

## 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.

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## PREFACE TO THE SECOND EDITION

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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  $20 \times 15$  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 (542½ 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  $abdef$ .

**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,

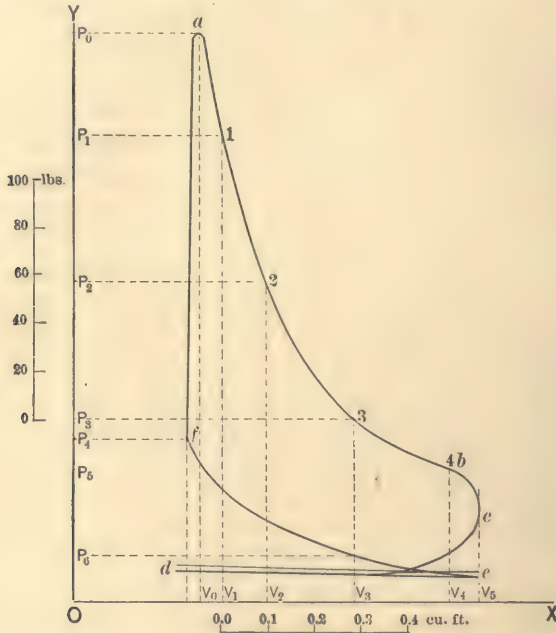


FIG. 1-1. — Pressure-volume or Work Diagram.

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 similar 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 CO<sub>2</sub> 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

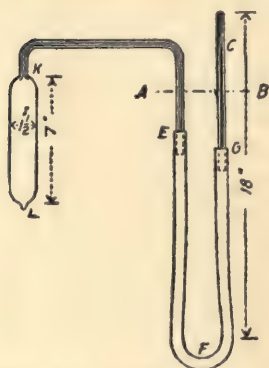


FIG. 1-4. — Jolly Air Thermometer.

leg *C F* 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 *A B*. The increase of pressure is measured by a scale attached to *C G*.

#### 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;

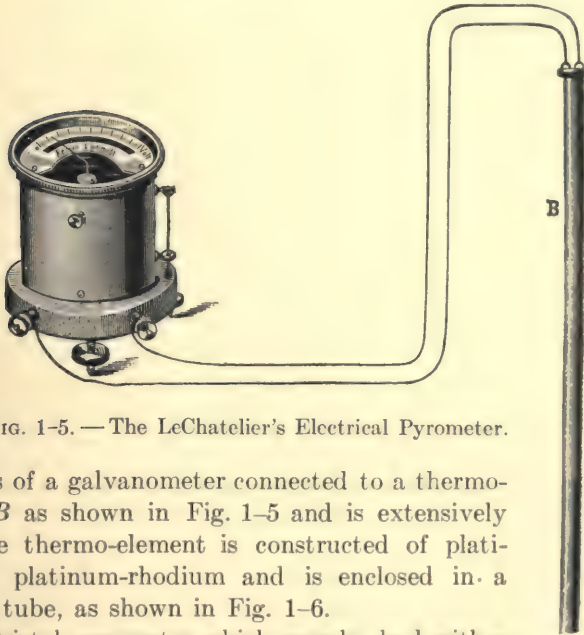


FIG. 1-5. — The LeChatelier's Electrical Pyrometer.

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                      | $C_p$  | $C_v$  | $\gamma$ | Gas                           | $C_p$  | $C_v$  | $\gamma$ |
|--------------------------|--------|--------|----------|-------------------------------|--------|--------|----------|
| H                        | 3.4090 | 2.4177 | 1.41     | CO                            | 0.2479 | 0.1738 | 1.41     |
| O                        | 0.2175 | 0.1543 | 1.41     | CO <sub>2</sub>               | 0.2169 | 0.157  | 1.29     |
| N                        | 0.2438 | 0.1729 | 1.41     | CH <sub>4</sub>               | 0.5929 | .....  | .....    |
| Air                      | 0.2375 | 0.1684 | 1.41     | C <sub>2</sub> H <sub>4</sub> | 0.4040 | 0.320  | 1.26     |
| H <sub>2</sub> 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  $\phi$  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.

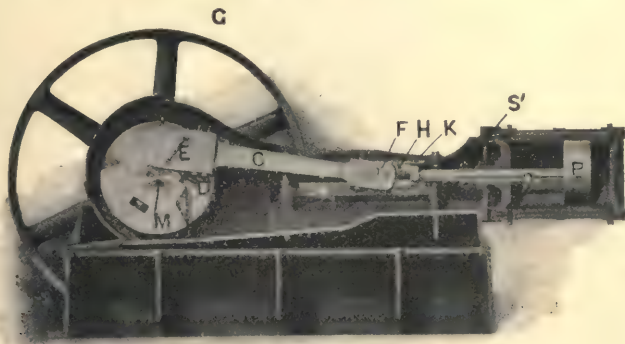


FIG. 1-7. — Double Acting Steam Engine.

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

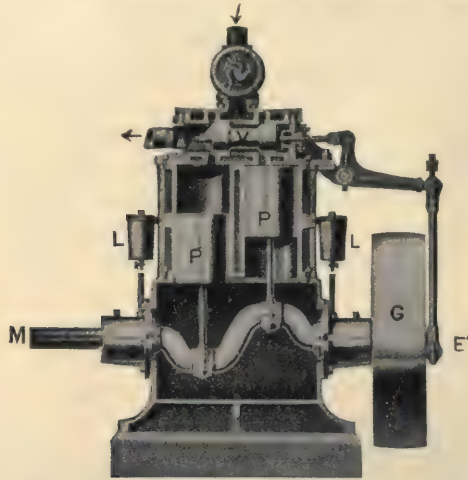


FIG. 1-8. — Two-cylinder Single-acting Steam Engine.

