

The advantages and disadvantages of mechanical supercharging are summarized in the list below.

- Advantages:
 - Spontaneous response behavior thanks to immediately available charge air in case of a sudden load change when a reduction of the compressor power can be quickly eliminated at partial load.
 - High boost pressures already available at low engine speeds (limitation: surge limit in the case of the mechanically powered turbo compressor).
 - Response behavior is hardly affected by exhaust gas aftertreatment systems.
 - No changes are required in the exhaust apparatus since only the inlet side is affected.
 - The exhaust back pressure does not influence the gas exchange/lower gas exchange work.
 - No adverse effects on heating and thus on the response behavior of exhaust gas aftertreatment systems (catalytic converters, soot filters etc.) in case of a cold start or unsteady processes (no heat extraction through the turbines).
 - No heating of the fresh gas via heat transfer from the exhaust (as in the case of the turbocharger).
 - No influence on the bearings or materials of the compressor due to exhaust heat.
 - There is no metal mass in the engine room which can heat up and radiate heat to the environment.
- Disadvantages:
 - Specific fuel consumption at partial load without corresponding control can be higher than that of the natural aspirated engine.
 - High costs for design and control is required to lower the compressor power (coupling, speed shifting and valve control).
 - Under certain conditions, response time is delayed because of shifting or positioning delays in case of a sudden load requirement.
 - High costs due to complex charging aggregates, transmissions with sometimes their own belt drives and difficult controls.
 - The smaller engine installation volume with the same power is compensated by the installation space for the charging aggregate and the drive.
 - Higher noise properties in the case of positive displacement chargers due to the unsteady operation.
 - In the case of the mechanically coupled compressor, the interaction of the engine and the charging aggregate also causes a reduction of the boost pressure at low engine speeds and of the useful engine torque due to the high compressor drive power at full engine load and due to the often high leakage losses of the charging aggregate at low speeds.
 - In the case of positive displacement chargers with abrasive sealing gaskets, there are high friction losses and a resultantly low overall efficiency and often a large amount of wear.
 - To obtain small gap widths, a considerable amount of manufacturing cost is required for positive displacement chargers.

In the case of engines with high efficiency requirements (slow and average-speed large diesel engines), the turbocharger has become preferred to the supercharger due to the high boost ratio and the better overall efficiency. For commercial and diesel passenger applications as well, mechanical supercharging has not become widespread despite its advantages in unsteady conditions due to its complexity and lower efficiency. Only in the field of SI passenger cars has mechanical charging been applied in some niche applications without cost pressure or in combination with turbocharging in order to improve unsteady behavior in the case of extreme downsizing concepts. Here too, the overall efficiency of mechanical charging is at most comparable with turbocharging with moderate boost grades.

8.1.3 Turbocharging

In turbocharging, a turbo compressor is powered with an exhaust turbine using energy from the exhaust gas, i.e. the reciprocating engine is only fluidically connected to the turbo machine “turbocharger” (Fig. 8.8). The exhaust gas of the internal combustion engine slows through the exhaust manifold to the turbine and powers it. A compressor sitting on the main shaft of the turbine, which can in fixed operation convert the drive power of the turbine into compressor power (minus bearing friction), itself compresses fresh air, which must be cooled by a charge air cooler. Via the manifold, the compressed air enters the cylinder. In the case of highly unsteady applications (e.g. vehicle operation), the power of the turbine must be controlled. This is done in the schematic shown here with a so-called wastegate.

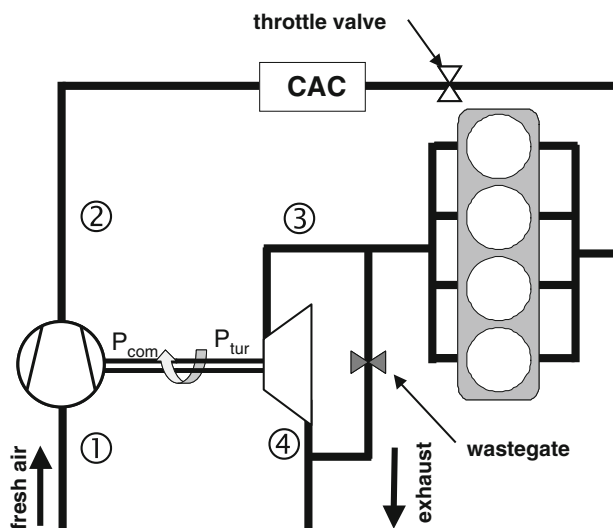


Fig. 8.8 A reciprocating engine and turbo machine. Indexes: 1: b.com, 2: a.com, 3: b.tur, 4: a.tur

This is the “second fundamental equation of turbocharging”

$$\dot{m}_{tur} \frac{\sqrt{T_{b,tur.}}}{p_{b,tur.}} = A_{is,tur} f\left(\frac{p_{a,tur.}}{p_{b,tur.}}, \kappa_{tur}\right). \quad (8.15)$$

With the help of this relation, we can form reference or dimensionless variables. This will be dealt with in more detail in Sect. 8.2.

The isentropic turbine substitute cross section is approximately constant and depends only on the geometrical turbine cross section; we thereby ignore its dependence on the speed of the turbocharger and on the pressure ratio $p_{b,tur}/p_{a,tur}$.

8.1.3.3 Interaction with the Exhaust Dynamics of the Internal Combustion Engine

Strictly speaking, turbocharging is classified as either constant-pressure turbocharging or pulse turbocharging.

In constant-pressure turbocharging, only the thermal energy of the exhaust is utilized, since the exhaust of a single cylinder of an internal combustion engine is guided into a large pipe volume between the engine and the turbine, in which it loses its momentum. This momentum results from the discontinuous working process of the internal combustion engine, in which the exhaust abruptly escapes into the manifold when the outlet valves open. There is thus a constant pressure before the turbine in this type of turbocharging, which also has an effect on the individual cylinders. For this reason, a free selection of the ignition sequence of the combustion engine is possible in the case of ideal constant-pressure turbocharging from the thermodynamic perspective. Also, there is no limitation on the cylinder number of the engine joined to the outlet volume. The constant pressure before the turbine has, on the one hand, a positive effect on the average efficiency and suction behavior of the turbine. On the other hand, the velocity contained in the individual impulses of exhaust gas is first converted into thermal energy and thus into static pressure. This energy must then again be converted in the turbine into the more valuable kinetic energy (nozzle, guide apparatus). Since each process of energy conversion is bound with losses however, corresponding losses arise both in the uniformization of the flow and the reacceleration of the flow. Besides the high demands on the package, which can only be met for a stationary application, the exhaust also loses a considerable amount of energy due to the large surface of the volume connected between the engine and the turbine. A large volume with highly unstable engine operation is especially problematic, e.g. as in an airplane, since filling the volume with exhaust in order to obtain a corresponding pressure and thus a corresponding amount of turbine power takes a long time. This is especially true at low speed and loads, since in this case there is already a low pressure level with low mass flows.

In pulse turbocharging on the other hand, the aim is to direct the exhaust to the turbine using as short a path as possible with as little loss as possible. Both the

thermal and kinetic energy of the exhaust is exploited. The exhaust system of a pulse-turbocharged engine is therefore characterized by its compactness. The flow of the exhaust is however very inconsistent, since the pressure waves in the exhaust system propagate with sonic speed and, due to the compactness of the pipe system, quickly affect the exhaust counterpressure of the other cylinders joined to the pipe system. This has adverse effects on the turbine on the one hand, which is applied with highly erratic mass and energy flows. The turbine blades are not optimally impacted with flow at every moment in the process. This in turn has a negative effect on efficiency and eliminates part of the benefits. The latter also has a negative effect on the mechanical and thermal strength of the turbine.

On the other hand, the high fluctuations in pressure also have a great effect on the engine gas exchange. Depending on the ignition sequence and the outlet valve control times of the cylinders combined in the exhaust system, it is possible that gas exchange during the push-out phase can be considerably impaired by the previous or subsequent cylinder. The result is that either the pushing-out work becomes very high or that residual gas remains in the cylinder and the combustion of the next working cycle is negatively affected. In the case of a four-stroke engine, three cylinders can be joined assuming a control time of max. 260°CA for one outlet valve. This means that in the case of three or six-cylinder engines (or all cylinder numbers dividable by 3), an aggregation of cylinders that is optimal with respect to gas exchange and turbocharging is possible in principle.

The six-cylinder engine requires however an arrangement of three cylinders each on either two turbines or on one double-flow turbine (twin-scroll), which makes flow separation possible (see below). In the case of the four-cylinder engine, there is an ignition distance of 180°CA , which is why only two cylinders can be joined optimally, which again means two exhaust flows (twin-scroll turbine). Nevertheless, four-cylinder engines in particular also have single-flow aggregates, but to significant disadvantage at low speeds and high loads and in their response behavior. In the V8 engine, the usual ignition sequence necessitates an ignition distance of only 90°CA on every bank, which can only be poorly compensated by flow combination. In this case, usually only combination of four cylinders on one bank remains.

Since complete reduction of the pressure pulse itself is not possible in the case of slowly running engines, while on the other hand an ideal utilization of the exhaust pulses is tied to a real volume of the exhaust system, no pure form of constant-pressure or pulse turbocharging exists.

8.1.3.4 Flow Turbine Designs

In the previous section, the necessity of different flow turbine designs was already pointed out. The basic in this section will be on radial turbines, which can be subdivided into monoscroll – and twin-scroll – or twin turbines. As the name already suggests, monoscroll turbines have only one entry snail which directs the exhaust to the rotor. Figure 8.15 (left) shows a cross-section through a monoscroll

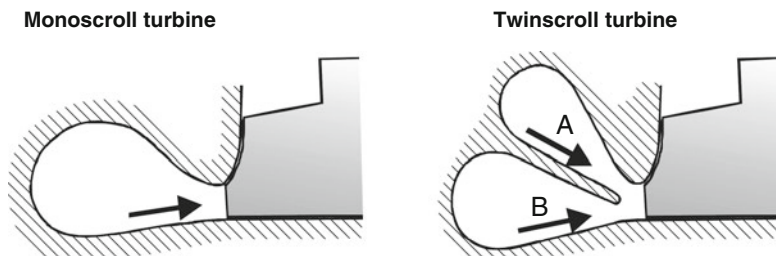


Fig. 8.15 Cross-section through a monoscroll and a twin-scroll turbine

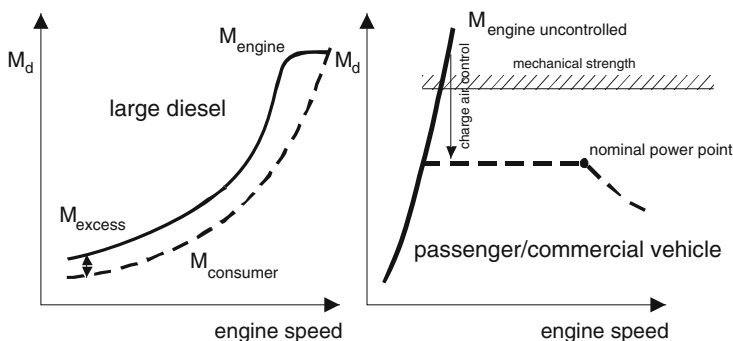


Fig. 8.16 Engine torque and load torque for large diesel and vehicle engines

turbine. On the right is a twin-scroll turbine. We can recognize the two parallel snails that direct the exhaust flows of each cylinder aggregate completely separately to the rotor of the turbine. To fully exploit the functionality of twin-scroll turbines, a complete and consistent separation of the flows until entering the rotor must be assured in order to avoid crosstalk of the gas pulses.

8.1.3.5 Charge Pressure Control Measures

Due to the characteristics of turbo machines, a turbocharger designed for full load without any external control has an approximately quadratic dependence of the boost pressure on the engine speed. At low engine speeds, only a small boost pressure and thus only a small engine torque can be created as a rule. Figure 8.16 (left) clarifies this behavior. This tendency is not as critical for medium speed diesel engines, whose time response is usually not decisive because of the stationary operation near the nominal power point, than for small, high-speed engines, which require high torque already at very low engine speeds due to their dynamic operation. For this reason, measures must be taken in such engines to increase the boost pressure and thus the torque availability at low engine speeds. In the basic design of

the turbine, value is placed on high pile-up pressure and thus on high turbine performance already with low mass flows. That means that the size of the turbine is not optimally designed for the nominal power point, but for a much lower engine speed ($< 2,000$ rpm). With a smaller turbine size, the pressures and turbocharger speeds would become very high with increasing engine speeds due to the almost quadratic increase of mass flow through the compressor and turbine, which would exceed the mechanical strength of the charging system and the combustion engine. Moreover, the charge pressure would increase dramatically, which would lead to serious knocking problems in SI engines due to the resultant high temperatures. For these reasons, boost pressure, exhaust counterpressure and the speed of the turbocharger must be limited by suitable measures in the charging system.

Until now, two methods have become established on the turbine side of the exhaust turbocharger. On the one hand, it is possible to control the exhaust mass flow through the turbine with a wastegate (Fig. 8.8). In this case, part of the exhaust mass flow is guided past the turbine and is thus no longer available for the production of drive power, which is why the compressor conveys a smaller mass flow. In this way, the boost pressure before the engine can be controlled and thus the engine torque as well. For wastegates, the control elements are either pressure sensors that are applied with boost pressure or negative pressure, or electric control elements.

It is also possible to limit the power of the turbine by means of variability in the guide vanes by changing the cross-section by turning the guide vanes. When the cross-section is reduced, this causes a higher pile-up pressure and thus larger turbine power, and if the cross-section is enlarged, it is possible to reduce the pile-up pressure and turbine power.

Nearly all modern turbocharged car diesel engines have charge pressure control using an adjustable guide apparatus. Turbocharged SI engines as a rule require a wastegate because of their much higher exhaust temperatures, since adjustable guide vanes would represent a much larger function risk.

Moreover, SI engines with conventional air-mass load control require, for acoustic reasons, a “diverter valve”. By means of reverse flow at the compressor, this valve makes possible a rapid reduction of the charge pressure in case of a sudden intrusion of mass flow due to load reduction. Without a diverter valve, the operating point of the compressor in the characteristic map would shift to the left over the surge limit (see Fig. 8.17) due to the inertia of the exhaust turbocharger. If one keeps the mass flow through the compressor to a higher level with the diverter valve, it is possible not to exceed the surge limit until the speed of the compressor has been reduced.

Figure 8.17 shows the characteristic map of a turbo compressor with its limits (surge, speed and choke limit). The structure of the map is described in more detail in Sect. 8.2.

The full load operating line is drawn in the characteristic map, which runs almost quadratically in the left part. We can clearly see the kink in the operating line, which is caused by the opening of the wastegate due to the charge pressure control described above.

8.1.3.7 Interaction with the Internal Combustion Engine

Figure 8.17 shows the interaction of a internal combustion engine and a turbocharger with the help of the stationary full load operating line in the compressor characteristic map. This operating line is drawn in Fig. 8.20 in the torque/speed diagram in comparison to a natural aspirated engine of the same capacity as well as a mechanically charged engine. We can clearly see that in the case of the steady-state operation of a turbocharged engine a high torque can be reached already at very low speeds (“low end torque” LET). The torque plateau can be held constant by the wastegate control described above until the desired power is reached (power hyperbola). The early position of LET is made possible by the variable control times in the SI engine caused by the inlet and outlet camshaft phasing. In the case of a positive pressure difference at the turbocharger, in which case the charge pressure is larger than the exhaust counterpressure, in combination with a large overlap of outlet and inlet valve opening during gas exchange, a complete purging of the residual gas from the cylinder and a filling of the cylinder with cold fresh air is possible. In combination with direct injection, the air fuel ratio in the cylinder can be set slightly rich, which causes very high combustion velocities. The result is a largely knock-free operation, even at low engine speeds along with high torque values.

Figure 8.21 shows the unsteady behavior of a turbocharged SI engine at a constant speed of 1,500 rpm and a load step from partial load to full load, from which we can see that the flow-type coupling of both sub-components results in a significantly delayed torque build-up compared with the steady-state behavior of the engine. Section 10.5 presents some simulation calculations for improving unsteady behavior and for explaining individual effects.

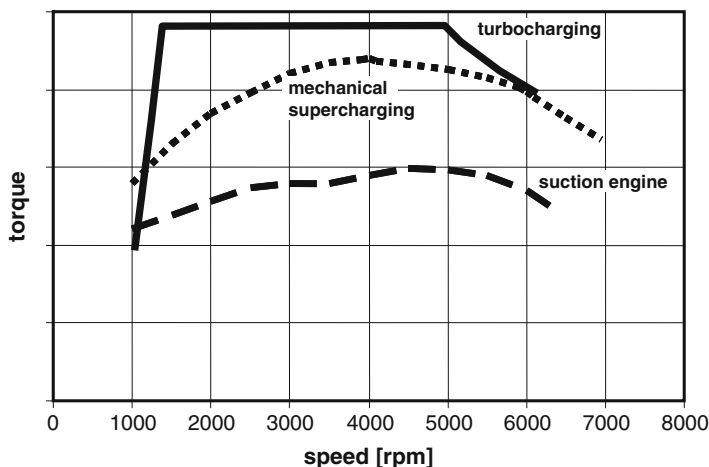
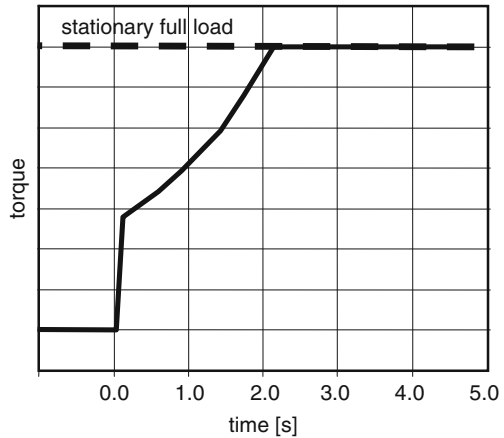


Fig. 8.20 The torque profile of various charging concepts

Fig. 8.21 Unsteady torque build-up in a turbocharged engine



The following summarizes the advantages and disadvantages of exhaust turbocharging.

- Advantages:
 - The exhaust counterpressure is reduced in case of partial load. A so-called “self-regulation effect” of the turbocharger: the specific fuel consumption is reduced with it as well.
 - Small turbine cross-sectional surfaces, small polar moments of inertia and high potential turbocharger speeds make for a fast, almost delay-free response time (no noticeable turbo lag).
 - No external drive necessary.
 - Economical manufacture and low manufacturing costs for the turbocharger with high production numbers, since they can be adjusted to the engine on the side of both the turbine and the compressor by means of a change of trim.
 - The small construction size of the turbocharger unit. Integration of the turbine into the exhaust manifold results in additional performance advantages and good response behavior.
 - Utilization of exhaust energy improves engine power and, under some conditions, even the indicated efficiency of the engine.
 - The continuous supply of the turbo compressor.
 - Lower thermal stress on the turbocharging system due to temperature reduction via the turbine.
- Disadvantages:
 - The surge limit restricts the charge pressure at low engine speeds.
 - In the case of large turbine housings (cross-sectional areas), the unsteady response time is clearly delayed (turbo lag).
 - High gas exchange losses and poor gas exchange success of the engine due to the high pile-up pressure before the turbine, especially in the case of a negative purge slope and small turbocharger designs.

EXHIBIT 1020



Introduction to Internal Combustion Engines



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Third Edition





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and this will reduce the ignition delay period. The modelling of the ignition delay and the subsequent combustion are treated in section 10.2.

To obtain maximum output the peak pressure should occur about $10\text{--}20^\circ$ after top dead centre. Sometimes the injection is later, in order to retard and to reduce the peak pressure. The fuel ignites spontaneously at many sites, and produces an intense flame. There is significant radiation from the flame front, and this is important for vaporising the fuel. Since the combustion occurs from many sites, compression ignition engines are not susceptible to cyclic variation or cyclic dispersion.

Compression ignition engines can operate over a wide range of air/fuel mixtures with equivalence ratios in the range 0.14–0.90. The power output of the engine is controlled by the amount of fuel injected, as in the combustion region the mixture is always approximately stoichiometric. This ensures good part load fuel economy as there are no throttling losses. The fuel/air mixture is always weaker than stoichiometric, as it is not possible to utilise all the air. At a given speed the power output of an engine is usually limited by the amount of fuel that causes the exhaust to become smoky.

3.7 Fuels and additives

The performance and in particular fuel economy of internal combustion engines should not be considered in isolation, but also in the context of the oil refinery. Oil refining and distribution currently has an overall efficiency of about 88 per cent (Francis and Woollacott, 1981). This efficiency would be changed if the demand for the current balance of products was changed. Oil refining can be compared to the work of a butcher; each process has a raw material (a barrel of oil or a carcass) that has to be cut in such a way that all the products can be sold at a competitive price. Just as there are different animals, there are also different types of crude oil, depending on the source. However, the oil refinery can change the type of product in additional ways, such as cracking, although there is an energy cost associated with this.

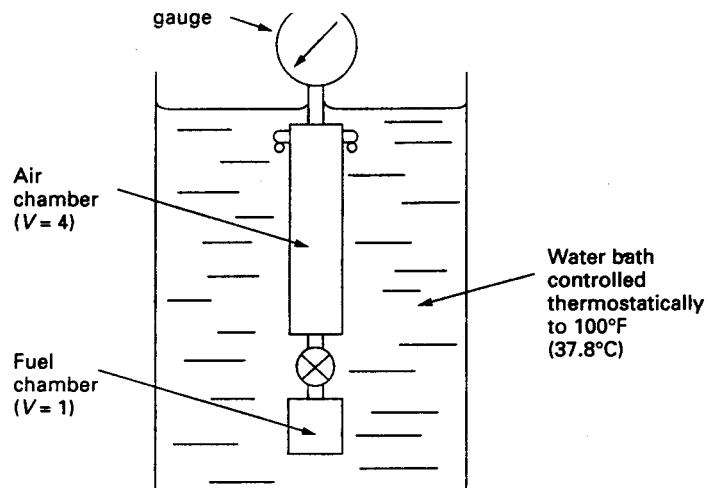
The energy content of a typical petrol is 44 MJ/kg or 31.8 MJ/litre and of a typical diesel fuel 42 MJ/kg or 38.15 MJ/litre; associated with these is an energy content at the refinery of typically 2.7 MJ/kg and 1.6 MJ/kg (Francis and Woollacott, 1981). This gives an effective primary energy of 46.7 MJ/kg or 35.5 MJ/litre for petrol and 43.65 MJ/kg or 39.95 MJ/litre for diesel fuel. There may be circumstances in which it is more appropriate to use primary energy density than to use calorific value. These figures also highlight the difference in energy content of unit volume of fuel. This is often overlooked when comparing the fuel economy of vehicles on a volumetric basis.

3.7.1 Characteristics of petrol

The properties of petrol are discussed thoroughly by Blackmore and Thomas (1977). The two most important characteristics of petrol are its volatility and octane number (its resistance to self-ignition).

Volatility is expressed in terms of the volume percentage that is distilled at or below fixed temperatures. If a petrol is too volatile, when it is used at high ambient temperatures the petrol is liable to vaporise in the fuel lines of carburetted engines

Reid vapour pressure apparatus (from Goodger, 1975).



and form vapour locks. This problem is most pronounced in vehicles that are being restarted, since under these conditions the engine compartment is hottest. If the fuel is not sufficiently volatile the engine will be difficult to start, especially at low ambient temperatures. The volatility also influences the cold start fuel economy. Spark ignition engines are started on very rich mixtures, and continue to operate on rich mixtures until they reach their normal operating temperature; this is to ensure adequate vaporisation of fuel. Increasing the volatility of the petrol at low temperatures will evidently improve the fuel economy during and after starting. Blackmore and Thomas (1977) point out that in the USA as much as 50 per cent of all petrol is consumed on trips of 10 miles or less. Short journeys have a profound effect on vehicle fuel economy, yet fuel consumption figures are invariably quoted for steady-state conditions.

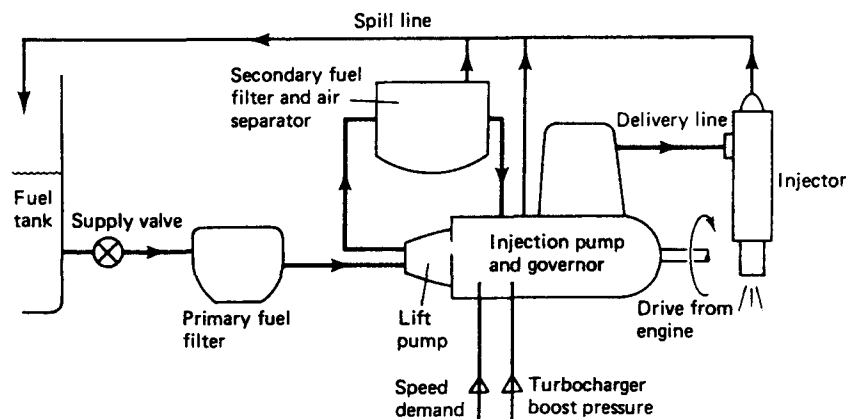
The European Standards Organisation (CEN) has defined a common standard for its 18 member countries, and for unleaded gasoline this standard has the designation EN 228: 1993, and this includes eight different volatility specifications. Members of CEN can then use their national standards (BS EN 228 in the case of the UK), to specify particular volatilities for different seasons. The volatility is also expressed in terms of the Reid Vapour Pressure (RVP). The Reid apparatus is shown in figure 3.12, and it consists of a fuel chamber connected to an air chamber of four times its volume. The fuel chamber is filled with chilled fuel, connected to the air chamber, and then immersed in a water bath at 37.8°C and shaken. The method is defined in ASTM Procedure D323. The final pressure has to be corrected for atmospheric pressure and the effect of heating the air in the air chamber, and the result is an absolute value.

The fuel volatility specified in BS EN 228: 1993 is compared with typical fuel specifications from Blackmore and Thomas (1977) in table 3.3. This is plotted with further data in figure 3.13.

Table 3.3 shows how the specification of petrol varies to suit climatic conditions. Petrol stored for a long time in vented tanks is said to go stale; this refers to the loss of the more volatile components that are necessary for easy engine starting.

The octane number of a fuel is a measure of its anti-knock performance. A scale of 0–100 is devised by assigning a value of 0 to n-heptane (a fuel prone to knock),

Figure 5.11
Fuel injection system for a
compression ignition
engine.



injection systems have a high-pressure fuel pump which produces a controlled and steady pressure, and the injector has to control the start and end of injection. Charlton (1998) provides a comprehensive overview of all types of electronically controlled fuel injection equipment.

After the overview in section 5.5.1 there is a discussion of the fuel injection nozzles (section 5.5.2), since these are common to all types of fuel injection system.

5.5.1 Injection system overview

A traditional pump-line-injector (PLI) system is shown in figure 5.11. Before discussing this, it is important to realise that the low-pressure part of the system is common to all diesel engines.

In general the fuel tank is below the injector pump level, and the lift pump provides a constant-pressure fuel supply (at about 0.75 bar) to the injector pump. The secondary fuel filter contains the pressure-regulating valve, and the fuel bleed also removes any air from the fuel. If air is drawn into the injection pump it cannot provide the correctly metered amount of fuel. It is essential to remove any water or other impurities from the fuel because of the fine clearances in the injection pump and injector. The injection pump contains a governor to control the engine speed. Without a governor the idle speed would vary and the engine could over-speed when the load on the engine is reduced.

The injection pump is directly coupled to the engine (half engine speed for a four-stroke engine) and the pump controls the quantity and timing of the fuel injection (figure 5.12). The quantity of fuel injected will depend on the engine load (figure 5.13). The maximum quantity of fuel that can be injected before the exhaust becomes smoky will vary with speed, and in the case of a turbocharged engine it will also vary with the boost pressure. The injection timing should vary with engine speed, and also load under some circumstances. As the engine speed increases, injection timing should be advanced to allow for the nearly constant ignition delay. As load increases, the ignition delay would reduce for a fixed injection timing, or for the same ignition delay the fuel can be injected earlier. This ensures that injection and combustion are complete earlier in the expansion stroke. In engines that have the injection advance limited by the maximum permissible cylinder pressure, the injection timing can be advanced as the load reduces. Injection timing should be accurate to 1° crank angle.

