GAS · GASOLINE AND OIL ENGINES

DESIGN CONSTRUCTION MANAGEMENT

STATIONARY MOTOR VEHICLE
MARINE AERIAL

HISCOX-PAGÉ

A COMPLETE SELF EDUCATOR
5 EDITION, REVISED - ENLARGED

This, the twenty-first edition of Gas, Gasoline and Oil-Engines, by Gardner D. Hiscox, revised and enlarged by Victor W. Pagé, is a net book, and should be sold by the book trade at the retail price of $2.50.

Gas, Gasoline and Oil-Engines

A COMPLETE, PRACTICAL WORK

DEFINING CLEARLY THE ELEMENTS OF INTERNAL COMBUSTION ENGINEERING. TREATING EXHAUSTIVELY ON THE DESIGN, CONSTRUCTION AND PRACTICAL APPLICATION OF ALL FORMS OF GAS, GASOLINE, KEROSENE AND CRUDE PETROLEUM-OIL ENGINES. DESCRIBES MINUTELY ALL AUXILIARY SYSTEMS, SUCH AS LUBRICATION, CARBURETION AND IGNITION. CONSIDERS THE THEORY AND MANAGEMENT OF ALL FORMS OF EXPLOSIVE MOTORS FOR STATIONARY AND MARINE WORK, AUTOMOBILES, AEROPLANES AND MOTORCYCLES; INCLUDES ALSO

PRODUCER GAS AND ITS PRODUCTION

By GARDNER D. HISCOX, M.E.
Author of "Mechanical Movements," "Compressed Air," Etc.

REVISED, ENLARGED AND BROUGHT UP TO DATE

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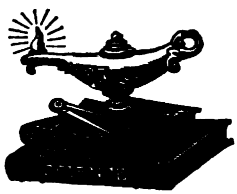

TWENTY-FIRST EDITION

Invaluable Instructions for all Students, Gas-Engine Owners, Gas-Engineers, Patent Experts, Designers, Mechanics, Draftsmen and all having to do with this modern power. Illustrated by 435 engravings, many especially made from engineering drawings, all in correct proportion.

AN INCOMPARABLE TREATISE AND WORK OF REFERENCE

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1915

COMPOSITION, ELECTROTYPING AND PRESSWORK
BY MACGOWAN & SLIPPER, NEW YORK, U. S. A.

PREFACE TO THE TWENTY-FIRST EDITION

A BOOK representing and illustrating the details of design, manufacture, and management of a new and progressive prime-moving power, falls behind its time by age, and, therefore, needs rearrangement and additions to bring its text and illustrations up to date in all the departments of such progressive industry. There is probably no more important mechanical development involving the production of motive power for all purposes within the age of steam than that of the explosive motor and its far-reaching effect in the promotion of industry by supplying a cheap, reliable, easily understood source of power.

So quickly has this new power expanded to almost universal usefulness as a labor-saving element in the lesser industries, that the literature of the past is found lacking in its up-to-date needs. Progress and improvement are the drift of genius in this advanced age. The advancement evidenced in adapting crude petroleum as fuel for explosive power, together with the rapid development of the producer-gas industry, have given a new economy in the production of power, while the use of the hitherto neglected gaseous elements of the blast-furnace and coke manufacture have added new sources of power production at a nominal cost.

Discussions of the producer, blast-furnace, and coke-oven gases, which are now coming to the front on a large scale for economic power, are included in this work, while crude petroleum and its conversion into power is also described and illustrated at length. It has a growing usefulness as the cheapest power fuel where the erection of gas-plants is not convenient. The insurance interests have formulated rules and regulations for the safe installation of gasoline-motors and producer-gas plants, which are given a place in this edition as a much needed matter of reference.

In the revised 1915 edition of this standard work every effort has been made to include striking examples of all recent developments in the field of internal combustion engineering. These include the new types of automobile, aircraft and motorcycle power plants, farm and gas tractor motors, marine engines and their use. Considerable descriptive matter is included relative to large ship motors and representative types of Diesel engines. The section on marine motors has been enlarged, and entirely new matter has been added on ignition, carburetion, lubrication, and motor testing. New tables, formulæ, and data pertinent to the subject have been secured and many new illustrations made. A portion of the matter relating to old type engines not manufactured at the present time has been retained, not only because of its historical value, but also on account of the number of such motors still in daily use.

VICTOR W. PAGÉ.

October, 1914.

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Gas, Gasoline and Oil Engines

including

Producer Gas and Its Production

CHAPTER I

INTRODUCTION — HISTORICAL — TYPES OF ENGINES — OPERATING PRINCIPLES

MUCH attention is now being given by mechanical engineers to the economical results that may be developed in the working of gas, gasoline, and oil-engines for higher powers from producer and other cheap gases and from petroleum and its products. In an economical sense, for small and intermediate power, steam has been left far behind. It now becomes a question as to how to adapt the design of the new prime mover to a wider range of usefulness and economy.

The best condensing steam-engines now made run with a consumption of about one and one-quarter pounds of coal per horse-power hour; while from two and one-half to seven pounds is the cost per horse-power hour in the various kinds of non-condensing engines now in use. This only covers the cost of fuel; the attendance required in the use of small steam-power is often far greater in cost than the fuel. When we come to require the larger powers by steam, in which economy may be obtained by compounding and condensing, the facility for obtaining the requisite water-supply is often a bar to its use. The direction in which lies the line of improvement for larger powers with the utmost economy, is as yet a moot point of discussion in engineering construction, as to steam or explosive-motor power.

The expansion of single-cylinder dimensions for explosive motors involves practical problems in the progress of ignition of the charge, as well as the thoroughness of mixture of the combustibles; the interference of the products of the previous combustion by producing areas of imperfect mixture or stratification, as discussed in the earlier publications, and which are not yet fully solved; but good progress has been made in this line. The enlargement of cylinder-area is a source of engine-friction economy, while, on the contrary, the multiplication of cylinders involves numbers and complexity of moving parts, which go to show a disparity between the indicated and brake horse-power, which is the measure of machine efficiency.

An impulse at every stroke, so desirable in an explosive motor and so satisfactorily carried out in the steam-engine, seems to have as yet but a limited counterpart in the explosive motor, as trials of motors with explosion at every stroke have not yet proved entirely satisfactory in service, although double-acting motors are in use, and in order to accomplish fully the desired result, resort has been had to the duplication of single-acting cylinders. This class of explosive-motors seems to fill the bill in effect; yet the complication of a two-cylinder engine as a moving mechanism must compete with a single-cylinder steam-engine.

The principal types of explosive motors seem to have gone through a series of practical trials during the past forty years, which have finally reduced the principles of action to a few permanent forms in the design of motors that have shown by their long-continued use the prospect of their staying qualities and efficiency.

For a gas, gasoline, or oil-explosive power to approximate an ideal standard as a prime mover, it should be simple in design and not liable to get out of order; the parts must be readily accessible, the ignition of the charge must be positive and controllable, the governing close; the motor must run quietly, and must be durable and economical in the use of fuel. These points of excellence have been striven for by many designers and builders, with varying success; but to get the entire combination without the sacrifice of some good point is not an easy matter.

But for all, the internal-combustion engine has come seemingly like an avalanche of a decade; but it has come to stay, to take its well-deserved position among the powers for aiding labor. Its ready adaptation to road and marine service has made it a wonder of the age in the development of speed not before dreamed of as a possibility; yet in so short a time, its power for speed has taken rank on the common road against the locomotive on the rail with its century's progress. It has made aerial navigation possible and practical, it furnishes power for all marine craft from the light canoe to the transatlantic liner. It operates the machine tools of the mechanic, tills the soil for the farmer and provides healthful recreation for thousands by furnishing an economical means of transport by land and sea. It has been a universal mechanical education for the masses, and in its present forms represents the great refinement and development made possible by the concentration of the world's master minds on the problems incidental to internal combustion engineering.

HISTORICAL.

Although the ideal principle of explosive power was conceived some two hundred years ago, at which time experiments were made with gunpowder as the explosive element, it was not until the last years of the eighteenth century that the idea took a patentable shape, and not until about 1826 (Brown's gas-vacuum engine) that a further progress was made in England by condensing the products of combustion by a jet of water, thus creating a partial vacuum.

Brown's was probably the first explosive engine that did real work. It was clumsy and unwieldy and was soon relegated to its place among the failures of previous experiments. No approach to active explosive effect in a cylinder was reached in practice, although many ingenious designs were described, until about 1838 and the following years. Barnett's engine in England was the first attempt to compress the charge before exploding. From this time on to about 1860 many patents were issued in Europe and a few in the United States for gas-engines, but the progress was slow, and its practical introduction for power came with spasmodic effect and low efficiency. From 1860 on, practical improvement seems to have been made, and the Lenoir motor was produced in France and brought to the United States. It failed to meet expectations, and was soon followed by further improvements in the Hugon motor in France (1862), followed by Beau de Rocha's four-cycle idea, which has been slowly developed through a long series of experimental trials by different inventors. In the hands of Otto and Langdon a further progress was made, and numerous patents were issued in England, France, and Germany, and followed up by an increasing interest in the United States, with a few patents.

From 1870 improvements seem to have advanced at a steady rate, and largely in the valve-gear and precision of governing for variable load. The early idea of the necessity of slow combustion was a great drawback in the advancement of efficiency, and the suggestion of de Rocha in 1862 did not take root as a prophetic truth until many failures and years of experience had taught the fundamental axiom that rapidity of action in both combustion and expansion was the basis of success in explosive motors.

With this truth and the demand for small and safe prime movers, the manufacture of gas-engines increased in Europe and America at a more rapid rate, and improvements in perfecting the details of this cheap and efficient prime mover have finally raised it to the dignity of a standard motor and a dangerous rival of the steam-engine for small and intermediate powers, with a prospect of largely increasing its individual units to many hundred, if not to the thousand horse-power in a single cylinder. The unit size in a single cylinder has now reached to about 700 horse-power and by combining cylinders in the same machine, powers of from 1,500 to 2,000 horse-power are now available for large power-plants.

The application of the gasoline and oil-motor to marine propulsion, to the horseless vehicle, the automobile, cyclecar, and bicycle, has had a most stimulating effect in adapting ways and means for applying this power to so many uses. For launches and as auxiliary power for yachts and larger sailing vessels, the explosive motor has overreached its steam competitor for economy and convenience and is now the leading power for the smaller craft; even aerial navigation has come in for its share in motor-power for air-ships.

Although the denser population of Europe claims a very large representation of explosive motors in use for all purposes, the manufacture in the United States is fast forging ahead in its output of this cheap power, for there are now thousands of establishments engaged in their manufacture, and the motors in operation number many hundred thousands. Their safety and easy management as well as their economy have made in their adoption as agricultural helpers a marvelous inroad on the old-fashioned hand and horse-powers and are now reaching a new and prominent place as a ready means of power for pumping water for the farm and for irrigation, and for driving threshing machines and wood-saws; the operation of mowers and reapers, are some of its late innovations.

Its adaptability as a power for generating electricity for all purposes, is fast expanding the use of lighting and power in fields that the higher cost of small steam-power has precluded, and is now in its newer phases, due to the use of the cheap producer-gas fuel, extending its usefulness to the largest electrical plants.

The expiration of patents in Europe and the United States has now cast loose many of the bonds that have in a measure retarded the freedom of manufacture in the explosive-motor line, so that the fundamental principles of construction are no longer a hindrance to anyone desiring to build a motor without infringing on patents in force.

MAIN TYPES OF GAS, GASOLINE, AND OIL-ENGINES

This form of prime mover has been built in so many different types, all of which have operated with some degree of success that the diversity in form will not be generally appreciated unless some attempt is made to classify the various designs that have received practical application. Obviously the same type of engine is not universally applicable, because each class of work has individual peculiarities which can best be met by an engine designed with the peculiar conditions present in view. The following tabular synopsis will enable the reader to judge the extent of the development of what is now the most popular prime mover for all purposes.

