a spray. Bubbling up through the gasoline it carries with it some of the vaporized liquid and goes through *d* and *e* to the engine,

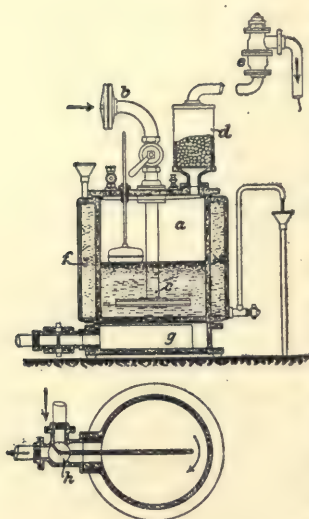


FIG. 8-1.

where a normal combustion mixture is formed by the addition of more air. The jacket water from the engine may be circulated through *f*, assisting in the vaporization. In special cases a part of the exhaust gases may also be circulated through the bottom, *g*.

Figures 8-2 and 8-3 are properly termed surface carbureters. In the former, the Reithman carbureter, the gasoline is fed from the reservoir *a* in drops into the chamber *b*, where it is absorbed by broad suspended wicks. From the surfaces of these the gasoline vaporizes, adding itself to the air which strikes through chamber *b* on every suction stroke. Chamber *b* is filled with clean gravel or wire

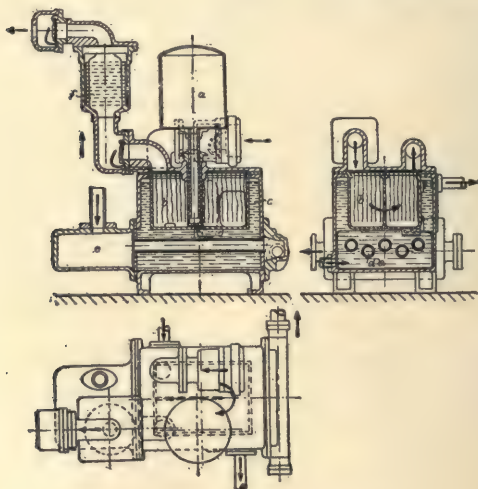


FIG. 8-2 — Reithman Carbureter.

screens, to act as a safety device against possible back firing. Chamber *b* is surrounded by a water jacket, the water in which is heated by the exhaust gases passing through the tubes *d d*.

The Petreano carbureter, Fig. 8-3, will serve for either gasoline or alcohol. The exhaust gases pass through a central pipe, *r*, which is surrounded by a jacket, *V*. The pipe *r* has a covering, *d*, of spongy asbestos film which is constantly kept moist with the liquid to be vaporized. The liquid fuel enters by one opening in the top, the air by another. The chamber *V* has four cones, as shown, two of which are also partially covered with asbestos. By means of these cones the fuel vapor and air are thoroughly mixed before entering the chamber *M* and finally the suction pipe to the engine. The small openings, *o*, are for the purpose of drawing off the heavier parts of the liquid, those that vaporize less easily, if there be any, in order not to interfere with the regularity of carbureting.

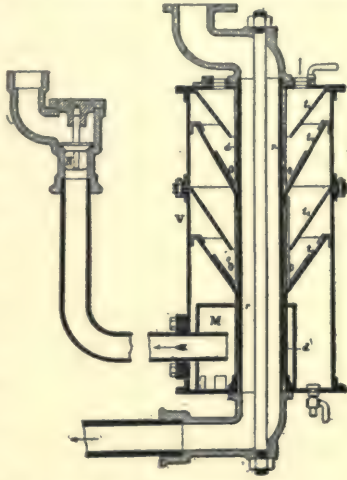


FIG. 8-3. — Petreano Carbureter.

Atomizing or spraying carbureters are, however, much more frequently employed, and the following to be described are all of this type.

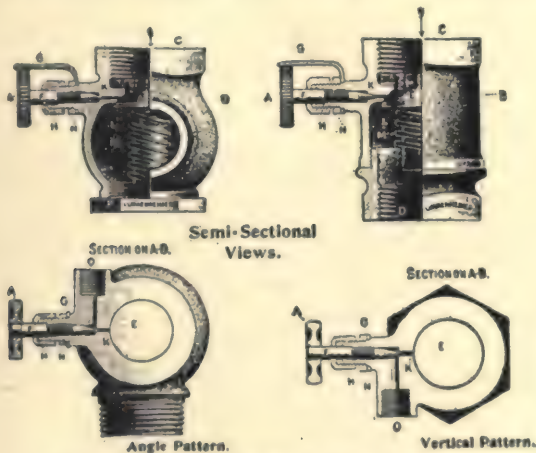
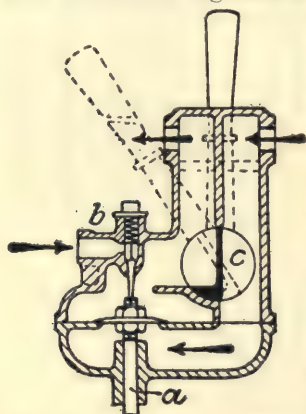


FIG. 8-4. — Lunkenheimer Mixing Valves.