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

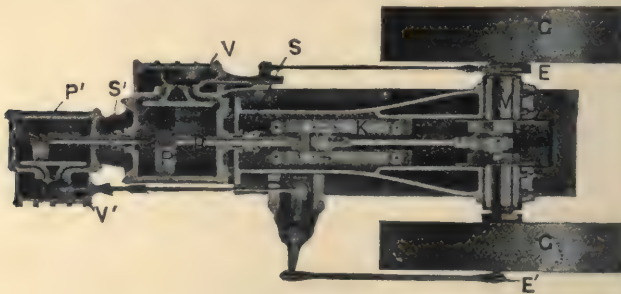


FIG. 1-9. — Tandem Compound Steam Engine.

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

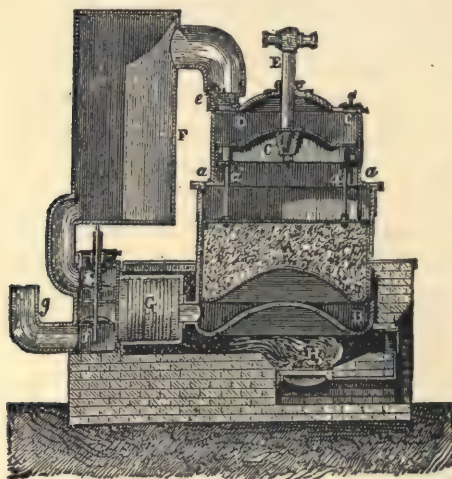


FIG. 1-10. — Ericsson Hot Air Engine.

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

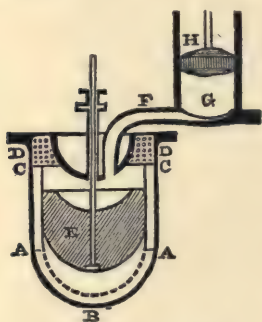


FIG. 1-11. — Stirling Hot Air Engine.

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

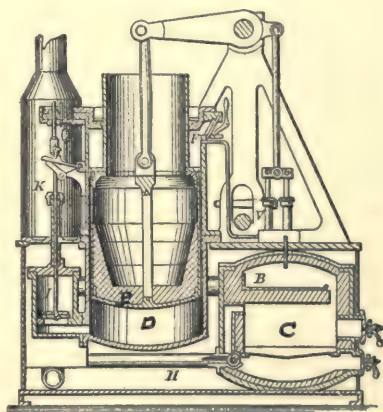


FIG. 1-12. — Wilcox Hot Air Engine.

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.

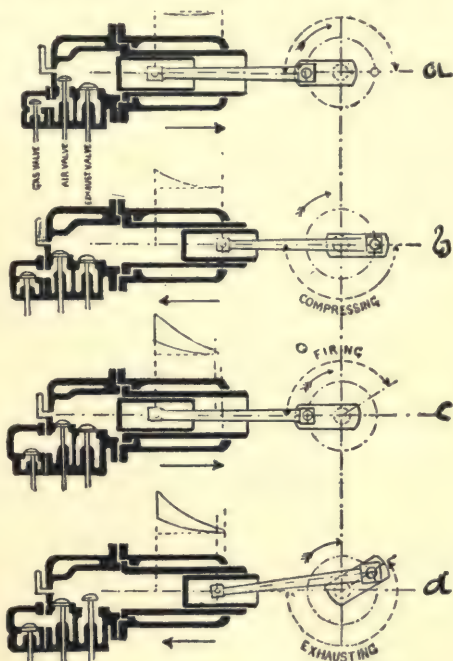


FIG. 1-13. — Diagram of Four-cycle Engine.

(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

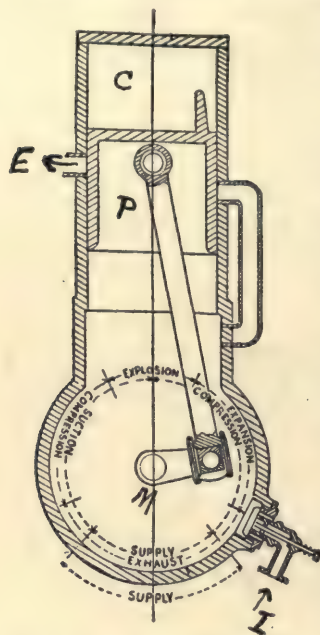


FIG. 1-14. — Two-cycle Engine.

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*.

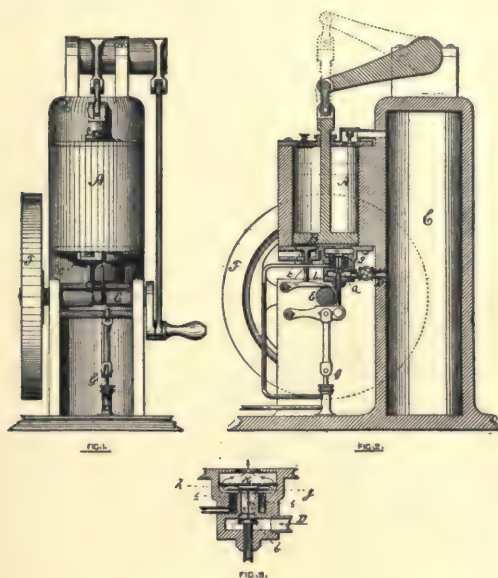


FIG. 1-15. — The Brayton Engine.