A. Internal Combustion (Standard Type)

1. Single Acting (Standard Type)
2. Double Acting (For Large Power Only)
3. Simple (Universal Form)
4. Compound (Rarely Used)
5. Reciprocating Piston (Standard Type)
6. Turbine (Revolving Rotor, not fully developed)

A1. Two-Stroke Cycle

a. Two Port
b. Three Port
c. Combined Two and Three Port
d. Fourth Port Accelerator
e. Differential Piston Type
f. Distributor Valve System

A2. Four-Stroke Cycle

a. Automatic Inlet Valve

b. Mechanical Inlet Valve
c. Poppet or Mushroom Valve
d. Slide Valve
  d 1. Sleeve Valve
  d 2. Reciprocating Ring Valve
  d 3. Piston Valve

e. Rotary Valves
  e 1. Disc
  e 2. Cylinder or Barrel
  e 3. Single Cone
  e 4. Double Cone

f. Two Piston (Balanced Explosion)
g. Rotary Cylinder, Fixed Crank (Aerial)
h. Fixed Cylinder, Rotary Crank (Standard Type)

A3. Six-Stroke Cycle

B. External Combustion (Practically Obsolete)
  a. Turbine, Revolving Rotor
  b. Reciprocating Piston

CLASSIFICATION BY CYLINDER ARRANGEMENT

Single Cylinder
  a. Vertical
  b. Horizontal
  c. Inverted Vertical

Double Cylinder
  a. Vertical
  b. Horizontal (Side by Side)
  c. Horizontal (Opposed)
  d. 45 to 90 Degrees V (Angularly Disposed)
  e. Horizontal Tandem (Double Acting)

Three Cylinder
  a. Vertical
  b. Horizontal
  c. Rotary (Cylinders Spaced at 120 Degrees)

d. Radially Placed (Stationary Cylinders)
e. One Vertical, One Each Side at an Angle
f. Compound (Two High Pressure, One Low Pressure)

Four Cylinder
a. Vertical
b. Horizontal (Side by Side)
c. Horizontal (Two Pairs Opposed)
d. 45 to 90 Degrees V
e. Twin Tandem (Double Acting)

Five Cylinder
a. Vertical (Five Throw Crankshaft)
b. Radially Spaced at 72 Degrees (Stationary)
c. Radially Placed Above Crankshaft (Stationary)
d. Placed Around Rotary Crankcase (72 Degrees Spacing)

Six Cylinder
a. Vertical
b. Horizontal (Three Pairs Opposed)
c. 45 to 90 Degrees V

Seven Cylinder
a. Equally Spaced (Rotary)

Eight Cylinder
a. Vertical
b. Horizontal (Four Pairs Opposed)
c. 45 to 90 Degrees V

Twelve Cylinder
a. Vertical
b. Horizontal (Six Pairs Opposed)
c. 45 to 90 Degrees V

Fourteen Cylinder
a. Rotary

Sixteen Cylinder
a. 45 to 90 Degrees V
b. Horizontal (Eight Pairs Opposed)

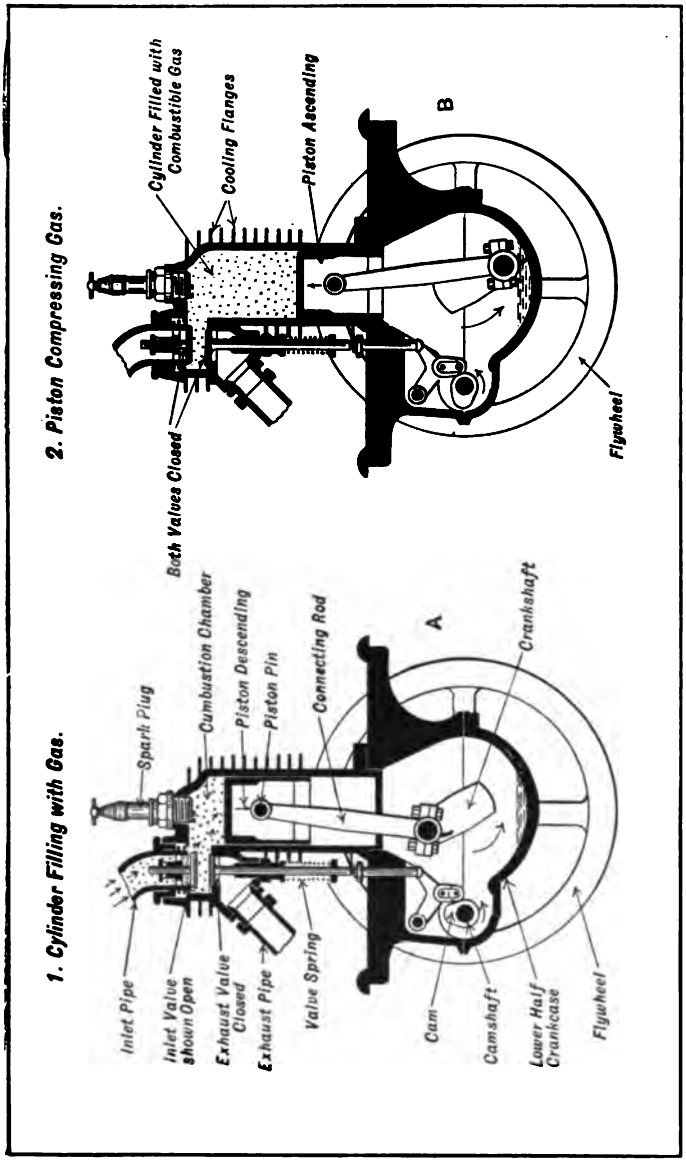


FIG. 1.—Views of simple four-cycle engine, showing methods of charging cylinder with explosive gas and compressing it prior to ignition.

Of all the types enumerated above engines having less than eight cylinders are the most popular. The four-cylinder vertical is without doubt the most widely used of all types owing to the large number employed as automobile and marine power plants as well as for stationary power in the larger sizes. Stationary engines in small and medium powers are invariably of the single or double form. Three-cylinder engines are seldom used at the present time, except in marine work and in some stationary forms. Eight and twelve-cylinder motors have received but limited application and practically always in racing automobiles, motor-boats or in air-craft. The only example of a fourteen-cylinder motor to be used to any extent is incorporated in aeroplane construction. This is also true of the sixteen-cylinder forms. All of the various types enumerated will be fully described in proper sequence and the advantages of the different constructions considered in detail under the proper chapter headings.

OPERATING PRINCIPLES OF TWO AND FOUR-STROKE CYCLE ENGINES

Before discussing the construction of the various forms of internal combustion engines it may be well to describe the operating cycle of the types most generally used. The two-cycle engine is the simplest because there are no valves in connection with the cylinder as the gas is introduced into that member and expelled from it through ports cored into the cylinder walls. These are covered by the piston at a certain portion of its travel and uncovered at other parts of its stroke. In the four-cycle engine the explosive gas is admitted to the cylinder through a port at the head end closed by a valve, while the exhaust gas is expelled through another port controlled in a similar manner. These valves are operated by mechanism distinct from the piston.

The action of the four-cycle type may be easily understood if one refers to illustrations at Figs. 1 and 2. It is called the "four-stroke engine" because the piston must make four strokes in the cylinder for each explosion or power impulse obtained. The principle of the gas-engine of the internal combustion type is similar to that of a gun, i.e., power is obtained by the rapid combustion of some explosive or other quick burning substance. The bullet is driven out of the gun barrel by the pressure of the gas evolved when the charge of powder is ignited. The piston or

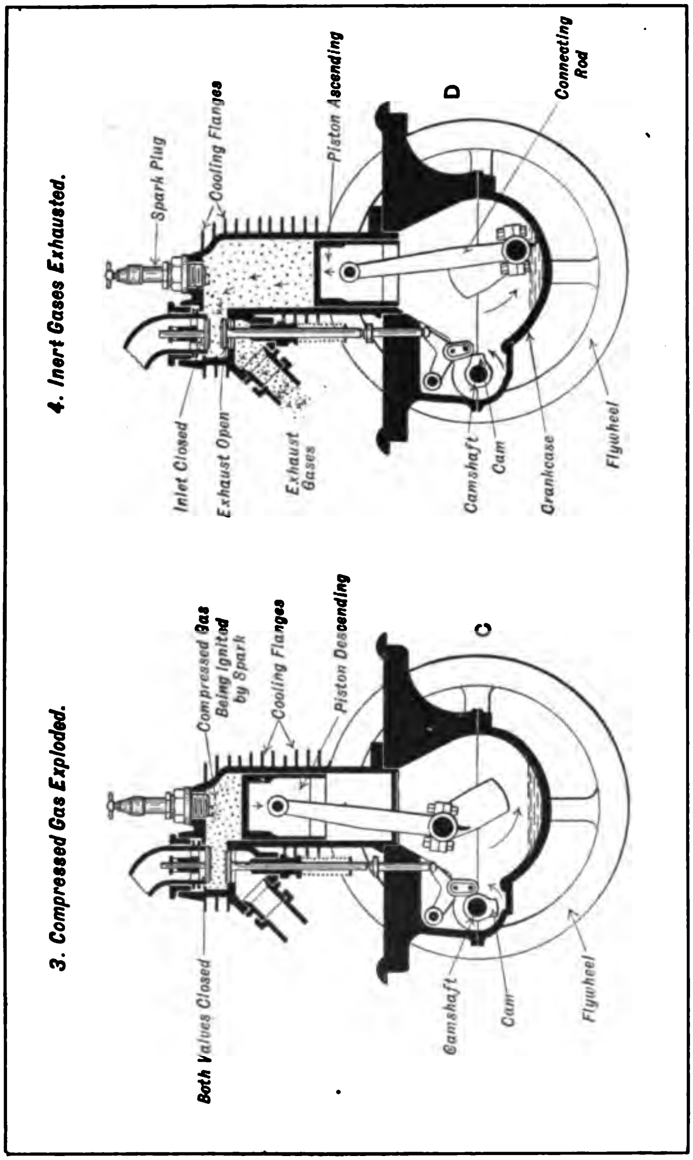


FIG. 2.—Views of simple four-cycle engine, showing effect of explosion and how burnt gases are exhausted from cylinder.

movable element of the gas-engine is driven from the closed or head end to the crank end of the cylinder by a similar expansion of gases resulting from combustion. The first operation in firing a gun or securing an explosion in the cylinder of the gas-engine is to fill the combustion space with combustible material. This is done by a down stroke of the piston during which time the inlet valve opens to admit the gaseous charge to the cylinder interior. This operation is shown at Fig. 1A. The second operation is to compress this gas which is done by an upward stroke of the piston as shown at Fig. 1B. When the top of the compression stroke is reached, the gas is ignited and the piston is driven down toward the open end of the cylinder, as indicated at Fig. 2C. The fourth operation or exhaust stroke is performed by the return upward movement of the piston as shown at Fig. 2D during which time the exhaust valve is opened to permit the burnt gases to leave the cylinder. As soon as the piston reaches the top of its exhaust stroke, the energy stored in the flywheel rim during the power stroke causes that member to continue revolving and as the piston again travels on its down stroke the inlet valve opens and admits a charge of fresh gas and the cycle of operations is repeated.

The two-cycle engine works on a different principle, as while only the combustion chamber end of the piston is employed to do useful work in the four-cycle engine, both upper and lower portions are called upon to perform the functions necessary to two-cycle engine operation. Instead of the gas being admitted into the cylinder as is the case with the four-stroke engine, it is first drawn into the engine base where it receives a preliminary compression prior to its transfer to the working end of the cylinder. The views at Fig. 3 should indicate clearly the operation of the two-port, two-cycle engine. At A the piston is seen reaching the top of its stroke and the gas above the piston is being compressed ready for ignition, while the suction in the engine base causes the automatic valve to open and admits mixture from the carburetor to the crank case. When the piston reaches the top of its stroke, the compressed gas is ignited and the piston is driven down on the power stroke, compressing the gas in the engine base.