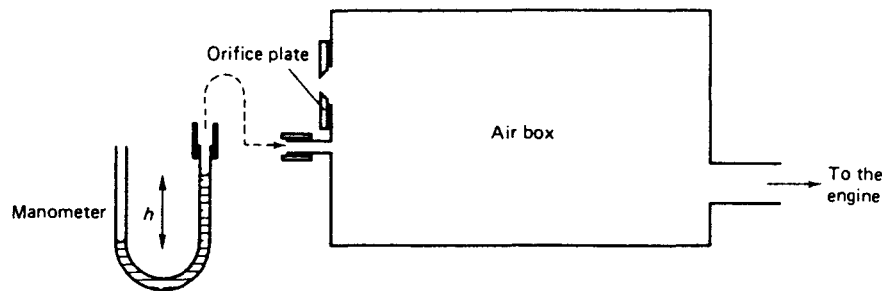


Figure 13.11
Air flow rate tank.

where $\dot{m}_a$ = mass flow rate of air (kg/s)
 C_d = discharge coefficient of the orifice
 A_o = cross-sectional area of the orifice (m)
 g = acceleration due to gravity (m/s²)
 h = height difference between liquid levels in the manometer (m)
 ρ_f = density of manometer fluid (kg/m³)
 ρ_a = density of air = $(p/R_a T)$ (kg/m³).

The accuracy depends on knowing the discharge coefficient for the orifice; this should be checked against a known standard.

In practice, it can be very difficult to make the air box work satisfactorily. Apart from making the air box act as a Helmholtz resonator (chapter 6, section 6.6.2), coupling a long tube to the inlet of an engine can have a significant effect on the engine performance. It should be remembered that even without a tuned induction system, there can be flow reversals occurring at the air inlet to the engine. A way of minimising any coupling effects and checking for their effect is shown in figure 13.12. If the flexible pipe is of large enough diameter, and the air entry protrudes far enough into the pipe, then the unsteady flow sensor will show no change to the air flow pattern when the flexible pipe is connected. The unsteady flow sensor is most likely to be a form of hot wire anemometer (see chapter 8, section 8.2.2).

An alternative approach is the viscous flowmeter. As no damping is required it is much more compact than an air box. To obtain viscous flow a matrix of passages is used in which the length is much greater than a typical diameter. Since the flow is viscous the pressure drop is proportional to the velocity or volumetric flow rate. The flowmeter has to be calibrated against a known standard.

However, in its usual form the viscous flowmeter is not well suited to pulsating flows. The instantaneous pressure drop will be a function of the unsteady flow,

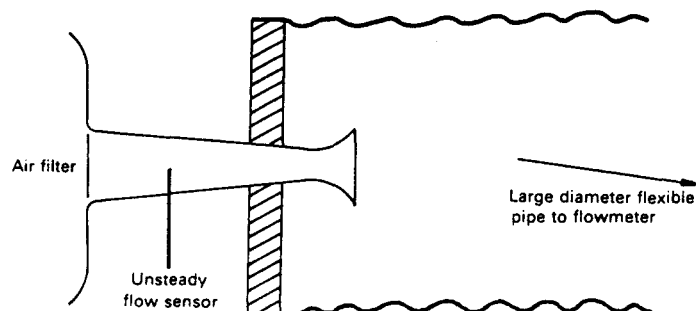


Figure 13.12
Connection of an air
flowmeter to minimise any
effects on engine
performance.

For example, if 300 cycles of a four-stroke engine are to be recorded from two channels, with readings every $\frac{1}{2}^\circ$ and 2 bytes are required for each reading (for a 12-bit reading), then about 3.4 MB of memory will be required.

The trade-offs between the sampling rate and the storage capacity have been illustrated by table 13.4 for a particular system. In this case, the 3.4 MB of memory would require the use of the RAM or the hard disk, and the engine speeds corresponding to the maximum sampling rates would be 8333 or 4166 rpm. These two examples illustrate that the system described in table 13.4 is capable of acquiring large amounts of data very quickly. This can then lead to analysis and archiving problems unless the combustion analysis software is quick to run.

There are other issues to be considered with computer-based data acquisition systems. Firstly, when channels are being multiplexed, the channels are not all being read at the same time. For example, table 13.4 shows that the Computerscope system has a delay of 1 microsecond between reading successive channels. This becomes significant with high sampling rates, but it can be corrected for in software that either assigns the correct angle to the reading or interpolates between readings to give the value at a specific crank angle. Secondly, it is essential that the ADC is coupled to a sample/hold circuit. If this is not the case, slight changes in the signal during the analogue-to-digital conversion process can lead to large errors.

When the data are being analysed, it must be remembered that they were originally a continuous analogue signal that has been assigned to a progression of digital levels. For example, when such a signal is differentiated numerically by looking at the difference in successive values, the result is very noisy. A more satisfactory result can be obtained by using a higher-order finite difference approach and smoothing the signal.

Taylor series can be used to represent the signal, and the following example calculates the first derivative, but includes the terms up to the fourth derivative. Consider a function $f(a)$ in which the interval between values is h :

$$\begin{aligned} f(a_{n+2}) &= f(a_n) + \frac{2hf'(a_n)}{1!} + \frac{2^2h^2f''(a_n)}{2!} + \frac{2^3h^3f'''(a_n)}{3!} + \frac{2^4h^4f^{(4)}(a_n)}{4!} + \dots \\ f(a_{n+1}) &= f(a_n) + \frac{hf'(a_n)}{1!} + \frac{h^2f''(a_n)}{2!} + \frac{h^3f'''(a_n)}{3!} + \frac{h^4f^{(4)}(a_n)}{4!} + \dots \\ f(a_n) &= f(a_n) \\ f(a_{n-1}) &= f(a_n) - \frac{hf'(a_n)}{1!} + \frac{h^2f''(a_n)}{2!} - \frac{h^3f'''(a_n)}{3!} + \frac{h^4f^{(4)}(a_n)}{4!} - \dots \\ f(a_{n-2}) &= f(a_n) - \frac{2hf'(a_n)}{1!} + \frac{2^2h^2f''(a_n)}{2!} - \frac{2^3h^3f'''(a_n)}{3!} + \frac{2^4h^4f^{(4)}(a_n)}{4!} - \dots \end{aligned} \quad (13.44)$$

These equations can be combined to eliminate the second, third and fourth derivatives ($f''(a_n)$, $f'''(a_n)$ and $f^{(4)}(a_n)$), to give

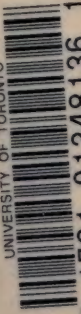
$$f'(a_n) = (f(a_{n-2}) - 8f(a_{n-1}) + 8f(a_{n+1}) - f(a_{n+2}))/12h \quad (13.45)$$

However, figure 13.34 shows that the derivative will still be noisy, and it is advantageous to apply smoothing. The following smoothing algorithm for $(2b + 1)$ values is widely used:

$$\begin{aligned} a_n &= \frac{1}{b^2} [a_{n-(b-1)} + 2a_{n-(b-2)} + 3a_{n-(b-3)} + \dots + ba_n \\ &\quad + \dots + 3a_{n+(b-3)} + 2a_{n+(b-2)} + a_{n+(b-1)}] \end{aligned} \quad (13.46)$$

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INTERNAL COMBUSTION ENGINES

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INTERNAL COMBUSTION ENGINES

THEIR THEORY, CONSTRUCTION
AND OPERATION

BY
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WITH 379 ILLUSTRATIONS

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PREFACE

THE intention of the authors in the preparation of this book has been to present in as simple terms as possible the fundamental and theoretical principles relating to the internal combustion engine, and to describe the various methods of applying these principles to practical construction. The book does not in any way treat of the proportioning and the strength of the various machine parts.

The general treatment of the subject is indicated by the various chapter headings. Thus the first five chapters relate to definitions and theoretical considerations, the subjects being as follows:

- CHAPTER I. DEFINITIONS AND CLASSIFICATION.
- CHAPTER II. THERMODYNAMIC PRINCIPLES.
- CHAPTER III. THEORETICAL DISCUSSION OF VARIOUS CYCLES.
- CHAPTER IV. THEORETICAL CYCLES MODIFIED BY PRACTICE.
- CHAPTER V. THE TEMPERATURE-ENTROPY DIAGRAM.

In the discussion on theoretical cycles in Chapter III, very little reference has been made to cycles not in actual use. The cycles are considered principally with reference to their practical application and any danger of confusing the mind of the student by a multiplicity of theoretical cycles of no practical value is avoided. The main idea of Chapter IV is to show how the lines of the real cycles differ from those of the theoretical cycles laid down in the previous chapter, and to discuss briefly the reasons for such difference.

The five chapters following, VI to X inclusive, take up the phenomena of combustion, the various gas-engine fuels, and the formation and properties of the fuel mixture. Thus, Chapter VI treats of combustion in general and discusses the most important properties of the gases usually found in gas-engine practice. Tables are given embodying the important constants for many of

these gases, and the chapter ends with some type computations on fuel mixture and exhaust gas constants which may serve as a guide for similar work by the student.

The question of gas-engine fuels is treated in the next three chapters by dividing all the fuels into three classes: the solid, the liquid and the gas fuels. Chapters VII and VIII take up the first two of these classes. Broadly speaking, neither class of fuel is directly available for gas-engine work, hence it was thought desirable to show in these chapters also the means by which these fuels are rendered available. Accordingly, Chapter VII discusses producer gas and describes the construction and operation of the most important types of producers, while Chapter VIII consists largely of a description of the various vaporizing devices used for crude oil, gasoline, kerosene, and alcohol. The latter fuel, recognizing its growing importance, has been treated in some detail. Chapter IX relates to industrial gases, pointing out briefly the method of their manufacture and giving the most important gas constants.

In Chapter X, after discussing the fuel mixture and its most important properties, an attempt has been made to collaborate the most important experiments on the variation of the specific heat of the fuel gases with a view to ascertaining the present state of our knowledge on this point.

Chapter XI gives a brief outline of the historical development of the gas engine, the various types being described only where they are of importance in connection with the development of modern forms. This is followed in Chapter XII by an extended description of the most important forms of internal combustion engines found in the market at the present time. The aim of this chapter has been not only to show how the various manufacturers have solved what is fundamentally the same problem, but also to familiarize the student, by means of a large number of illustrations, with the main constructive features of the gas engines at the disposal of the man requiring power.

Two of the important problems connected with the gas engine are the questions of ignition and of governing. The former is taken up in detail in Chapter XIII. This chapter, after mentioning briefly other methods of ignition, concerns itself mainly with ignition by electric spark, as being the most important method

used to-day. The chapter also takes up briefly two other gas-engine auxiliaries: mufflers and starting apparatus.

Chapter XIV treats the governing problem by discussing first the principles of the various systems of governing employed, and afterward shows the mechanical details of the governors used. It is beyond the scope of this book, however, to treat of the principles of governor design.

Chapter XIV discusses various methods in use for determining the necessary cylinder dimensions of a gas engine to develop a certain given power, or conversely, to determine the probable power for a given engine. The most reliable of these methods appears to be based upon the necessary charge volume for the given power. This method, the authors believe, was originally due to Güldner, and was adapted from that author's hand-book "*Entwerfen und Berechnen der Verbrennungsmotoren.*"

For the determination of the power of automobile engines, two additional semi-empirical formulæ are given, and the results of the computations by the various methods are compared by means of a type example.

The remaining three chapters of the book treat of what might be called the economic side of the internal combustion engine.

Chapter XVI takes up the methods used in the testing of gas engines. The rules followed in this country will be of course the code laid down by the American Society of Mechanical Engineers. The Code of the German Society of Engineers, however, is appended, because it gives additional information upon some of the points involved and because it treats in greater detail of gas producers.

The results of tests on engines and producers are discussed in Chapter XVII. The various factors affecting economy are taken up in somewhat greater detail than has been done in any of the previous chapters. Tables are given showing the results of numerous tests and these should prove valuable in furnishing a guide as to what may be expected of other installations in the matter of fuel economy.

Finally, while Chapter XVII treats only of the question of fuel economy, Chapter XVIII takes up the entire financial problem relating to the gas engine. It shows in brief, as far as the information is available, the probable capital cost of the installation,

the cost of erection, the operating expenses, etc. It is shown most strongly that the question of fuel cost is not always the important item of the problem, a point which is often lost sight of in discussions regarding the comparative merits of various prime movers. It must be confessed that reliable information regarding some of the factors in this financial problem is still very scarce, owing, of course, directly to the comparative youth of the gas engine. It should also be pointed out that many so-called comparisons between various prime movers, as between steam and gas, are often based upon hypothetical assumptions that fit only the particular case under discussion, and any generalization of the results obtained often leads to serious misrepresentation.

The book is of course largely compiled from different sources and is in the main an outgrowth of a course of lectures on the internal combustion engine delivered to students of Sibley College for the past three years. Acknowledgments are due to numerous writers upon the subject for the facts and statements presented. The acknowledgments have generally been made in the body of the book, but the authors desire to extend herewith special acknowledgment to the following European authorities: Messrs. H. Güldner, Dugald Clerk, Bryan Donkin, and Aimé Witz, and to Professor C. E. Lucke of Columbia University and Mr. F. E. Junge of New York.

Thanks are also due to various manufacturers for kindly furnishing the text and illustrations of Chapter XII.

ITHACA, N. Y.,
April 23, 1908.

PREFACE TO THE SECOND EDITION

In presenting the second edition of this book the authors wish to express their obligation to all those who kindly aided in pointing out errors, and especially to Professor H. P. de Schweinitz of Lehigh University, for a thorough revision of the first two chapters.

February 1, 1909.

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CHAPTER I

INTRODUCTION, DEFINITIONS, CLASSIFICATION, AND FORM OF INTERNAL COMBUSTION ENGINES, INDICATED AND BRAKE HORSE-POWER

1. Mechanical Work. — Work is done when resistance is overcome; it is measured by the product of the resisting force and the distance through which that force is moved. If one pound is lifted one foot high in opposition to the force of gravity, a quantity of work, measured by the product of one pound by one foot, is performed, which quantity is known as a *foot-pound*, and is the unit of measurement for mechanical work in countries where the pound is a unit of weight and the foot a unit of distance. If 20 pounds are lifted 15 feet the work performed would be similarly 20×15 foot-pounds = 300 foot-pounds.

In countries where the metric system is used mechanical work is measured by the product of the resisting force in kilograms (2.2046 pounds) multiplied by the distance in meters (3.2808 feet); the product is expressed in *kilogrammeters* (7.233 foot-pounds).

The foot-pound or kilogrammeter is a gravity measure which depends on the intensity of the force of gravity at the place, and varies with that force. The variation is, however, so slight for different positions on the earth's surface that for all practical engineering work no sensible error is produced by considering it a constant quantity.

The unit of measurement usually employed by engineers for expressing *the rate of work*, or the quantity of work done in a given time as one second or one minute, is the *horse-power*, which has been arbitrarily defined as equivalent to 550 foot-pounds per second or 33,000 foot-pounds per minute. This quantity is considerably greater than the power a horse can exert, at least for any considerable length of time; it was first

used by James Watt in defining the power of the steam engine and has been established by long use as a definite measure of power. In France the term *Force de Cheval* is applied to a rate of work of 75 kilogrammeters per second ($542\frac{1}{2}$ ft. lbs.) or 4500 kilogrammeters per minute (32549 ft. lbs.).

In general, if W be the work performed against the pressure or resisting force p while moving through the space or volume v ,

$$W = pv. \quad (1)$$

Work is done when force is applied so as to produce motion in the direction of action of the force, and also when force is employed in changing the velocity of a body already in motion. The latter condition is of considerable practical importance and can be considered as follows: Suppose a body whose mass is M be moving in a certain direction with a velocity u , and let a force exerting a momentum P be applied in the direction of motion, required to find the effect produced by this force acting through the small time t , during which the body moves through the distance v , and has at the end of the time the velocity u' .

The momentum produced by the force in one unit of time is P , and in t units of time it is Pt . Since this is equal to the increase of momentum produced, we have

$$Pt = M (u' - u).$$

As the distance is equal to the mean velocity multiplied by the time, we have

$$v = \frac{1}{2} (u' + u) t.$$

By multiplying the above equations,

$$Pvt = \frac{1}{2} (Mu'^2 - Mu^2) t;$$

dividing by t ,

$$Pv = \frac{1}{2} (Mu'^2 - Mu^2) \quad (2)$$

Pv is the mechanical work done in overcoming a resistance; the expression $\frac{1}{2} Mu^2$ is the kinetic energy. From this it is seen that the mechanical work done is measured by the increase in the kinetic energy produced.

The mechanical work done by a fluid during a change of volume from v to v' is equal to the mean resistance overcome, or

pressure exerted, p , multiplied by the change of volume. That is, in general,

$$W = \int_v^v p dv \quad (3)$$

$$W = p (v' - v)$$

In the operation of an engine, the working fluid expands and contracts as the piston moves forward and backward, and in one or more revolutions returns to its initial condition, so far as pressure, volume and temperature are concerned, and then passes through the same stages of expansion and contraction as before. The period through which these changes take place is termed a *cycle*.

The work performed in a cycle would be equal to the mean pressure, p , exerted, multiplied by the total volume, v' , swept through; that is

$$W = pv'. \quad (4)$$

Mechanical work can be represented by a diagram in which the pressure exerted or resistance overcome, p , is represented by the ordinates, and the volume v by the abscissa. Such a diagram is called a *pressure-volume diagram*; its area is equal to $\int p dv$, and is proportional to the work performed.

Thus in Fig. 1-1, if the distances parallel to OY represent the pressure at any given point, and the distances parallel to OX the corresponding volume, then will the total work done in changing from the highest to the lowest pressure and from least to greatest volume be represented by the area of the figure $a b d e f$.

2. Heat. — Heat is a peculiar form of energy; it may be generated by the application of mechanical work, the amount so produced being exactly proportional to the mechanical energy which disappears. Conversely, mechanical work may be done by the action of heat, and for every foot-pound of work so done a definite amount of heat is put out of existence. Heat is also produced by a form of chemical action known as *combustion*, during which operation fuels are burned.

3. Temperature. — The *temperature* of a body is defined by Maxwell* as "its thermal state with reference to its power of communicating heat to other bodies."

* Theory of Heat.

A body transmitting heat to another is at a higher temperature and is said to be hotter; conversely, one receiving heat is at a lower temperature and is said to be colder.

Heat flows from a hotter to a colder body, but not conversely,

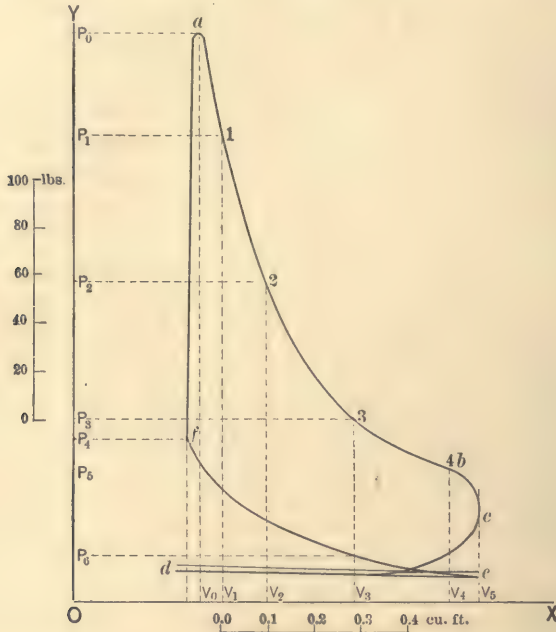


FIG. 1-1. — Pressure-volume or Work Diagram.

and the rate of flow increases with the difference of temperature, although probably not exactly in the same ratio. The difference of temperature thus causes a flow of heat in a manner somewhat smiliar to that caused by a difference of pressure in the case of water.

The terms *hotter* and *colder* are relative ones commonly applied to distinguish substances having relatively a higher or lower temperature. It should be noted that temperature is that property of heat which refers to its intensity or transmission power also, that heat energy may exist at different temperatures, and, furthermore, in one condition may be much colder than in another.

The following scales of temperatures are in common use in which the temperatures, of freezing and boiling water under a barometric pressure of 29.92 inches are taken as points of reference.

The Centigrade scale was introduced by Celsius, professor of astronomy in the University of Upsala about 1742; in it the freezing-point is marked 0 degrees and called zero, and the boiling-point is marked 100 degrees. The simplicity of dividing the distance between the points of reference into 100 parts and calling each of them a degree has caused it to be generally adopted along with the Metric System for scientific use, especially on the Continent of Europe. The other scales are called by the names of those who introduced them.

Fahrenheit of Dantzig, about 1714, introduced a thermometer scale in which the freezing-point was marked 32 degrees and the boiling-point 212 degrees, the space between the reference points being divided into 180 equal parts called degrees, and the graduation extended above and below the points of reference. A point 32 degrees below freezing was called zero. Despite the inconvenience of the scale of the Fahrenheit thermometer it is in general use by English-speaking people for commercial and business purposes, and for that reason will be used principally in this treatise.

Réaumur introduced a thermometer scale about 1730 in which the freezing-point is marked 0 degrees and the boiling-point 80 degrees, which is used to some extent on the Continent of Europe for medical purposes.

The following table gives the comparative value of the three thermometric scales:

THERMOMETRIC SCALES

	Fahrenheit	Centigrade	Réaumur
Degrees between freezing and boiling..	180	100	80
Assumed temperatures at freezing-point	32	0	0
Assumed temperatures at boiling-point	212	100	80
Comparative length of a degree.....	1	1.80	2.25
Comparative length of a degree.....	$\frac{5}{9}$	1	$\frac{4}{5}$
To transform into absolute temperature add	460°	273°	218°

4. **Absolute Temperature** is an expression for the value of temperature measured from an ideal point called the *absolute zero*, which is assumed to be the lowest possible point on any scale of temperature. The position of the absolute zero can be calculated by the law of expansion of a perfect gas, which is expressed by the simple equation

$$\frac{pv}{T} = R \quad (5)$$

in which p = the pressure, v = the volume of a given mass of gas, T = the absolute temperature, and R = a constant which varies only with the different kinds of gas. This equation can be considered as the characteristic equation of a permanent gas, from which T can be computed if p , v , and R are known, which is the case with most of the gases.

It is evident that the value of one degree of absolute temperature can be taken at pleasure as equal either to that on the Centigrade or Fahrenheit scale.

The exact location of the position of absolute zero is somewhat in doubt since it is determined by the relative expansion of air, nitrogen, or hydrogen under a constant pressure; these gases are not perfect gases and the expansion in volume per degree of increase in temperature may not be exactly the same as for a gas which could not be liquefied for any conditions of pressure or temperature. Preston in his work on the Theory of Heat states that the most trustworthy observations indicate that the absolute temperature of freezing water is 273.14 Centigrade, which would correspond to 491.65 Fahrenheit. It is sufficiently near for all practical purposes to consider the temperature of freezing water on the absolute scale as 273 degrees Centigrade or 492 degrees Fahrenheit, and these numbers will be used in this treatise in reducing to the absolute scale.

From this it is seen that to reduce to the absolute scale it is necessary to add to the temperature, if expressed in degrees, Fahrenheit 460, or if in degrees Centigrade 273.

5. **Thermometers.** — Instruments for measuring temperature are called *Thermometers*.

The expansion of a gaseous, liquid or solid body under con-

stant pressure is almost, if not exactly, proportional to the increase of temperature, estimated from absolute zero.

In the thermometer in common use the temperature is measured by the expansion of mercury, confined in a glass tube, from which the air has been exhausted. Such a thermometer is quite satisfactory within the range of temperature through which mercury will remain liquid. In the better grades of mercurial thermometers the graduations are cut with extreme care directly on the stem. The glass is carefully selected and is permitted to season or age until

molecular changes have stopped before graduation. The general appearance of such thermometers is shown in Fig. 1-2. When thermometers are likely to be used in temperatures which would send the mercury column above the limits of the graduations, it is desirable to have an extra bulb, called a *safety-bulb*, at the top, to prevent breaking from overheating. Mercurial thermometers can be used from a temperature about 40 degrees below zero to 600 degrees above zero Fahrenheit. By filling the space above the mercury with some neutral gas as N or CO₂ under pressure the upper limit may be raised some hundred degrees; but as the melting-point of glass is low, the upper limit can scarcely ever exceed 800 degrees to 900 degrees Fahrenheit.

FIG. 1-2.—
Mercurial
Thermom-
eter.



FIG. 1-3.—
Metallic
Pyrometer.

METALLIC THERMOMETERS in which the expansion of a metal, or the difference in expansion of metals of two different kinds, is multiplied by a system of levers so as to move a hand over a dial are frequently used for the measurement of temperature. Such thermometers are sometimes called *pyrometers*. An illustration of such a thermometer is shown in Fig. 1-3.

The metallic thermometer can be used for temperatures not exceeding 1200 degrees to 1500 degrees Fahrenheit, but it is sel-



dom an instrument of accuracy and is extremely liable to accident. The scale of these instruments should be frequently compared with the boiling-point, and adjusted if not found correct.

It has already been shown that air or permanent gases like nitrogen and hydrogen when under a constant pressure will expand in volume in proportion to the absolute temperature, or when confined so as to have a constant volume will increase in pressure in proportion to the absolute temperature.

It follows from this that if air be maintained at a constant volume and heated, its absolute pressure will increase with the absolute temperature, or *vice versa*, if it be maintained at constant pressure, its volume will vary with absolute temperature.

The air thermometer constructed in accordance with either principle is used as a standard way of measuring temperature, but because of the extreme difficulty of maintaining constant pressures or constant volumes it is an awkward instrument to use and is employed very little in the ordinary measurement of temperature. The Jolly form of constant volume air thermometer is shown in Fig. 1-4. The leg

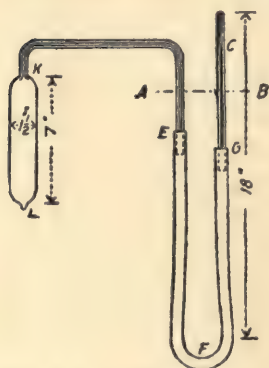


FIG. 1-4. — Jolly Air Thermometer.

leg *CF* has a flexible connection at the bottom and may be raised to maintain a constant volume of air from the bulb *L* to the line *AB*. The increase of pressure is measured by a scale attached to *CG*.

ELECTRICAL THERMOMETERS. —

Temperature may be measured by electrical thermometers, of which there are two classes. In one class a conducting circuit is formed of two different metals, such a construction being frequently termed a *thermo-element*, a number of these connected together is known as

a *thermopile*. In this construction an electro-motive force is produced which is proportional to the difference of temperature of the junctions and may be measured by a sensitive galvanometer. If one of the junctions be maintained at a constant or known temperature, the temperature of the other may be computed from the reading of the galvanometer. For the measurement of

high temperature, metals having a high melting-point, such as platinum and a platinum-iridium alloy, may be used for the elements.

The LeChatelier pyrometer is an instrument of this class;

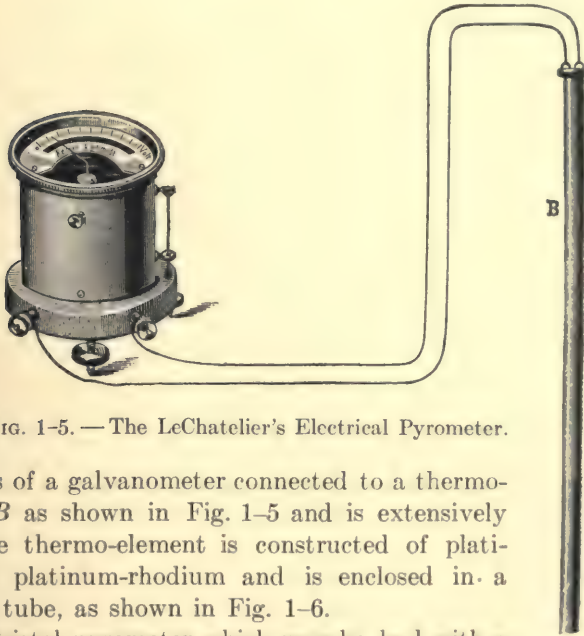


FIG. 1-5. — The LeChatelier's Electrical Pyrometer.

it consists of a galvanometer connected to a thermo-element *B* as shown in Fig. 1-5 and is extensively used; the thermo-element is constructed of platinum and platinum-rhodium and is enclosed in a porcelain tube, as shown in Fig. 1-6.