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.

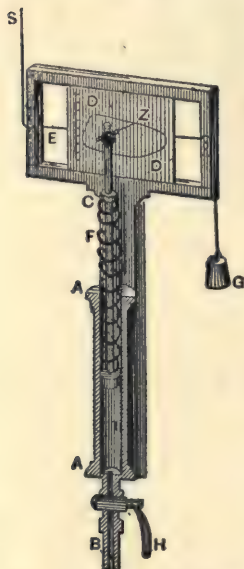


FIG. 1-16. — The Watt Engine Indicator.

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.

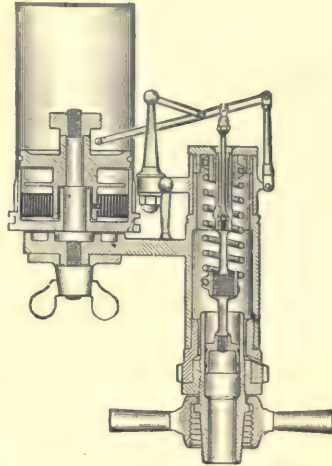


FIG. 1-17. — The Thompson Indicator.

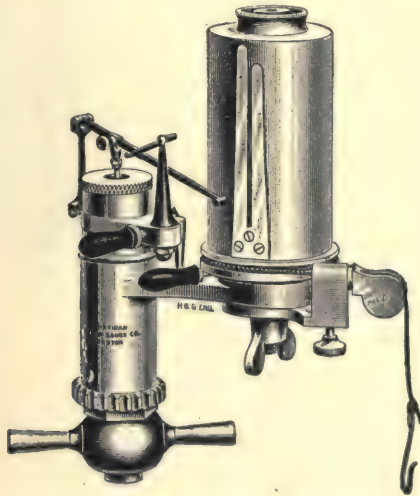


FIG. 1-18. — The Thompson Indicator.

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

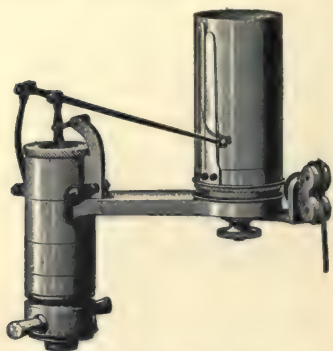


FIG. 1-19. — The Crosby Indicator.

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.

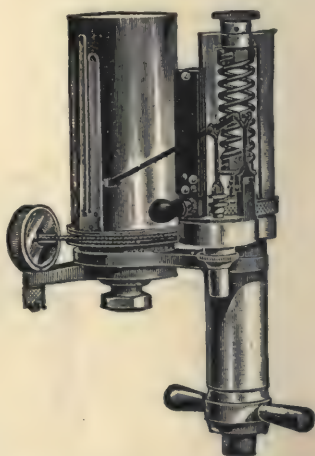


FIG. 1-20. — The Tabor Indicator with External Spring.

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.

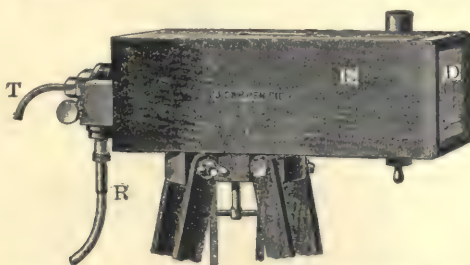


FIG. 1-21. — Perspective View of Manograph.

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

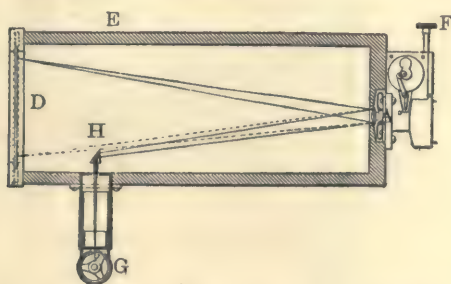


FIG. 1-22. — Horizontal Section of Manograph.

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.

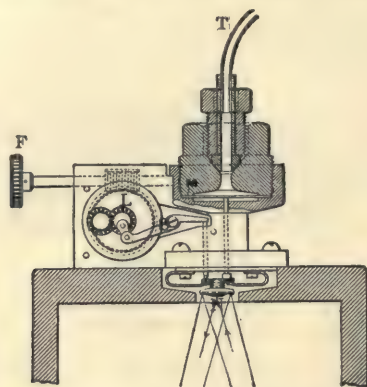


FIG. 1-23. — Section of Manograph Mirror.

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

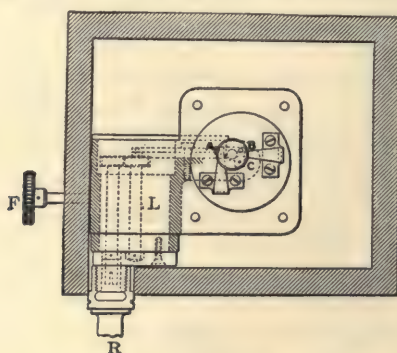


FIG. 1-24. — Manograph Engine Connection.