When the top of the piston uncovers the exhaust port the flaming gas escapes because of its pressure. A downward movement of the piston uncovers the inlet port opposite the exhaust

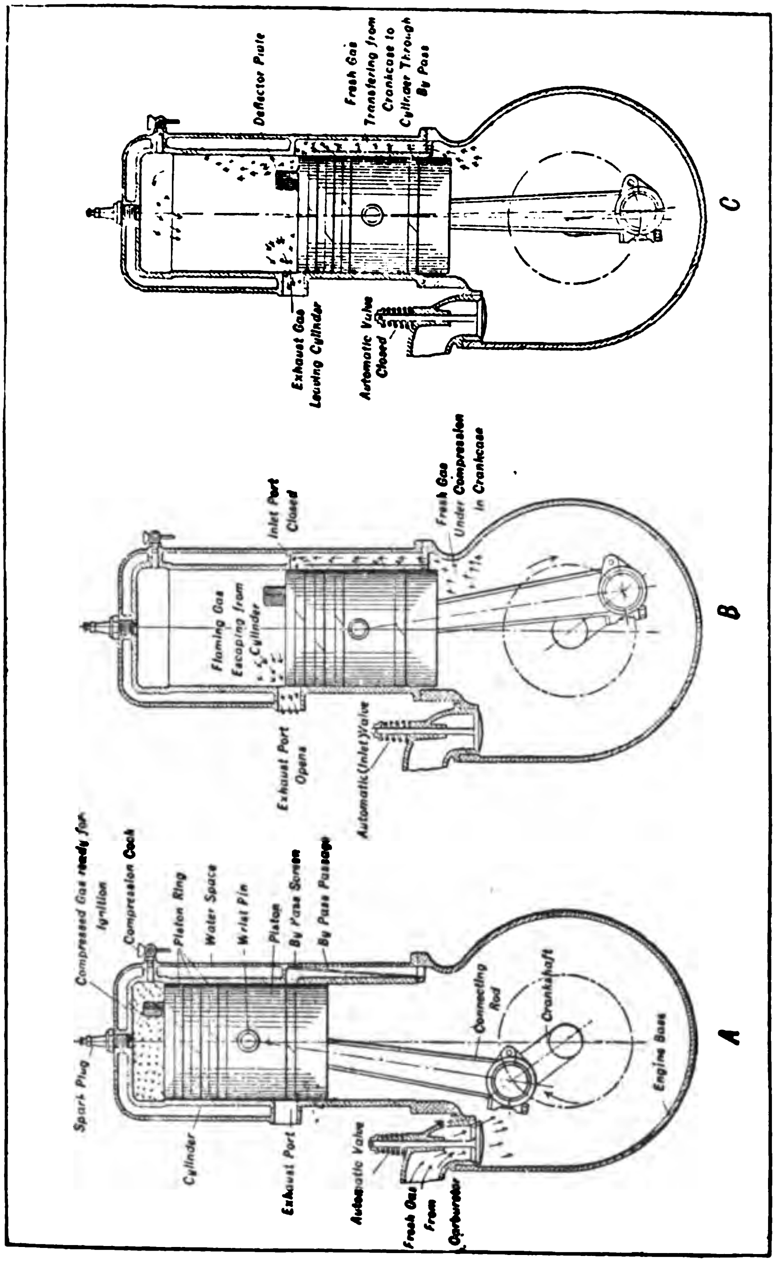


FIG. 3.—Showing two-port, two-cycle engine operation.

and permits the fresh gas to bypass through the transfer passage from the engine base to the cylinder. The conditions with the intake and exhaust port fully opened are clearly shown at Fig. 3C. The deflector plate on the top of the piston directs the entering fresh gas to the top of the cylinder and prevents the main portion of the gas stream from flowing out through the open exhaust port. On the next upstroke of the piston the gas in the cylinder is compressed and the inlet valve opened, as shown at A to permit a fresh charge to enter the engine base.

The operating principle of the three-port, two-cycle engine is practically the same as that previously described with the exception that the gas is admitted to the crank case through a third port in the cylinder wall, which is uncovered by the piston when that member reaches the end of its up stroke. The action of the three-port form can be readily ascertained by studying the diagrams given at Fig. 4. Combination two- and three-port engines have been evolved and other modifications made to improve the action. These forms will be fully described in the chapter on marine motors.

THE TWO-CYCLE AND FOUR-CYCLE TYPES

In the earlier years of explosive-motor progress, was evolved the two types of motors in regard to the cycles of their operation. The early attempts to perfect the two-cycle principle were for many years held in abeyance from the pressure of interests in the four-cycle type, until its simplicity and power possibilities were demonstrated by Mr. Dugald Clerk in England, who gave the principles of the two-cycle motor a broad bearing leading to immediate improvements in design, which has made further progress in the United States, until at the present time it has an equal standard value as a motor-power in some applications as its ancient rival the four-cycle or Otto type, as demonstrated by Beau de Rocha in 1862.

Thermodynamically, the methods of the two types are equal as far as combustion is concerned, and compression may favor in a small degree the four-cycle type as well as the purity of the charge. The cylinder volume of the two-cycle motor is much smaller per unit of power, and the enveloping cylinder surface is therefore greater per unit of volume. Hence more heat is carried off by the jacket water during compression, and the higher compression avail-

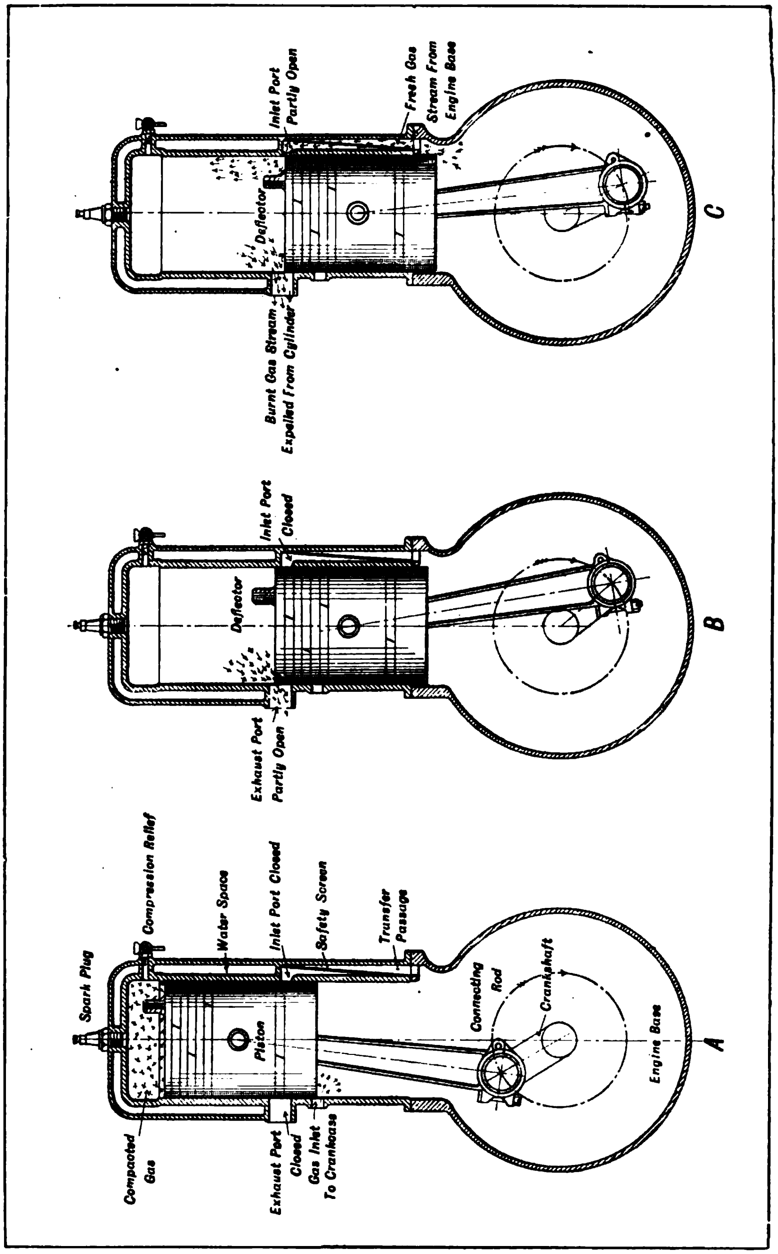


FIG. 4.—Defining three-port, two-cycle engine action.

able from this tends to increase the economy during compression which is lost during expansion.

In the two-cycle motor a scavenging may be obtained to a small extent under the conditions of a crank-chamber pressure charge, while in a four-cycle motor the charge is made by the suction stroke of the main piston and at less than atmospheric pressure, and no scavenging can be made possible except by the momentum of the exhaust in a long exhaust-pipe, which is not always available. The result of these conditions is that the two-cycle type has a denser charge and a gain in power per unit of volume.

From the above considerations it may be safely stated that a *lower* temperature and higher pressure of charge at the beginning of compression is obtained in the two-cycle motor, greater weight of charge and greater specific power of higher compression resulting in higher thermal efficiency. The smaller cylinder for the same power of the two-cycle motor gives less friction surface per impulse than of the other type; although the crank-chamber pressure may, in a measure, balance the friction of the four-cycle type. Probably the strongest points in favor of the two-cycle type are the lighter fly-wheel and the absence of valves and valve gear, making this type the most simple in construction and the lightest in weight for its developed power. Yet, for the larger power units, the four-cycle type will no doubt always maintain the standard for efficiency and durability of action.

The distribution of the charge and its degree of mixture with the remains of the previous explosion in the clearance space, has been a matter of discussion for both types of explosive motors, with doubtful results. In Fig. 5A we illustrate what theory suggests as to the distribution of the fresh charge in a two-cycle motor, and in Fig. 5B what is the probable distribution of the mixture when the piston starts on its compressive stroke. The arrows show the probable direction of flow of the fresh charge and burnt gases at the crucial moment.

In Fig. 5C is shown the complete out-sweep of the products of combustion for the full extent of the piston stroke of a four-cycle motor, leaving only the volume of the clearance to mix with the new charge and at D the manner by which the new charge sweeps by the ignition device, keeping it cool and avoiding possibilities of pre-ignition by undue heating of the terminals of the sparking

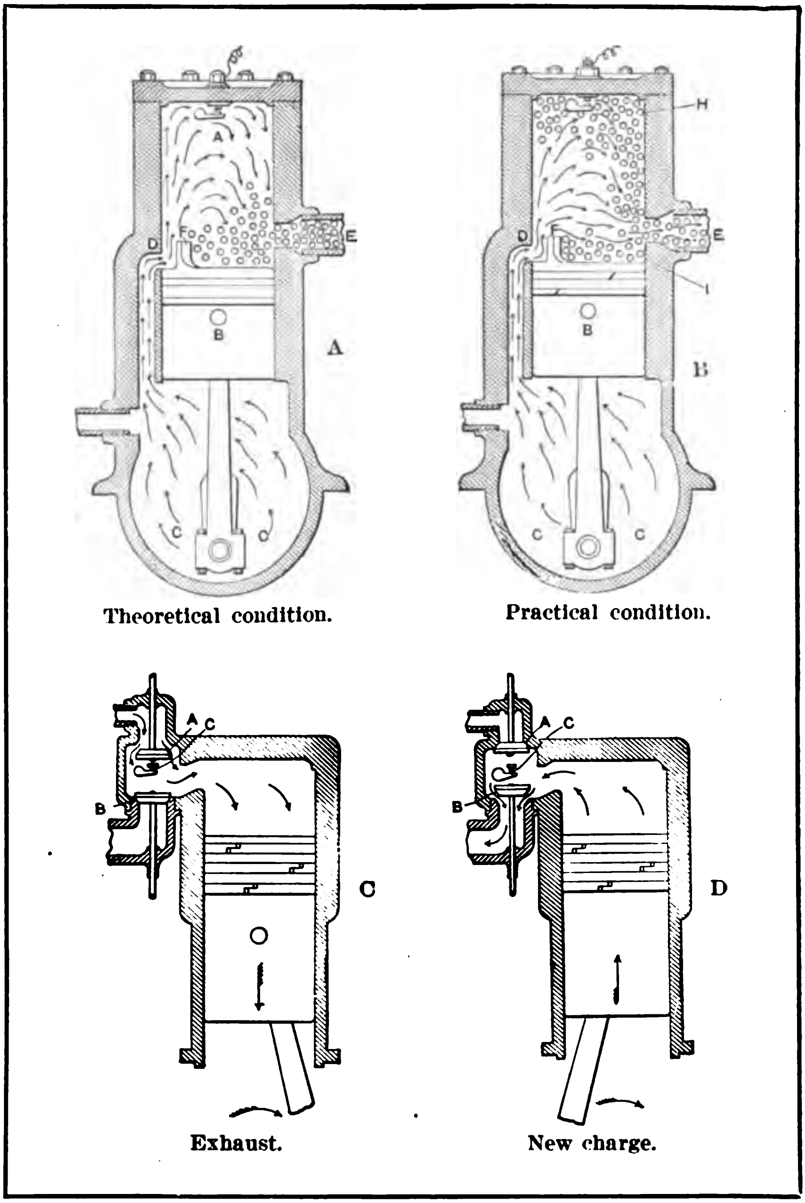


FIG. 5.—Diagrams contrasting action of two- and four-cycle cylinders on exhaust and intake stroke.

device. Thus, by enveloping the sparking device with the pure mixture, ignition spreads through the charge with its greatest possible velocity, a most desirable condition in high-speed motors with side-valve chambers and igniters within the valve chamber. An igniter in the cylinder-head in this design would be one of the sources of unseen trouble from uncertain ignition.