The Bristol pyrometer which may be had with a recording device belongs to the above class.

Another class of electrical thermometers is based on the law of increase of electric resistance of metals due to the rise of temperature. With this class of thermometers the difference in temperature can be determined from the measurement of the drop in potential for a known current passing through a coil. This method is employed in the platinum thermometers of Siemens, Calendar, and various others. The electrical thermometers are superior to all others for many uses.

OPTICAL PYROMETERS. — The approximate temperature of incandescent bodies may be determined by the color of the radiant rays. Pouillet, as the result of a large number of experiments, concluded that all incandescent bodies have a definite and fixed

color corresponding to each temperature, as shown in the following table:

Color	Temp. C.	Temp. F.
Faint red	525	927
Dark red	700	1292
Faint cherry	800	1482
Cherry	900	1652
Bright cherry	1000	1932
Dark orange	1160	2120
Bright orange	1200	2192
White heat	1300	2372
Bright white	1400	2552
Dazzling white	1500	2732

The fixed relation between color and temperature is due to the fact that the color of an incandescent body varies with the wave length which is a function of the temperature. A number of optical pyrometers have been devised which determine the temperature by the appearance of the heated body. The Mesuré and Noel pyrometer changes the wave lengths by the rotation of the plane of polarization of light passing through a quartz plate cut perpendicularly to its axis. In the use of the instrument the temperature is measured by noting the angle through which the analyzer is turned in order to produce a lemon yellow color.



FIG. 1-6. — The Le Chatelier Thermo-Element.

The Morse thermo-gage, which is extensively used in the steel industry, consists of an incandescent lamp with a rheostat arranged so that the current flowing through it and its consequent brightness may be regulated. When the film of the incandescent lamp becomes the same color as that of the object the temperature is computed from the reading of a milli-voltmeter arranged to measure the current. The temperature corresponding to a given electrical reading is determined by calibration.

The optical pyrometers are convenient and of approximate

accuracy in determining high temperatures of incandescent bodies. They are of no value in determining temperatures of combustible objects.

VAPOR THERMOMETERS. — The pressure produced by a saturated vapor confined in a closed vessel increases with the temperature, in accordance with a known law or a law which may be determined. By providing a suitable pressure gage and attaching it to a closed vessel of the proper shape the temperature may be obtained from the pressure readings, the value of which are known, or from the dial readings of the pressure gage, which may be graduated by trial into degrees of temperature. An instrument of this type is made by Schaeffer and Budenberg and called by them a *Thalpotasimeter*.

CALOMETRIC THERMOMETER. — The temperature can also be measured by calometric methods, by heating a body of known weight and specific heat to the temperature which it is desired to measure, then transferring this body with as little cooling as possible to a vessel containing a known weight of water.

The equation for this operation will be expressed as follows:

$$W(t - t') = ws(t_x - t),$$

in which W = equivalent weight of water and its containing vessel.

w = weight of heated body.

s = specific heat of hot body.

t' = original temperature of water.

t = final temperature of water.

t_x = temperature of hot body.

A ball of platinum, copper, porcelain, or burned fire clay answers nicely for the body to be heated.

FUSION THERMOMETERS. — The temperature can be approximately determined from the known melting-points of metallic bodies, on the principle that the temperature will be higher than the melting-point of a body that melts, and lower than the melting temperature of one that does not melt. In place of metallic bodies a series of fusible clay cones called "Seger cones," whose melting-points are known, are often employed in the same manner.

6. Specific Heat. — Different materials of the same weights have different capacities for absorbing heat for a corresponding change of temperature; thus one pound of water will absorb about nine times as much heat as one pound of wrought iron for the same change of temperature. This peculiar heat capacity of bodies compared with water is termed *specific heat*, which is usually defined as follows:

The specific heat of a body is the ratio of the quantity of heat required to raise that body one degree in temperature, to the quantity required to raise an equal weight of water at standard temperature one degree.

The specific heat of water is not quite constant, being nearly three fourths of one per cent higher at the boiling and freezing temperatures than at fifteen degrees Centigrade; this is shown by the following table from the book of Steam Tables by Professor Peabody:

SPECIFIC HEAT OF WATER

Centigrade	Fahrenheit	Specific Heat
0°—5°	32°—41°	1.0072
5 — 10	41 — 50	1.0044
15 — 20	59 — 68	1.00
20 — 25	68 — 77	0.9984
25 — 30	77 — 86	0.9948
30 — 35	86 — 95	0.9954
40 — 45	104 — 113	1.00
45 — 155	113 — 311	1.008
155 — 200	311 — 392	1.046

7. Specific Heat of a Permanent Gas. — In general the conditions under which the change of temperature occurs should be distinctly specified, for the temperature of a body may be varied by the mechanical work done in the compression or expansion which occurs. The change of volume due to increase of temperature is so small for solids and liquids that the external work in change of volume may be neglected, but such is not the case for gases. For this reason the conditions under which the heating of a gas takes place must be stated when referring to its specific heat, and it has become customary to speak of two spe-

cific heats in connection with any gas, namely, the specific heat at *constant volume*, and the specific heat under *constant pressure*. The former is the quantity of heat required to raise the temperature of a unit mass of the gas one degree when its volume is kept constant, and the latter the quantity of heat required to raise the temperature of a unit mass one degree when the pressure is kept constant, compared with that of a unit mass of water. In the first case the pressure increases while the volume is kept constant and no external work is done; in the latter the volume increases under constant pressure, and an amount of external work is done which is measured by the product of the pressure by the change of volume.

The specific heat of a gas at constant volume bears a known relation to the specific heat at constant pressure, so that if one be determined experimentally the other may be computed. It will be shown later that the specific heat at constant pressure equals the specific heat at constant volume plus a constant which depends on the nature of the gas. That is,

$$C_p = C_v + R \text{ when expressed in heat units, and}$$

$$K_p = K_v + JR \text{ when expressed in foot-pounds.}$$

The following table gives the specific heats of constant pressure and volume of the principal gases:

TABLE OF SPECIFIC HEATS

Gas	C_p	C_v	γ	Gas	C_p	C_v	γ
H	3.4090	2.4177	1.41	CO	0.2479	0.1738	1.41
O	0.2175	0.1543	1.41	CO ₂	0.2169	0.157	1.29
N	0.2438	0.1729	1.41	CH ₄	0.5929		
Air	0.2375	0.1684	1.41	C ₂ H ₄	0.4040	0.320	1.26
H ₂ O (vapor)	0.4805	0.3585	1.34				

In the above table C_p = Specific heat of constant pressure.

C_v = Specific heat of constant volume.

$\gamma = C_p \div C_v$.

The specific heat of a perfect gas is considered constant by most English writers who discuss the subject. Quite a number

of experiments have shown, however, that it varies with both the pressure and the temperature.

In the French work by A. Witz the following formula is given for the change of specific heat with pressure:

$C_p = a + b(p - 1)$, in which a and b are constants for each gas and p the pressure in kilos per square centimeter.

The specific heat of all vapors which are liquefied at moderate temperature undoubtedly increases with the increase of temperature. This subject is fully discussed in Chapter X of this work.

In this work the specific heat of gases will be considered constant unless otherwise mentioned. Any error caused by such consideration will not usually be serious, and by so doing the various formulas which express the heat capacities under actual working conditions are much simplified.

The specific heat of a mixture of various gases constituting a known weight or mass is equal to the mean specific heat of the mixture, and is found by multiplying the weight of each component part by its specific heat and dividing the sum of the products by the total weight.

8. The Heat Unit. — Heat is measured by its capacity to raise the temperature of a known weight of water. The unit of measurement is termed a *calorie* in the metric system, and a *British thermal unit* (B. T. U.) in the English system. A calorie is commonly defined as the heat required to raise one kilogram of water from the freezing-point to one degree Centigrade, and a British thermal unit (B. T. U) that required to raise one pound of water from 32 to 33 degrees Fahrenheit. Because of the variation in the specific heat of water near the freezing-point Professor Peabody in his Tables of Saturated Steam defines the thermal unit as that required to raise the temperature from 62 to 63 degrees Fahrenheit, or from about 15 to 16 degrees Centigrade.

The specific heat of water changes slightly for different temperatures as already noted, but it can be considered as constant without sensible error for all ordinary purposes in the measurement of heat when the temperature is maintained between the freezing and boiling points.

It is noted from the above statement that heat is measured not by the temperature alone but by its ability to heat a mass of water from one temperature to another; the total heat expressed

in thermal units being equal to the product of the weight of water by the change of temperature. Thus if 20 pounds of water be heated 25 degrees Fahrenheit, the heat required is the product of 20 times 25 = 500 B. T. U.

9. Mechanical Equivalent of Heat. — The mechanical equivalent of heat is the amount of work expressed in mechanical units which may be performed by the transformation of one heat unit into mechanical work. The value of the mechanical equivalent of heat was determined experimentally by Joule who found that 772 foot-pounds was equivalent to one B. T. U. or 425.6 kilogrammeters to one calorie. The determinations made later and with more accurate instruments by Rowland, reduced to the sea level and to 45 degrees of latitude, give the following values which are now generally adopted:

Expressed in calories, $J = 426.9$ kilogrammeters.

Expressed in B. T. U., $J = 778$ foot-pounds.

In this work J is used as the symbol for the mechanical equivalent of heat, and A as the reciprocal of J . That is, $J = 1/A$.

The experiment by means of which the equivalent value of the heat unit was determined in units of mechanical work serve to prove the general mechanical principle of the conservation of energy, which had been previously stated by Clausius as follows: "*In all cases where work is produced by heat, the quantity of heat consumed is proportional to the work done; and conversely, by the expenditure of the same amount of work the same quantity of heat may be produced.*" This principle is often called the first law of Thermodynamics.

10. Entropy. — One of the qualities or properties of heat which cannot be measured by any simple physical apparatus is termed "Entropy." This same quality was named by Rankine "the Thermodynamic Function"; its value is such that its change during a given time multiplied by the absolute temperature equals the total heat which may be transformed into mechanical work. From this definition it is noted that the product of absolute temperature by change of entropy is the measure of the capacity of heat for performing mechanical work.

As an illustration, if a gaseous body under pressure be allowed

to expand without receiving or giving off heat its entropy would remain constant and any mechanical work performed would be done at the expense of the heat existing in the body. A change which takes place without gain or loss of heat is termed *adiabatic*, which condition corresponds to that of constant entropy.

Expansion or compression of a body taking place without change of temperature is called *isothermal* expansion or compression, and lines drawn in a diagram indicating this property are termed "isothermal" lines.

Change of entropy with respect to heat is in many respects analogous to change of volume in respect to mechanical work; it has already been shown that the mechanical work is equal to the change in volume multiplied by the resistance or pressure overcome. If we denote the total heat capable of being transformed into mechanical work by Q , the change of entropy by ϕ and the absolute temperature by T , we shall have the following equations:

$$\begin{aligned} W &= (v - v') p. \\ Q &= (\phi - \phi') T. \end{aligned}$$

From the first of the above equations it is noted that no mechanical work can be done without a change in volume, and further, that the amount of work done is measured by the change of volume multiplied by the *pressure*.

From the second it is seen that no change in the amount of heat which a body contains can take place without a change in entropy, for when $\phi = \phi'$, $Q = 0$. The amount of heat transferred is measured by the change of entropy multiplied by the absolute temperature.

From the above

$$v - v' = \frac{W}{p}$$

From which is seen that the change in volume is equal to the mechanical work performed divided by the mean pressure.

$$\phi - \phi' = \frac{Q}{T}$$

From which it is seen that the change in entropy is equal to the total heat transformed into work divided by the absolute temperature.

11. Classification of Engines—The action of an engine is, in general, to produce motion against a resistance or to perform work. Engines are popularly classified in accordance with the nature of the working fluid, as hydraulic engines, steam engines, gas engines, oil engines, etc. They may be more scientifically classified in accordance with the nature of the working process as pressure engines and heat engines. In the *pressure engine* work is produced by change of pressure without change of temperature, as illustrated in the piston water engine. In the *heat engine* work is produced by transforming heat into mechanical work, which process is accompanied with change of temperature, and usually, also, a change of pressure.

In mechanical structure engines are of two classes, *reciprocating* and *rotary*. In the *reciprocating engine* a piston free to move in a cylinder is pushed backward and forward by alternate changes in pressure of the fluid against either face. The reciprocating motion of the piston is converted into a continuous rotary motion by a mechanism usually consisting of crank and fly-wheel which will be described later. In the *rotary engine* a rotary motion is directly produced by the force due to pressure, impulse, or reaction acting upon revolving blades or pistons arranged in a suitable casing. The term is often confined to a structure with a revolving piston in which motion is produced by a difference of pressure, whereas the term *turbine* is applied to the structure when rotation is produced by impulse or reaction of the jet. In a general way the turbine is a species of rotary engine. This treatise will be principally confined to internal combustion engines having a piston with reciprocating motion.

Engines are classified as *single acting* when the propelling force is applied to one side of the piston only, and as *double acting* when it is applied alternately to both sides.

Engines are classified as *simple*, *compound*, *triple*, *expansion*, etc., depending on the number of cylinders through which the working fluid passes in succession as it expands from highest to lowest pressure.

12. Classification of Heat Engines.—Heat available for use in a heat engine is usually produced by a species of chemical action termed *combustion*. Heat engines may be classified in accordance with the location of the place of combustion with

respect to the working cylinder as *external combustion engines* and *internal combustion engines*.

The external combustion engines include steam and other vapor engines, hot air engines, and some forms of gas or oil engines; the internal combustion engines include all the usual forms of gas and oil engines in which the fuel is consumed in the working cylinder. In this work the term *gas engine* will be frequently used as including all forms of internal combustion engines adapted to burn gas or vapor irrespective of the nature of the fuel.

Of these various types of engines the steam and the gas engines are the only ones of practical commercial importance at the present time. The hot air engine was built extensively about fifty years ago, and its theory was thoroughly investigated at that time. It failed as a commercial machine because of the high cost of repairs and operation; it is principally useful at the present time as illustrating the practical application of certain thermodynamical principles.

As the steam engine and hot air engine have a practical bearing on the internal combustion engine, a short description is inserted.

13. The Steam Engine. — The mechanism of the steam engine and its mode of operation should be familiar to all students of the internal combustion engine. The term steam engine is used here in its broad sense, including the boiler, the engine proper, and all the accessories necessary for its operation. In the mode of operation of the steam engine, steam is produced at any desired pressure by the combustion of fuel in a furnace beneath the boiler, which latter is a strong closed vessel containing a certain amount of water and into which water is introduced by a feed pump as desired. The steam engine proper is provided with a cylinder in which is fitted a piston which is propelled by the steam pressure acting on one or both sides. The admission and discharge of the steam are controlled by a valve or valves moved by the mechanism of the engine, the form of which varies greatly with different types. In the more common form, a slide valve is used which is propelled backward and forward by a valve rod moved either by an eccentric or by a short crank attached to the main shaft. The valve is operated so as to admit steam at nearly boiler pressure back of the piston for a portion of the stroke, and

then to cut off communication with the boiler, after which the piston is pushed forward by the expansive force of the steam. The valve is moved at the end of the stroke so as to open communication between the cylinder and the exhaust pipe, which in the case of a non-condensing engine discharges into the atmosphere and in case of a condensing engine discharges into a more or less perfect vacuum.

The motion of the piston is communicated, by means of a piston rod which slides through a stuffing-box at the end of the cylinder, to a block called a *cross-head* which moves in guides, from which motion is communicated by a connecting rod to the crank of the main shaft so as to produce rotary motion.

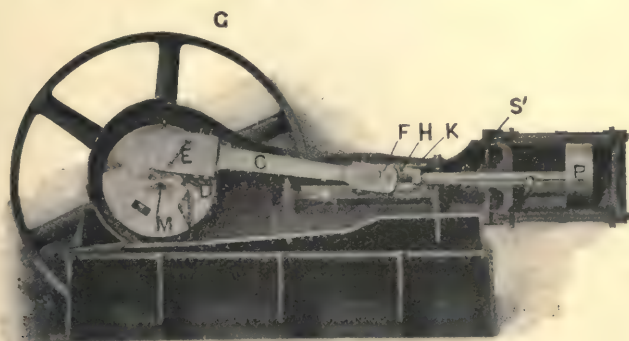


FIG. 1-7. — Double Acting Steam Engine.

The train of mechanism of the ordinary double-acting steam engine which is used to communicate motion from the piston to the fly-wheel is shown in Fig. 1-7, in which *P* represents the piston, *p* the piston rod, *H* the cross-head, *C* the connecting rod, *M E* the crank, *E* the crank pin, *F* the wrist pin, *M* the main shaft, *G* the fly-wheel, *S* the stuffing-box, which is used to prevent leakage of steam around the piston rod. The cross-head moves in guides *K* which direct its motion. One end of the connecting rod *E* has a circular motion, the other a rectilinear motion. In the view referred to the valves which admit steam alternately to the ends of the cylinder are not shown.

A view of a two-cylinder single-acting engine is shown in

Fig. 1-8 in section. In this engine the steam is admitted only at one end of the cylinder at the proper intervals of time by the sliding motion of the piston valve *V*, which is operated by the

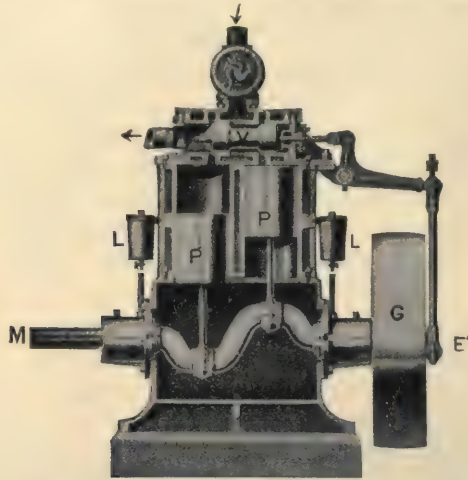


FIG. 1-8. — Two-cylinder Single-acting Steam Engine.

mechanism of the engine by means of a small crank attached to the main shaft and the various link connections shown.

Steam engines are frequently built compound, in which case the steam works in succession in a small and large cylinder, termed respectively the high-pressure and low-pressure cylinder. These cylinders may be arranged side by side as in the view of the single-acting engine, Fig. 1-8, or they may be arranged in tandem as shown in Fig. 1-9. In the tandem compound engine shown

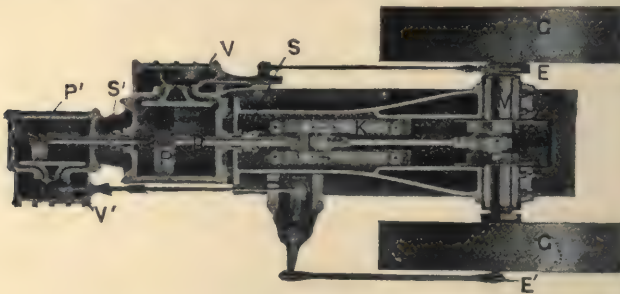


FIG. 1-9. — Tandem Compound Steam Engine.

the supply of steam is admitted, to the high-pressure cylinder by the slide valve V' operated from the small crank E' , and to the low-pressure cylinder by the slide valve V operated from the eccentric E .

Since the application of pressure to the piston as described above is periodic and not constant, a fly-wheel consisting of a heavy mass of metal must be applied as shown, to produce uniform motion. To regulate the speed a governor is used, which is constructed so that the variation in centrifugal force either tends to cut off the supply of steam or to close the steam valves if the speed become higher than desired, or the reverse.

An indicator or pressure volume diagram of a steam engine is shown in Fig. 1-35. In this diagram vertical distances are proportional to pressures per square inch acting on the piston, and horizontal distances to the space passed through by the piston. The diagram shows the relation of pressure and volume at any point. The work done on the piston is proportional to the area of the diagram.

The steam engine, as will be shown later, does not realize in the work performed as great an efficiency on the basis of the heat values of the fuels employed as the gas engine; but it has the advantage of being adapted for the burning of solid fuels under its boiler without converting the same into gas, and this in a measure sometimes compensates for the greater heat losses. The steam pressure is exerted on the piston for a much larger portion of the stroke than an explosion in a gas engine piston, and as a consequence the inertia of moving parts is depended upon less for uniform speed than in the gas engine, as will be afterwards shown.

In the past the steam engine has had a great advantage over the gas engine, due to the fact that it has been more reliable in operation and could produce a more nearly uniform motion. The development of the gas engine has, however, removed in a large measure such defects, and at the present time there is no great difference in respect to reliability and regulation on the part of these two classes of engines. The use of a producer in which gas can readily be made from solid fuels also equalizes any advantages which the steam engine has had from that source.

14. Hot Air Engines.—The hot air engine is principally of importance to-day for its scientific value. Its actual commercial use is confined to pumping small quantities of water under favorable conditions. It is of scientific interest because it is the only heat engine yet produced which represents almost perfectly the standard ideal cycle of Carnot, with which the operation of nearly all heat engines is compared.

In the hot air engine a mass of air is successively heated and cooled; during the time that it is heated either its volume or pressure increases, and during the time it is cooled the reverse operation takes place. Mechanical work is performed by the change of volume or pressure which is utilized for moving a piston.

The principal varieties of air engines may be classified by the following distinctive features: 1. Change of temperature at constant pressure. 2. Change of temperature at constant volumes. 3. Heat received and rejected at a pair of constant pressures.

Ericsson's engine, best known as the caloric engine, may be

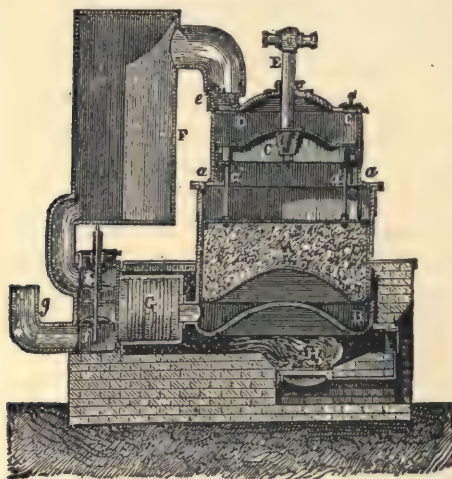


FIG. 1-10. — Ericsson Hot Air Engine.

taken as an example of the first class. In this engine, air is admitted from the atmosphere to the compressing pump at the lowest working temperature, and compressed, the temperature being maintained constant by the action of some refrigerating apparatus. The air when compressed enters a receiver. It is then admitted to the working cylinder, being heated on its passage to the higher temperature, so that its vol-

ume is increased and the pressure remains constant under the movement of the piston, then expands with its temperature maintained constant at the higher limit, and is finally expelled into the atmosphere, giving up its heat to the regenerator, to be used in heating the volume of air next introduced.

This engine is represented in Fig. 1-10. *B* is a working cylinder, placed over the furnace *H*. This cylinder consists of two parts; the upper part in which the piston works is accurately turned, and the lower part in which the air receives heat from the furnace is less accurately made. *A* is the piston of the cylinder, consisting of an upper part which is accurately fitted and provided with metallic packings so as to work air-tight in the upper part of the cylinder. The lower part is somewhat smaller than the cylinder, is hollow, and filled with brick dust, fragments of fireclay, or some slow conductor of heat. The cover of the cylinder *B* has holes in it marked *a* to admit the external air to the space above the piston.

D is the compressing pump with piston, *C*, which is connected to the piston *A* by three or four piston rods, of which two are shown at *d d*. The space between the piston *C* and *A* is open to the external air. In operation the air is drawn into the compressing pump through the valve *c*, and is forced out after being compressed through the valve *e* into a receiver marked *F*. It is admitted at proper intervals of time by the valve *b* into the working cylinder *B*, being heated in its passage by hot plates in the vessel *G*, termed the regenerator.

It is further heated while in the working cylinder by the heat from the furnace *H*, the effect of which is to increase the temperature and volume underneath the piston. The increase of volume drives the piston to the end of its stroke. The exhaust valve *f* is opened by a mechanism connected to the engine and the working gases are forced outward through the regenerator, *G*, and discharge into the atmosphere through the pipe *g*. The regenerator is a vessel nearly filled with metallic plates which are heated by the escaping gases and give up heat to the engine gases, thus reducing in large measure the heat wastes from the engine.

In some forms of the Eri sson engine the air entering the compressor through the valve *c* is taken from the exhaust opening *g*; with this arrangement the same mass of air is repeatedly warmed and cooled, the changes of temperature taking place at constant pressures.

Eri sson's caloric engine was employed to drive a ship across the Atlantic in 1853. The ship was 250 ft. long, had paddle

wheels 32 ft. in diameter. On its first trial trip the ship made twelve knots an hour with the wind, burning six tons of fuel per day. On the second trial the maximum speed was nine knots. After this unfavorable circumstances came to light and in 1855 the engine was taken out and a steam engine substituted.

Extended accounts of the Ericsson and other hot-air engines will be found in Knight's Mechanical Dictionary, 1873, in Appleton's Cyclopedia of Mechanics, 1878, in Bourne's work on the Steam, Hot Air, and Gas Engine, and in Rankine's Steam Engine.

Figure 1-11 represents the reciprocating parts of an air engine of the class in which temperature is changed at constant volume. Such an engine was designed and built by Dr. Robert Stirling about 1850, but improved by various other inventors, and is still built and sold for pumping small quantities of water as redesigned by Rider.

In the figure *D C A B* is the air receiver or heating and cooling vessel, which is provided with a furnace underneath, not shown in the diagram; *G* is the working cylinder with the working piston *H*. The receiver and cylinder communicate freely

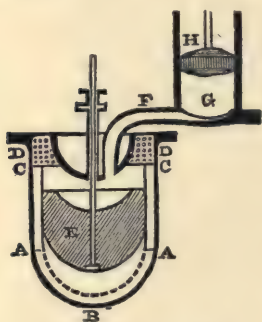


FIG. 1-11. — Stirling Hot Air Engine.

through the passage *F*, which is open at all times when the engine is working. Within the receiver is an inner receiver or lining of a similar figure, which has its bottom pierced with many small holes shown with dotted lines in the cut. The annular space between the receiver and its lining extending along the side of the receiver contains the regenerator, which consists of a series of oblong strips of metal with narrow passages between them. The inner surface of the cylindrical part of the lining from *A A* to *C C* is turned and the plunger *E* is fitted nicely in this portion. The upper portion of the receiver *D D* is supplied with a horizontal coil of fine copper tube, through which a current of cold water is forced or it is jacketed with cold water, and is termed the refrigerator. It is thus noted that the bottom of the receiver is kept hot while the top is kept cold. The plunger *E* is con-

nected to the working mechanism of the engine so as to transfer a certain mass of air, which may be called the working air, from the hot to the cold end of the receiver, and in so doing making it pass up and down through the regenerator. The mechanism for moving the plunger *E* is so adjusted that the up-stroke of that plunger takes place when the piston, *H*, is at or near the beginning of its forward stroke, and the down stroke of the plunger when the piston *H* is at or near the beginning of the back stroke.

An air engine of class 3, which received and rejected heat at constant pressures, was designed by Jewell, but probably was never put into practical use.

HOT AIR ENGINES OPERATED BY PRODUCTS OF COMBUSTION. — Another form of air engine, which Rankine in his *Steam Engine* terms a *furnace gas engine*, was first designed by Cayley of England and Barré of France, and was redesigned and improved and put on the market in this country about 1865 by Wilcox.

The engines of this class operate similar to a steam engine, pressure being produced by combustion in a closed furnace instead of in a steam boiler.

In the operation of this engine (see Fig. 1-12) a pump draws air from the atmosphere through valve *F*, compresses it, and forces it into a strong air-tight furnace *C* through pipe *H*, where its oxygen combines with the fuel; then the hot gas produced by the *combustion* mixed with air is admitted into the working cylinder through pipe *B* and valve *I*, under pressure produced by the temperature of combustion, where it drives the piston *P* through part of its stroke at full pressure and through the remainder by expansion, until it falls to atmospheric pressure. It is discharged through an exhaust valve not shown and a pipe *X*. The entering air is heated by a regenerator which is warmed by the exhaust gases. In the

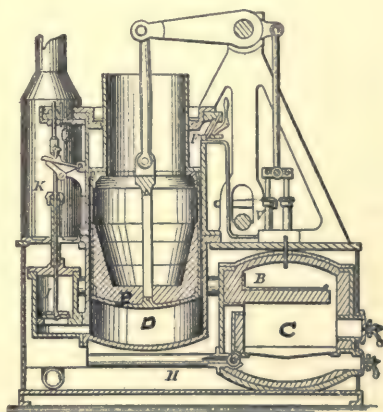


FIG. 1-12. — Wilcox Hot Air Engine.

form shown (U. S. patent May 19, 1865) the furnace is fed with coal through a double valve, which is so constructed that it can be introduced without permitting the escape of more than a very small quantity of the compressed air. In the Wilcox engine, patented Sept. 19, 1865, the furnace was fed with petroleum oil under pressure.