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

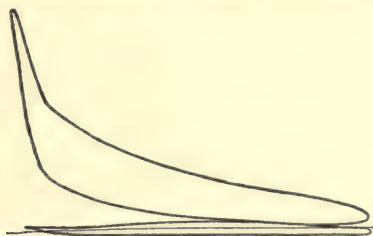


FIG. 1-25 — Diagram with Manograph.

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.

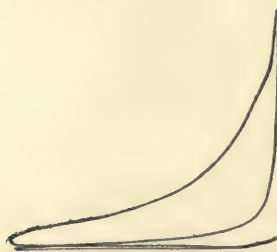


FIG. 1-27. — Diagram with Optical Indicator.

**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

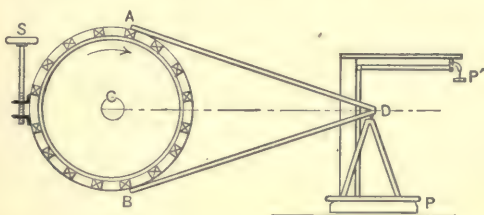


FIG. 1-28. — A Prony Brake.

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.

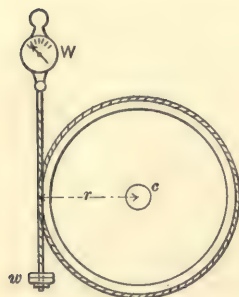


FIG. 1-29.—The Rope Brake.

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  $c d$

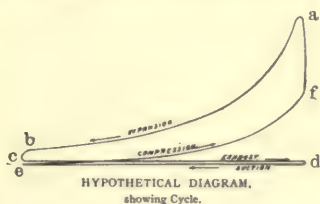


FIG. 1-30. — Four-cycle Engine.

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-

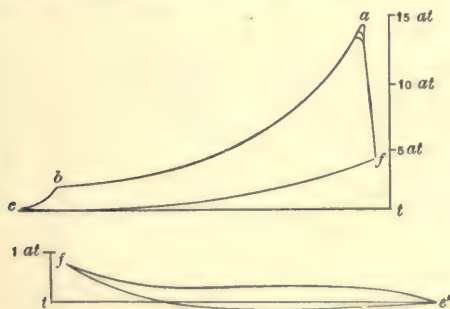


FIG. 1-31. — Two-cycle Engine Diagram.

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.

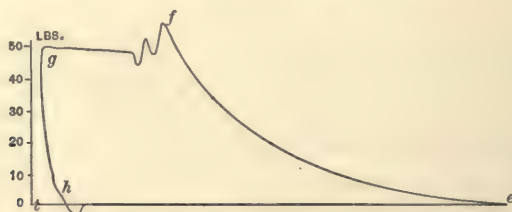


FIG. 1-33. — Diagram from Brayton Compressor.

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

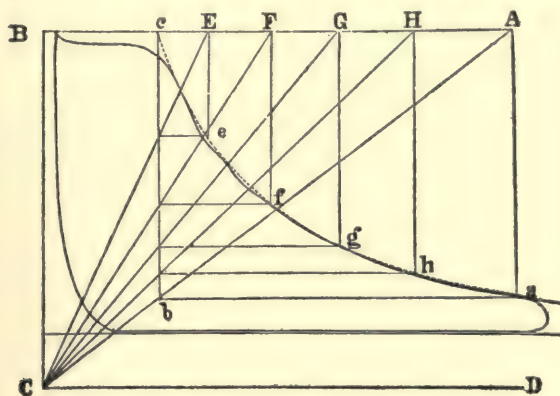


FIG. 1-35. — Method of Drawing Hyperbola.

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,

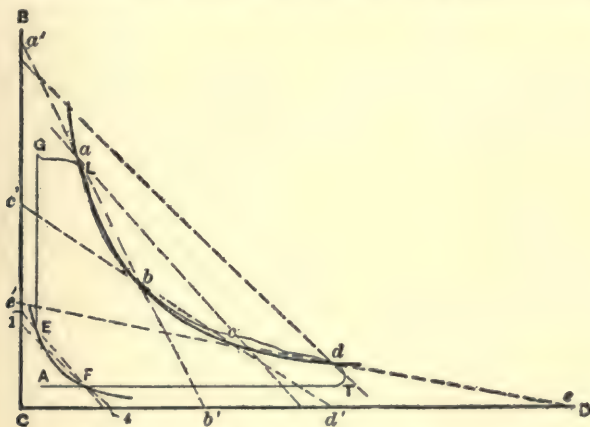


FIG. 1-36. — Method of Drawing an Hyperbola.