CHAPTER II

THEORY OF THE GAS AND GASOLINE-ENGINE

The laws controlling the elements that create a power by their expansion by heat due to combustion, when properly understood, become a matter of computation in regard to their value as an agent for generating power in the various kinds of explosive engines. The method of heating the elements of power in explosive engines greatly widens the limits of temperature as available in other types of heat-engines. It disposes of many of the practical troubles of hot-air, and even of steam-engines, in the simplicity and directness of application of the elements of power. In the explosive engine the difficulty of conveying heat for producing expansive effect by convection is displaced by the generation of the required heat within the expansive element and at the instant of its useful work. The low conductivity of heat to and from air has been the great obstacle in the practical development of the hot-air engine; while, on the contrary, it has become the source of economy and practicability in the development of the internal-combustion engine.

The action of air, gas, and the vapors of gasoline and petroleum oil, whether singly or mixed, is affected by changes of temperature practically in nearly the same ratio; but when the elements that produce combustion are interchanged in confined spaces, there is a marked difference of effect. The oxygen of the air, the hydrogen and carbon of a gas, or vapor of gasoline or petroleum oil are the elements that by combustion produce heat to expand the nitrogen of the air and the watery vapor produced by the union of the oxygen in the air and the hydrogen in the gas, as well as also the monoxide and carbonic-acid gas that may be formed by the union of the carbon of gas or vapor with part of the oxygen of the air. The various mixtures as between air and gas, or air and vapor, with the proportion of the products of combustion left in the cylinder from a previous combustion, form the elements to be considered in estimating the amount of pressure that may be obtained by their combustion and expansive force.

The working process of the explosive motor may be divided into three principal types: 1. Motors with charges igniting at constant volume without compression, such as the Lenoir, Hugon, and other similar types now abandoned as wasteful in fuel and effect. 2. Motors with charges igniting at constant pressure with compression, in which a receiver is charged by a pump and the gases burned while being admitted to the motor cylinder, such as types of the Simon and Brayton engine. 3. Motors with charges igniting at constant volume with variable compression, such as the later two- and four-cycle motors with compression of the indrawn charge; limited in the two-cycle type and variable in the four-cycle type with the ratios of the clearance space in the cylinder. This principle produces the explosive motor of greatest efficiency.

The phenomena of the brilliant light and its accompanying heat at the moment of explosion have been witnessed in the experiments of Dugald Clerk in England, the illumination lasting throughout the stroke; but in regard to time in a four-cycle engine, the incandescent state exists only one-quarter of the running time. Thus the time interval, together with the non-conductibility of the gases, makes the phenomena of a high-temperature combustion within the comparatively cool walls of a cylinder a practical possibility.

THE ISOTHERMAL LAW

The natural laws, long since promulgated by Boyle, Gay Lussac, and others, on the subject of the expansion and compression of gases by force and by heat, and their variable pressures and temperatures when confined, are conceded to be practically true and applicable to all gases, whether single, mixed, or combined.

The law formulated by Boyle only relates to the compression and expansion of gases without a change of temperature, and is stated in these words:

If the temperature of a gas be kept constant, its pressure or elastic force will vary inversely as the volume it occupies.

It is expressed in the formula $P \times V = C$, or pressure $\times$ volume $=$ constant. Hence, $\frac{C}{P} = V$ and $\frac{C}{V} = P$.

Thus the curve formed by increments of pressure during the expansion or compression of a given volume of gas without change of temperature is designated as the isothermal curve in which the volume multiplied by the pressure is a constant value in expansion, and inversely the pressure divided by the volume is a constant value in compressing a gas.

But as compression and expansion of gases require force for their accomplishment mechanically, or by the application or abstraction of heat chemically, or by convection, a second condition becomes involved, which was formulated into a law of thermodynamics by Gay Lussac under the following conditions: A given volume of gas under a free piston expands by heat and contracts by the loss of heat, its volume causing a proportional movement of a free piston equal to $\frac{1}{273}$ part of the cylinder volume for each degree Centigrade difference in temperature, or $\frac{1}{492}$ part of its volume for each degree Fahrenheit. With a fixed piston (constant volume), the pressure is increased or decreased by an increase or decrease of heat in the same proportion of $\frac{1}{273}$ part of its pressure for each degree Centigrade, or $\frac{1}{492}$ part of its pressure for each degree Fahrenheit change in temperature. This is the natural sequence of the law of mechanical equivalent, which is a necessary deduction from the principle that nothing in nature can be lost or wasted, for all the heat that is imparted to or abstracted from a gaseous body must be accounted for, either as heat or its equivalent transformed into some other form of energy. In the case of a piston moving in a cylinder by the expansive force of heat in a gaseous body, all the heat expended in expansion of the gas is turned into work; the balance must be accounted for in absorption by the cylinder or radiation.

This theory is equally applicable to the cooling of gases by abstraction of heat or by cooling due to expansion by the motion of a piston. The denominators of these heat fractions of expansion or contraction represent the absolute zero of cold below the freezing-point of water, and read — 273° C. or — 492.66° = — 460.66° F. below zero; and these are the starting-points of reference in computing the heat expansion in gas-engines. According to Boyle's law, called the first law of gases, there are but two characteristics of a gas and their variations to be considered, *viz.*, volume and pressure: while by the law of Gay Lussac, called the second law of gases, a third is added, consisting of the value of the absolute temperature, counting from absolute zero to the temperatures at which the operations take place. This is the *Adiabatic* law.

The ratio of the variation of the three conditions — volume, pressure, and heat — from the absolute zero temperature has a certain rate, in which the volume multiplied by the pressure and the product divided by the absolute temperature equals the ratio of expansion for each degree. If a volume of air is contained in a cylinder having a piston and fitted with an indicator, the piston, if moved to and fro slowly, will alternately compress and expand the air, and the indicator pencil will trace a line or lines upon the card, which lines register the change of pressure and volume occurring in the cylinder. If the piston is perfectly free from leakage, and it be supposed that the temperature of the air is kept quite constant, then the line so traced is called an *Isothermal line*, and the pressure at any point when multiplied by the volume is a constant according to Boyle's law,

$$pv = \text{a constant.}$$

If, however, the piston is moved very rapidly, the air will not remain at constant temperature, but the temperature will increase because work has been done upon the air, and the heat has no time to escape by conduction. If no heat whatever is lost by any cause, the line will be traced over and over again by the indicator pencil, the cooling by expansion doing work precisely equalling the heating by compression. This is the line of no transmission of heat, therefore, known as *Adiabatic*.

The expansion of a gas $\frac{1}{273}$ of its volume for every degree Centigrade, added to its temperature, is equal to the decimal .00366, the coefficient of expansion for Centigrade units. To any given volume of a gas, its expansion may be computed by multiplying the coefficient by the number of degrees, and by reversing the process the degree of acquired heat may be obtained approximately. These methods are not strictly in conformity with the absolute mathematical formula, because there is a small increase in the increment of expansion of a dry gas, and there is also a slight difference in the increment of expansion due to moisture in the atmosphere and to the vapor of water formed by the union of the hydrogen and oxygen in the combustion chamber of explosive engines.

The ratio of expansion on the Fahrenheit scale is derived from the absolute temperature below the freezing-point of water (32°) to correspond with the Centigrade scale; therefore $\frac{1}{492.66} = .0020297$, the ratio of expansion from 32° for each degree rise in temperature on the Fahrenheit scale. As an example, if the temperature of any volume of air or gas at constant volume is raised, say from 60° to 2000° F., the increase in temperature will be 1940°. The ratio will be $\frac{1}{520.66} = .0019206$. Then by the formula:

Ratio × acquired temp. × initial pressure = the gauge pressure; and $.0019206 \times 1940° \times 14.7 = 54.77$ lbs.

By another formula, a convenient ratio is obtained by $\frac{\text{absolute pressure}}{\text{absolute temp.}}$ or $\frac{14.7}{520.66} = .028233$; then, using the difference of temperature as before, $.028233 \times 1940° = 54.77$ lbs. pressure.

By another formula, leaving out a small increment due to specific heat at high temperatures:

$$\text{I. } \frac{\text{Atmospheric pressure} \times \text{absolute temp.} + \text{acquired temp.}}{\text{Absolute temp.} + \text{initial temp.}} =$$

absolute pressure due to the acquired temperature, from which the atmospheric pressure is deducted for the gauge pressure. Using the foregoing example, we have $\frac{14.7 \times 460.66° + 2000°}{460.66 + 60°}$ $= 69.47 - 14.7 = 54.77$, the gauge pressure, 460.66 being the absolute temperature for zero Fahrenheit.

For obtaining the volume of expansion of a gas from a given increment of heat, we have the approximate formula:

$$\text{II. } \frac{\text{Volume} \times \text{absolute temp.} + \text{acquired temp.}}{\text{Absolute temp.} + \text{initial temp.}} = \text{heated volume.}$$

In applying this formula to the foregoing example, the figures become:

$$\text{I.} \times \frac{460.66° + 2000°}{460.66 + 60°} = 4.72604 \text{ volumes.}$$

From this last term the gauge pressure may be obtained as follows:

III. $4.72604 \times 14.7 = 69.47$ lbs. absolute $-$ 14.7 lbs. atmospheric pressure = 54.77 lbs. gauge pressure; which is the theoretical pressure due to heating air in a confined space, or at constant volume from 60° to 2000° F.

By inversion of the heat formula for absolute pressure we have the formula for the acquired heat, derived from combustion at constant volume from atmospheric pressure to gauge pressure plus atmospheric pressure as derived from Example I., by which the expression

$$\frac{\text{absolute pressure} \times \text{absolute temp.} + \text{initial temp.}}{\text{initial absolute pressure}}$$

= absolute temperature + temperature of combustion, from which the acquired temperature is obtained by subtracting the absolute temperature.

Then, for Example, $\frac{69.47 \times 460.66 + 60}{14.7} = 2460.66$, and 2460.66 — 460.66 = 2000°, the theoretical heat of combustion. The dropping of terminal decimals makes a small decimal difference in the result in the different formulas.

HEAT AND ITS WORK

By Joule's law of the mechanical equivalent of heat, whenever heat is imparted to an elastic body, as air or gas, energy is generated and mechanical work produced by the expansion of the air or gas. When the heat is imparted by combustion within a cylinder containing a movable piston, the mechanical work becomes an amount measurable by the observed pressure and movement of the piston. The heat generated by the explosive elements and the expansion of the non-combining elements of nitrogen and water vapor that may have been injected into the cylinder as moisture in the air, and the water vapor formed by the union of the oxygen of the air with the hydrogen of the gas, all add to the energy of the work from their expansion by the heat of internal combustion. As against this, the absorption of heat by the walls of the cylinder, the piston, and cylinder-head or clearance walls, becomes a modifying condition in the force imparted to the moving piston.