The simplest design of this type of carbureter is shown in Fig. 8-4, which shows two of the Lunkenheimer carbureters. In either design, gasoline is furnished through the opening *O*, and its admission to the valve seat of the main suction valve *E* is regulated by the needle valve *A*. On the suction stroke of the engine the valve *E* automatically lifts, and the air rushing through this opening carries with it a certain amount of gasoline flowing from the small hole *K*. It is quite evident that this type of carbureter gives best service on hit-and-miss engines. For any other type a varying load would probably cause trouble, as there is no automatic regulation of the gasoline supply.



Still of very simple design, but capable of regulation of the gasoline supply, is the Sinto carbureter, Fig. 8-5. The lift of the gasoline needle valve *b* is regulated by the lift of the mechanically regulated suction valve *a*. By means of the cock *c* it is possible to admit part of the air uncarbureted to the cylinder.

Figures 8-6 and 8-7 are examples of the so-called float-feed type of carbureters.

The Daimler, Fig. 8-6, is well known. A float, *B*, in the chamber, *A*, operates a pair of counterweight levers, *E*, and through them the valve spindle, *D*, which controls the admission of gasoline at *C*, keeping it at a constant level in the nozzle. Gasoline enters at *N* from a tank under slight pressure or at a higher level. *O* is a cloth filter and the plug *P* serves to catch any grit that may be brought in. The float chamber is vented through a small hole in the cap over the valve spindle to relieve any pressure which may be formed due to varying positions of the float *B*. The action of the carbureter is as follows: At each charging stroke of the engine air is drawn into the annular chamber *H* and passes with great velocity through the drop tube *F*, surrounding the gasoline nozzle. The gasoline is drawn in a jet from the nozzle and, with the air, striking the deflector *K*, the two are very thoroughly mixed, passing to the engine through *M*. An aux-

iliary air supply can be admitted through the cap at the top, the openings through which can be regulated.

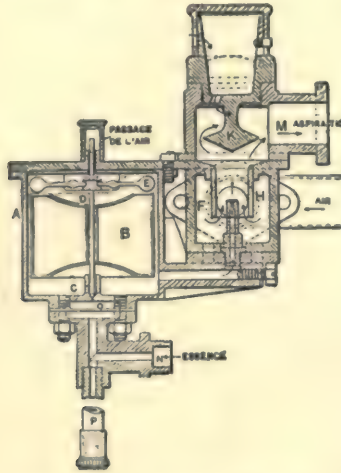


FIG. 8-6. — Daimler Carbureter.

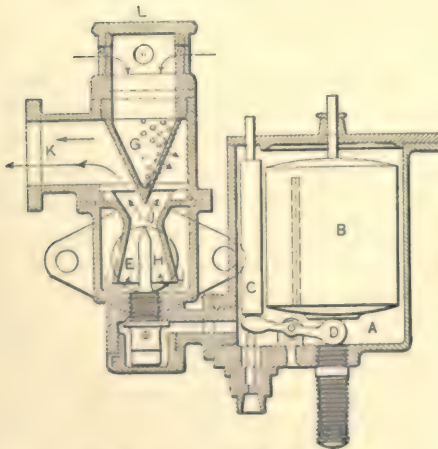


FIG. 8-7. — Abeille Carbureter.

The Abeille carbureter, Fig. 8-7, embodies the same idea as the Daimler. The float *B*, by operating the lever *D*, opens and closes a needle valve at the lower end of the weighted spindle *C*, to maintain a constant level just below the opening in the nozzle

E. On the suction stroke, air enters at *H*, and rushing up through the double cone draws the gasoline from the nozzle, atomizing it by striking the perforated cone *G*. A secondary air supply is admitted through the cap *L* and the holes in *G*, being regulated by the position of the cap *L*.

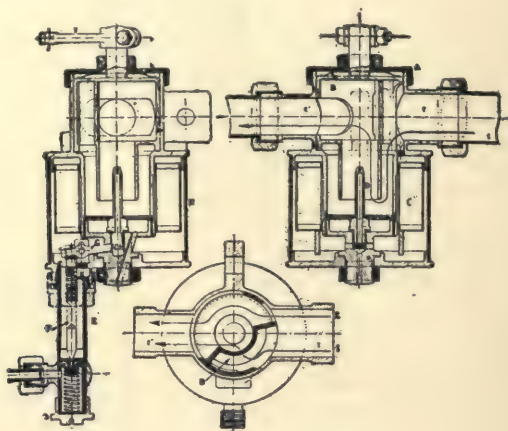


FIG. 8-8.— De Dion Carbureter.

Of more complicated design, but in principle the same as the other float-feed carbureters, is the De Dion, Fig. 8-8. Air is admitted through the tube *t*. By the position of the valve *V* (see horizontal section), which position can be regulated by the lever *I* (see vertical section), part of the air goes directly into the discharge pipe *t'*, while the rest is deflected downward to be carbureted. The course of this latter body of air is indicated in the right vertical section by the arrow. The level of the gasoline in the nozzle *D* is again kept constant by the annular float *C* in the chamber *H*, operating the lever *G* and through it the valve *F*.

It will be noted that none of the atomizing or spraying carbureters for gasoline so far described are heated in any manner. Occasionally, however, we find one in which the atomizing is assisted by heat. The W. Hay vaporizer, Fig. 8-9, is of this type. Gasoline enters the annular chamber, *a a*, through the pipe *d*. From this chamber a number of small openings lead through the seat of the suction valve *E*. Some of these openings are provided with adjusting screws as shown at the left. On