Respecting Cayley's engine, Rankine states: "The cylinder, piston, and valves of this engine were found to be so rapidly destroyed by the intense heat and the dust from the fuel that no attempt was made to bring it into general practical use." Wilcox's engine never met with commercial success.

Most forms of the gas turbine belong to a class, in which pressure is produced by the heat of combustion of the gases before entering the turbine, either in a closed combustion chamber or in the inlet pipe. As the supply to a turbine is continuous no inlet or exhaust valves are required, and hence the trouble experienced in the early forms of this engine is greatly reduced.

15. Structure and Mode of Operation of the Gas Engine.—The mechanism of the ordinary gas engine, using the term in its general sense as covering all internal combustion engines, is similar in most respects to that of the steam engine, which has already been briefly described. It consists of a cylinder containing a piston which is moved by the pressure produced by the explosion of a charge consisting of a mixture of gas or vapor and air in the cylinder. The motion of the piston is communicated to a main shaft by a connecting rod similar to that used in the steam engine. The valve mechanism of the gas engine serves to admit and discharge the charge at the proper interval of time and is operated by the mechanism of the engine. In the early development of the gas engine a slide valve was used to a considerable extent, but at the present time the poppet valve is commonly used as it has been found to withstand high temperature better than the other form.

In order to start the gas engine some external force must be provided to introduce the combustible charge into the working cylinder and give it the initial compression. This may be done in the first instance by revolving the engine by extraneous power, which puts the piston in motion and serves to draw in the necessary gas and air by suction and to compress the same. After the

engine is in operation the inertia of the moving parts keeps it in motion and serves to draw in and compress the charge.

It is quite evident that the amount of work performed will be proportional to the mass or weight of the charge. For this reason it is desirable that the charge be under as much compression as practicable at the time of ignition, and all modern internal combustion engines provide means for compressing the charge previous to ignition either outside or inside of the working cylinder. In nearly every case, in the modern engine, the compression is completed in the working cylinder.

The principal events in the operation of an internal combustion engine are as follows:

1. Charging or suction, during which time the charge is drawn into the cylinder.
2. Compression, during which time the charge is compressed.
3. Ignition, explosion, and expansion, during which time heat is supplied which causes the combustion or explosion followed by the expansion due to increase of volume caused by motion of the piston. Ignition may take place under conditions of (1) *constant volume*, (2) *constant pressure*, or (3) *constant temperature*.
4. Exhaust, during which time the products of combustion leave the cylinder.

16. Classification of Internal Combustion Engines. — Gas engines are scientifically classified in accordance with the mode of applying heat during ignition as follows:

1. Engines receiving heat with charge at constant volume; these will be called in this work *Explosion Engines*.
2. Engines receiving heat with charge at constant pressure; these will be called in this work *Pressure Engines*.
3. Engines receiving heat with charge at constant temperature; these will be called in this work *Constant Temperature Engines*.

In the first of the above classes of engines the charge may be ignited with or without previous compression; consequently this class may be subdivided into *non-compression* and *compression engines*, the non-compression engine has entirely gone out of use because of its low efficiency and small capacity for a given size. Engines of any class may be either two or four stroke cycle engines as explained.

This classification is based on the characteristic equation expressing the relation between pressure, volume, and temperature of a given weight of a perfect gas, which, as will be shown later, is

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

in which p = absolute pressure.

v = the volume.

T = absolute temperature.

R = a constant for any given gas.

It is evident that the heat may be received while any of the variables, pressure, volume, and temperature, remains constant, that is, at constant volume as in Class 1, constant pressure as in Class 2, or constant temperature as in Class 3.

Internal combustion engines are often unscientifically classified by the nature of the working fluid as gas engines, petrol engines, and oil engines. This classification gives no consideration to the fact that any of the above classes will operate with any of the fuels named. The term *gas engine* is frequently used in this work in its general sense, as applying to any form of internal combustion engine.

ENGINES IGNITING AT CONSTANT VOLUME, OR EXPLOSION ENGINES. — In these engines, which are the ones commonly used, the various operations are performed in the order mentioned above in each working cycle of the engine. These engines may be divided into two classes accordingly as they perform these operations, in one end of the cylinder, (a) in four strokes or (b) in two strokes.

In this class of engines the combustion is practically instantaneous and of the nature of an explosion, taking place under normal conditions while the volume of gas remains constant, thus producing an extremely rapid rise of pressure.

(a) *Four-Stroke Cycle Engine*. — The internal combustion engine most commonly used ignites the charge while its volume remains stationary, and requires four strokes for one cycle of operation. For this reason it is known as the *four-stroke cycle* or *four-cycle engine*. This engine as ordinarily built is a single-acting engine with all the operations performed on one side of

the working piston. A diagram showing its general construction and mode of operation for each stroke is shown in Fig. 1-13. In the operation of this engine the charge is drawn in during the first out stroke of the piston, is compressed during the return stroke, is ignited with the piston stationary at the end of the stroke and with the volume of the charge constant. It expands during the next out stroke and is exhausted and expelled from the cylinder during the next in stroke.

An engine of this kind was first described by Beau de Rochas in 1861, it was first built by Otto in 1876. The cycle on which it operates is for this reason often called the Beau de Rochas or Otto cycle.

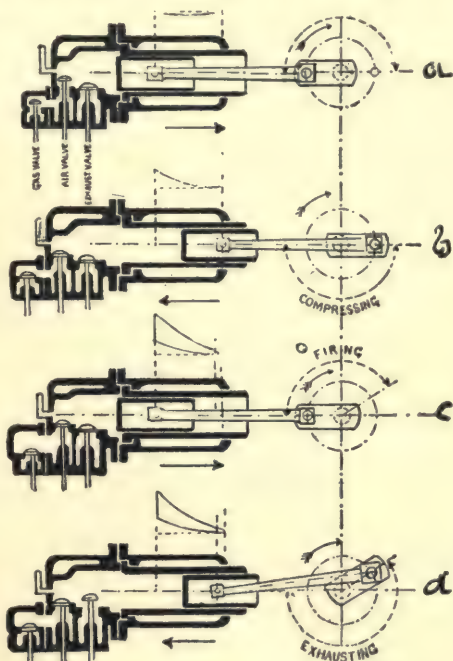


FIG. 1-13. — Diagram of Four-cycle Engine.

(b) *Two-stroke Cycle Engine.* — An internal combustion engine, igniting as before, which is used for many purposes, is designed to perform the four operations above referred to in two strokes and is known as a *two-stroke cycle engine* or a *two-cycle engine*. A common form of such engine is shown in Fig. 1-14, in which form the engine is in part a double-acting engine and both sides of the piston constitute closed chambers. The crank case is made tight by the use of stuffing-boxes on the main shaft, and the suction operation is performed by the inward stroke of the piston which draws the charge into the crank case through the inlet *I*, where it is partially compressed by the out or return stroke of the piston and transferred through port *a*, which is uncovered at the proper time by the piston, to the ignition side of

the piston. At the same time the charge of previously burned gases is escaping through the port *E*. The compression is completed in the working cylinder *C*, after the piston has

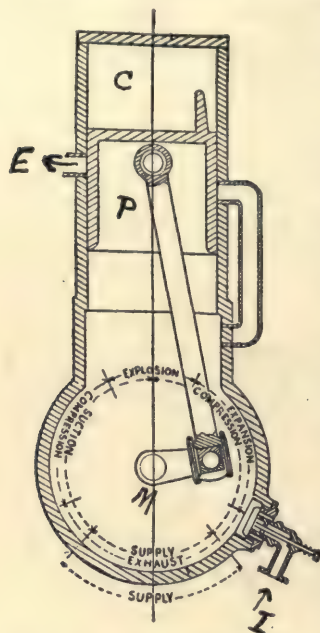


FIG. 1-14. — Two-cycle Engine.

closed the transfer port *A*. Ignition occurs at the beginning of the out stroke and when the piston and volume in the working cylinder are stationary as in the preceding case. In the action of this engine suction takes place below the piston at the same time that compression takes place above, and compression takes place below the piston at the time of expansion above, as shown by the diagram of the crank circle in the figure. In certain large engines which are now made to operate on the two-stroke cycle system the suction and preliminary compression of the charge, which is performed in the engine just described by the working piston, is performed in a separate cylinder, the compression being completed in the working cylinder.

The method of igniting the charge in common use will be described at length later in the book. In the class of engines referred to it consists of means for firing the charge instantaneously when under compression, and with its volume constant by an electric spark, a hot tube, or an open flame.

ENGINES IGNITING THE CHARGE AT CONSTANT PRESSURE. — The only engine of this class of practical importance is the Brayton, although under many conditions the Diesel engine ignites under constant pressure. The Brayton engine was at one time used extensively in America but is not now manufactured. In this engine the combustible and air for supporting combustion were supplied to the working cylinder under pressure which remained constant until the inlet valve closed; the combustion taking place during the admission of the air and com-

bustible. The work is performed by pressure acting during increase of volume in much the same manner as in the steam engine. In the Brayton engine, one form of which is shown in Fig. 1-15, the compression is performed in a compressor distinct from the power end of the working cylinder, and the heat is supplied at constant pressure. In the figure *B* is the working piston arranged to move in the cylinder *A*. The lower part of the cylinder *A* is the working cylinder, the upper part the air compressor which is arranged to deliver air into the reservoir *C*.

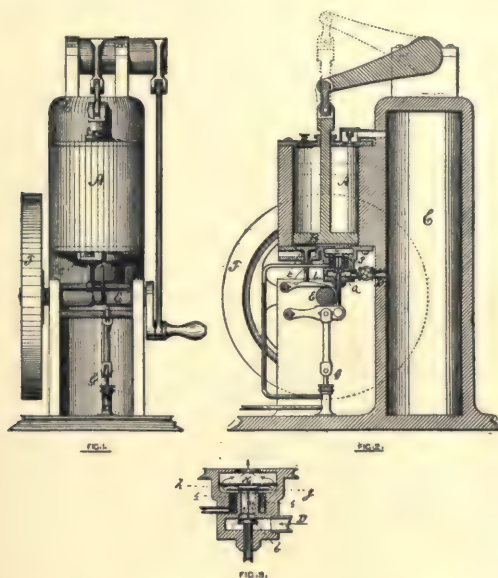


FIG. 1-15. — The Brayton Engine.

Oil is injected by the pump *G* into a vaporizing device (shown on a larger scale in 3 of Fig. 1-15) when it comes in contact with compressed air from the reservoir *C*. The inlet valve *b* and exhaust valve *c* are operated by suitable cams on the cam shaft *E*.

ENGINES IGNITING THE CHARGE AT CONSTANT TEMPERATURE. — Another class of engine, patented by Diesel, is arranged to supply the fuel during a portion of the working stroke at such a rate as to maintain the temperature constant, the working cylinder having previously been filled with air during

a suction stroke and compressed during a return stroke. In the Diesel engine the compression is sufficient to raise the air to a temperature high enough to ignite the fuel as it enters the working cylinder.

The Brayton engine as above described is a two-stroke cycle engine and the Diesel a four-stroke cycle engine, but both engines could be constructed to operate with either cycle.

17. The Engine Indicator.— This is an instrument designed to draw a diagram with ordinates proportional to the pressure which acts inside the cylinder at each point during the working and return stroke of the piston. It is briefly described here in order to give the student an idea of the method of obtaining the pressure volume diagrams which are frequently referred to in the work.

It consists essentially of (1) a part carrying a sheet of paper which is moved by proper mechanism in corresponding directions and proportional to the piston of the engine, and of (2) a part which carries a pencil which is moved a distance proportional to the pressure per square inch acting upon the piston.

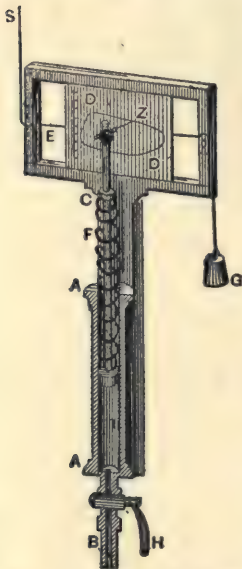


FIG. 1-16. — The Watt Engine Indicator.

The engine indicator was first designed by James Watt, substantially as shown in Fig. 1-16. This indicator was constructed with (1) a flat plate, *D B*, on which paper could be mounted which was moved in proportion to the motion of the engine piston, and (2) a cylinder *A A* which could be put in communication with the working cylinder of the engine by a three-way cock *H* and pipe *B*. In the cylinder *A A* was a piston whose motion was resisted by a spiral spring arranged to carry on its piston rod a pencil so constructed as to draw a diagram on the moving plate with ordinates proportional to the pressure.

The cock *H* can be turned so as to put the cylinder *A A* in communication with the air for the purpose of drawing a line showing the atmospheric pressure, which is called

the "atmospheric line." This simple form of indicator, although containing the essential elements of the modern indicator, was crude in construction and gave results which were far from accurate.

The modern indicator is an instrument of precision, and differs principally from that designed by Watt by the substitution (1) of an oscillating drum, called the paper drum, for the flat reciprocating plate, for carrying the paper, (2) of movable indicator springs in place of a fixed one, making it possible to regulate the length of the ordinates, and (3) a multiplying pencil motion in place of the direct one, whereby the motion of the pencil on the indicator drum is made greater than that of the indicator piston.

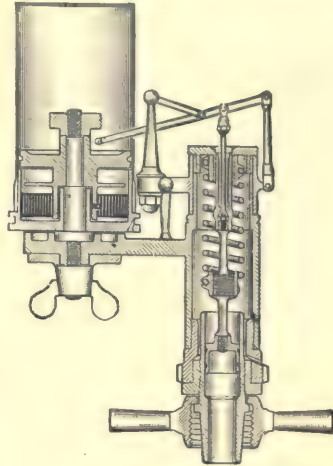


FIG. 1-17. — The Thompson Indicator.

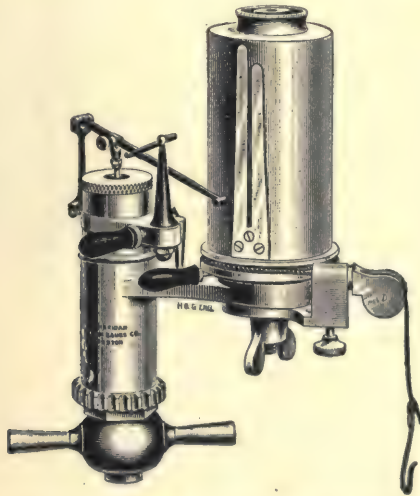


FIG. 1-18. — The Thompson Indicator.

A sectional and perspective view of the Thompson indicator, made by the American Steam Gauge & Valve Company, is

shown in Figs. 1-17 and 1-18. A perspective view of the Crosby indicator, which differs from the Thompson principally

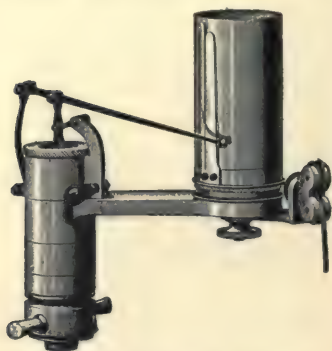


FIG. 1-19. — The Crosby Indicator.

in the construction of the indicator spring and pencil motion, is shown in Fig. 1-19. For gas engine work the indicator spring is liable to be injured by heat. To lessen these difficulties most of the makers supply indicators with external springs, as shown in the attached view of the Tabor indicator, Fig. 1-20.

The authors have found from an extensive experience in indicating gas engines that the indicator spring when arranged as in Fig.

1-17 can be kept from injury by surrounding the working cylinder with a water jacket or cup filled with water. The indicator cylinder is connected to the working cylinder by a pipe containing an indicator cock, which is arranged as in the Watt indicator to connect either with the air or the engine. The indicator drum is usually connected to some form of *reducing motion* by the indicator cord, which will move the surface of the drum proportional to the motion of the piston and not to exceed two or three inches, regardless of the stroke of the piston. Several forms of reducing motions with schemes for connecting will be shown in the chapter on the testing of gas engines, but will not be further referred to here.

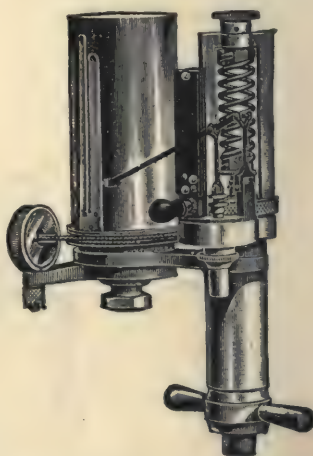


FIG. 1-20. — The Tabor Indicator with External Spring.

The class of engine indicator as described above is not adapted to take diagrams at extremely high speeds because of the inertia of the moving parts and because of the error due to stretching

of cords or flexible parts which connect the paper drum to the moving parts of the engine. For high speeds the optical indicator is preferable.

The *optical indicator* has been designed so that the only moving part is a small mirror which is arranged so as to project a ray of light on a ground glass screen or on a photographic plate.

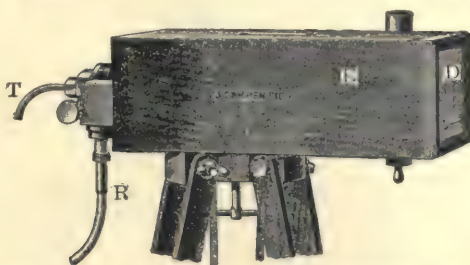


FIG. 1-21. — Perspective View of Manograph.

The mirror is moved in one direction an amount proportional to the pressure acting on the piston of the engine, and in a direction at right angles an amount in proportion to the motion of the piston, so that the joint movement is proportional to the pressure volume diagram, the area of which represents the mechanical

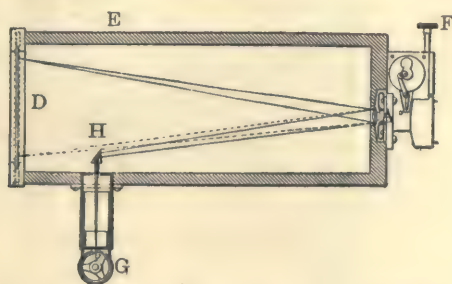


FIG. 1-22. — Horizontal Section of Manograph.

work performed by the working fluid in the engine cylinder. Such indicators are affected only to a slight degree by the rotative speed of the engine.

There are two instruments of this class on the market, one the *Manograph*, manufactured by J. Charpentier, Rue de Lambre, 20, Paris, the other the *Optische Indicator*, constructed by the

Elsaessische Electricitaets-Werke, Strassburg. A perspective view of the *Manograph* mounted on a tripod is shown in Fig. 1-21, a vertical section of same in Fig. 1-22; a detailed view of the mirror and engine connections are shown in Figs. 1-23 and 1-24.

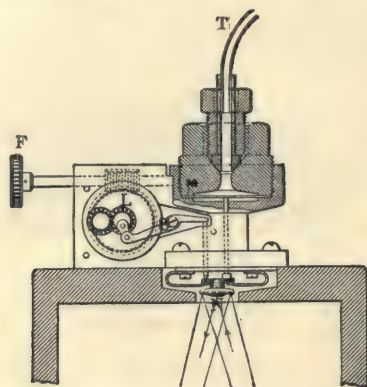


FIG. 1-23. — Section of Manograph Mirror.

The general construction is that of a photographic camera connected to the engine by tube *T*, and by flexible shaft *R* con-

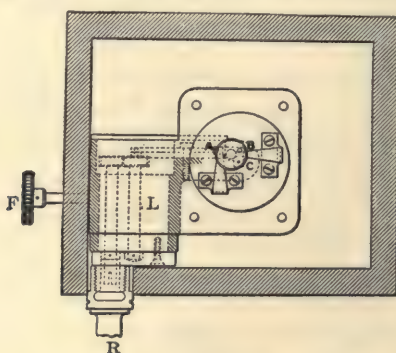


FIG. 1-24. — Manograph Engine Connection.

nected to the main shaft of the engine. An acetylene or electric lamp is located at *G* and its ray of light is projected through a perforated diaphragm to a prism, *H*, and thence to mirror *N*, at the back of the camera. The mirror *N*, supported on springs,

Fig. 1-23, is connected by a pin with the diaphragm *M*, which is in communication with the engine cylinder, so as to be tilted in one direction an amount proportional to the change of pressure in the engine cylinder. The mirror *N* is also tilted in a direction at right angles to the first motion by means of a crank with lever connection which is rotated from the shaft in proportion to the motion of the engine. The motion of the small crank can be set

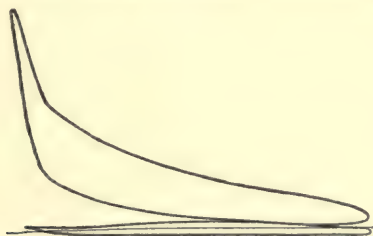


FIG. 1-25 — Diagram with Manograph.

in phase with that of the engine crank by the thumb screw *F* so that it will give a motion directly proportional to the piston of the engine. To make the errors as small as possible the angular motion of the mirror is made very small. The pressure scale of the manograph should be determined by carefully comparing the photographic diagram with a known pressure. The instrument



FIG. 1-26. — Optical Indicator.

is arranged to give a diagram which would have exactly correct proportions when the connecting rod has a length 4.5 times that of the crank, which is nearly the average proportions in actual practice and would make the resulting error small for other conditions.

Figure 1-25 represents a diagram taken with the Manograph from an engine making 1500 turns per minute and giving a maximum pressure corresponding to 158 pounds per square inch.

The Optische Indicator differs from the Manograph principally in details of construction. Its general appearance is shown in Fig. 1-26. A diagram taken from a motor operated with gasoline making 1000 turns per minute is shown in Fig. 1-27, the pressure scale of which is about 120 pounds per square inch.

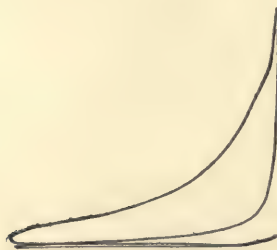


FIG. 1-27. — Diagram with Optical Indicator.

18. Indicated and Brake or Delivered Horse-Power. — The indicated horse-power which is generally denoted by the symbol I. H. P. is proportional to the area of the diagram obtained by use of the engine indicator, since this diagram has ordinates which are proportional to the pressures acting upon the engine piston at each point during the working and return strokes, and abscissa proportional to the corresponding space moved through by the engine piston.

The indicated horse-power (I. H. P.) is computed by use of the formula

$$\text{I. H. P.} = \frac{\text{plan}}{33000}$$

in which p = the mean effective pressure (m. e. p.) for the cycle of operation, acting on each square inch of the piston.

l = length of stroke in feet.

a = net area of piston in square inches.

n = number of cycles per minute.

33,000 = number of foot-pounds per minute in one horse-power.

The mean effective pressure (m. e. p.) is the mean ordinate

for all the strokes constituting the cycle multiplied by the proper pressure scale; it is best obtained by finding the net area by use of a *planimeter* (an instrument which will be described later), which is to be divided by the length of the diagram and multiplied by the scale of the indicator spring. The other quantities in the formulæ depend upon the dimensions and speed of the engine.

The *brake* or *dynametric horse-power*, for which the symbol in this work will be D. H. P., is that delivered from the main shaft of the engine and is consequently less than the indicated horse-power by an amount equal to the engine friction and internal losses.

The brake horse-power is usually measured by use of a special form of absorption dynamometer known as the Prony brake. Various forms of this brake have been employed, of which

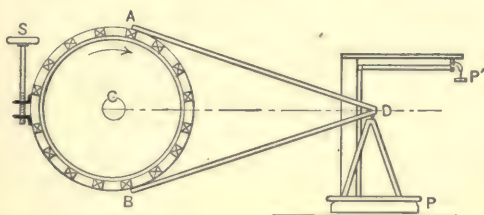


FIG. 1-28. — A Prony Brake.

two only are considered in this place. One form is shown in the diagram Fig. 1-28, which consists of a series of blocks connected by a leather strap or strip of iron and arranged so as to rub on the surface of a wheel attached to the main shaft. The brake is provided with two arms, the free end of which rests on a pair of scales. The amount of friction may be varied by use of a hand wheel or similar device as shown at *S*. The horizontal distance from the center of the wheel to the end of the arms is known as the arm of the brake and is denoted in the formulæ which follow by *a*. In the use of the brake the load is applied by turning the screw and is measured by the reading on the weighing scale. In a brake with the arm on one side only, as shown in the figure, the amount required to balance the overhanging brake arm must be deducted from the reading of the scales to give the net load.

The horse-power is calculated from the formula

$$\text{D. H. P.} = \frac{2 \pi a n W}{33000}$$

in which n = number of revolutions per minute.

a = the brake arm in feet.

W = the net load on scales corrected for unbalanced effect of brake.

Another form of brake is shown in diagram, Fig. 1-29, which is convenient for testing small engines. It consists of a rope or strap which makes one or more turns around the wheel, the tension or pull on both ends of which must be known or measured. In the form shown a weight of known amount, w , is applied at one end, and a spring balance is employed to measure the resistance at the other end.

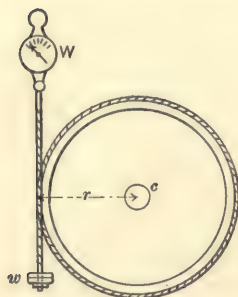


FIG. 1-29. — The Rope Brake.

The formula for this brake is as follows:

$$\text{D. H. P.} = \frac{2 \pi r n (W - w)}{33000}$$

in which r = radius of the wheel in feet to center of strap.

W = the principal scale reading.

w = the lesser scale reading or weight carried.

In a modification of this brake the principal tension is received by a framework resting on a pair of scales, and the smaller resistance is absorbed by an upward pull on the platform of the same scales. With this arrangement the scale reading gives directly the difference of weights $W - w$.

In the use of the Prony brake heat is generated equivalent to the mechanical work absorbed. If the load is heavy it will be necessary to circulate water or some other heat-removing fluid inside of the rim of the revolving wheel or in some equivalent place.

19. Forms of Indicator Diagrams. — A few of the typical forms of indicator diagrams are considered in this place for the purpose of making the student familiar with the subject. They will be discussed at length later in the work.

Figure 1-30 is a hypothetical diagram of a four-cycle explosion engine with the events which take place on the various strokes marked. Thus the suction stroke is represented by $d e$, the compression by $e f$, the explosion by $f a$, the expansion by $a b$, the exhaust by $b c d$. The atmospheric line not clearly shown in the diagram would occupy a position intermediate between $c d$

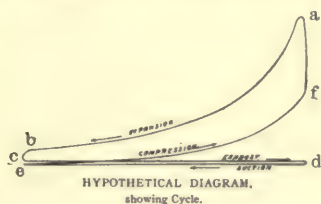


FIG. 1-30. — Four-cycle Engine.

and $e d$. The lines $c d$ and $e d$ on the indicator usually coincide with the atmospheric line for the reason that the spring used is too stiff to show such small variations in pressure as exist between the atmospheric line and the exhaust and suction lines.

Figure 1-31 shows diagrams of a two-cycle explosion engine in which the upper diagram, $a b c f$, is taken in the working cyl-

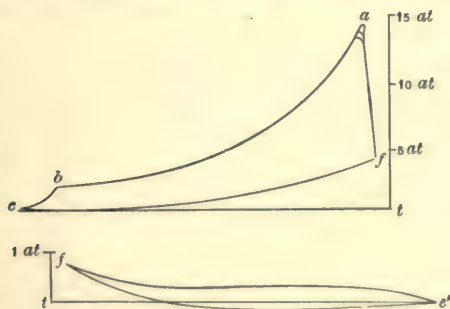


FIG. 1-31. — Two-cycle Engine Diagram.

inder, and the lower diagram, $e' f'$, in the compressor. In this diagram the net work is the difference between that shown on the first diagram and that on the second.