It is found that when any explosive mixture of air and gas or hydrocarbon vapor is fired, the pressure falls far short of the pressure computed from the theoretical effect of the heat produced, and from gauging the expansion of the contents of a cylinder. It is now well known that in practice the high efficiency which is promised by theoretical calculation is never realized; but it must always be remembered that the heat of combustion is the real agent, and that the gases and vapors are but the medium for the conversion of inert elements of power into the activity of energy by their chemical union. The theory of combustion has been the leading

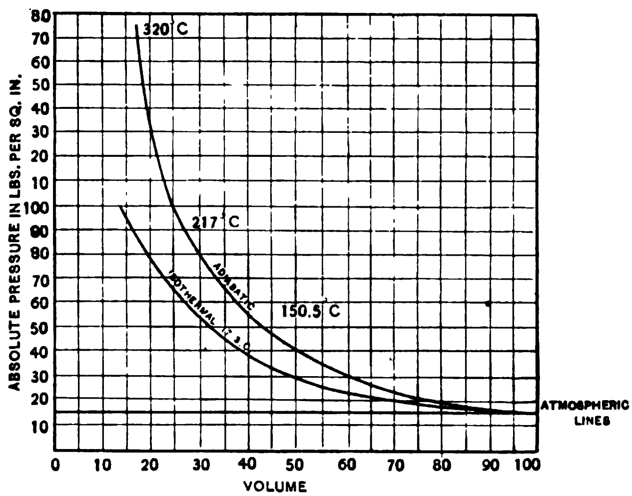


FIG. 6.—Diagram isothermal and adiabatic lines.

stimulus to large expectations with inventors and constructors of explosive motors; its entanglement with the modifying elements in practice has delayed the best development in construction, and as yet no positive design of best form or action seems to have been accomplished; although great progress has been made during the past decade in the development of speed, economy, and the size of the individual units of this new power.

One of the most serious difficulties in the practical development of pressure, due to the theoretical computations of the pressure value of the full heat, is probably caused by imparting the heat of the fresh charge to the balance of the previous charge that has been cooled by expansion from the maximum pressure to near the atmospheric pressure of the exhaust. The retardation in the velocity of combustion of perfectly mixed elements is now well known from experimental trials with measured quantities; but the principal difficulty in applying these conditions to the practical work of an explosive engine where a necessity for a large clearance space cannot be obviated, is in the inability to obtain a maximum effect from the imperfect mixture and the mingling of the products of the last explosion with the new mixture, which produces a clouded condition that makes the ignition of the mass irregular or chattering, as observed in the expansion lines of indicator cards; but this must not be confounded with the reaction of the spring in the indicator.

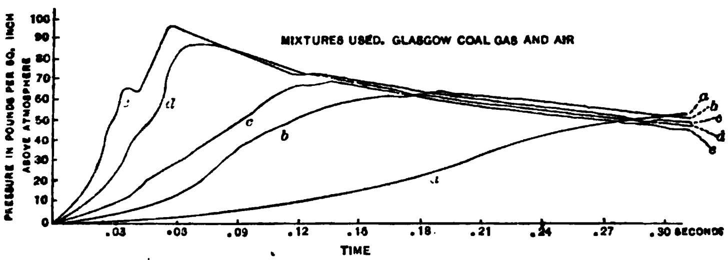


FIG. 7.—Diagram of moments of combustion in a closed chamber constant volume.

Stratification of the mixture has been claimed as taking place in the clearance chamber of the cylinder; but this is not satisfactory, in view of the vortical effect of the violent injection of the air and gas or vapor mixture. It certainly cannot become a perfect mixture in the time of a stroke of a high-speed motor of the two-cycle class. In a four-cycle engine, making 300 revolutions per minute, the injection and compression take place in one-fifth of a second — formerly considered far too short a time for a perfect infusion of the elements of combustion.

In an experimental way, the velocity of explosion of a perfect mixture of 2 volumes of hydrogen and 1 volume of oxygen has been found to approximate 65 feet per second; and for equal volumes of hydrogen and oxygen, 32 feet per second; with 1 volume coal-gas to 5 volumes air, 3¼ feet per second; 1 volume coal-gas to 6 volumes of air, 1 foot per secnod; and with an increasing proportion of air, 10 to 9 inches per second. These velocities were obtained in tubes fired at one end only. When the ignition was made in a closed tube, so that compression was produced by the expansion from combustion, the velocity was largely increased; and with compressed mixtures a great increase of velocity was obtained over the above-stated figures, as has been proved in motors running at 2000 revolutions per minute.

The different values of time, pressure, and computed heat of combustion are shown in Table I., and graphically compared in the diagram (Fig. 7).

The mixtures were Glasgow, Scotland, coal-gas and air. The table and the diagram (Fig. 7) make an excellent study of the conditions of time and pressure, as well as also of the control of the work of a gas-engine, by varying the proportion of the mixture.

TABLE I.—EXPLOSION AT CONSTANT VOLUME IN A CLOSED CHAMBER.

Diagram curve Fig. 7.	Mixture injected.	Time of explosion. Second.	Gauge pressure. Pounds per square inch.	Computed temperature, Fahr.
a	1 volume gas to 13 volumes air.	0.28	52	1,916°
b	1 " " " 11 " "	0.18	63	2,309°
c	1 " " " 9 " "	0.13	69	2,523°
d	1 " " " 7 " "	0.07	89	3,236°
e	1 " " " 5 " "	0.05	96	3,484°

The irregularity of the explosive curves in the diagram is fair evidence of imperfect diffusion of the gas and air mixture at the moment of combustion, assuming that the indicator was in perfect action.

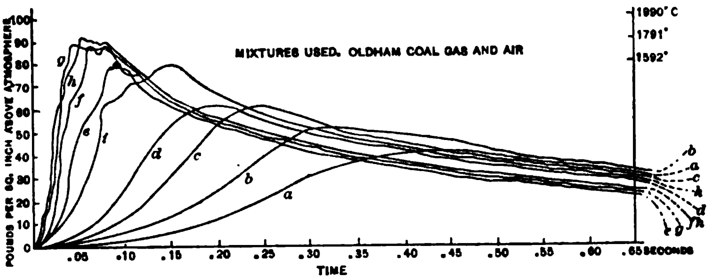


FIG. 8.—Diagram of moments of combustion in a closed chamber constant volume.

Experiments with mixtures of coal-gas and air (Fig. 8), made at Oldham, England, show a slight variation of effect, which is probably due to different proportions of hydrogen and carbon in the Oldham gas, with the same elements in the Glasgow gas. In Table II. the injection temperature is given, which in itself is not important further than as a basis for computing the theoretical temperature of combustion. A record of the hygrometric state of the atmosphere in its extremes would be valuable in showing the variation in explosive effect due to the vapor of water derived from the air under different hygrometric conditions.

TABLE II.—EXPLOSION AT CONSTANT VOLUME IN A CLOSED CHAMBER.

Diagram curve Fig. 8.	Mixture injected.	Temp. of injection Fahr.	Time of explosion. Second.	Observed gauge pressure. Pounds.	Computed temp. Fahr.
a	1 volume gas to 14 volumes air.	64°	0.45	40.	1,483°
b	1 " " " 13 " "	51°	0.31	51.5	1,859°
c	1 " " " 12 " "	51°	0.24	.60.	2,195°
d	1 " " " 11 " "	51°	0.17	61.	2,228°
e	1 " " " 9 " "	62°	0.08	78.	2,835°
f	1 " " " 7 " "	62°	0.06	87.	3,151°
g	1 " " " 6 " "	51°	0.04	90.	3,257°
h	1 " " " 5 " "	51°	0.055	91.	3,293°
i	1 " " " 4 " "	66°	0.16	80.	2,871°

In an examination of the times of explosion and the corresponding pressures in both tables, it will be seen that a mixture of 1 part gas to 6 parts air is the most effective and will give the highest mean pressure in a gas-engine. In this diagram the undulations of the rising curves due to irregular firing of the mixture are well marked. There is a limit to the relative proportions of illuminating gas and air mixture that is explosive, somewhat variable, depending upon the proportion of hydrogen in the gas. With ordinary coal-gas, 1 of gas to 15 parts of air; and on the lower end of the scale, 1 volume of gas to 2 parts air, are non-explosive. With gasoline vapor the explosive effect ceases at 1 to 16, and a saturated mixture of equal volumes of vapor and air will not explode, while the most intense explosive effect is from a mixture of 1 part vapor to 9 parts air. In the use of gasoline and air mixtures from a carburetor, the best effect is from 1 part saturated air to 8 parts free air.

TABLE III.—PROPERTIES AND EXPLOSIVE TEMPERATURE OF A MIXTURE OF ONE PART OF ILLUMINATING GAS OF 660 THERMAL UNITS PER CUBIC FOOT WITH VARIOUS PROPORTIONS OF AIR WITHOUT MIXTURE OF CHARGE WITH THE PRODUCTS OF A PREVIOUS EXPLOSION.

Proportion, Air to Gas, by Volumes.	Pounds in One Cubic Foot of Mixture	Specific Heat. Heat Units Required to Raise 1 lb. 1 deg. Fahrenheit.		Heat to Raise One Cubic Foot of Mixture 1 deg. Fahr.	Heat Units Evolved by Combustion.	Ratio. Col. $\frac{6}{5}$	Usual Combustion Efficiency.	Usual Rise of Temperature due to Explosion at Constant Volume.
		Constant Pressure.	Constant Volume.					
6 to 1...	.074195	.2668	.1913	.014189	94.28	6644.6	.465	3090
7 to 1...	.075012	.2628	.1882	.014116	82.	5844.4	.518	3027
8 to 1...	.075647	.2598	.1858	.014059	73.33	5216.1	.543	2832
9 to 1...	.076155	.2575	.1846	.014013	66.	4709.9	.56	2637
10 to 1...	.076571	.2555	.1825	.013976	60.	4293.	.575	2468
11 to 1...	.076917	.2540	.1813	.013945	55.	3944.	.585	2307
12 to 1...	.077211	.2526	.1803	.013922	50.77	3646.7	.58	2115

The weight of a cubic foot of gas and air mixture as given in Col. 2 is found by adding the number of volumes of air multiplied by its weight, .0807, to one volume of gas of weight .035 pound per cubic foot and dividing by the total number of volumes; for example, as in the table $6 \times .0807 = \frac{.5192}{7} = .074195$ as in the first line, and so on for any mixture or for other gases of different specific weight per cubic foot. The heat units evolved by combustion of the mixture (Col. 6) are obtained by dividing the total heat units in a cubic foot of gas by the total proportion of the mixture, $\frac{660}{7} = 94.28$ as in the first line of the table. Col. 5 is obtained by multiplying the weight of a cubic foot of the mixture in Col. 2 by the specific heat at constant volume (Col. 4), $\frac{\text{Col. 6}}{\text{Col. 5}} =$ Col. 7 the total heat ratio, of which Col. 8 gives the usual combustion efficiency — Col. 7 $\times$ by Col. 8 gives the absolute rise in temperature of a pure mixture, as given in Col. 9.

The many recorded experiments made to solve the discrepancy between the theoretical and the actual heat development and resulting pressures in the cylinder of an explosive motor, to which much discussion has been given as to the possibilities of dissociation and the increased specific heat of the elements of combustion and non-combustion, as well, also, of absorption and radiation of heat, have as yet furnished no satisfactory conclusion as to what really takes place within the cylinder walls. There seems to be very little known about dissociation, and somewhat vague theories have been advanced to explain the phenomenon. The fact is, nevertheless, apparent as shown in the production of water and other producer gases by the use of steam in contact with highly incandescent fuel. It is known that a maximum explosive mixture of pure gases, as hydrogen and oxygen or carbonic oxide and oxygen, suffers a contraction of one-third their volume by combustion to their compounds, steam or carbonic acid. In the explosive mixtures in the cylinder of a motor, however, the combining elements form so small a proportion of the contents of the cylinder that the shrinkage of their volume amounts to no more than three per cent. of the cylinder volume. This by no means accounts for the great heat and pressure differences between the theoretical and actual effects.

CHAPTER III

THE UTILIZATION OF HEAT AND ITS EFFICIENCY IN EXPLOSIVE MOTORS

THE utilization of heat in any heat-engine has long been a theme of inquiry and experiment with scientists and engineers, for the purpose of obtaining the best practical conditions and construction of heat-engines that would represent the highest efficiency or the nearest approach to the theoretical value of heat, as measured by empirical laws that have been derived from experimental researches relating to its ultimate value. It is well known that the steam-engine returns only from 12 to 18 per cent. of the power due to the heat generated by the fuel, about 25 per cent. of the total heat being lost in the chimney, the only use of which is to create a draught for the fire; the balance, some 60 per cent., is lost in the exhaust and by radiation. The problem of utmost utilization of force in steam has nearly reached its limit.

The internal-combustion system of creating power is comparatively new in practice, and is but just settling into definite shape by repeated trials and modification of details, so as to give somewhat reliable data as to what may be expected from the rival of the steam-engine as a prime mover. For small powers, the gas, gasoline, and petroleum-oil engines are forging ahead at a rapid rate, filling the thousand wants of manufacture and business for a power that does not require expensive care, that is perfectly safe at all times, that can be used in any place in the wide world to which its concentrated fuel can be conveyed, and that has eliminated the constant handling of crude fuel and water.