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.
- $\alpha$  = the coefficient of expansion of a perfect gas = the reciprocal of  $a$ .
- $\gamma$  = 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. $45^\circ$      |                              |                         |
|----------------------------------------|------------------------------|-------------------------|
|                                        | Cubic meters per<br>Kilogram | Cubic feet<br>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 <sub>2</sub> ) ..... | 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 <sub>2</sub> ) ..... | 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              |                          | $\frac{C_p}{C_v}$<br>$\gamma$ |
|-------------------------|--------------------------------|----------------------------|--------------------------|-------------------------------|
|                         |                                | Constant Pressure<br>$C_p$ | Constant Volume<br>$C_v$ |                               |
| 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 <sub>2</sub>                | 0.2000                     | 0.1550                   | 1.261                         |
| Steam .....             | H <sub>2</sub> O (saturated)   | 0.4805                     | 0.3700                   | 1.298                         |
| Methane .....           | CH <sub>4</sub>                | 0.5930                     | 0.4680                   |                               |
| Acetylene .....         | C <sub>2</sub> H <sub>2</sub>  | 0.3460                     | 0.2700                   |                               |
| Disulphide carbon ..... | CS <sub>2</sub>                | 0.1569                     | 0.1310                   | 1.198                         |
| Olefiant gas .....      | C <sub>2</sub> H <sub>4</sub>  | 0.4040                     | 0.3330                   | 1.125                         |
| Ammonia .....           | NH <sub>3</sub>                | 0.5084                     | 0.3910                   | 1.300                         |
| Alcohol .....           | CH <sub>2</sub> O <sub>6</sub> | 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, CO<sub>2</sub>, H<sub>2</sub>O, 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:

$$\begin{aligned} dQ &= \frac{\delta U}{\delta v} dv + \frac{\delta U}{\delta p} dp + A p dv \\ &\text{or} \\ dQ &= \frac{\delta U}{\delta p} dp + \left( \frac{\delta U}{\delta v} + A p \right) dv \end{aligned} \quad (7)$$

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),  $p v = R T$ , hence  $v dp = R dt$  when  $v$  is constant, and  $p dv = R dt$  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

$$p dv + v dp = R dt, \text{ from which } v dp = R dt - p dv$$

Substituting this in eq. (11) we may write

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

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  $\gamma$  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,  $\phi$ , 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,  $\phi$ .

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

$$\begin{aligned} dQ &= T d\phi \\ d\phi &= \frac{dQ}{T} \\ \phi - \phi_1 &= \int \frac{dQ}{T} \end{aligned}$$

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  $\gamma$  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.

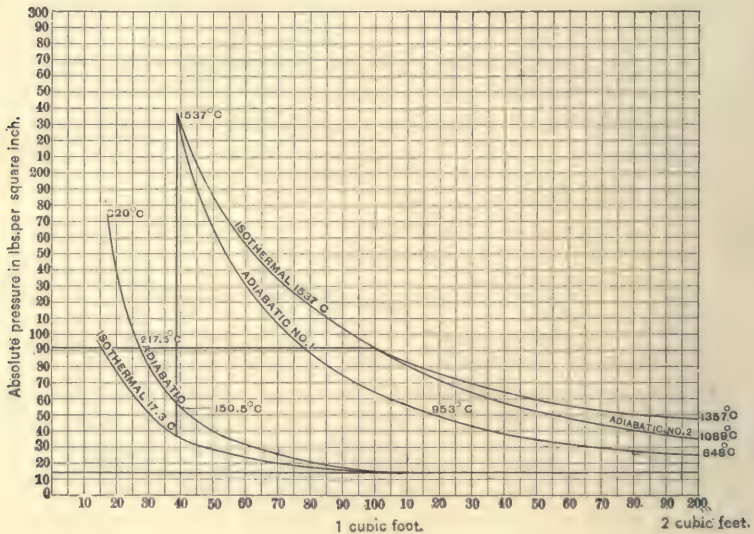


FIG. 2-1.—Relations of Isothermal and Adiabatic Curves.

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.

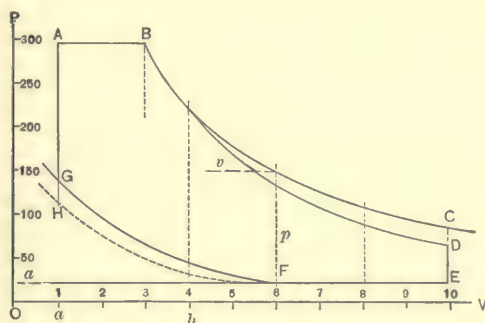


FIG. 2-2.— Isothermal and Adiabatic Changes Compared.

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  $ABCDEFGH$  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.

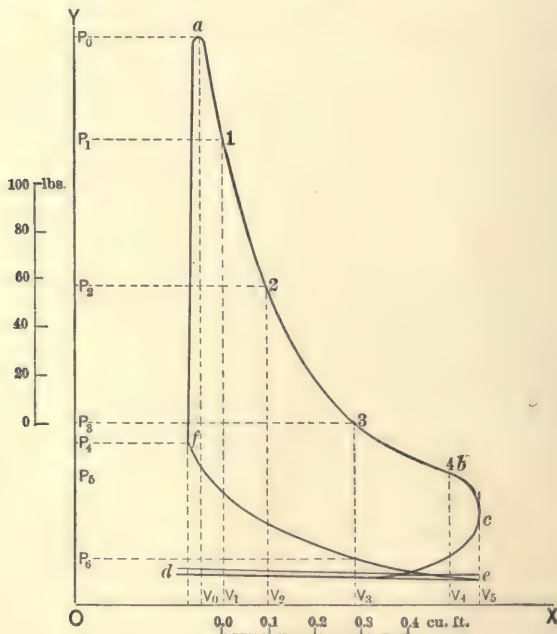


FIG. 2-3. — Four-cycle Engine Diagram.

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  $af$  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.