Figures 1-32 and 1-33 are diagrams from a Brayton or constant pressure engine, Fig. 1-32 being taken from the working cylinder and Fig. 1-33 from the compression cylinder. It will be noted

from Fig. 1-32 that the pressure remains constant from a to a' , at which time communication is cut off from the compressor, after which the fluid expands from a' to c in much the same



FIG. 1-32. — Diagram from Brayton Working Cylinder.

manner as in the steam engine. The net work is proportional to the difference of the areas of the two diagrams.

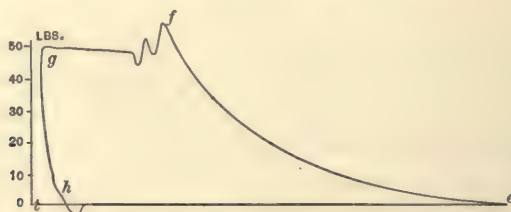


FIG. 1-33. — Diagram from Brayton Compressor.

Figure 1-34 is a diagram from a Diesel engine in which the temperature is supposed to be constant during that portion of the stroke represented by $a b$, during which time fuel is being supplied to the working cylinder.



FIG. 1-34. — Diesel Engine Diagram.

Figure 1-35 is a diagram from a steam engine, the expansion line of which as referred to an hyperbola, $c e f g h$, which is asymptotic to the line of no pressure, $C D$, and of no volume, $C B$. Points in the hyperbola are obtained if the initial point, c , is

known, by drawing a vertical from c ; then from C draw diagonals crossing cb and AB . The intersection of a horizontal line from the intersection of the diagonal and cb , with a vertical line from

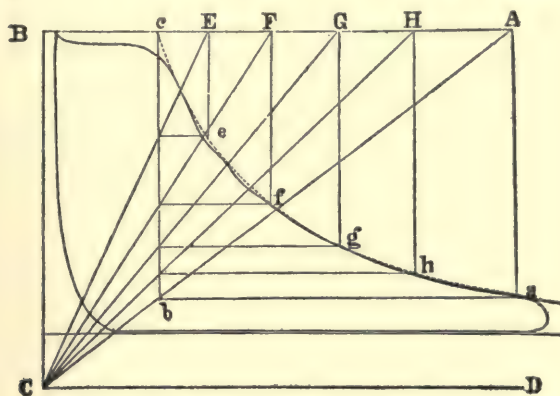


FIG. 1-35. — Method of Drawing Hyperbola.

the intersection of the same diagonal and the line AB , give points in the hyperbola.

Another method of drawing an hyperbola is shown in Fig. 1-36,

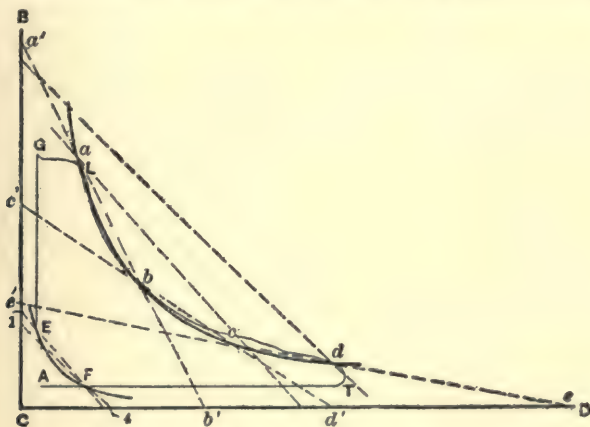


FIG. 1-36. — Method of Drawing an Hyperbola.

which represents an indicator diagram referred to lines of no volume and no pressure. This method is founded on the principle that the intercepts made by a straight line intersecting an

hyperbola and its asymptotes are equal. Beginning at any point as a , draw the straight line, $a' b'$, and lay off from the line $C D$, $b' b$ equal to $a' a$, then will b be a point in the hyperbola. Draw a similar line through b as $c' d'$ and find another point as c . Repeat the method until all the points required for drawing the curve are found.

The hyperbola is a useful line of reference in connection with indicator diagrams. As will be shown later, it represents the condition of isothermal expansion in the gas engine.

CHAPTER II

THERMODYNAMICS OF THE GAS ENGINE

1. Notation. — In order to comprehend the limitations of the actual engine it is necessary to understand how the working fluid behaves when subject to definite changes in an engine unaffected by friction or mechanical limitations, and we will for that reason give attention to the theoretical considerations relating to the “internal combustion engine” which forms part of the science of Thermodynamics.

In the consideration of the theoretical action of a perfect engine the following symbols will be used:

- A = the reciprocal of the mechanical equivalent of heat.
- a = the absolute temperature of the freezing-point = 273 C. or 492 F.
- C_p = the specific heat of constant pressure in heat units.
- C_v = the specific heat of constant volume in heat units.
- J = mechanical equivalent of heat = 778 in foot-pounds = 426 K. G. M.
- K_p = specific heat of constant pressure in mechanical units.
- K_v = the mechanical heat at constant volume in mechanical units.
- p = absolute pressure for condition denoted by postscript.
- p_0 = absolute pressure at freezing-point.
- Q = the total heat of a given mass.
- R = a constant for a given gas.
- T = absolute temperature.
- t = temperature Fahrenheit or Centigrade as marked.
- v = volume of a given mass of gas, its condition being denoted by postscript.
- v_0 = the volume of a given mass of gas at freezing-point.
- W = the mechanical work performed in mechanical units.
- W' = the mechanical work in heat units.
- α = the coefficient of expansion of a perfect gas = the reciprocal of a .
- γ = specific heat of constant pressure divided by the specific heat of constant volume.

2. Characteristics of Perfect Gases. — In an internal combustion engine the work is produced by the change of volume and pressure of a gaseous mixture composed of atmospheric air

and the various products of combustion. This gas mixture, within the working limits of temperature, is obedient to the laws of the perfect gases, and for that reason in any theory of the internal combustion engine we are principally concerned with such laws and with the changes of volume and pressure in a perfect gas, in relation to its change of temperature.

By combining Boyle's law with Gay-Lussac's law it is learned that the product of pressure and volume of a given mass of a perfect gas varies directly as its absolute temperature, that is:

$$pv = p_0v_0 (1 + a t) \quad (1)$$

in which p equals the absolute pressure and v the volume of a given mass of gas at a temperature, t , above the freezing-point, p_0 and v_0 represent the pressure and volume of the same mass at the freezing-point, and a represents the coefficient of expansion of the gas per degree of absolute temperature. As has already been shown a will equal when expressed in the Centigrade system the reciprocal of 273 and when expressed in the Fahrenheit system the reciprocal of 492. If we denote the number of degrees between the freezing-point and absolute zero by a , then from the preceding explanation a will equal the reciprocal of a . If we denote the absolute temperature by T we shall have

$$T = a + t. \quad (2)$$

substituting $\frac{1}{a}$ for a in equation (1) we have

$$pv = \frac{p_0v_0}{a} (a + t) = \frac{p_0v_0}{a} T \quad (3)$$

in the above equation $\frac{p_0v_0}{a}$ is a constant for each gas.

$$\text{Let } R = \frac{p_0v_0}{a} \quad (4)$$

then we have

$$pv = RT. \quad (5)$$

The above equation may be considered the characteristic equation of a perfect gas, since it shows the relations between the pressure, volume and absolute temperature.

R is a constant which depends on the nature of the gas and

can be computed if the specific pressure, p_0 , and specific volume, v_0 at standard pressures and temperatures are known.

It follows from the above that, $p_1 v_1 = RT_1$, from which by comparing with (5)

$$pv: p_1 v_1 :: T_1 : T \quad (5a)$$

The specific pressure p_0 is the weight of the atmosphere at the freezing-point under normal conditions; it is equivalent to that of a column of mercury 760 mm. high (29.921 inches). This reduced to pressure per unit of area is

$$p_0 = 10333 \text{ kilograms per square meter;}$$

or in English units,

$$\begin{aligned} p_0 &= 2146.32 \text{ pounds per square foot.} \\ &= 14.696 \text{ pounds per square inch.} \\ &= 29.921 \text{ inches of mercury.} \end{aligned}$$

The specific volume is determined from the density of the gas. The following table gives the specific volume in Metric and English measures for some of the more common gases:

VALUE OF v_0 AT LAT. 45°		
	Cubic meters per Kilogram	Cubic feet per Pound
Air	0.7735327	12.3909
Nitrogen (N)	0.7963291	12.7561
Oxygen (O)	0.6996231	11.2070
Hydrogen (H)	1.116705	178.881
Carbonic Acid (CO ₂)	0.5058741	8.10324

By substituting the values of $p_0 v_0$, and a in the equation $R = \frac{p_0 v_0}{a}$ the value of R can be found.

Thus for air, in French units

$$R = 10333 \times 0.77353 \div 273 = 29.20 ;$$

in English units

$$R = 2116.3 \times 12.391 \div 492 = 53.22.$$

The following table gives values of R for a few gases:

	VALUES OF R	
	English	Metric
Hydrogen (H)	770.3	422.68
Oxygen (O)	48.74	26.475
Carbon-dioxide (CO ₂)	35.41	19.43
Air	53.22	29.20

The following table showing the specific heat of the ordinary gases is inserted here for convenience.

TABLE OF SPECIFIC HEATS

NAME OF GAS	SYMBOL	SPECIFIC HEAT		$\frac{C_p}{C_v}$ γ
		Constant Pressure C_p	Constant Volume C_v	
Air		0.2375	0.1684	1.406
Oxygen	O	0.2175	0.1552	1.403
Nitrogen	N	0.2438	0.1727	1.416
Hydrogen	H	3.4090	2.4110	1.414
Nitric oxide	NO	0.2317	0.1652	1.402
Carbonic oxide	CO	0.2450	0.1736	1.413
Carbon dioxide	CO ₂	0.2000	0.1550	1.261
Steam	H ₂ O (saturated)	0.4805	0.3700	1.298
Methane	CH ₄	0.5930	0.4680	
Acetylene	C ₂ H ₂	0.3460	0.2700	
Disulphide carbon	CS ₂	0.1569	0.1310	1.198
Olefiant gas	C ₂ H ₄	0.4040	0.3330	1.125
Ammonia	NH ₃	0.5084	0.3910	1.300
Alcohol	CH ₂ O ₆	0.4534	0.4100	1.150

As before noted, the specific heat of gases increases with the temperature and possibly also with the pressure, which law will be referred to in discussing the application to special cases. The expanding products of combustion are composed of N, CO₂, H₂O, O, and possibly a trace of NO. For approximate computations the value of C_p/C_v for the burned gases may be taken at 1.37.

3. General Relations of Heat Transmission to Changes of Volume and Pressure.—When a quantity of heat dQ is supplied to a mass of gas it produces a complex result; the heat

warms the gas and raises its temperature, at the same time performing internal work by overcoming the molecular forces; it then develops external work, which is made apparent by the expansion of the gas against an external resistance. Denoting by dU the quantity of heat employed in warming the gas and in molecular work we will have

$$dQ = dU + A p dv,$$

in which $p dv$ is the external work expressed in mechanical units and $A p dv$ its equivalent in heat units.

In the operation of a gas engine the mass of gas constituting the working substance expands during each cycle from one volume to another, passing through a series of successive changes of volume and pressure, and finally returns to its initial state. It is evident that when the mass returns to its initial state that the quantity represented by $\int dU$ is equal to zero. For this condition we shall have

$$dQ = A p dv.$$

Calling U the internal heat of the gas, which has already been shown to be a function of the volume pressure and temperature, we have

$$U = f(v, p, t).$$

Since t can be determined from the values of p and v as indicated in equation 5, we can consider that in practice U is a function only of v and p and may write

$$U = f(v, p).$$

The increase of internal heat for a variation of volume dv and of pressure dp may be expressed as a total differential of the function, f , and we have

$$dU = \frac{\delta U}{\delta v} dv + \frac{\delta U}{\delta p} dp \quad (6)$$

By substituting the above value in the expression dQ the thermal state of the gas may be represented as follows:

$$dQ = \frac{\delta U}{\delta v} dv + \frac{\delta U}{\delta p} dp + A p dv$$

or

$$dQ = \frac{\delta U}{\delta p} dp + \left(\frac{\delta U}{\delta v} + A p \right) dv \quad (7)$$

It may be noted that the above equation cannot be integrated in its present form unless Q can be expressed as a function of the initial and final volumes and pressures of the gas. This demonstrates that the expenditure of heat required to make a gas pass from one state to another cannot be deduced from a knowledge of the extreme states, if one does not know the order and relation of the intermediate states.

The heat transferred at constant volume is equal to the specific heat C_v multiplied by the change of temperature. That is, $Q = C_v (T_1 - T)$, from which $dQ = C_v dt$. For a similar reason that transferred at constant pressure

$$\begin{aligned} Q &= C_p (T_1 - T) \\ dQ &= C_p dt. \end{aligned}$$

For the condition of *transfer of heat at constant volume*, dv of equation (7) will equal 0, and equation (7) will become

$$dQ = \frac{\delta U}{\delta p} dp$$

Since the heat interchange for this case takes place at constant volume, it has been shown that

$$dQ = C_v dt.$$

By placing these two values of dQ equal we have

$$C_v \frac{dt}{dp} = \frac{\delta U}{\delta p} \quad (8)$$

from which the value of $\frac{\delta U}{\delta p}$ for heat interchanges at constant volume can be found.

For the *transfer of heat at constant pressure* the temperature changes take place without change of pressure, in which case dp of equation (7) = 0, and we have by substitution

$$dQ = \left(\frac{\delta U}{\delta v} + Ap \right) dv$$

Since the heat interchange for this case takes place at constant pressure, $dQ = C_p dt$, and by substitution

$$C_p \frac{dt}{dv} = \frac{\delta U}{\delta v} + Ap \quad (9)$$

from which the value of $\frac{\delta U}{\delta v}$ for heat transformation at constant pressure can be obtained. Substituting the values of eq. (8) and (9) in eq. (7) we have

$$dQ = C_v \frac{\delta t}{\delta p} dp + C_p \frac{\delta t}{\delta v} dv \quad (10)$$

which gives the value of the heat interchange for successive change of pressure and volume. This can be reduced as follows:

From eq. (5), $pv = RT$, hence $vdv = Rdt$ when v is constant, and $pdv = Rdt$ when p is constant. From this it follows that

$$\frac{\delta t}{\delta p} = \frac{v}{R}, \text{ and } \frac{\delta t}{\delta v} = \frac{p}{R}$$

Substituting the above values in eq. (10) we have

$$dQ = \frac{1}{R} (C_v v dp + C_p p dv) \quad (11)$$

Now, from eq. (5), we obtain by complete differentiation

$$pdv + vdp = Rdt, \text{ from which } vdp = Rdt - pdv$$

Substituting this in eq. (11) we may write

$$\begin{aligned} dQ &= \frac{1}{R} [C_v (Rdt - pdv) + C_p p dv] \\ &= C_v dt - C_v \frac{p}{R} dv + C_p \frac{p}{R} dv \end{aligned}$$

But from eq. (5)

$$\frac{p}{R} = \frac{T}{v}$$

hence finally

$$dQ = C_v dt + (C_p - C_v) \frac{T}{v} dv \quad (11a)$$

4. Transformation to Different States.—The modes of transformation from one state to another depend upon the relation of p and v at different points, T being always determined, by the relation of p to v , from eq. (5).

For simplicity of treatment and for producing a standard for comparison it is assumed that the changes in the relations of volume, pressure, and temperature take place with one of the variables constant in the general equation (5a), p. 47.

Thus, if the volume remains constant during the change, we have $v = v_1$, and

$$\frac{p}{p_1} = \frac{T}{T_1} \quad (12)$$

which is the equation for constant-volume conditions. If the pressure remain constant, $p = p_1$ and

$$\frac{v}{v_1} = \frac{T}{T_1} \quad (13)$$

which is the equation for constant-pressure conditions. If the temperature remain constant, which latter is termed an *isothermal* condition, $T = T_1$.

$$\frac{p}{p_1} = \frac{v_1}{v} \quad (14)$$

which is the *equation of an isothermal line* for a perfect gas in a pressure-volume diagram.

Another standard of comparison is the transformation in pressure, volume or temperature which takes place without gain or loss of heat. This latter condition is called *adiabatic* and corresponds to that of constant *entropy*. It represents the conditions of the equation (11) when $dQ = 0$, in which case

$$\begin{aligned} C_v dp + C_p p dv &= 0 \\ v dp + \frac{C_p}{C_v} p dv &= 0 \end{aligned}$$

Substitute γ for $\frac{C_p}{C_v}$, then

$$v dp + \gamma p dv = 0.$$

which integrated between the limits $p_1 v_1$ and $p v$ gives

$$\log_e \frac{p}{p_1} = \gamma \log_e \left(\frac{v_1}{v} \right)$$

from which $p_1 v_1^\gamma = p v^\gamma = \text{constant}$ (15)

which is the *equation of an adiabatic line* for a perfect gas in a pressure-volume diagram.

The equation for adiabatic transformation in terms of v and T can be obtained by substituting for p and p_1 the values as given in (5a) and reducing, which will give

$$T v^{\gamma-1} = T_1 v_1^{\gamma-1} \quad (16)$$

In a similar manner the adiabatic equation in terms of p and T can be obtained, which is as follows:

$$T p^{\frac{1-\gamma}{\gamma}} = T_1 p_1^{\frac{1-\gamma}{\gamma}} \quad (17)$$

5. Work Performed in Isothermal Expansion. — The work, W , performed when the gas expands isothermally from an initial volume, v to a volume v_1 can be calculated as follows:

The general formula for mechanical work is

$$W = \int p dv$$

but as $pv = p_1v_1$ for isothermal expansion

$$p = \frac{p_1v_1}{v}$$

$$W = p_1v_1 \int_v^{v_1} \frac{dv}{v} = p_1v_1 \log_e \frac{v_1}{v} \quad (18)$$

Since $pv = p_1v_1 = RT$ this may be written

$$W = RT \log_e \frac{v_1}{v} = pv \log_e \frac{v_1}{v}$$

The heat applied during isothermal expansion can be obtained by making $dt = 0$ and T a constant in (11a) and integrating. We will have

$$Q = (C_p - C_v)T \int_v^{v_1} \frac{dv}{v} = (C_p - C_v)T \log_e \frac{v_1}{v} = ART \log_e \frac{v_1}{v} \quad (19)$$

This value being the same as that of the external work indicates that *the heat applied during isothermal expansion is equivalent to the external work performed.*

It will be noted from (18) that an infinite increase in isothermal expansion will lead to an infinite amount of work. Thus in the equation

$$W = pv \log_e \frac{v_1}{v}$$

if v_1 be made equal to infinity the value of W also becomes infinite.

6. Work Performed in Adiabatic Expansion. — The work performed when the gas expands adiabatically from an initial volume, v_1 to a volume, v_2 , can be found by substituting the value

of $p = \frac{p_1v_1^\gamma}{v^\gamma}$ from formula (15) as follows:

$$W = \int_v^{v_1} p dv = p_1v_1^\gamma \int_v^{v_1} \frac{dv}{v^\gamma} = -\frac{p_1v_1^\gamma}{\gamma-1} \left(\frac{1}{v_1^{\gamma-1}} - \frac{1}{v^{\gamma-1}} \right)$$

Therefore
$$W = \frac{p_1 v_1}{\gamma - 1} \left\{ 1 - \left(\frac{v_1}{v} \right)^{\gamma - 1} \right\} \quad (20)$$

For infinite adiabatic expansion the work, W , does not become infinite as in isothermal expansion, since for this case $\frac{v_1}{v} = 0$, and

$$W = \frac{p_1 v_1}{\gamma - 1}$$

7. Relations of Heat to Entropy. — The heat transformations can be expressed as a function of the absolute temperature and entropy, which expression possesses some advantages for tracing heat interchanges over the pressure-volume equations.

For this case the variables in the equation become temperature, T , and entropy, ϕ , instead of v and p as in the preceding cases; in the diagram representing such conditions, horizontal lines would represent equal temperatures or isothermal conditions, while vertical lines would represent equal entropy or adiabatic conditions. The ordinates in such a diagram would then represent temperature, T , and the abscissa entropy, ϕ .

Since the heat interchange dQ is equal to the product of the absolute temperature into the corresponding change of entropy, we have

$$\begin{aligned} dQ &= T d\phi \\ d\phi &= \frac{dQ}{T} \end{aligned}$$

$$\phi - \phi_1 = \int \frac{dQ}{T}$$

For gases, if heat is supplied at constant pressure $dQ = C_p dt$

$$\therefore \phi - \phi_1 = \int \frac{C_p dt}{T} = C_p \log_e \frac{T}{T_1} \quad (21)$$

8. Carnot or Reversible Engine, Second Law of Thermodynamics. — A reversible engine is one that may be run in one direction so as to transform heat into work, or in the opposite direction so as to transform work into heat.

No actual heat engine is built in this manner, since such a hypothesis requires that all the gases exhausted shall pass through all the states in a reversed direction during compression and return to the initial state, which, because of the chemical

changes during combustion, is impossible in the internal combustion engine. The internal combustion engine can be considered as approximating the theoretical reversible engine, which thus becomes useful as a standard of comparison.

For the cycle of a reversible engine

$$\int \frac{dQ}{T} = 0$$

This is the highest attainable result with any heat engine, since it indicates that the heat transferred into work from motion in one direction would be returned to its source by an equal amount of work applied to drive the engine in an opposite direction. The above statement is *Carnot's principle*, which is often called the *Second Law of Thermodynamics*. It follows from this:*

(1) All reversible engines working between the same source of heat and refrigerator have equal efficiencies.

(2) The efficiency of a reversible engine is independent of the working substance.

(3) A self-acting machine cannot transfer heat from one body to another at a higher temperature.

It further follows from this that for any irreversible engine cycle the work for a given expenditure of heat is less than for a reversible engine; that is,

$$\int \frac{dQ}{T} = N$$

in which N represents the mechanical results of the work performed.

9. Graphical Relations. — The relations of the heat interchanges to the transformations of pressure, volume, and temperature will be more clearly understood by reference to a diagram. The pressure-volume diagram, which has for its ordinates lines corresponding to pressure and for its abscissæ distance corresponding to volumes, shows the conditions of uniform pressure by a horizontal line and of constant volume by a vertical line. On this diagram an isothermal line is represented by equation (14), $pv = p_1v_1$ which is the equation of an equilateral hyperbola of which the axes are the lines of zero volume and the line of zero pressure. Two methods of drawing the hyperbola have been

*Peabody's Thermodynamics, page 30

given in Art. 19, Chapter I. In the case of a steam engine it will be remembered that an isothermal condition is represented by an equal pressure line.

The equation of an adiabatic line on a pressure-volume diagram, as given in (15) is $\log \frac{p}{p_1} = \gamma \log \frac{v_1}{v}$. The values of the coordinates for drawing this curve can be found by assuming values of $\frac{v_1}{v}$ and finding the corresponding values, by use of a table of Napierian logarithms, of $\frac{p}{p_1}$. A table giving the values of γ for different gases has been given. It is usually assumed as 1.37 for gas engines and is subject to some correction for changes due to rise of temperature. The general relations of isothermal and adiabatic lines to pressure, volume and temperature is shown on the diagram Fig. 2-1.

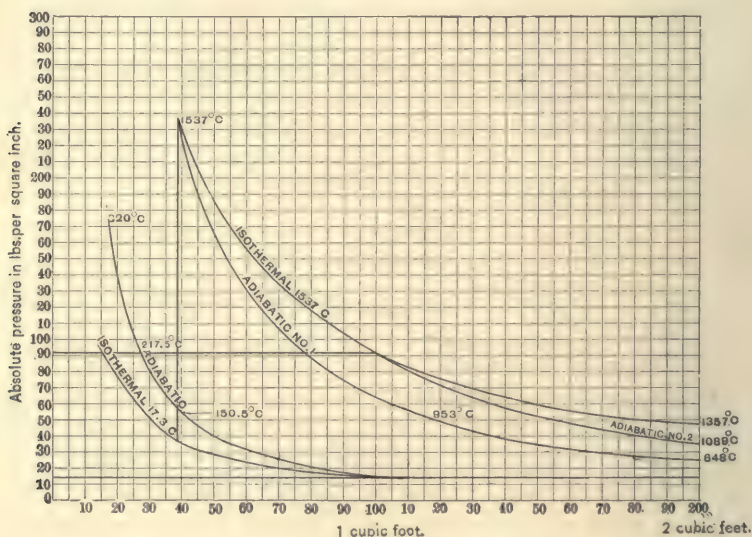


FIG. 2-1.—Relations of Isothermal and Adiabatic Curves.

Figure 2-1 shows isothermal and adiabatic expansion and compression lines drawn from the same points. This diagram shows that for a given number of expansions the adiabatic line falls

below the isothermal line, and also that the area between the adiabatic line and the base line is less than that between the isothermal and the base line. As this area represents the external work performed, it indicates that for a given number of expansions the work is greater in isothermal than in adiabatic expansion, which also follows from the demonstrations which have been given.

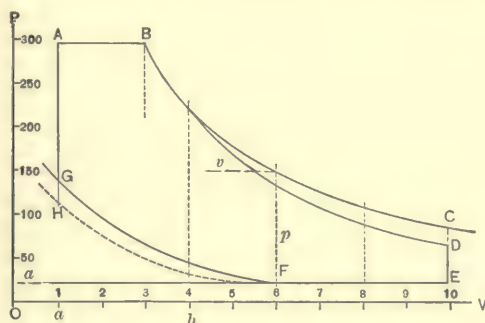


FIG. 2-2.— Isothermal and Adiabatic Changes Compared.

The various fundamental changes which may take place in a perfect heat engine are represented by the diagram, Fig. 2-2, in which OV is the base line from which pressures are measured and OP the zero volume line from which volumes are measured. In the diagram GA is a vertical line and represents the condition of receiving heat at constant volume; AB , a horizontal line, represents the condition of receiving heat at constant pressure; BC , a plain hyperbola, represents isothermal expansion, and BD , a logarithmic curve, represents adiabatic expansion; DE , a vertical line, represents the discharge of heat at constant volume; EF , a horizontal line, represents the discharge of heat at constant pressure; FG , a logarithmic curve, represents adiabatic compression, and FH , a plain hyperbola, represents isothermal compression. The area of the diagram, $ABDEFG$, represents the external work done with adiabatic expansion and compression, and $ABCDEFG$ represents the work done with isothermal expansion and compression.

During the period of receiving heat at constant volume, represented by AG , the absolute temperature may be computed

at A , from (12), provided it is known at G , since it is proportional to the pressures at those points.

During the period of receiving heat at constant pressure, represented by AB , the absolute temperature increases in proportion to the volume, as shown in equation (13), and if known at one point may be computed at any other. As the volume is proportional to the distance from the line OP , the temperature on the line AB will be proportional to that distance.

If the expansion is isothermal the temperature would remain constant from B to C . If the expansion is adiabatic, as from B to D , the temperature could be calculated from either equation (16) or (17) for various points of the curve.

The actual pressure-volume diagram as taken with the indicator shows lines which only approximate those for constant pressure, constant temperature, or constant volume as shown in Fig. 2-2. The corners of actual diagrams are likely to be rounded to a considerable extent and many of the transformations indicated may not appear.

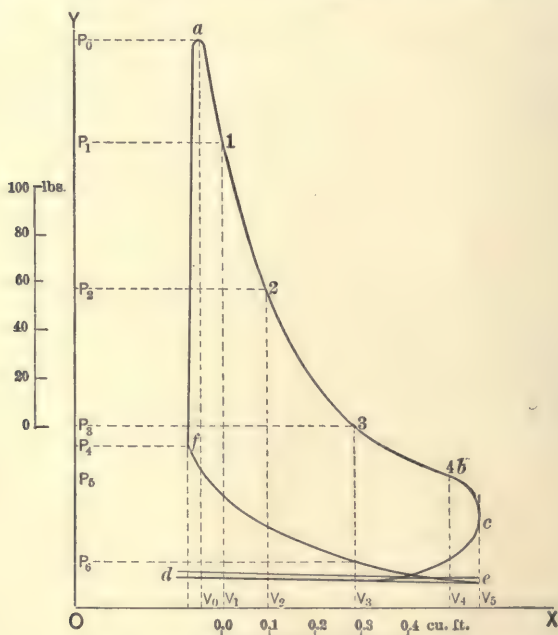


FIG. 2-3. — Four-cycle Engine Diagram.