The utilization of heat in a gas-engine is mainly due to the manner in which the products entering into combustion are distributed in relation to the movement of the piston. The investigation of the foremost exponent of the theory of the explosive motor was prophetic in consideration of the later realization of the best conditions under which these motors can be made to meet the requirements of economy and practicability. As early as 1862, Beau de Rocha announced, in regard to the coming power, that four requisites were the basis of operation for economy and best effect. 1. The greatest possible cylinder volume with the least possible cooling surface. 2. The greatest possible rapidity of expansion. Hence, *high speed.* 3. The greatest possible expansion. *Long stroke.* 4. The greatest possible pressure at the commencement of expansion. *High compression.*

In the two-cycle motors of the early or Lenoir type, the gas or vapor and air mixtures were drawn in during a part of the stroke, fired, expanded with the motion of the piston, and exhausted by the return stroke. The proportions of the indraught to the stroke of the piston, and the volume of the clearance or combustion chamber, as it is usually called, have been subject to a vast amount of experiment and practical trial, in an endeavor to bring the heat value of their power up to its highest possible limit.

The earlier engines of this class used as high as 96 cubic feet of illuminating gas per horse-power per hour. The consumption of gas fell off by improvements to 70 cubic feet, and finally dropped to 44 and 36 cubic feet per indicated horse-power per hour in the various modifications following the early trials, all of which have dropped out of use. The efficiency of this class of gas-engine seldom reached 20 per cent. of the heat value of the gas used, while in the compression types of two- and four-cycle motors there are possibilities of over 40 per cent. The total efficiency of the gas or vapor entering into combustion in an internal-heat engine is variable, depending upon its constituent-combining elements and the degree of temperature produced. The efficiency due to heat only varies as the difference between the initial temperature of the explosive mixture and the temperature of combustion; and as this varies in actual practice from 1400° to 2500° F., then the reciprocal of the absolute heat of the initial charge, divided by the assumed heat of combustion, would represent the total efficiency. The formula $\frac{H - H^1}{H}$ represents this condition, "in which H is the absolute heat of combustion, and H^1 is the absolute initial temperature," so that if the operation of the heat cycle was between 60° and 1400° F., the equation would be $\frac{60 + 460}{1400 + 460} = .279$ and

$1 - .279 = .72$ per cent. But this cannot represent a working cycle from the change in the specific heat of the gaseous contents of a cylinder while undergoing expansion by the movement of a piston.

The specific heat of air at constant volume is .1685, and at constant pressure is .2375. Their ratio $\frac{.2375}{.1685} = 1.408$. The ratios of the other elements entering into combustion in a gas-engine are slightly less than for air; but the ratio for air is near enough for all practical operations. The formula for the application of the condition of work with complete expansion is $1 - \left(1.408 \frac{H^1}{H}\right)$; or, as for above example, $1 - \left(1.408 \frac{60 + 460}{1400 + 460}\right) = .3928$, and $1 - .3928 = .6071$, or 60 per cent. As the temperature cannot be utilized for work from the excess of heat in the products of combustion when the expansion has reached the atmospheric line, then the practical amount of expansion and the heat of combustion at the point of exhaust must be considered. In practice, the measured heat of the exhaust at atmospheric pressure, plus the additional heat due to the terminal pressure, becomes a factor in the equation; and assuming this to be 950° F. in a well-regulated motor, the equation for the above example becomes $1 - \left(1.408 \times \frac{950 - 460}{1400 - 460}\right) = \frac{490}{940} = .521 \times 1.408 = .733$, and $1 - .733 = .26$, or an efficiency of 26 per cent. The greater difference in temperature, other things being equal, the greater the efficiency.

In this way efficiencies are worked out through intricate formulas for a variety of theoretical and unknown conditions of combustion in the cylinder: ratios of clearance and cylinder volume, and the uncertain condition of the products of combustion left from the last impulse and the wall temperature. But they are of but little value, except as a mathematical inquiry as to possibilities. The real commercial efficiency of a gas or gasoline-engine depends upon the volume of gas or liquid at some assigned cost, required per actual brake horse-power per hour, in which an indicator card should show that the mechanical action of the valve gear and ignition was as perfect as practicable, and that the ratio of clearance, space, and cylinder volume gave a satisfactory terminal pressure and compression: *i.e.*, the difference between the power figured from the indicator card and the brake power being the friction loss of the engine.

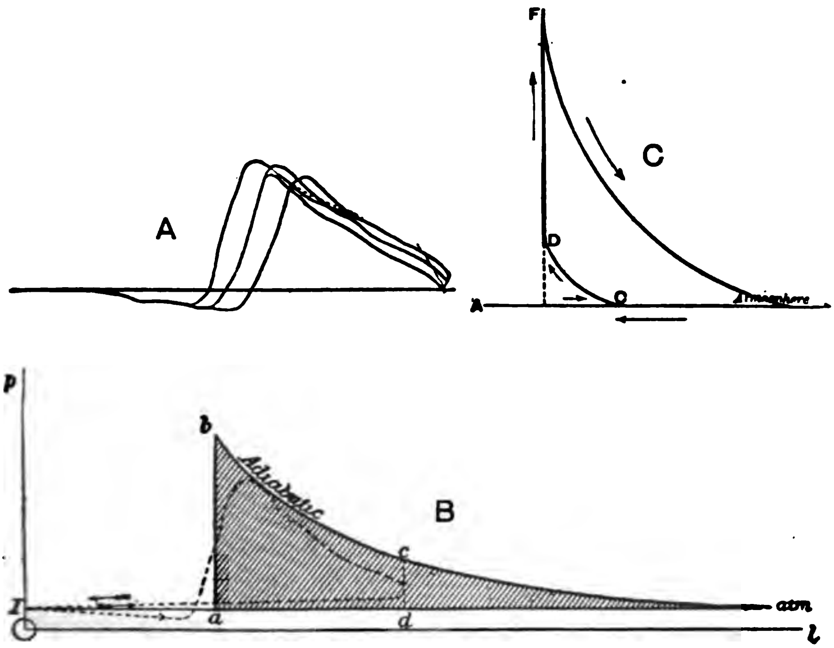


FIG. 9.—*A*. Lenoir type of indicator card. *B*. Comparative card, Lenoir and perfect expansion. *C*. Diagram of a perfect cycle with compression.

In practice, the heat value of the gas per cubic foot may vary from 30 per cent. with illuminating and natural gases to 75 or 80 per cent. as between good illuminating gas and producer gas; then, in order that a given size engine should maintain its rating, a larger volume of a poorer gas should be swept through the cylinder. This requires adjustment of the areas in all the valves to give an explosive motor its highest efficiency for the kind of fuel that is to be used.

The practical effect of the work done by the half-cycle in the earlier Lenoir type of the two-cycle engine is graphically shown in Fig. 9B, in which *I, d* represents the stroke of the piston; the dotted line, the indicator card; and the space in the lines, *a, b, c, d,* the ideal diagram of a perfect gas exhausting at the point *d,* in its incomplete adiabatic expansion. In the valuation of such a card, the depression of the indraught below the atmospheric line and the pressure of the exhaust line should have due consideration as negative quantities to be deducted from the pressure values above the atmospheric line. This class of engines is fast becoming obsolete as a type.

In two-cycle motors of the compression type and in four-cycle motors of the same type, the efficiencies are greatly advanced by compression, producing a more complete infusion of the mixture of gas or vapor and air, quicker firing, and far greater pressure than is possible with the two-cycle type just described. In the practical operation of the gas-engine during the past twenty years, the gas-consumption efficiencies per indicated horse-power have gradually risen from 17 per cent. to a maximum of 40 per cent. of the theoretical heat, and this has been done chiefly through a decreased combustion chamber and increased compression — the compression having gradually increased in practice from 30 lbs. per square inch to above 100; but there seems to be a limit to compression, as the efficiency ratio decreases with greater increase in compression. It has been shown that an ideal efficiency of 33 per cent. for 38 lbs. compression will increase to 40 per cent. for 66 lbs., and 43 per cent. for 88 lbs. compression. On the other hand, greater compression means greater explosive pressure and greater strain on the engine structure, which will probably retain in future practice the compression between the limits of 40 and 80 lbs.

In experiments made by Dugald Clerk, in England, with a combustion chamber equal to 0.6 of the space swept by the piston, with a compression of 38 lbs., the consumption of gas was 24 cubic feet per indicated horse-power per hour. With 0.4 compression space and 61 lbs. compression, the consumption of gas was 20 cubic feet per indicated horse-power per hour; and with 0.34 compression space and 87 lbs. compression, the consumption of gas fell to 14.8 cubic feet per indicated horse-power per hour — the actual efficiencies being respectively 17, 21, and 25 per cent. This was with a Crossley four-cycle engine.

In Fig. 9C is represented an ideal card of the work of a perfect compression cycle in which the gases are compressed. Additional pressure is instantly developed by combustion or heat at constant volume, and then allowed to expand to atmospheric pressure — the curves of compression and expansion being adiabatic, as for a dry gas. In this diagram the lines follow Carnot's cycle, in which the whole heat energy is represented in work. The piston stroke commencing at O, compression completed at D, pressure augmented from D to F, expansion doing work from F to B, and exhausting along the atmospheric line B A. The gases in this case expand till their pressure falls to the atmospheric line, and their whole energy is supposed to be utilized. In this imaginary cycle, no heat is supposed to be lost by absorption of walls of a cylinder or by radiation, and no back-pressure during exhaust or friction are taken into account.

The efficiences in regard to power in a heat-engine may be divided into four kinds, as follows: I. The first is known as the *maximum theoretical efficiency* of a perfect engine (represented by the lines in the indicator diagram, Fig. 9C). It is expressed by the formula $\frac{T_1 - T_0}{T_1}$ and shows the work of a perfect cycle in an engine working between the received temperature + absolute temperature (T_1) and the initial atmospheric temperature + absolute temperature (T_0). II. The second is the *actual heat efficiency,* or the ratio of the heat turned into work to the total heat received by the engine. It expresses the *indicated horse-power.* III. The third is the ratio between the second or *actual heat efficiency* and the first or *maximum theoretical efficiency* of a perfect cycle. It represents the greatest possible utilization of the power of heat in an internal-combustion engine. IV. The fourth is the *mechanical efficiency.* This is the ratio between the actual horse-power delivered by the engine through a dynamometer or measured by a brake (brake horse-power), and the indicated horse-power. The difference between the two is the power lost by engine friction. In regard to the general heat efficiency of the materials of power in explosive engines, we find that with good illuminating gas the practical efficiency varies from 25 to 40 per cent.; kerosene-motors, 20 to 30; gasoline-motors, 20 to 32;

acetylene, 25 to 35; alcohol, 20 to 30 per cent. of their heat value. The great variation is no doubt due to imperfect mixtures and variable conditions of the old and new charge in the cylinder; uncertainty as to leakage and the perfection of combustion. In the Diesel motors operating under high pressure, up to nearly 500 pounds, an efficiency of 36 per cent. is claimed.

On general principles the greater difference between the heat of combustion and the heat at exhaust is the relative measure of the heat turned into work, which represents the degree of efficiency without loss during expansion. The mathematical formulas appertaining to the computation of the element of heat and its work in an explosive engine are in a large measure dependent upon assumed values, as the conditions of the heat of combustion are made uncertain by the mixing of the fresh charge with the products of a previous combustion, and by absorption, radiation, and leakage. The computation of the temperature from the observed pressure may be made as before explained, but for compression-engines the needed starting-points for computation are very uncertain, and can only be approximated from the exact measure and value of the elements of combustion in a cylinder charge.

Then theoretically the absolute efficiency in a perfect heat-engine is represented by $\frac{T - T_1}{T}$, in which T is the acquired temperature from absolute zero; T_1, the final absolute temperature after expansion without loss. Then, for example, supposing the acquired temperature of combustion in a cylinder charge was raised 2000° F. from 60°: the absolute temperature would be $2000 + 60 + 460 = 2520°$, and if expanded to the initial temperature of 60° without loss the absolute temperature of expansion will be $60 + 460 = 520$, then $\frac{2520 - 520}{2520} = .79$ per cent., the theoretical efficiency for the above range of temperature. In adiabatic compression or expansion, the ratio of the specific heat of air or other gases becomes a logarithmic exponent of both compression and expansion. The specific heat of air at constant volume is .1685 and at constant pressure, .2375 for 1 lb. in weight; water = 1. for 1 lb. Then $\frac{.2375}{.1685}$ = the ratio $y = 1.408$.