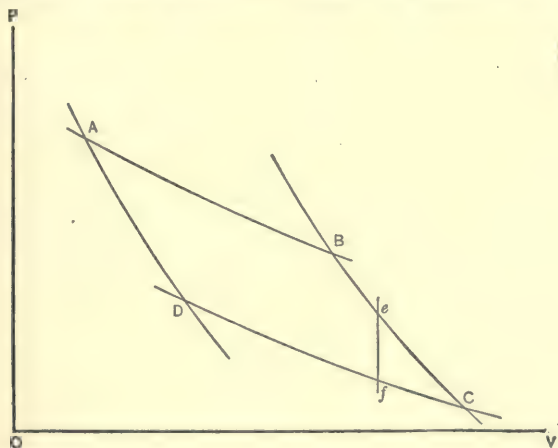


FIG. 2-4. — Pressure Volume Diagram of Isothermal and Adiabatic Changes.

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.

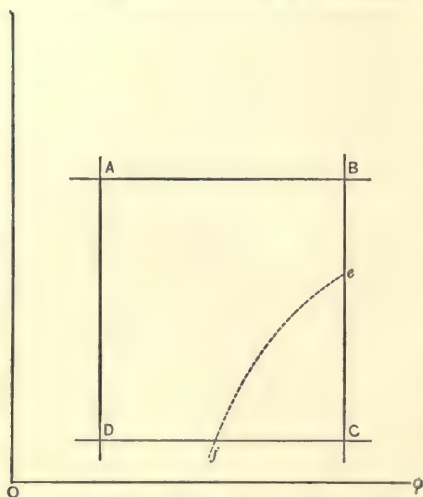


FIG. 2-5. — Entropy-Temperature Diagram. The area of this diagram shows the heat transferred into work. For the case considered this is a rectangle and its area represents the maximum amount of heat available for work within the temperature limits  $AB$  and  $DC$ .

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.

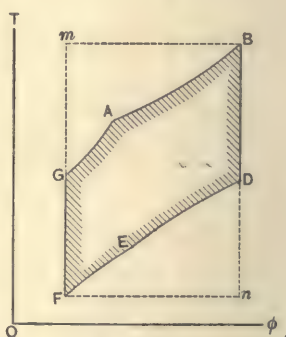


FIG. 2-6.

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

volume diagram we have isothermal expansion  $BC$  and isothermal compression  $FH$ . The entropy-temperature diagram for this case is represented by the diagram, not drawn to scale, in Fig. 2-7,  $ABCEFH$ , which by inspection is smaller and shows less heat available for work than the diagram  $mCnH$ , 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.

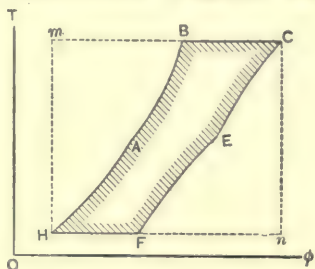


FIG. 2-7.

#### 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.

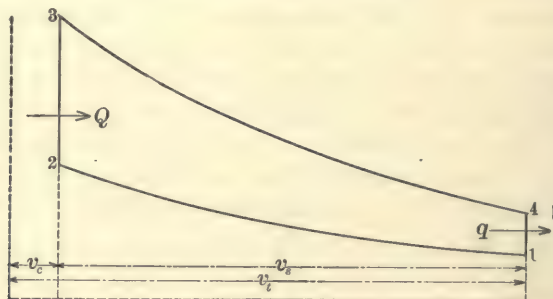


FIG 3-1.

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

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

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

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

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

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  $\gamma$  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  $\gamma$ , 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  $\gamma$  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,

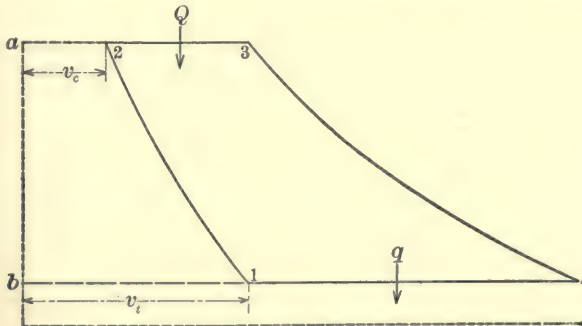


FIG. 3-2.

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

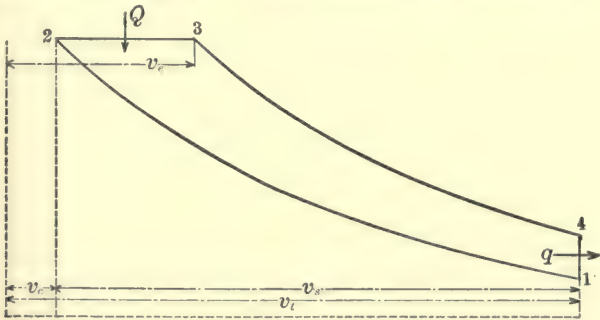


FIG. 3-3.

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,  $\delta$  = 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  $\gamma$ , but also upon  $\delta$ , 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  $\delta$  about  $= 2.5$ . To show the influence of the factor  $\delta$  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  $\delta$ , 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  $\delta q_1$ , supplied at a constant temperature  $T_1$ , and the heat element  $\delta q_2$  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,

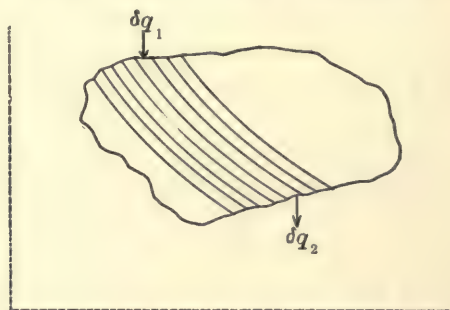


FIG. 3-4.

for each of which we may with very small error assume that the heat element  $\delta q_1$  is supplied at a constant temperature  $T_1$ , and the heat element  $\delta q_2$  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  $\delta q_1$  should be supplied to the working fluid at the highest temperature possible, and each element  $\delta q_2$ , 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  $\delta q_1$  must be supplied isothermally, or that the sum total of the elements  $\delta q_2$  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.