Figure 2-3 represents a diagram of a 4-cycle engine in which the ordinates have been enlarged relatively with respect to the abscissæ and on which there have been drawn a line of no volume, often called the clearance line, and a line of zero pressure. The scale of ordinates is attached to the diagram. If we assume that the line $a f$ is a vertical line and that the temperature at the lower end of that line is 567 degrees absolute, then we find by computation, as explained above, that for the point a it is 1995 degrees absolute. As absolute temperature F is 460 degrees higher than that shown on a thermometer, the temperature F at these two points would correspond to 107 degrees and 1535 degrees.

The heat transformations may also be represented by the entropy temperature diagram, in which case the ordinates become temperature and entropy instead of pressure and volume.

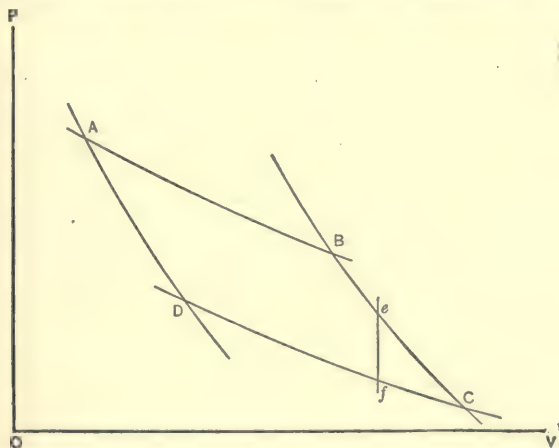


FIG. 2-4. — Pressure Volume Diagram of Isothermal and Adiabatic Changes.

A simple illustration is shown in Figs. 2-4 and 2-5. In Fig. 2-4 is shown a pressure-volume diagram of a working material which expands isothermally from A to B , then adiabatically from B to C . It is then compressed isothermally from C to D and adiabatically from D to A , when it reaches the initial condition. The mechanical work performed during these operations is proportional to the area of the diagram $A B C D$. This diagram may be

transformed into a temperature entropy diagram very easily, since for that case the isothermal lines AB and BC , would be horizontal and parallel to the line of zero temperature, and the adiabatic lines AD and DC would be vertical and parallel to the line from which entropy is reckoned. Fig. 2-5 shows the temperature-entropy diagram constructed as described.

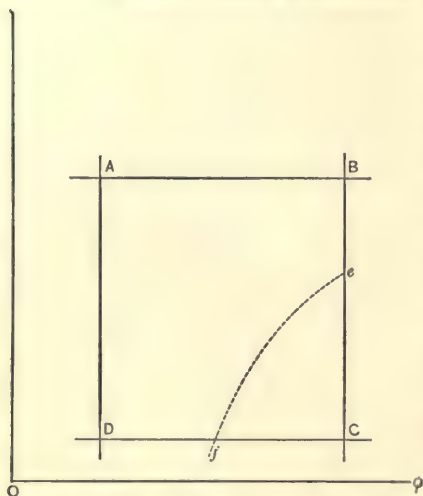


FIG. 2-5. — Entropy-Temperature Diagram. The area of this diagram shows the heat transferred into work. For the case considered this is a rectangle and its area represents the maximum amount of heat available for work within the temperature limits AB and DC .

The case considered is that of the perfect reversible engine which operates in a Carnot cycle, as already described. It appears from the diagram, Fig. 2-5, as well as from the demonstration, that an engine working on this principle can transform the maximum amount of heat into work.

An engine working on any other principle, as for instance that shown in the Fig. 2-2 $ABDEFGA$, will transform a less amount of heat into work. For this case there is both adiabatic expansion BD and adiabatic compression FG . The entropy-temperature diagram for this case is represented, but not to scale, in Fig. 2-6 by the diagram $ABDEFG$, which by inspection is smaller and shows less heat available for work than the diagram $mBnF$, which is drawn between the same temperature limits.

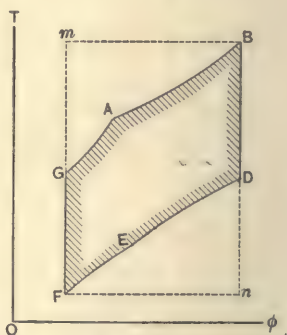


FIG. 2-6.

For the figure $ABCEFH$, shown in Fig. 2-2, as a pressure-

volume diagram we have isothermal expansion $B C$ and isothermal compression $F H$. The entropy-temperature diagram for this case is represented by the diagram, not drawn to scale, in Fig. 2-7, $A B C E F H$, which by inspection is smaller and shows less heat available for work than the diagram $m C n H$, which is drawn between the same temperature limit. The transformation of the pressure-volume diagram as taken on the indicator into entropy-temperature diagram is given at length later in the book.

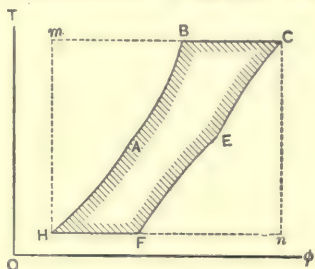


FIG. 2-7.

10. Comparison of Theoretical and Actual Heat Engines.

— When a gas after a series of transformations of pressure-volume and temperature passes through a series of intermediate states and of physical and chemical changes and returns to the same condition, in all respects, which it possessed at the beginning of the transformations, it is said to operate in a *closed cycle*.

It is evident that if a change of composition occurs during the course of the cycle the body may return to its initial condition so far as pressure and volume are concerned without returning to its initial condition in other respects; for example, a mixture composed of hydrogen, carbon monoxide, methane, carbon dioxide and nitrogen, with which the cylinder is charged in the initial condition before combustion, may be changed during combustion to the vapor of water, carbon dioxide, and nitrogen, which change would not be shown on the indicator diagram.

In the actual operation of the internal combustion engine the gas or vapor is subjected to periodical changes, as outlined above which do not constitute rigorously closed cycles. The operation is, however, approximately a closed cycle since the state of the working fluid after each series of changes returns to its initial state, from which all the operations as to chemical and physical changes are reproduced as in the preceding phase.

The machine in which the changes take place will be a perfect heat engine if all of the heat disappearing has been transformed into work; it has been demonstrated practically, however, that it

is not possible to utilize all of the heat, consequently it is necessary to expend in heat a larger quantity than its equivalent in units of work. It has already been shown in effect that it is not only necessary to have a place of combustion which produces a high temperature at the beginning of the period of movement, but it is also necessary to have a point of lower temperature which will act as a refrigerator and permit the flow of heat from a higher to a lower temperature level. The quantity of heat Q supplied by the combustion less the amount q taken up by the refrigerator leaves a difference $Q - q$ which may be utilized. From this statement it would appear that the amount of work possible would be increased either by increasing Q or by diminishing q . The temperature of the discharge heat q must evidently be considerably above that of absolute zero because of the difficulty of obtaining a refrigerant of low temperature and of disposing of the heat which would be discharged under such a condition. Generally in practice the temperature of the discharge heat is considerably above that of the surrounding air, which is much in excess of absolute zero.

The ratio of $Q - q$ to Q measures the perfection of the heat engine. This we will call the cyclic efficiency

$$\frac{Q - q}{Q} = \text{cyclic efficiency}$$

It is of great practical interest to know how to determine the value of this coefficient for any case, but we should at first establish the maximum value that can be obtained.

In this connection the cycle of Carnot which is formed of two *isothermal* lines and two *adiabatic* lines, Figs. 2-4 and 2-5, should receive consideration, since it is the one which gives the maximum work returned for the heat expended.

The Carnot cycle is represented in the pressure-volume diagram Fig. 2-4, in which a mass of fluid in its initial state $p_0 v_0$, with temperature T_0 as shown at A , expands in volume isothermally to B , at which point it has pressure and volume $p_1 v_1$. From B it expands to C without gain or loss of heat, following the adiabatic BC . From C to D there is a discharge of heat into a colder body or refrigerator at constant temperature, during which time the volume is reduced from v_2 to v_3 . From D to A compression takes

place without gain or loss of heat, which raises the temperature to that of the initial state at A .

In order to carry out the cycle of operation described, sufficient heat must be supplied during the isothermal expansion from A to B to keep the temperature constant; this amount by equation (19) will be $Q = A R T \log \frac{v_1}{v_o}$. In the adiabatic expansion from B to C , during which there is neither gain nor loss of heat, the relations of the volumes to the temperatures are expressed by equation (16)

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1} \right)^{\gamma - 1}$$

During the third period from C to D heat is discharged at constant temperature and can be expressed as before $q = A R T \log \frac{v_2}{v_3}$. During the fourth period the quantity of heat remains constant and we have

$$\frac{T_o}{T_3} = \left(\frac{v_3}{v_o} \right)^{\gamma - 1}$$

The cycle as above described is a closed one, and as the heat received and discharged is of constant temperature it follows that

$$\frac{Q}{q} = \frac{T_1}{T_2}$$

The work produced is equal to the area $A B C D$. The efficiency of the cycle is

$$\frac{Q - q}{Q} = \frac{T_1 - T_2}{T_1}$$

This is equal to the ratio of the fall of temperature to the absolute temperature of the initial condition.

The Carnot cycle may also be represented, as already shown, by the temperature-entropy diagram, in which case the diagram will be a rectangle, Fig. 2-5.

It is doubtless true that no better method exists for utilizing the heat furnished by combustion than that of supplying it at constant temperature, permitting the body to expand without gain or loss of heat, discharging it to the refrigerator at a constant temperature, which should be as low as possible, and compressing without gain or loss of heat to the original temperature.

Theoretically it is possible to equal but not to surpass the return from the Carnot cycle. The maximum effect that can be

obtained from a heat engine working between the temperatures T and T' , and in which Q is the heat expended, is expressed by the equation

$$Q\left(\frac{T-T'}{T}\right)$$

In order to judge the theoretical value of a cycle on which any heat engine operates it is desirable to calculate at first the coefficient of economy of the proposed cycle, then compare this coefficient with the Carnot coefficient between the same temperature limits.

The knowledge which is given by comparing the cycle of the engine with the Carnot cycle is not sufficient to determine the practical value of the engine, since the effect of friction, shocks due to inertia, and the passive resistance of the various mechanical parts consume a portion of the work supplied by the transformation of heat and do not appear in the useful work delivered by the machine. It is quite possible that machines which have a high degree of perfection for the transformation of heat will still give small return as practical, useful machines.

The hot-air engines for example, which have a perfect cycle of operation, have proved in practice of little value because of the small amount of heat that would pass through a metallic wall in a given time, and as a consequence the return in useful work is small in proportion to the expense of construction. There are few engines of any kind which operate in a cycle approximating that of Carnot, among these should be mentioned the Sterling air engine, which theoretically operates on the Carnot cycle.

The cycle of operation of the steam engine is incomplete in many respects; it however resembles that of Carnot in that heat is received into the engine cylinder, until the valve closes connection with the boiler, at practically constant temperature. After cut-off the steam is expanded approximately without gain or loss of heat. It is then discharged through the condenser at constant temperature. The adiabatic compression is sometimes considered as being performed by the feed pump which supplies water to the boiler. The steam engine cycle is thus seen, when the boiler furnace and boiler feed pump are included as a part, to approximate that of the Carnot cycle.

The Diesel motor is the only gas engine which approximates in the theory of its operation to the Carnot cycle.

CHAPTER III

THEORETICAL COMPARISON OF VARIOUS TYPES OF INTERNAL COMBUSTION ENGINES

1. Throughout the following discussion of the theoretical cycles it will be assumed that the specific heats at constant volume C_v , and at constant pressure C_p , do not vary either with pressure or temperature. It has been shown that they vary, but the question is unsettled. If the variation is such as determined by the experiments of Mallard and Le Chatelier, which are extensively quoted, our present-day gas engine does not admit thermally of any further improvement. In view of this unsettled condition it is best to assume the specific heats constant. It is further assumed, in this theoretical discussion, that the value of $\gamma = \frac{C_p}{C_v}$ is the same for the burned gases as for the fresh fuel mixture.

It is further understood that, wherever heat supplied to a cycle is mentioned, it refers to the lower heating value of the fuel concerned, either per pound or per standard cubic foot, as stated.

2. The cycle receiving heat at constant volume, Beau de Rochas or Otto cycle.

The principles upon which the present-day constant volume combustion engine is based, and which helped it to its commercial success, were first clearly enunciated by Beau de Rochas in a written pamphlet in 1862. It remained for Otto, however, to construct the first practically successful machine operating with this cycle, hence the cycle receiving heat at constant volume is more often known as the Otto cycle.

In what follows let

Q = quantity of heat received by the theoretical cycle,

q = quantity of heat rejected by the theoretical cycle,

then E_c = the cycle efficiency = $\frac{Q-q}{Q}$.

It shows the highest efficiency which an engine could possibly show if it could follow exactly the lines of the theoretical cycle. Practically this can never be realized, but the conditions which determine why the actual thermal efficiency of an engine is always less than the cyclic efficiency will be treated in detail later on.

In Fig. 3-1, at the end of the charging stroke, whether that be in the two- or in the four-cycle, the charge is under a pressure, temperature, and volume determined by the point 1. Adiabatic compression then takes place to 2. A quantity of heat, Q , is next received at constant volume to 3. From 3 to 4 adiabatic expansion takes place, and finally the quantity of heat, q , is rejected along line 4-1 at constant volume.

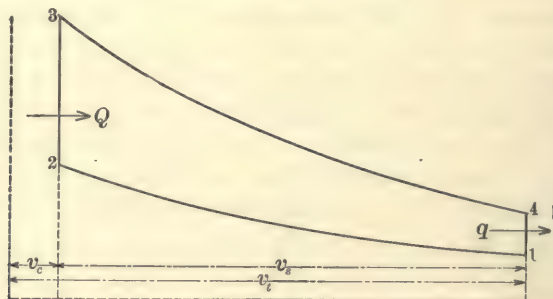


FIG 3-1.

Let the total charge weight be G lbs. This consists of G_a lbs. air, G_g lbs. gaseous fuel, and G_r lbs. burned gases from the previous cycle.

Then

$$Q = GC_v (T_3 - T_2) \quad (1)$$

and

$$q = GC_v (T_4 - T_1) \quad (2)$$

Hence the heat utilized by the cycle is

$$Q - q = GC_v (T_3 - T_2 - T_4 + T_1) \quad (3)$$

and the cyclic efficiency

$$E_c = \frac{Q - q}{Q} = \frac{GC_v (T_3 - T_2 - T_4 + T_1)}{GC_v (T_3 - T_2)} \quad (4)$$

$$= \frac{T_3 \left(1 - \frac{T_4}{T_3}\right) - T_2 \left(1 - \frac{T_1}{T_2}\right)}{T_3 - T_2} \quad (5)$$

Again, in Fig. 3-1, let v_c = clearance volume, and v_s = stroke volume, so that $v_t = v_c + v_s$ = total volume. We may write, since compression and expansion are assumed adiabatic,

$$p_2 v_c^\gamma = p_1 v_t^\gamma \quad (6)$$

and

$$p_3 v_c^\gamma = p_4 v_t^\gamma. \quad (7)$$

Dividing (6) by (7), $\frac{p_2}{p_3} = \frac{p_1}{p_4}$. (8)

Now $\frac{T_2}{T_3} = \frac{p_2}{p_3}$ and $\frac{T_1}{T_4} = \frac{p_1}{p_4}$

hence finally $\frac{T_2}{T_3} = \frac{T_1}{T_4}$ or $\frac{T_4}{T_3} = \frac{T_1}{T_2}$ (9)

With the aid of (9), equation (5) may be written

$$\begin{aligned} E_c &= \frac{T_3 \left(1 - \frac{T_1}{T_2}\right) - T_2 \left(1 - \frac{T_1}{T_2}\right)}{T_3 - T_2} \\ &= 1 - \frac{T_1}{T_2} \end{aligned} \quad (10)$$

There are two other ways of stating the cyclic efficiency which will be developed next.

Again we can write

$$p_1 v_t^\gamma = p_2 v_c^\gamma \quad (11)$$

also

$$\frac{p_1 v_t}{T_1} = \frac{p_2 v_c}{T_2} \quad (12)$$

Dividing (11) by (12)

$$T_1 v_t^{\gamma-1} = T_2 v_c^{\gamma-1}$$

or

$$\frac{T_1}{T_2} = \left(\frac{v_c}{v_t}\right)^{\gamma-1} = \frac{1}{r^{\gamma-1}} \quad (13)$$

in which r = the compression ratio $\frac{v_t}{v_c}$.

Hence

$$E_c = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{r^{\gamma-1}} \quad (14)$$

Finally, raise equation (12) to the γ power

$$\frac{p_1^\gamma v_1^\gamma}{T_1^\gamma} = \frac{p_2^\gamma v_2^\gamma}{T_2^\gamma} \quad (15)$$

and divide (11) by (15). We shall have

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2} \right)^{\frac{\gamma-1}{\gamma}} \quad (16)$$

from which

$$E_c = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{p_1}{p_2} \right)^{\frac{\gamma-1}{\gamma}} \quad (17)$$

From an examination of equations (14) and (17) it will be seen that E_c depends upon r or p_2 , and $\gamma = \frac{C_p}{C_v}$.

To make clear the influence of the value of γ , assume that in two given cases the value of $r = 5$, but that in the first case a rich gas mixture with $\gamma = 1.35$, in the second case a lean gas mixture with $\gamma = 1.39$ be employed. Then for the two cases we shall have

$$E_c = 1 - \frac{1}{(5)^{.35}} = .431$$

and

$$E_c = 1 - \frac{1}{(5)^{.39}} = .466$$

The advantage in favor of the lean gas mixture is therefore

$$\frac{.466 - .431}{.431} = 8.1 \%$$

Besides this the use of lean gas mixtures in practice usually shows a smaller jacket water loss, owing to the lower mean temperature of the entire cycle.

The value of E_c also depends upon the ratio $\frac{p_1}{p_2}$, according to eq. (17). The smaller this ratio the greater the efficiency. But the value of p_1 , the suction pressure, is almost entirely out of our control, so that the problem narrows down to making p_2 , the compression pressure, as high as possible. This brings out clearly the value of high compression. The practical limits to this statement will be pointed out later on.

To show the combined influence upon E_c of r or p_2 and γ the following table, from Güldner, is given:

CYCLIC EFFICIENCIES, E_c , FOR THE OTTO CYCLE

$r =$	2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0	10.0
$\gamma = 1.20$	.129	.167	.197	.221	.242	.260	.275	.301	.322	.340	.356	.369
$= 1.25$	.159	.205	.240	.269	.290	.313	.331	.361	.385	.405	.423	.438
1.30	.188	.241	.281	.313	.343	.363	.383	.416	.442	.464	.483	.499
1.35	.216	.274	.319	.355	.384	.409	.431	.466	.494	.517	.537	.553
1.40	.248	.313	.363	.402	.434	.460	.483	.520	.550	.574	.594	.611

3. The cycle receiving heat at a constant pressure, the Brayton cycle, and the approximate Diesel cycle of to-day.

The Brayton Cycle.

In the older machines of the Brayton type, suction and compression of the fuel mixture or of air took place in one cylinder,

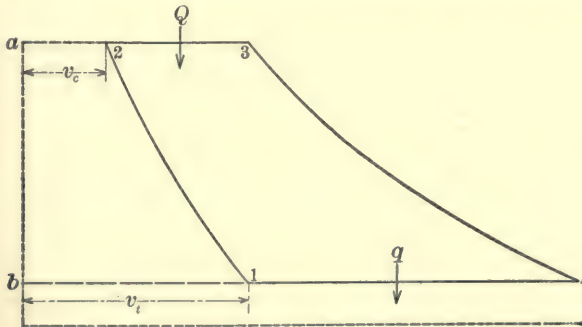


FIG. 3-2.

while the combustion, expansion, and exhaust took place in another. In Fig. 3-2, area $b\ 1\ 2\ a\ b$ represents the pump diagram, area $a\ 3\ 4\ b\ a$ the diagram from the power cylinder. For the purpose of theoretical discussion of this cycle, the two diagrams may be combined as shown, giving in area $1\ 2\ 3\ 4$ the useful work developed.

The compression line, 1-2, and the expansion line, 3-4, are again assumed adiabatic. Expansion is carried to exhaust pressure. The quantity of heat Q is received along 2-3, the amount q is rejected along 4-1, at constant pressure in both cases.

Let the charge weight be G pounds made up as in the previous case. Then

$$Q = GC_p (T_3 - T_2) \text{ B. T. U.}$$

and

$$q = GC_p (T_4 - T_1) \text{ B. T. U.}$$

Hence

$$\begin{aligned} E_c &= \frac{Q - q}{Q} = \frac{GC_p(T_3 - T_2 - T_4 + T_1)}{GC_p(T_3 - T_2)} \\ &= 1 - \frac{T_4 - T_1}{T_3 - T_2} \end{aligned} \quad (18)$$

But from the adiabatic law we may write

$$p_1 v_1^\gamma = p_2 v_2^\gamma. \quad (19)$$

and

$$p_4 v_4^\gamma = p_3 v_3^\gamma. \quad (20)$$

Divide (19) by (20).

$$\left(\frac{v_1}{v_4}\right)^\gamma = \left(\frac{v_2}{v_3}\right)^\gamma \text{ or } \frac{v_1}{v_4} = \frac{v_2}{v_3} \quad (21)$$

Also,

$$\frac{p_1 v_1}{T_1} = \frac{p_4 v_4}{T_4}, \text{ or since } p_1 = p_4, \frac{v_1}{T_1} = \frac{v_4}{T_4} \quad (22)$$

and similarly

$$\frac{v_2}{T_2} = \frac{v_3}{T_3} \quad (23)$$

From (22)

$$\frac{v_1}{v_4} = \frac{T_1}{T_4} \quad (24)$$

From (23)

$$\frac{v_2}{v_3} = \frac{T_2}{T_3} \quad (25)$$

Equations (25) and (24) in combination with (21) finally give

$$\frac{T_1}{T_4} = \frac{T_2}{T_3} \text{ or } \frac{T_1}{T_2} = \frac{T_4}{T_3} \quad (26)$$

Substituting (26) in (18), we have

$$E_c = 1 - \frac{T_1}{T_2} = 1 - \frac{T_4}{T_3} \quad (27)$$

But it has already been shown, equation (13), that for adiabatic compression

$$\frac{T_1}{T_2} = \left(\frac{v_c}{v_t} \right)^{\gamma-1} = \frac{1}{r^{\gamma-1}}$$

so that finally the cyclic efficiency for combustion at constant pressure is

$$E_c = 1 - \frac{1}{r^{\gamma-1}} \quad (28)$$

which is the same as for combustion at constant volume.

The Diesel Cycle of to-day.

The Diesel cycle of to-day approximates the constant-pressure form outlined in Fig. 3-3. Compression line 1-2 and expansion

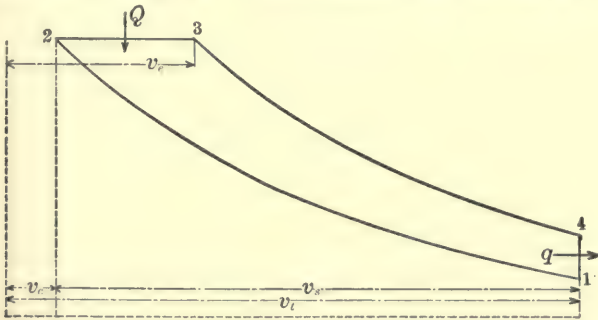


FIG. 3-3.

line 3-4 are assumed adiabatic. Heat is received at constant pressure along line 2-3, and rejected at constant volume along line 4-1. In the Otto cycle, with the machine at full load, the ratio of compression is equal to the ratio of expansion. In the Diesel, the ratio of compression $\frac{v_t}{v_c}$, (see Fig. 3-3), is always greater than the ration of expansion, $\frac{v_t}{v_e}$.

Let the charge weight again be G pounds. Diesel machines, as constructed, are oil engines, so that the increase of charge weight along line 2-3 is small and may be neglected without serious error. That is, we may assume G constant for the cycle. If the investigation, however, is carried through for a gas, especially a lean gas, this assumption is not permissible.

To develop the efficiency formula, we may write

$$Q = GC_p (T_3 - T_2). \quad (29)$$

$$q = GC_v (T_4 - T_1). \quad (30)$$

Now from Fig. 3-3

$$\frac{v_c}{T_2} = \frac{v_e}{T_3} \text{ or } T_3 = T_2 \frac{v_e}{v_c} = T_2 \delta \quad (31)$$

where, δ = ratio of cut-off volume to clearance volume.

Also

$$\frac{p_4}{T_4} = \frac{p_1}{T_1} \text{ from which } T_4 = T_1 \frac{p_4}{p_1} \quad (32)$$

But from the adiabatic law

$$p_4 v_4^\gamma = p_3 v_3^\gamma \text{ from which } p_4 = \frac{p_3 v_3^\gamma}{v_4^\gamma}$$

and

$$p_1 v_1^\gamma = p_2 v_2^\gamma \text{ from which } p_1 = \frac{p_2 v_2^\gamma}{v_1^\gamma}$$

Hence (32) may be written

$$T_4 = T_1 \frac{\frac{p_3 v_3^\gamma}{v_4^\gamma}}{\frac{p_2 v_2^\gamma}{v_1^\gamma}} = T_1 \left(\frac{v_3}{v_2} \right)^\gamma = T_1 \left(\frac{v_e}{v_c} \right)^\gamma = T_1 \delta^\gamma \quad (33)$$

Substituting (31) and (33) in (29) and (30) respectively,

$$Q = GC_p T_2 (\delta - 1).$$

$$q = GC_v T_1 (\delta^\gamma - 1).$$

The cyclic efficiency for the Diesel cycle consequently is-

$$\begin{aligned} E_c &= \frac{Q - q}{Q} = 1 - \frac{q}{Q} = 1 - \frac{GC_v T_1 (\delta^\gamma - 1)}{GC_p T_2 (\delta - 1)} \\ &= 1 - \frac{T_1}{T_2} \frac{(\delta^\gamma - 1)}{(\delta - 1)\gamma}. \end{aligned}$$

But it has shown, equation (13), that $\frac{T_1}{T_2} = \frac{1}{r^{\gamma-1}}$

hence finally

$$E_c = 1 - \frac{1}{r^{\gamma-1}} \frac{(\delta^\gamma - 1)}{\gamma(\delta - 1)} \quad (34)$$

Equation (34) shows that the expression for the theoretical efficiency of the Diesel cycle is the same as that for the Otto and the Brayton, with the exception of the factor $\frac{\delta^\gamma - 1}{\gamma(\delta - 1)}$. The efficiency thus depends not only upon r and γ , but also upon δ , that is, finally, on the volume at cut-off.

In actual practice the cut-off volume v_c is about 10 per cent of the stroke at full load. With a compression ratio of $r = 13$, this makes δ about $= 2.5$. To show the influence of the factor δ upon E_c , assume $\gamma = 1.35$. The efficiencies for full load, $\delta = 2.5$, an overload, $\delta = 3.0$, and some partial load, $\delta = 1.5$, will be as follows:

Cyclic Efficiency E_c for the Diesel cycle.

$$\text{For } \delta = \frac{v_e}{v_c} = \begin{matrix} 1.5 & 2.5 & 3.0 \end{matrix}$$

and for $r = 13$, and $\gamma = 1.35$, $E_c = \begin{matrix} .560 & .509 & .487. \end{matrix}$

It appears from this that, other conditions remaining the same, the smaller the value of δ , the greater E_c . This result is actually borne out in practice, within limits, where a large number of tests of Diesel engines have often shown a greater thermal efficiency at three-quarters than at full load. That this condition does not hold for still lower loads is due to other circumstances.

4. Comparison of Various Cycles.

The question of the best gas-engine cycle has often been discussed. In general, there is no best gas-engine cycle, but that cycle should be chosen which will give the best return for the practical conditions existing.

To give some insight into the problem of choosing the cycle best adapted to given conditions, we will first obtain some theoretical basis of comparison, and show afterwards how this is affected in practice.

Many methods of comparison have been employed by various writers, but the following, due originally to E. Meyer* seems to the writer to be the clearest and most comprehensive.

* *Zeitschrift des Vereins deutscher Ingenieure*, 1897, p. 1108.