Then for the following formulas the specific heat $= K_v = .1685$ constant volume, and $K_p = .2375$ constant pressure. The quantity of heat in thermal units given by an impulse of an explosive engine is $K_v\ (T - t) =$ heat units. Then using the figures as before, $.1685 \times (2520 - 520) = 337$ heat units per pound of the initial charge.

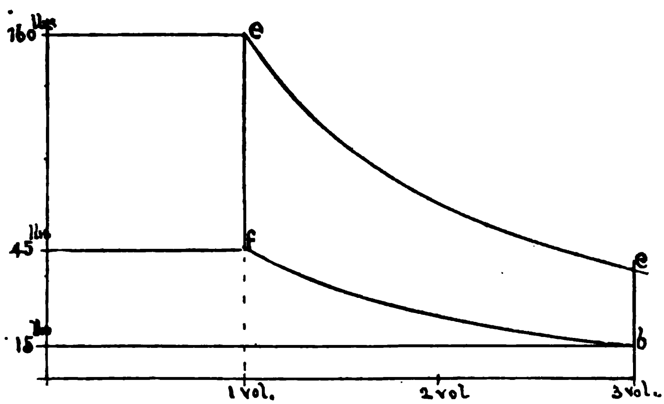


FIG. 10.—The four-cycle compression card. Theoretical.

The heat in thermal units discharged will be $K_p\ (T_1 - t)$, $T_1 = t\left(\frac{T}{t}\right)^{\frac{1}{y}}$; $t =$ absolute initial temperature, say 520°.

Then using again the figures as before and assuming that $T =$ 2520° F., then $T_1 = 520 \left(\frac{2520}{520}\right)^{\frac{1}{1.408}} = 520 \times (\log. 4.846 \times .7102)$ $= 1594°$ absolute, and $1594 - 520 = 1074°$ F. Then the heat in thermal units discharged will be $.2375 \times (1594 - 520) = .2375 \times 1074 = 255$ heat units.

With the absolute temperature at the moment of exhaust known, the efficiency of the working cycle may be known, always excepting the losses by convection through the walls of the cylinder.

The formula for this efficiency is: eff. $= 1 - y\ \frac{T_1 - t}{T - t}$; then by substituting the figures as before, $1 - 1.408\ \frac{1594 - 520}{2520 - 520} = \frac{1074}{2000}$ $= .537 \times 1.408 = .756$, and $1 - .756 = 24$ per cent.

To obtain the adiabatic terminal temperature from the relative volumes of clearance and expansion, we have the formula $\left(\frac{V_0}{V}\right)^{-y-1} = \frac{T_1}{T}$, in which $\frac{V_0}{V}$ is the ratio of expansion in terms of the charging space in engines of the Lenoir type to the whole volume of the cylinder, including the charging space, so that if the stroke of the piston is equal to the area of the charging or combustion space, the expansion will be twice the volume of the charging space and $\frac{V}{V} = \frac{1}{2}$. Then $\frac{T_1}{T} = \left(\frac{1}{2}\right)^{.408}$ and $T_1 = T\left(\frac{1}{2}\right)^{.408}$. Using the same value as before, $T_1 = 2520\left(\frac{1}{2}\right)^{.408}$ and using logarithms for $\frac{1}{2}$, log. 2 $= 0.30103 \times {}^{.408} =$ log. 0.12282 = index 1.32, and $\frac{2520°}{1.32} = 1908°$, the absolute temperature T_1 at the terminal of the stroke. Then 1908° − 460° = 1448° F., temperature at end of stroke.

For obtaining the efficiency from the volume of expansion from a known acquired temperature we have $\frac{V}{V_0} t = \frac{2}{1} \times 520° =$ 1040° absolute $= t_1$. Then

$$\text{the efficiency} = 1. - \frac{(T_1 - t_1) + y\,(t_1 - t)}{T - t}$$

Then using the values as above:

$$\text{efficiency} = 1. - \frac{(1908 - 1040) + 1.408\,(1040 - 520)}{2520 - 520} = 868 +$$

$$1.408 \times 520 = 732 + 868 = \frac{1600}{2000} = .80, \text{ and } 1 - .80 = .20 \text{ per cent.}$$

For a four-cycle compression-engine with compression say to 45 lbs., the efficiency is dependent upon the temperature of compression, the relative volume of combustion chamber and piston stroke, and the temperatures. Fig. 10 is a type card of reference for the formulæ for efficiencies of this class of explosive motors, in which:

t = abs. temp. at *b* normal.
t_c = abs. temp. of compression *f*.
T = abs. acquired temp. *e*.
T_1 = abs. temp. at *c*.

P = abs. pressure at *b*.
P_c = abs. pressure at *f*.
P_o = abs. pressure at *c*.
V_o = volume at *b*.
V = volume at *c*.
V_c = volume at *f*.
vo = V or volume at compression = volume at exhaust.
K_v = .1685 specific heat at constant volume.
Let T = abs. acquired temp. = 2520° F. as before.
t = abs. normal temp. = 520° or 60° F.

$$t_c = \text{abs. temp. of compression} = t\left(\frac{P_c}{P}\right)^{\frac{y-1}{y}} = \frac{1.408-1}{1.408} = 0.29.$$

Then $520°\left(\frac{60}{15}\right)^{0.29}$ = 777° absolute temperature of compression.

$$T_1 = \text{abs. temp. of expansion} = \frac{T\,t}{t_c} \text{ or } \frac{2520° \times 520}{777} = 1686°.$$

The terms being assumed and known from assumed data, the

$$\text{efficiency} = 1\,\frac{K_v\,(T - t_c) - K_v\,(T_1 - t)}{K_v\,(T - t_c)}$$

Reducing, efficiency $= 1 - \frac{T_1 - t}{T - t_c}$; substituting figures as above found, $1 - \frac{1686 - 520}{2520 - 777} = .333$ per cent.; also $1 - \frac{T_1}{T} = \frac{1686}{2520} =$ .333 and $1 - \frac{t}{t_c} = \frac{520}{777} = .333$ approximately.

For obtaining the efficiency from the relative volumes at both ends of the piston stroke, with an expansion in the cylinder equal to twice the clearance space, by which the total volume at the end of the stroke will be three times the volume of the clearance space, — efficiency in this case may be expressed by the formula $1-\left(\frac{V_o}{V_c}\right)^{y-1}$; substituting, the values become $1 - \left(\frac{1}{3}\right)^{.408}$; using logarithms as before, log. 3 = 0.477121 × .408 = 0.194665, the index of which is 1.565, and $\frac{1}{1.565} = .639$. Then 1 − .639 = .36 per cent.

TEMPERATURES AND PRESSURES

Owing to the decrease from atmospheric pressure in the in-drawing charge of the cylinder, caused by valve and frictional obstruction, the compression seldom starts above 13 lbs. absolute, especially in high-speed engines. Col. 3 in the following table represents the approximate absolute compression pressure for the clearance percentage and ratio in Cols. 1 and 2, while Col. 4 indicates the gauge pressure from the atmospheric line. The

Table IV.—Gas-Engine Clearance Ratios, Approximate Compression, Temperatures of Explosion and Explosive Pressures with a Mixture of Gas of 660 Heat Units per Cubic Foot and Mixture of Gas 1 to 6 of Air.

Clearance Per Cent. of Piston Volume.	Ratio $\frac{V}{V_c} = \frac{P + C \text{ Vol.}}{\text{Clearance.}}$	Approximate Compression from 13 pounds Absolute.	Approximate Gauge Pressure.	Absolute Temperature of Compression from 560 deg. Fahr. in Cylinder.	Absolute Temperature of Explosion. Gas, 1 part; Air, 6 parts.	Approximate Explosion Pressure Absolute.	Approximate Gauge Pressure.	Approximate Temperature of Explosion, Fahrenheit.
1	2	3	4	5	6	7	8	9
		Lbs.		Deg.	Deg.	Lbs.	Lbs.	Deg.
.50	3.	57.	42.	822.	2488	169	144	2027
.444	3.25	65.	50.	846.	2568	197	182	2107
.40	3.50	70.	55.	868.	2638	212	197	2177
.363	3.75	77.	62.	889.	2701	234	219	2240
.333	4.	84.	69.	910.	2751	254	239	2290
.285	4.50	102.	88.	955.	2842	303	288	2381
.25	5.	114.	99.	983.	2901	336	321	2440

temperatures in Col. 5 are due to the compression in Col. 3 from an assumed temperature of 560° F. in the mixture of the fresh charge of 6 air to 1 gas with the products of combustion left in the clearance chamber from the exhaust stroke of a medium-speed motor. This temperature is subject to considerable variation from the difference in the heat-unit power of the gases and vapors used for explosive power, as also of the cylinder-cooling effect. In Col. 6 is given the approximate temperatures of explosion or a mixture of air 6 to gas 1 of 660 heat units per cubic foot, for the relative values of the clearance ratio in Col. 2 at constant volume. The formulæ for the above approximate table, avoiding decimal values, are as follows:

$\frac{\text{Col. } 1 + 1}{\text{Col. } 1} = \text{Col. } 2.$ $1.35 \log. \frac{V}{V_c} = \log. \frac{p_c}{P} = \text{Col. } 3.$

$p_c + P =$ absolute pressure Col. 3.

$.35 \log. \text{Ratio} = \log. \frac{t_c}{t}$ Col. 5.

$\frac{p_c T}{t_c} = P$ absolute pressure Col. 7. $P - p =$ Col. 8. $T - 461° =$ Col. 9.

$p_c =$ absolute pressure of compression.

$p =$ initial absolute pressure in cylinder before compression, 13 lbs.

$P =$ absolute pressure of explosion.

$T =$ absolute explosion temperature.

$t =$ initial absolute temperature in cylinder after charge 560° F.

$t_c =$ absolute temperature of compression.

The explosive absolute temperature in Col. 6 decreases in proportion to the dilution of the gas with air, until with the proportion of 12 air to 1 gas, but 69 per cent. of the temperature given in Col. 6 is available. The decrease in pressure follows in a like proportion. In Col. 7 is given the absolute explosive pressure due to the conditions in the preceding columns and computed from the formula $\frac{p_c T}{t} =$ P, in which $p_c =$ absolute compression pressure Col. 3. T = absolute explosive temperature Col. 6. $t =$ absolute compression temperature Col. 5, for each ratio in Col. 2. Col. 8 is the gauge pressure derived from the absolute pressures in Col. 7. Col. 9 is the explosive temperature on the Fahrenheit scale, T — 461°, or Col. 6 — 461°.

The following table and diagram show the approximate resulting temperatures usual in gas-engines, in consideration of the heat values of each element in the gas and its distribution to the air and heated contents of the clearance space from a previous explosion, and the estimated absorption of heat by the walls of the clearance space at the moment of combustion, for gas of 660 thermal units per cubic foot:

TABLE V.

Clearance Per Cent. of Piston Volume.	Ratio $\frac{P + C}{\text{Clearance.}}$	Usual rise in temperature of explosion of various air and gas mixtures, due to the ratio of compression in column 2.						
		6 to 1	7 to 1	8 to 1	9 to 1	10 to 1	11 to 1	12 to 1
.50	3.	2,029	1,922	1,845	1,739	1,629	1,524	1,398
.444	3.25	2,111	2,001	1,918	1,807	1,693	1,584	1,452
.40	3.50	2,183	2,069	1,981	1,866	1,748	1,635	1,500
.363	3.75	2,245	2,127	2,036	1,917	1,795	1,679	1,540
.333	4.	2,300	2,178	2,084	1,961	1,837	1,718	1,578
.285	4.5	2,390	2,269	2,165	2,036	1,907	1,783	1,636
.25	5.	2,462	2,343	2,225	2,098	1,963	1,836	1,683
.222	5.5	2,522	2,404	2,282	2,145	2,008	1,878	1,722
.20	6.	2,572	2,456	2,326	2,186	2,046	1,914	1,755

Diagram of the rise in temperature of various mixtures of air and gas of 660 thermal units per cubic foot at ratios of compression of $\frac{\text{P} + \text{C Vol.}}{\text{Clearance Vol.}}$ and of piston-stroke volume, less the estimated loss of temperature due to the clearance volume of a previous combustion and wall-cooling.