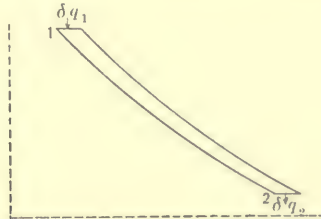


FIG. 3-5.

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  $\delta q$ , 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,

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

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.

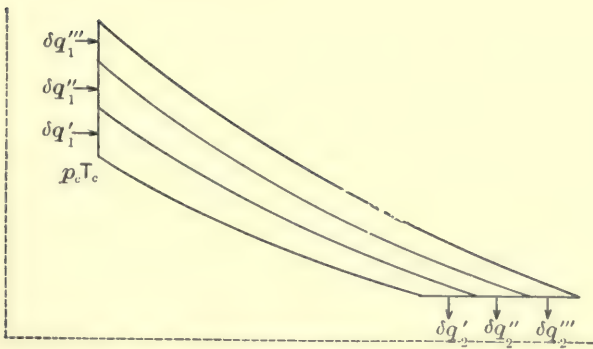


FIG. 3-6.

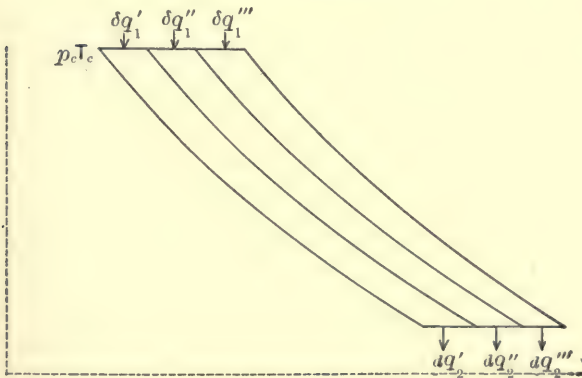


FIG. 3-7.

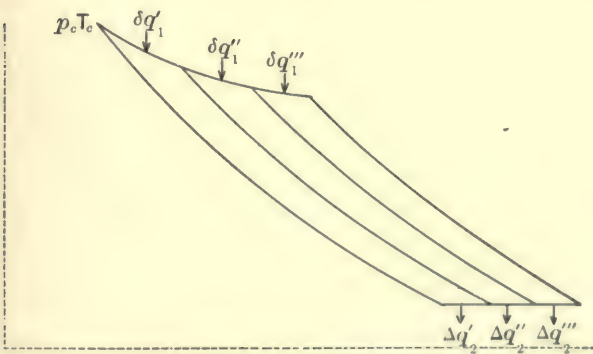


FIG. 3-8.

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

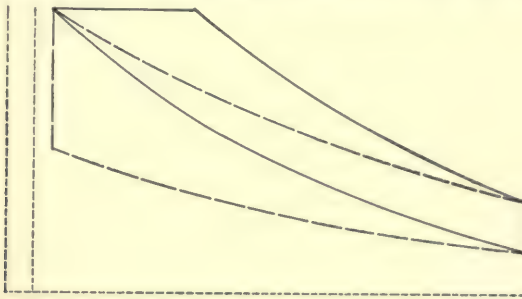


FIG. 3-9.

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.

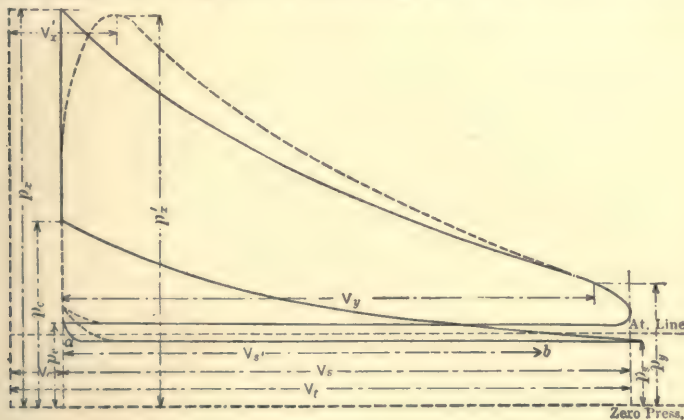


FIG. 4-1.