Using the Carnot cycle as a basis of comparison, it is clear that the amount of heat transferred into mechanical work depends only upon the total amount of heat and the temperature difference in the cycle. But to become available as a basis for the comparison of gas-engine cycles, only that Carnot cycle can be used for which the heat element δq_1 , supplied at a constant temperature T_1 , and the heat element δq_2 rejected at a constant temperature T_2 , are of infinitesimal amount, so that the two adiabatics forming the rest of the cycle are infinitely close together. Then every closed cycle may, by a number of adiabatics, be divided into an infinite number of elementary cycles, Fig. 3-4,

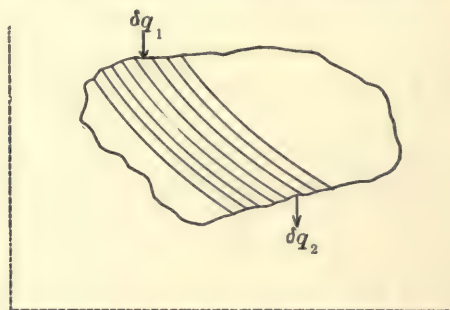


FIG. 3-4.

for each of which we may with very small error assume that the heat element δq_1 is supplied at a constant temperature T_1 , and the heat element δq_2 is rejected at the constant temperature T_2 . The efficiency of one of these elementary Carnot cycles may be expressed for say the n th cycle by

$$E_{c(n)} = \frac{T_{1(n)} - T_{2(n)}}{T_{1(n)}}$$

From the above equation we at once derive the important requirement that for best efficiency each heat element δq_1 should be supplied to the working fluid at the highest temperature possible, and each element δq_2 , should be rejected at the lowest possible temperature, that is, the temperature limits should be as wide as possible.

It should be observed, however, that the above requirement

does not mean that the sum total of all the heat elements δq_1 must be supplied isothermally, or that the sum total of the elements δq_2 rejected must be taken up isothermally. For in general in gas engines the various successive heat elements are not available at constant temperature, and in fact in following out the requirement above outlined for best efficiency, one would be led to constantly change the temperature of supply and of rejection to widen the temperature limits as far as possible. It is clear, therefore, that only in the case where all the heat elements supplied are available at constant temperature, and all those rejected can be taken up at constant temperature, as is the case in a steam engine, will the Carnot cycle represent the ideal.

Now consider one of the elementary Carnot cycles, into which the gas-engine cycle has been divided, by itself, Fig. 3-5.

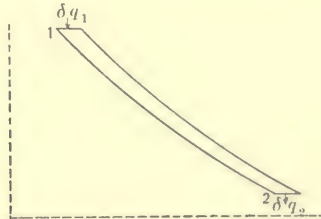


FIG. 3-5.

For the point 1 we may write

$$T_1 = \frac{p_1 v_1}{R}$$

and since during the supply of the infinitesimal heat element δq , the values of p_1 and v_1 increase only by infinitesimal amounts, we may assume without great error that the volume and pressure at the end of the heat supply are also represented by v_1 and p_1 respectively. Similarly we may write for the point 2

$$T_2 = \frac{p_2 v_2}{R}$$

The efficiency of the elementary cycle under discussion is then

$$E_c = \frac{T_1 - T_2}{T_1} = \frac{p_1 v_1 - p_2 v_2}{p_1 v_1} = 1 - \frac{p_2 v_2}{p_1 v_1}$$

But since the other enclosing lines are adiabatics, we have

$$p_1 v_1^\gamma = p_2 v_2^\gamma, \text{ from which}$$

$$\frac{v_2}{v_1} = \left(\frac{p_1}{p_2} \right)^{\frac{1}{\gamma}}$$

Finally, therefore,

$$E_c = 1 - \frac{p_2 \left(\frac{p_1}{p_2} \right)^{\frac{1}{\gamma}}}{p_1} = 1 - \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} \quad (35)$$

In connection with the last equation it should be observed that while the expression $E_c = \frac{T_1 - T_2}{T_1}$ is based upon the general laws of thermodynamics, and is therefore applicable to all working fluids without exception, equation (35) has been derived from

this by the use of equations $\frac{p v}{T} = R$, and $p v^\gamma = \text{constant}$.

Equation (35) is therefore strictly applicable to gas only.

Equation (35) leads to the following important deduction: *To obtain the best efficiency in a closed cycle, it is necessary to supply each heat element at such a pressure p_1 and to reject the part not transformed into work at such a pressure p_2 , that the ratio $\frac{p_1}{p_2}$ shall be as large as possible.*

As previously pointed out, however, in the case of the gas engine, the pressure p_2 cannot well be below atmosphere because the use of a vacuum is neither practicable nor economical. Hence the meaning of equation (35) narrows down to the requirement *to introduce each heat element at highest possible pressure p_1 , and to reject the part not used at a pressure as close to atmosphere as possible.*

With this knowledge it becomes easy to compare the various gas-engine cycles among themselves to determine which of the pressure lines at which heat is supplied gives the best guarantee of efficiency.

Figure 3-6 represents the cycle with combustion at constant volume, Fig. 3-7 the constant pressure cycle, and Fig. 3-8 the cycle with isothermal combustion. The heat is rejected in each case along a constant-pressure line as nearly as possible to atmosphere. It is assumed also in each case that the pressure and temperature at the end of compression shall be the same, that is, that p_c and T_c in Fig. 3-6 shall be equal to p_c and T_c in Fig. 3-7, etc.

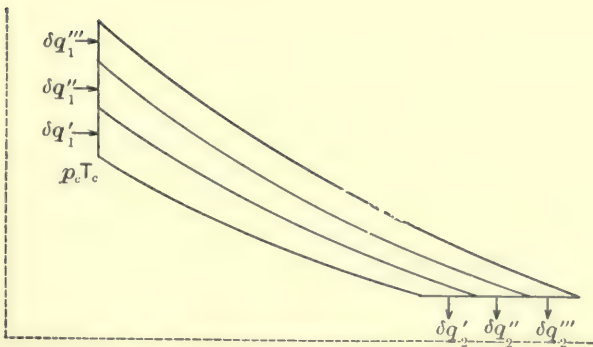


FIG. 3-6.

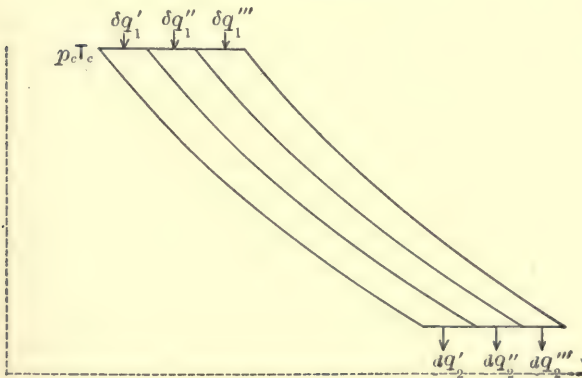


FIG. 3-7.

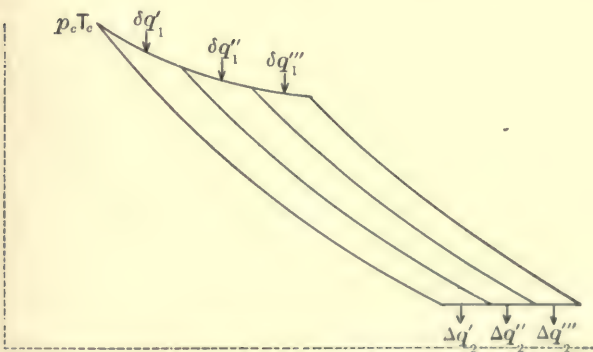


FIG. 3-8.

Now assume that each cycle is divided by adiabatics into an infinite number of elementary cycles. For convenience only three of these are indicated in each figure. With combustion at constant volume Fig. 3-6, each heat element, $\delta q_1'$, $\delta q_1''$, etc., supplied serves to raise the temperature and especially the pressure for the one succeeding it, and since expansion is carried to atmosphere in each case, it is plain that this method of combustion tends of itself to fulfil the requirement above outlined. On account of the increasing pressure ranges, the efficiency of each elementary cycle is in this case greater than that of the one just preceding it.

The same course of reasoning applied to combustion at constant pressure, Fig. 3-7, will show that each heat element, $\delta q_1'$, $\delta q_1''$, etc., while it serves to raise the temperature does not raise the pressure for the next succeeding element. The expansion being again in each case to atmosphere, the pressure ranges for all the elementary cycles, and hence also the efficiency, is the same.

The case of combustion at constant temperature, Fig. 3-8, is even less favorable. Here each heat element, $\delta q_1'$, $\delta q_1''$, etc., does not raise the temperature for the succeeding element, but is accompanied by a decrease of pressure. The exhaust pressure being unchanged, this means a smaller pressure range for each succeeding elementary cycle, and consequently a steadily decreasing efficiency.

Looking next at the heat discharged from each elementary cycle, we may write:

$$\begin{aligned}\delta q_2' &= dq_2' = \Delta q_2' \\ \delta q_2' &> \delta q_2'' > \delta q_2''' \\ dq_2' &= dq_2'' = dq_2''' \\ \Delta q_2' &< \Delta q_2'' < \Delta q_2'''\end{aligned}$$

From all this it follows that, starting with the same pressure of compression, the constant-volume combustion is more efficient than that at constant pressure; and this in turn is more efficient than isothermal combustion.

Isothermal combustion, being so obviously inferior to the other two in theory, is also difficult to carry into operation practically, and for these reasons is practically obsolete.

5. Practical conditions affecting the choice of best cycle for any given case.

It is evident from the preceding article that the most important item in the efficiency question is the pressure at the end of compression. The aim should be to use as high a pressure as possible in order to introduce the first heat element at the highest possible efficiency. But the compression pressure governs the maximum pressure and temperature occurring in the cycle, and that brings us to a consideration of the pressure and temperature limits.

Of these two the temperature limit plays but a secondary part, because it is possible to operate on a cycle whose maximum temperature may be 3000 degrees Fahrenheit for the reason that

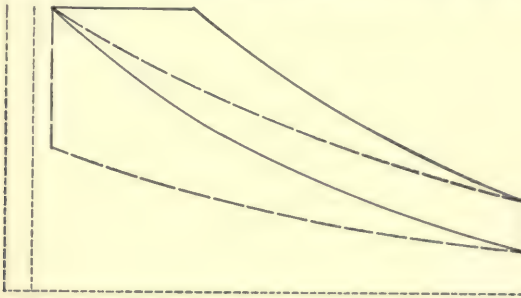


FIG. 3-9.

these maximum temperatures exist but for a very short time. The pressure limit is more important, because no matter how short a time the maximum pressure lasts, the driving mechanism of the machine must be built for this pressure. In modern practice 550-600 pounds seems to be about the maximum pressure limit that can be economically handled. Herein we find a condition which may modify the conclusion arrived at in the previous article as regards comparative efficiency of combustion at constant volume and at constant pressure.

Assuming the more practical condition, that the maximum pressures in the two cycles, instead of the compression pressures, shall be the same, the diagrams would be placed as shown in Fig. 3-9, in which the broken line represents the Otto cycle. It is evident in such a case that only the last heat element of the

Otto cycle is introduced at the same efficiency as the first, and consequently all, of the heat elements of the cycle at constant volume. Hence, upon these premises, the constant-pressure cycle is more efficient than the constant volume cycle. Under some conditions, however, favorable to the Otto cycle, this advantage of the constant-pressure cycle is not very great. Thus in a Diesel engine cutting off at full load at 10 per cent, $\delta = 2.5$, assume $r = 13$, and $\gamma = 1.41$, then E_c will be .564. An Otto engine cycle of the same maximum pressure limit, = about 460 pounds, would show a compression pressure of from 190–225 pounds, r would be about equal to 7, and with $\gamma = 1.41$, E_c would be .550. This shows a gain for the Diesel engine of only 1.5 per cent, but it should be pointed out that a compression pressure of 200 pounds in an Otto cycle can only be reached with extremely lean fuel gases, or with separate fuel and air compression. However that may be, the advantage of the constant-pressure cycle over the constant-volume cycle, presupposing equal maximum pressures in each, is comparatively small, and hence the undoubted gain that the Diesel engine shows in practice over the average constant-volume engine is by some writers attributed not so much to the cycle as to the greater perfection of combustion. That this is approximately true has been proven by Gldner who constructed engines operating on the Otto cycle upon the most advanced ideas, and obtained efficiencies fully as good as those obtained by Diesel.

A second limit set to the compression pressure that can be carried is due to pre-ignition. All fuel mixtures will ignite spontaneously if the temperature becomes high enough, but the critical temperature varies greatly for the different fuels. This fact directly governs the compression pressure. While it is hardly advisable to use more than say 80–90 pounds in the case of a gasoline mixture, a blast furnace gas mixture will easily stand 150 pounds without pre-ignition. Since the efficiency of the cycle depends directly upon the compression pressure, as above shown, we should expect a better efficiency for blast furnace gas than for gasoline, and this is actually so in practice. The difference, however, is due to the nature of the fuel. A remedy for this state of affairs would be to compress the air separately and introduce the gasoline or other fuel oil only at the moment

combustion is desired. This leads to the constant-pressure cycle, and thus the fuel to be employed should be considered in the choice of cycle.

A final point to be considered is this: By theory, the greater the compression pressure, the higher the efficiency. This holds for either type of cycle. But how far can this compression be carried, outside of the questions of upper pressure limit and pre-ignition above considered, before the added gain due to higher compression is in practice balanced or overbalanced by attendant losses.

The following discussion of this question by Güldner, as applied to combustion at constant volume, is instructive.

The efficiency of any engine should not be judged upon cylinder performance, but upon the performance at the shaft. Usually this is called the "thermal efficiency per brake horse-power," but a shorter and more expressive term which the authors prefer and will use in this treatise, is "Economic Efficiency E_e ."

Now

$$E_e = E_t E_m = X E_c E_m.$$

where

E_t = thermal efficiency per indicated horse-power.

E_m = mechanical efficiency of engine = $\frac{\text{Brake H. P.}}{\text{Indicated H. P.}}$

E_c = cyclic efficiency as derived in the preceding articles = $= 1 - \frac{1}{r^{\gamma-1}}$ for combustion at constant volume.

and

X = a factor such that $E_t = X E_c$, so that X is always less than 1.

The mechanical efficiency may be expressed by

$$E_m = \frac{p_i - p_f}{p_i}$$

where

p_i = mean indicated pressure per square inch of piston,

and

p_f = pressure lost in friction per square inch of piston.

Hence

$$E_e = X E_c \frac{p_i - p_f}{p_i} = X \left(1 - \frac{1}{r^\gamma - 1} \right) \frac{p_i - p_f}{p_i}$$

The value of E_e therefore depends upon the relation between the factors X , p_i , and p_f , as modified by a variation in r , that is, in the pressure of compression. The change in the value of X due to a variation in r is quite unknown, and is in any case so small that it may be neglected. Regarding the relation between p_i , p_f , and r , numerous tests have shown that, as r increases, the mechanical efficiency $\frac{p_i - p_f}{p_i}$, is at first quite constant, but that after a certain point it commences to decrease quite rapidly. This is due to the fact that for the first part of the range any increase in p_f , due to an increase in r , is counterbalanced by a corresponding gain in p_i . Beyond a certain point, however, there is a loss in p_i , owing to the fact that the fuel mixture has to be made more and more lean to prevent pre-ignition, and the mechanical efficiency consequently decreases. *Hence we have the net result that as long as an increase in p_f , due to an increase in r , is met by a proportionate gain in p_i , the Economic Efficiency E_e will increase. Just as soon, however, as, with increase in r , the gain in p_i is no longer sufficient to overcome the increase in p_f , E_e will commence to decrease, and the economic compression limit will have been passed.*

Tests and computations have shown that up to $r = 6$, which means a compression pressure of about 160 pounds, the mechanical efficiency does not change materially, but that beyond this it commences to decrease. This is probably due to the necessary increase in the size of machine parts due to the greater pressures and to the fact that leaner mixtures than would ordinarily be used must be employed to prevent pre-ignition, with a corresponding loss in the value of p_i . Beyond $r = 6$ the gain in E_e is small and at $r = 10$ it practically ceases to increase. *Hence we conclude that for combustion at constant volume the use of values of r greater than about 8, for which the compression pressure equals about 225 pounds, is no longer accompanied by any useful gain in the economic efficiency of the machine. This statement does not apply to combustion at constant pressure because its E_e follows*

a different law, and E_m decreases more slowly on account of the generally greater value of p_i .

Finally, in laying out an Otto cycle, to obtain high engine capacity per unit volume of cylinder means making the maximum pressure as high as possible to obtain a high value of p_i . With fuel mixtures which cannot stand a high amount of compression this would mean the use of rich gas mixtures. Where the compression can be made higher, the ratio of $\frac{\text{max. pressure}}{\text{comp. pressure}}$, instead of being from 4-5, may be profitably made from 2.5 to 3 by the use of leaner mixtures, so that the maximum pressure shall be in the neighborhood of say 450 pounds. The chances are, as the gas engine develops, that higher maximum pressures will be employed, but according to Güldner the use of working pressures exceeding 600 pounds is neither economical nor safe.

CHAPTER IV

THE VARIOUS EVENTS OF THE CONSTANT VOLUME AND THE CONSTANT-PRESSURE CYCLES AS MODIFIED BY PRACTICAL CONDITIONS

1. IN the previous chapter the various cycles were discussed and compared on theoretical grounds. For this purpose several things were assumed which in practice are only approximately true; thus compression and expansion lines were assumed adiabatic and the surrounding walls impermeable to heat. It is, however, true that the heat interchange between the charge and the walls may be such as to give a line on a diagram which strictly follows the adiabatic law. Such a line is by some writers called a false or pseudo-adiabatic. Again it has been assumed in the theoretical discussion that ignition is perfect, that the composition of the charge is uniform, and that combustion is complete and perfect; none of these things quite obtain in practice and all the variations have their influence upon engine performance. Thus it happens that the cyclic efficiency above computed is in any given case never realized, but that the actual thermal efficiency is always less than E_c .

The following paragraphs will point out these modifications more in detail.

2. The four-stroke Otto cycle.

(a) THE SUCTION STROKE. At the end of the exhaust stroke, the clearance volume V_c , Fig. 4-1, is filled with burned gases under a pressure p_e and a temperature T_e . The weight of these gases can only be approximately computed, since nothing definite is known of the temperature T_e . It is in most cases probably between 12-1400 degrees Fahrenheit, while the pressure p_e in well-designed machines may be from 16-18 pounds absolute, but circumstances may alter these figures considerably. At the commencement of the suction stroke the pressure falls from p_e to the suction pressure p_s along a curve determined by the re-expansion of the burned gases in the clearance spaces. Only after this re-expansion will the fresh charge be drawn into the cylinder. It is thus seen that the

volumetric efficiency, E_v , of the cylinder, that is, the ratio $\frac{\text{volume of fresh mixture}}{\text{volume of piston displacement}}$, depends directly upon the weight of burned gases remaining, thus affecting cylinder capacity. If through bad form of combustion chamber, too small an exhaust opening, or a restricted exhaust pipe, the exhaust pressure should be kept too high, or too much burned gas remain behind, this effect of re-expansion will be more marked than above stated. A too early closure of the exhaust valve may have the effect shown in Fig. 4-1 in dotted line. It is evident, however, that in the ordinary four-cycle engine without scavenging this loss of volumetric efficiency will always be present, depending upon the clearance volume.

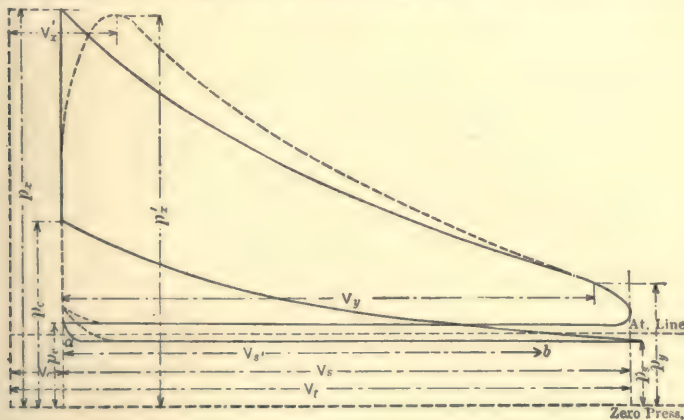


FIG. 4-1.

A second factor affecting the volumetric efficiency of the cylinder is the suction pressure p_s . At the end of the suction stroke the cylinder contains a volume of gas, v_r , made up partly of burned gas and partly of fresh mixture, under a pressure p_s . The compression stroke raises this amount of gas to the pressure p_c with a volume v_c . The compression curve crosses the atmospheric line when the stroke volume is only v_s and not the full volume v_s . Hence the volume $v_s - v_s$ represents a loss in volumetric efficiency, and this loss is the greater the smaller p_s . It follows that the inlet pipes and valves should be so designed as to cause a minimum suction "under-pressure," that is, to keep p_s as close to atmospheric pressure as possible. This also makes clear why vaporizers and carbureters always decrease engine capacity somewhat.

Thus there is a loss of volumetric efficiency, and hence of engine capacity, at each end of the suction line. The real volumetric efficiency, E_v , is found in any given case by dividing the volume represented by the line $a-b$, measured along the atmospheric line, by the volume v_s of the piston displacement.

Attention should at this point be called to the fact that certain systems of speed regulations depend upon the variation in the suction pressure. By throttling the mixture from the beginning of the stroke, or by cutting off the supply completely at a given point in the stroke, p_s is increased or decreased depending on the load on the engine, thus directly controlling the charge volume, and hence the engine capacity. This is explained more in detail in the chapter on governing.

Since it is quite evident that neither the loss by re-expansion or that due to the suction pressure at the end of the suction stroke can be entirely avoided, it becomes interesting, at least from the point of design, to know approximately what values of E_v to expect in different types of engines. Naturally E_v decreases as engine speed increases, because since high-speed engines are usually small engines, the difficulty of placing valve openings large enough to prevent serious throttling of the charge becomes greater as speeds increase. Computations are of little avail in this matter since little is definitely known of the temperature of the charge at the end of the suction stroke. For that reason more reliance is to be placed in figures based upon practical experience. The following table is due to Güldner:

	E_v	p_s lbs. per sq. inch absolute
1. Slow-speed engines with mechanically operated inlet valve.....	.88 - .93	12.9 - 13.7
2. Slow-speed engines with automatic inlet valve.....	.80 - .87	12.5 - 13.2
3. High-speed engines with mechanically operated inlet valve.....	.78 - .85	11.7 - 12.5
4. High-speed engines with automatic inlet valve.....	.65 - .75	11.4 - 12.2
5. Very high-speed engines with automatic inlet valves and air cooling.....	.50 - .65	8.8 - 11.0

Suction gas generators and vaporizers may in unfavorable cases decrease the above figures for E_v as much as .05.

(b) THE COMPRESSION STROKE. — The compression line may

be taken to follow the general law $pv^n = \text{constant}$. During the first part of the stroke there is probably a flow of heat from the walls to the comparatively cool charge, but this is soon overbalanced by the heat of compression, so that during the last and greatest part of the stroke the flow is into the walls. The compression curve is therefore rarely an adiabatic $pv^\gamma = \text{constant}$, where $\gamma = \frac{C_p}{C_v}$, as computed for the charge. In most cases the line is intermediate between an adiabatic and an isothermal, and strictly also the value of the exponent, n , is not constant along the entire line. The value of n in actual cases lies between 1.30 and 1.38, with an average of about 1.35. In cases of very ineffective cooling n may exceed $\gamma = \frac{C_p}{C_v}$. It should also be noted that leaky pistons and valves cause a flattening of the compression curve, which apparently decreases the true value of n .

From the equation $pv^n = \text{constant}$, we may derive the following equation for the absolute pressure p_c at the end of compression, see Fig. 1.

$$p_c = p_s \left(\frac{v_s}{v_c} \right)^n = p_s r^n$$

The absolute temperature at the pressure p_c will be

$$T_c = T_s \left(\frac{p_c}{p_s} \right)^{\frac{n-1}{n}} = T_s r^{\frac{n-1}{n}}$$

This equation requires an assumption for the value of T_s , the temperature at the end of the suction stroke. As already stated, not a great deal is known about this. S. A. Moss states that experiments have shown it to be between 200–300 degrees Fahrenheit, that is, 660–760 degrees absolute, but gives no details. Schöttler in his examples on type-cycles assumes in most cases 350 degrees Centigrade absolute, = 632 degrees Fahrenheit absolute.

The clearance volume required to produce a pressure p_c and a temperature T_c will be

$$v_c = v_s \left(\frac{p_s}{p_c} \right)^{\frac{1}{n}} = v_s \left(\frac{T_s}{T_c} \right)^{\frac{1}{n-1}}$$

The following table shows values for the absolute compression pressure, p_c and the end temperature T_c for various values of n , of r , the ratio of compression, and of T_s . The value of p_s has been assumed at 12.5 pounds absolute:

Ratio of Compression $r =$		2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0	10.0
$n = 1.25$	For $p_s = 12.5$, $p_c =$ $\left\{ \begin{matrix} 600 \\ 650 \end{matrix} \right.$	29.7	39.2	49.3	59.9	70.7	81.8	93.2	117.5	142.0	168.0	195.0	222.0
	$T_s =$ $\left\{ \begin{matrix} 700 \\ 750 \end{matrix} \right.$	690	720	747	770	792	811	828	858	885	910	930	951
	$T_c =$ $\left\{ \begin{matrix} 700 \\ 750 \end{matrix} \right.$	748	780	811	835	858	877	896	930	958	983	1010	1028
	$\left\{ \begin{matrix} 700 \\ 750 \\ 800 \end{matrix} \right.$	805	840	875	900	924	945	965	1002	1032	1062	1088	1110
$n = 1.30$	For $p_s = 12.5$, $p_c =$ $\left\{ \begin{matrix} 600 \\ 650 \end{matrix} \right.$	30.7	41.2	52.0	63.7	75.7	88.3	101.2	128.2	156.7	186.5	217.5	250.0
	$T_s =$ $\left\{ \begin{matrix} 700 \\ 750 \end{matrix} \right.$	703	740	772	800	825	847	868	905	938	966	1000	1018
	$T_c =$ $\left\{ \begin{matrix} 700 \\ 750 \end{matrix} \right.$	763	803	837	866	895	917	942	980	1015	1043	1076	1102
	$\left\{ \begin{matrix} 700 \\ 750 \\ 800 \end{matrix} \right.$	821	864	900	933	964	987	1012	1055	1093	1128	1160	1187
$n = 1.35$	For $p_s = 12.5$, $p_c =$ $\left\{ \begin{matrix} 600 \\ 650 \end{matrix} \right.$	31.9	43.0	55.1	67.8	81.2	95.2	109.5	140.5	162.4	206.5	242.2	300
	$T_s =$ $\left\{ \begin{matrix} 700 \\ 750 \end{matrix} \right.$	718	760	797	832	860	885	912	955	995	1030	1063	1092
	$T_c =$ $\left\{ \begin{matrix} 700 \\ 750 \end{matrix} \right.$	778	825	864	900	932	963	987	1035	1078	1115	1152	1182
	$\left\{ \begin{matrix} 700 \\ 750 \\ 800 \end{matrix} \right.$	838	888	932	970	1002	1035	1064	1114	1157	1200	1240	1275
$n = 1.41$	For $p_s = 12.5$, $p_c =$ $\left\{ \begin{matrix} 600 \\ 650 \end{matrix} \right.$	33.1	45.5	58.8	73.1	88.3	103.0	120.6	156.5	194.0	234.0	277.0	321.0
	$T_s =$ $\left\{ \begin{matrix} 700 \\ 750 \end{matrix} \right.$	735	783	826	864	897	927	956	1013	1055	1095	1135	1170
	$T_c =$ $\left\{ \begin{matrix} 700 \\ 750 \end{matrix} \right.$	795	847	893	936	972	1003	1033	1097	1140	1187	1228	1268
	$\left\{ \begin{matrix} 700 \\ 750 \\ 800 \end{matrix} \right.$	857	912	962	1008	1048	1080	1113	1180	1228	1278	1320	1363
		917	976	1030	1077	1122	1157	1194	1265	1316	1368	1415	1462
		977	1142	1098	1149	1195	1232	1273	1348	1402	1458	1510	1558

It has already been shown in the previous chapter how the cyclic efficiency and consequently also the thermal efficiency of an engine depends upon the compression pressure. It has also been shown that there are commercial limits to the compression pressure due to pre-ignition of charge. As regards pre-ignition of charge, which is to be distinguished from back firing or explosions in the exhaust pipe, the greater danger of this exists with fuel mixtures high in hydrogen. Hydrogen, next to acetylene, possesses the lowest ignition point of any of the gaseous fuels commonly employed. Hence illuminating gas with about 45 per cent by volume of hydrogen is much more liable to pre-explosion under the temperature of compression than a producer gas with 15 per cent. Hence we may use a higher compression pressure in the case of the latter gas, and may expect, and actually obtain, a higher thermal efficiency in practice. With gases very rich in CO and low in hydrogen there is little danger of pre-ignition even up to the commercial limit of high pressures. Lucke estimates that for every 5 per cent of hydrogen that the gas contains 15 pounds should be subtracted from the otherwise allowable compression pressure. In general it should be borne in mind, and this applies to all fuel mixtures, the better and more effective the cooling of the cylinder, the higher can be the compression without danger of pre-ignition. Anything which draws down the temperature during compression, as water injection, is also favorable to the same thing. A case in point is Banki's method of water injection with gasoline as fuel, by which means compression pressures could be employed which gave thermal efficiency results equal to the best obtained on lean gases.

Premature explosions are sometimes directly due to faulty design of the combustion chamber. Any projecting point or edge in the chamber which cannot be effectively water-cooled may become red hot under compression, and thus locally raise the temperature high enough to cause pre-ignition. This action is so certain that it has been proposed to use it as a method of ignition by placing a projection on the piston face. It was found that while this scheme would work, it was not susceptible of control, and was abandoned.

The following table shows the safe compression pressures in use with the fuels commonly employed, as given by Lucke. The

second column gives the percentage of clearance required to produce the pressure p_c in terms of the piston displacement. This column has been computed assuming $p_s = 13$ lbs. absolute and $n = 1.35$.

Fuel	Compression Pressure Lbs. by gage	% Clearance in terms of Piston Displacement
Gasoline:		
Auto-engines with carbureters, cooling not very effective, high speeds.....	45 - 95 Ave., 65	35
Gasoline:		
Stationary, slow-speed engines, more effec- tive cooling but usually less simple com- bustion chamber	60 - 85 Ave., 70	32
Kerosene:		
Hot bulb injection and ignition	30 - 75	35 - 40
Kerosene:		
Previously vaporized in devices not requir- ing a vacuum	45 - 85 Ave., 65	35
Natural Gas	75 - 130	
Natural Gas:		
Average for large and medium engines...	Ave., 115	22
Illuminating gas	60 - 100 Ave., 80	26
Producer Gas	100 - 160	
Producer Gas:		
In large engines water-cooled on pistons and valves	Ave., 135	20
Blast Furnace Gas	120 - 190 Ave., 155	17

(c) THE COMBUSTION LINE. — The shape of the combustion line depends primarily upon the interrelation of three things: composition of charge, point of ignition, and piston speed.

For every fuel there is a certain fuel-air mixture which gives the greatest rate of flame propagation, *i.e.*, the most rapid combustion. Any further admixture of neutral gases, whether these be air or burned gases, results in a slower combustion, until there comes a time when ignition fails. Suppose, therefore, that in any given engine with full throttle, constant speed, and proper igni-

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

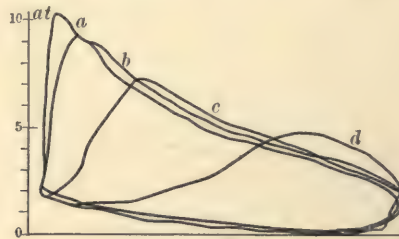


FIG. 4-3.

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.

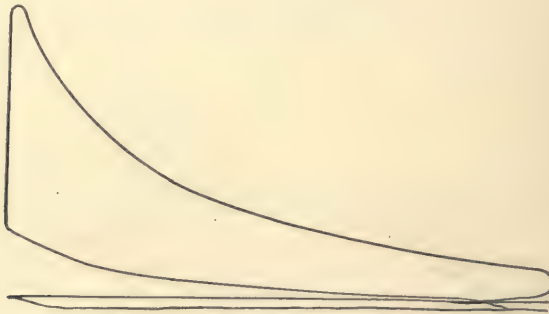


FIG. 4-4.

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 $\frac{3.5}{1} = 3.5$.

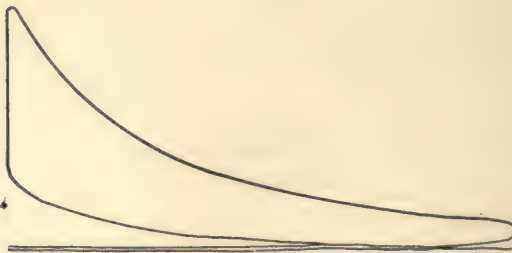


FIG. 4-5.

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 $\frac{3.1}{1} = 3.1$.

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.

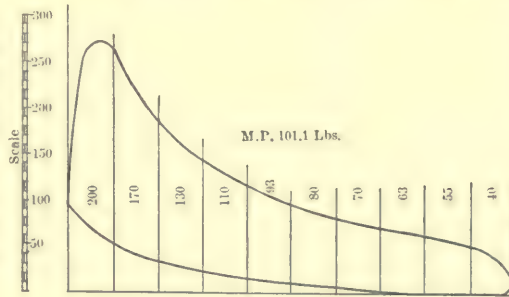


FIG. 4-6.

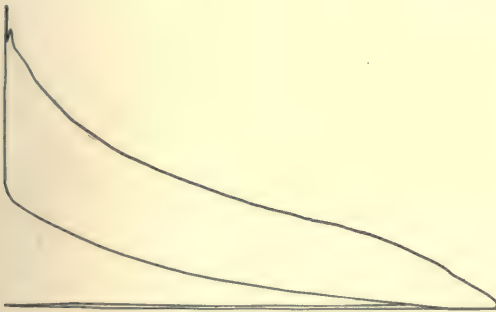


FIG. 4-7.

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 $\frac{372}{80} = 4.07$.

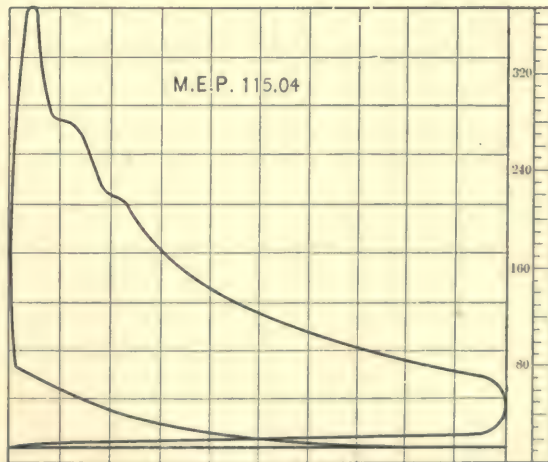


FIG. 4-8.



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 = $\frac{257}{142} \div 1.8$.

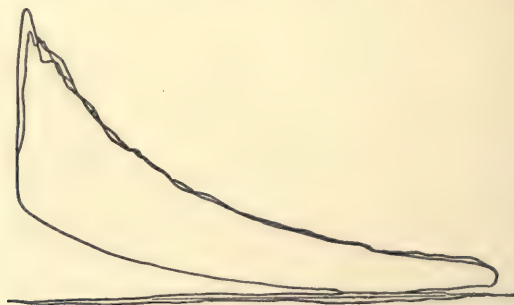


FIG. 4-10.

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:

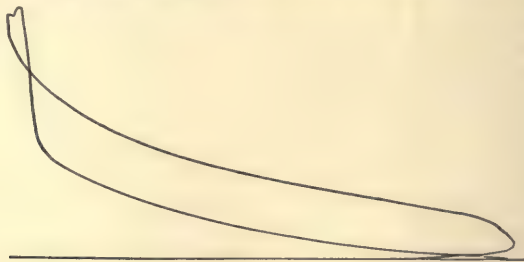


FIG. 4-11.

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.

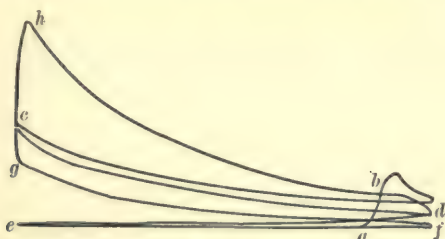


FIG. 4-12.

FIG. 4-12. Case of back-firing as distinguished from pre-ignition. Power, Oct. 15, 1900. Explosion in the suction pipe at *a* during the suction stroke. Observed in an engine which attempted scavenging by means of oscillations of burned gases in the exhaust pipe. Back-firing more often occurs in the exhaust pipe due to accumulation of unburned gas.

(*d*) THE EXPANSION LINE. — The expansion line, like the compression line, may be taken to follow a general law $p v^n = \text{constant}$, in which n is rarely equal to $\gamma = \frac{C_p}{C_v}$, for the burned gases. Assuming that combustion is complete when the piston starts forward, the loss of heat to the jacket during expansion should cause the expansion line to lie below the adiabatic, that is, n should be greater than γ . (It should be remembered in this connection that leaky pistons and valves cause an increase in the true value of n .) Now it is very often found that the expansion line falls off much more slowly than this, coinciding now and then with the adiabatic, but very often lying between this and the isothermal, that is, n is less than γ . The explanation of this phenomenon has been held to be an evolution of heat along the expansion line equal to or exceeding the loss of heat to the jacket during this period. Several theories have been advanced to explain this.

The older machines of Lenoir and Hugon showed values of the exponent n for the expansion line between 1.4 and 1.6, while the earlier Otto machines showed values approximating 1.3. To explain this, Otto, and after him, Slaby, at least for a time, advanced the *theory of stratification*. It was supposed that the charge of a 4-cycle machine could be so arranged as to have practically nothing but burned gases against the piston, next practically air, then a layer of poor mixture, and finally near the igniter the fuel mixture in its full strength. It was further

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

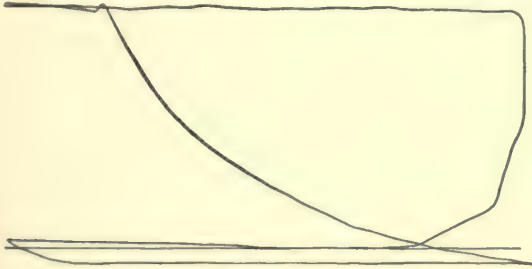


FIG. 4-13.

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 $16\frac{1}{2} \times 22$

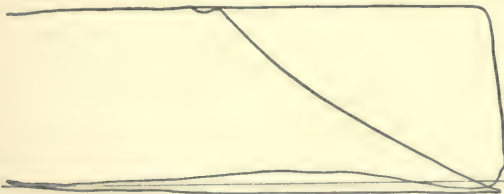


FIG. 4-14.

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

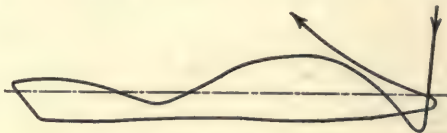


FIG. 4-15.

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

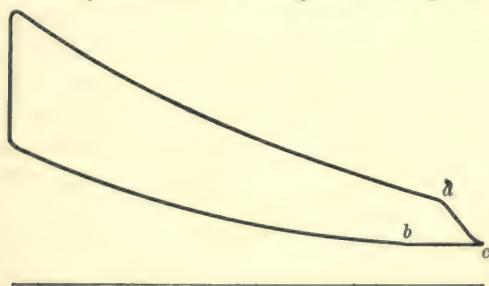


FIG. 4-16.

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.

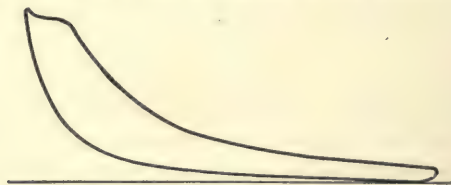


FIG. 4-17.

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.)

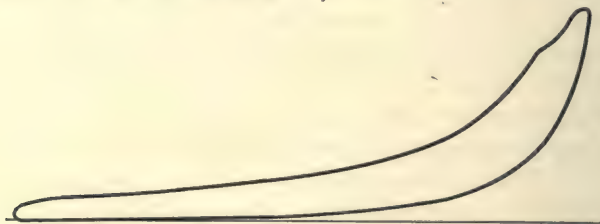


FIG. 4-18.

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.

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

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

Substituting (4) for p in equation (2)

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

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

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 $p v^n = \text{constant}$.

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

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

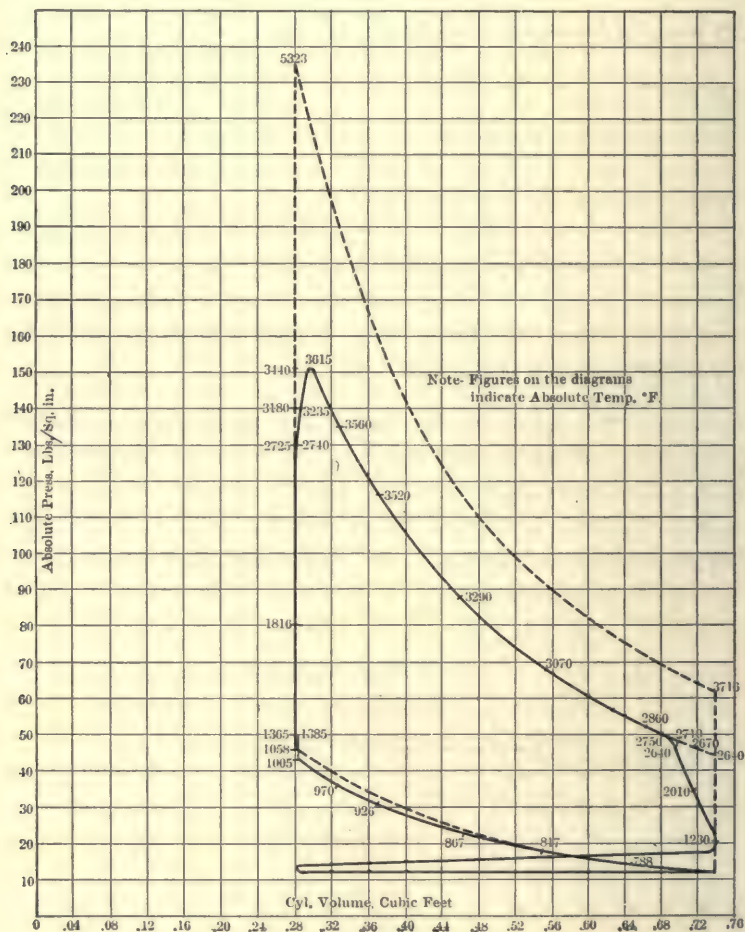


FIG. 5-1.

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.

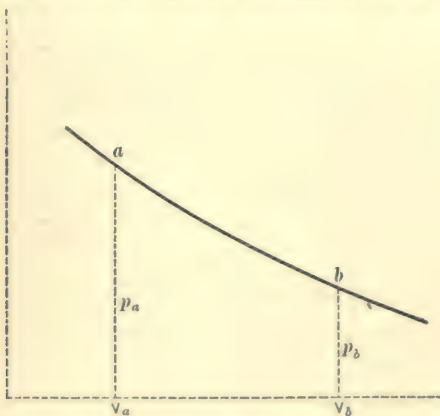


FIG. 5-2.

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.

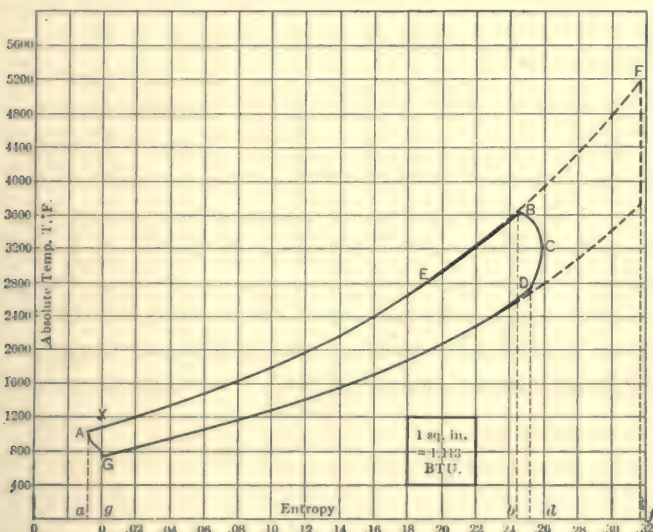


FIG. 5-3.

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).

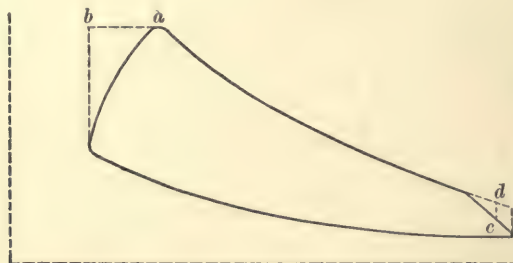


FIG. 5-4.

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.

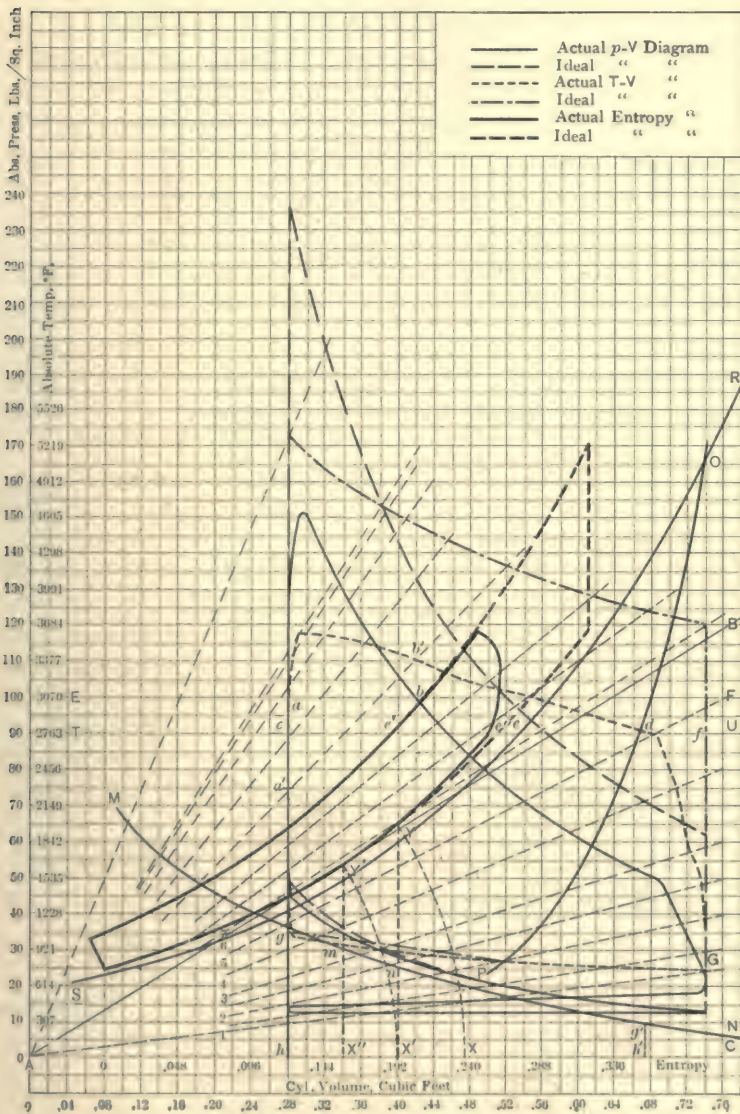


FIG. 5-5.

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 $A B$ 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, $B A C$, and hence $\frac{p}{R}$ is constant for any point along $A B$. 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 $E F = 100$ 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 $A C$ has been taken. With A as a center and any radius AX , draw the arc XY to cut any convenient line, as $A B$, 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 $A B$, 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

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

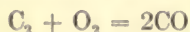
may be employed for a change of either pressure and temperature separately or a simultaneous change of both. The weight of a cubic foot is inversely proportional to the volume.

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

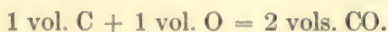
COMBINING WEIGHTS AND VOLUMES. — All elements, when they do combine, unite only in certain fixed proportions, although there may be several proportions for any given pair of elements. Thus carbon forms two combinations with oxygen, CO and CO_2 .

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.

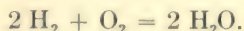
Thus take the combination of C and O to CO. Always remembering to use the combining weights of the various elements we can write



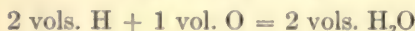
that is



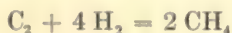
Similarly,



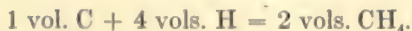
that is,



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

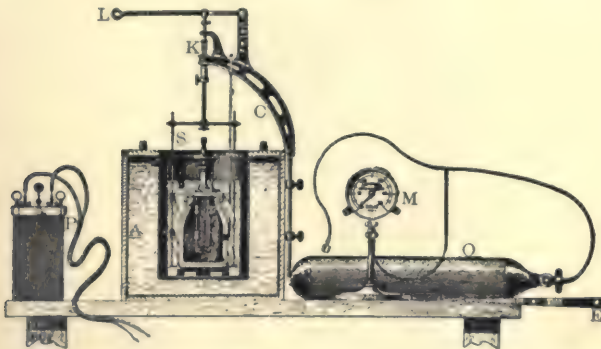


FIG. 6-1. — Mahler Bomb Calorimeter.

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

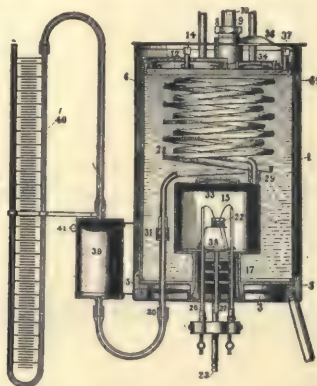


FIG. 6-2. — Carpenter Coal Calorimeter.

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 $L K S$, 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-

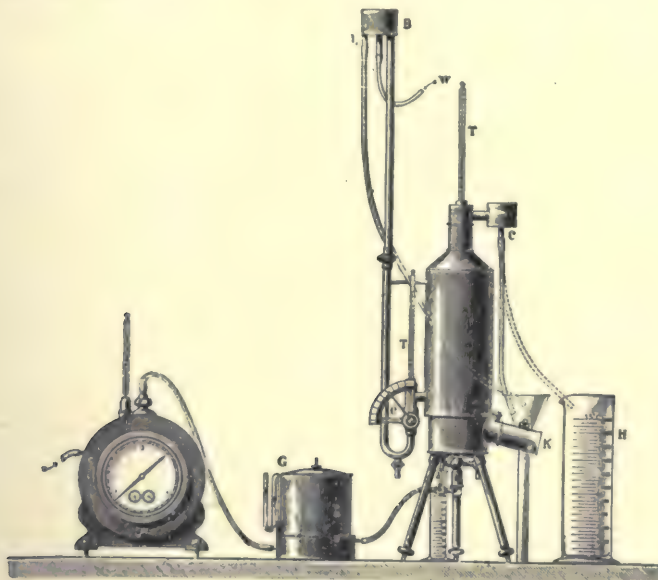


FIG. 6-3. — Junker's Gas Calorimeter.

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

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

$$= 20016 \text{ B. T. U. per lb.}$$

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
Trevelton..	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 K in equation $\frac{p_v}{T} = R$
	Lb.	Cu. ft.		Oxygen for 1 lb., lbs.	Air for 1 lb., lbs.	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	.593	.468	97.25
Ethylene	20023	1564	.07809	3.43	14.95	14.58	3.14	.404	.333	55.23
Acetylene	20673	1499	.07251	3.07	13.35	11.99	3.38	.346	.270	59.13
Butylene	19512	3040	.15590	3.43	14.95	28.86	3.14	.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 V	Weight W		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

$$\frac{\frac{.2507 + .1873}{2} + (2 \times .0031) + (3 \times .0031) - .0003}{.21} = \frac{.2342}{.21} = 1.12 \text{ cubic feet.}$$

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).

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

$\text{H}_2\text{O} = (.1873 + .0062 + .0062) = .1997$ 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_2O	.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/V_1	.896	.912	.926
% contraction	10.4	8.8	7.4
Principal Burned Gas Constants $\left\{ \begin{array}{l} R_e = R \frac{V_2}{V_1} \\ C_p \\ C_v \end{array} \right\}$	52.1	52.5	52.7
per pound	.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

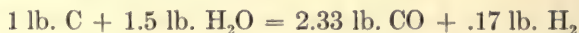
$$1 \text{ lb. of C} + 1.33 \text{ lb. O} = 2.33 \text{ lb. CO}$$

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

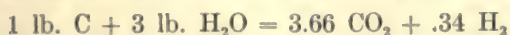
$$\frac{.17}{2.33 + .17} = .068 \text{ lb. Hydrogen}$$

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:

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

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 per cent CO, 15.9 per cent H, 44.3 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.

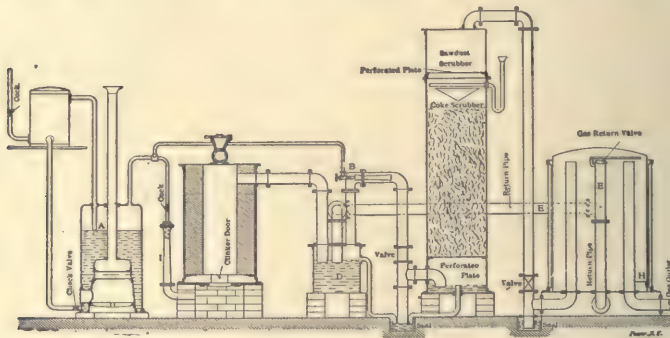


FIG. 7-3. — Wile Producer Installation.

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.

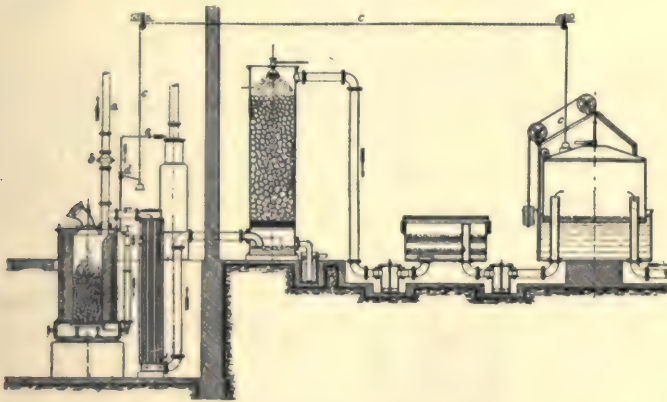


FIG. 7-4. — Koerting Pressure Producer.

*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.

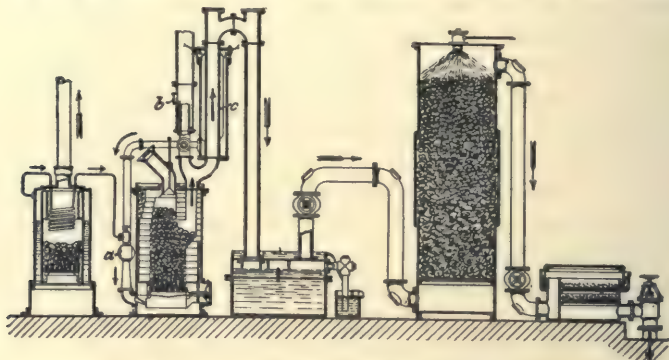


FIG. 7-5. — Deutz Pressure Producer.

*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.

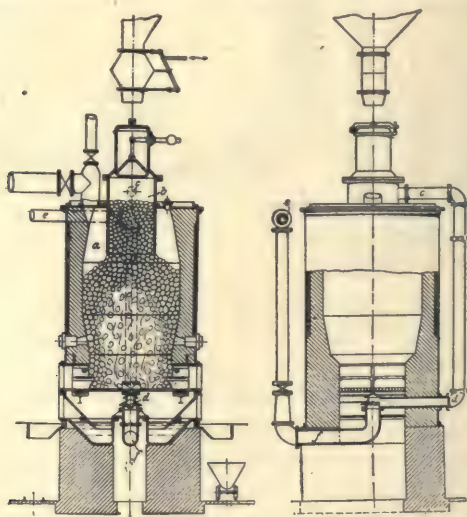


FIG. 7-6. — Poetter Producer.

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