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  $v_r$ . Hence the volume  $v_r - v_s$  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:

|                                                                             | $E_v$     | $p_s$<br>lbs. per sq. inch<br>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

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

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 $p_s = 12.5$ ,<br>$p_c =$<br>$\left\{ \begin{array}{c} 600 \\ 650 \\ 700 \\ 750 \\ 800 \end{array} \right.$<br>when $T_s =$ | 29.7 | 39.2 | 49.3 | 59.9 | 70.7 | 81.8  | 93.2  | 117.5 | 142.0 | 168.0 | 195.0 | 222.0 |
|                            |                                                                                                                                 | 690  | 720  | 747  | 770  | 792  | 811   | 828   | 858   | 885   | 910   | 930   | 951   |
|                            |                                                                                                                                 | 748  | 780  | 811  | 835  | 858  | 877   | 896   | 930   | 958   | 983   | 1010  | 1028  |
|                            |                                                                                                                                 | 805  | 840  | 875  | 900  | 924  | 945   | 965   | 1002  | 1032  | 1062  | 1088  | 1110  |
|                            |                                                                                                                                 | 863  | 900  | 937  | 956  | 990  | 1013  | 1036  | 1073  | 1107  | 1137  | 1163  | 1188  |
|                            |                                                                                                                                 | 921  | 960  | 1000 | 1027 | 1056 | 1080  | 1105  | 1144  | 1180  | 1212  | 1240  | 1267  |
| $n = 1.30$                 | For $p_s = 12.5$ ,<br>$p_c =$<br>$\left\{ \begin{array}{c} 600 \\ 650 \\ 700 \\ 750 \\ 800 \end{array} \right.$<br>when $T_s =$ | 30.7 | 41.2 | 52.0 | 63.7 | 75.7 | 88.3  | 101.2 | 128.2 | 156.7 | 186.5 | 217.5 | 250.0 |
|                            |                                                                                                                                 | 703  | 740  | 772  | 800  | 825  | 847   | 868   | 905   | 938   | 966   | 1000  | 1018  |
|                            |                                                                                                                                 | 763  | 803  | 837  | 866  | 895  | 917   | 942   | 980   | 1015  | 1043  | 1076  | 1102  |
|                            |                                                                                                                                 | 821  | 864  | 900  | 933  | 964  | 987   | 1012  | 1055  | 1093  | 1128  | 1160  | 1187  |
|                            |                                                                                                                                 | 880  | 925  | 965  | 1000 | 1030 | 1058  | 1087  | 1130  | 1170  | 1210  | 1242  | 1272  |
|                            |                                                                                                                                 | 937  | 987  | 1028 | 1065 | 1100 | 1130  | 1157  | 1206  | 1250  | 1290  | 1322  | 1355  |
| $n = 1.35$                 | For $p_s = 12.5$ ,<br>$p_c =$<br>$\left\{ \begin{array}{c} 600 \\ 650 \\ 700 \\ 750 \\ 800 \end{array} \right.$<br>when $T_s =$ | 31.9 | 43.0 | 55.1 | 67.8 | 81.2 | 95.2  | 109.5 | 140.5 | 162.4 | 206.5 | 242.2 | 300   |
|                            |                                                                                                                                 | 718  | 760  | 797  | 832  | 860  | 885   | 912   | 955   | 995   | 1030  | 1063  | 1092  |
|                            |                                                                                                                                 | 778  | 825  | 864  | 900  | 932  | 963   | 987   | 1035  | 1078  | 1115  | 1152  | 1182  |
|                            |                                                                                                                                 | 838  | 888  | 932  | 970  | 1002 | 1035  | 1064  | 1114  | 1157  | 1200  | 1240  | 1275  |
|                            |                                                                                                                                 | 897  | 952  | 997  | 1038 | 1075 | 1105  | 1139  | 1193  | 1242  | 1284  | 1328  | 1364  |
|                            |                                                                                                                                 | 958  | 1012 | 1062 | 1108 | 1146 | 1180  | 1214  | 1264  | 1323  | 1372  | 1418  | 1455  |
| $n = 1.41$                 | For $p_s = 12.5$ ,<br>$p_c =$<br>$\left\{ \begin{array}{c} 600 \\ 650 \\ 700 \\ 750 \\ 800 \end{array} \right.$<br>when $T_s =$ | 33.1 | 45.5 | 58.8 | 73.1 | 88.3 | 103.0 | 120.6 | 156.5 | 194.0 | 234.0 | 277.0 | 321.0 |
|                            |                                                                                                                                 | 735  | 783  | 826  | 864  | 897  | 927   | 956   | 1013  | 1055  | 1095  | 1135  | 1170  |
|                            |                                                                                                                                 | 795  | 847  | 893  | 936  | 972  | 1003  | 1033  | 1097  | 1140  | 1187  | 1228  | 1268  |
|                            |                                                                                                                                 | 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<br>Lbs. by gage | % Clearance in<br>terms of Piston<br>Displacement |
|---------------------------------------------------------------------------------------------------------|--------------------------------------|---------------------------------------------------|
| Gasoline:                                                                                               |                                      |                                                   |
| Auto-engines with carbureters, cooling not very effective, high speeds.....                             | 45 - 95<br>Ave., 65                  | 35                                                |
| Gasoline:                                                                                               |                                      |                                                   |
| Stationary, slow-speed engines, more effective cooling but usually less simple combustion chamber ..... | 60 - 85<br>Ave., 70                  | 32                                                |
| Kerosene:                                                                                               |                                      |                                                   |
| Hot bulb injection and ignition .....                                                                   | 30 - 75                              | 35 - 40                                           |
| Kerosene:                                                                                               |                                      |                                                   |
| Previously vaporized in devices not requiring a vacuum .....                                            | 45 - 85<br>Ave., 65                  | 35                                                |
| Natural Gas .....                                                                                       | 75 - 130                             |                                                   |
| Natural Gas:                                                                                            |                                      |                                                   |
| Average for large and medium engines...                                                                 | Ave., 115                            | 22                                                |
| Illuminating gas .....                                                                                  | 60 - 100<br>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<br>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-