The ratio of compression is obtained by the stroke volume of the piston, which may be represented by 1. to which is added the percentage of the volume for clearance, and the sum divided by the clearance equals the *ratio*. For example:

$\frac{1 + .50}{.50} = 3.$ and $\frac{1 + .20}{.20} = 6.$ the ratios as in the diagram.

Then using the *ratio* for obtaining both stroke and clearance $\frac{3.}{3.} = 1$ and $3 - 1 = 2$ the stroke and $2 - 1 = 1$, the clearance. At the other end, for example, $\frac{6}{6} = 1.$ and $6. - 1 = 5.$ the stroke and $6 - 5 = 1.$ the clearance in parts of the stroke.

In view of the experiments in this direction, it clearly shows that in practical work, to obtain the greatest economy per effective brake horse-power, it is necessary: 1st. To transform the heat into work with the greatest rapidity mechanically allowable. This means high piston speed. 2d. To have high initial compression. 3d. To reduce the duration of contact between the hot gases and

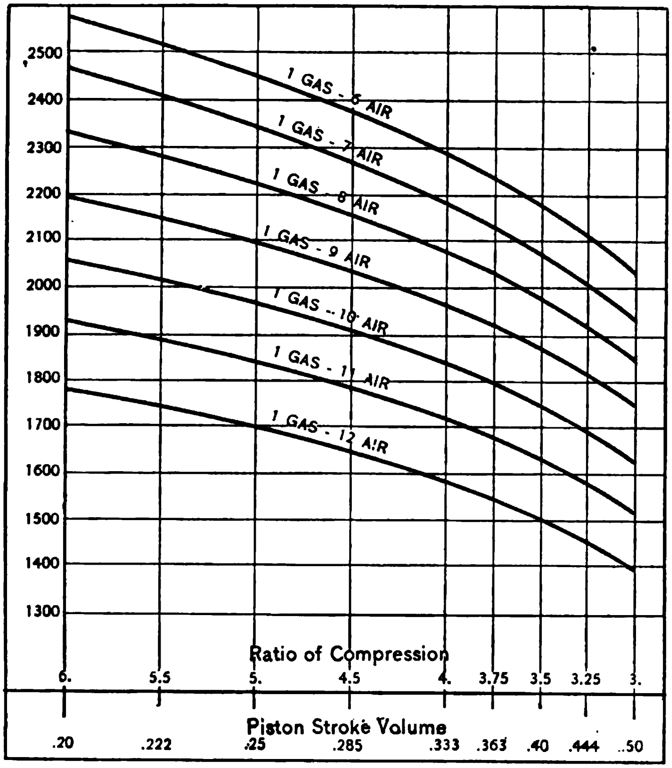


FIG. 11.—Diagram of heat in the gas-engine cylinder.

the cylinder walls to the smallest amount possible; which means short stroke and quick speed, with a spherical cylinder head. 4th. To adjust the temperature of the jacket water to obtain the most economical output of actual power. This means water-tanks or water-coils, with air-cooling surfaces suitable and adjustable to the most economical requirement of the engine, which by late trials requires the jacket water to be discharged at about 200° F. 5th. To reduce the wall surface of the clearance space or combustion chamber to the smallest possible area, in proportion to its required volume. This lessens the loss of the heat of combustion by exposure to a large surface, and allows of a higher mean wall temperature to facilitate the heat of compression.

CHAPTER IV

RETARDED COMBUSTION, WALL-COOLING, AND COMPRESSION EFFICIENCIES

SOME of the serious difficulties in practically realizing the condition of a perfect cycle in an internal-combustion engine are shown in the diagram Fig. 12, taken from an Otto gas-engine, in which the cooling effect of the walls is shown by the lagging of the explosion curve, by the missing of several explosions when the cylinder walls have been unduly cooled by the water jacket. The same delay is experienced in starting a gas-engine. The indicator card I A D representing the normal condition of constant work in the cylinder; the curve I B D an interruption of explosions for several revolutions; and I C D a still longer interruption in the explosions with the engine in continuous motion.

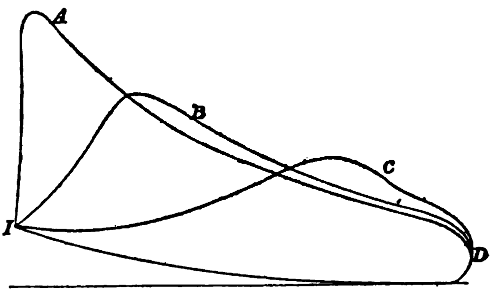


FIG. 12.—Variable card from wall cooling.

In an experimental investigation of the efficiency of a gas-engine under variable piston speeds made in France, it was found that the useful effect increases with the velocity of the piston — that is, with the rate of expansion of the burning gases with mixtures of uniform volumes; so that with variations of time of complete combustion at constant pressure, and the variations due to speed, in a way compensate in their efficiencies. The dilute mixture, being slow burning, will have its time and pressure quickened by increasing the speed.

TABLE VI.—TRIAL EFFICIENCIES DUE TO INCREASED PISTON SPEED.

$$\text{Efficiency} = \frac{\text{work of indicator diagram}}{\text{theoretical work.}}$$

Mixture.	Time of Explosion, Second.	Piston Speed Foot per Second.	Computed work diagram. Foot-pounds.	Theoretical Work of the Gas. Foot-pounds.	Efficiency.
1 volume coal-gas to 9.4 volumes air (.1093 cubic feet mixture).........	.53	1.181	70.8	4917	1.44
1 volume coal-gas to 9.4 volumes air	.40	1.64	85.3	4917	1.70
1 volume coal-gas to 9.4 volumes air	.25	3.01	105.5	4917	2.10
1 volume coal-gas to 9.4 volumes air	.16	4.55	125.8	4917	2.60
1 volume coal-gas to 6.33 volumes air (.073 cubic feet mixture)..........	.15	5.57	127.2	4793	2.60
1 volume coal-gas to 6.33 volumes air	.09	9.51	289.9	4793	6.00
1 volume coal-gas to 6.33 volumes air	.06	14.1	364.4	4793	7.50

These trials give unmistakable evidence that the useful effect increases with the velocity of the piston — that is, with the rate of expansion of the burning gases. The time necessary for the explosion to become complete and to attain its maximum pressure depends not only on the composition of the mixture, but also upon the rate of expansion. This has been verified in experiments with a high-speed motor, at speeds from 500 to 1,000 revolutions per minute, or piston speeds of from 16 to 32 feet per second. The increased speed of combustion due to increased piston speed is a matter of great importance to builders of gas-engines, as well as to the users, as indicating the mechanical direction of improvements to lessen the wearing strain due to high speed and to lighten the vibrating parts with increased strength, in order that the balancing of high-speed engines may be accomplished with the least weight.

From many experiments made in Europe and in the United States, it has been conclusively proved that excessive cylinder cooling by the water-jacket results in a marked loss of efficiency. In a series of experiments with a simplex engine in France, it was found that a saving of 7 per cent. in gas consumption per brake horse-power was made by raising the temperature of the jacket water from 141° to 165° F. A still greater saving was made in a trial with an Otto engine by raising the temperature of the jacket water from 61° to 140° F. — it being 9.5 per cent. less gas per brake horse-power.

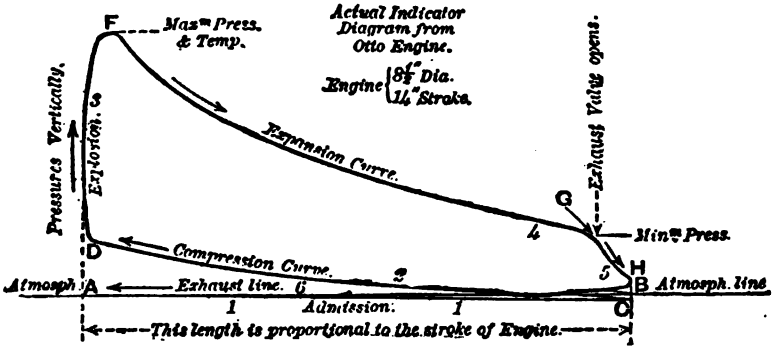


FIG. 13.—Otto four-cycle card.

It will be noticed that volumes of similar cylinders increase as the cube of their diameters, while the surface of their cold walls varies as the square of their diameters; so that for large cylinders the ratio of surface to volume is less than for small ones. This points to greater economy in the larger engines. The study of many experiments goes to prove that combustion takes place gradually in the gas-engine cylinder, and that the rate of increase of pressure or rapidity of firing is controlled by dilution and compression of the mixture, as well as by the rate of expansion or piston speed. The rate of combustion also depends on the size and shape of the exploding chamber, and is increased by mechanical agitation of the mixture during combustion, and still more by the mode of firing. A small intermittent spark gives the most uncertain ignition, whereas a continuous electric spark passed through an explosive mixture, or a large flame as the shooting of a mass of lighted gas into a weak mixture, will produce rapid ignition.

The shrinkage of the charge of mixed gas and air by the union of its hydrogen and oxygen constituents by the production of the vapor of water in a gas-engine cylinder, using 1 part illuminating gas to 6.05 parts air, is a notable amount, and of the total volume of 7.05 in cubic feet, the product will be:

1.3714	cubic feet	water vapor.
.5714	" "	carbonic acid.
.0050	" "	nitrogen derived from the gas.
4.8000	" "	" " " " air.
6.7428	" "	products of combustion.

Then 7.05 cubic feet of the mixture charge will have shrunk by combustion to 6.7428 cubic feet at initial temperature, or 4.4 per cent. This difference in the computed shrinkage at initial temperature is manifested in the reduced pressure of combustion due to the computed shrinkage, and amounts to about 2 per cent. in the mean pressure, as shown on an indicator card. With the less rich gas, as water, producer, and Dowson gas, the shrinkage by conversion into water vapor is equal to 5.5 per cent.

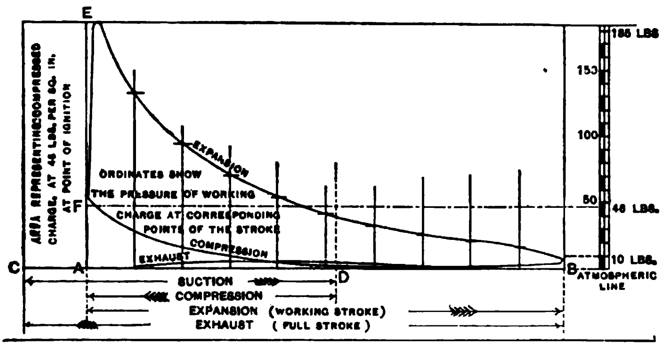


Fig. 14.—Indicator card, Atkinson type.

In Fig. 14 is represented a card from the Atkinson gas-engine. The peculiar design of this engine enables the largest degree of expansion known in gas-engine practice. In Fig. 15 is shown an actual indicator diagram from an Otto or four-cycle engine, in which the sequences of operation are delineated through two of its four cycles. The curve of explosion shows that firing commenced slightly before the end of the compression stroke, and that combustion lagged until a moment after reversal of the stroke. The expansion line is somewhat higher than the adiabatic curve, indicating a partial combustion taking place during the stroke of the piston, showing an irregularity in firing the charge, and probably an irregular progress of combustion by defective mixture.

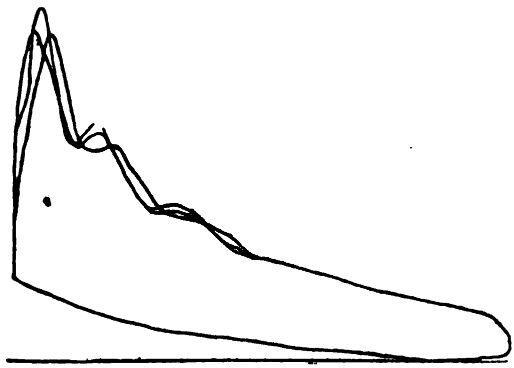


FIG. 15.—Indicator card. full load. Four-cycle.

FIG. 16.—Indicator card, half load.

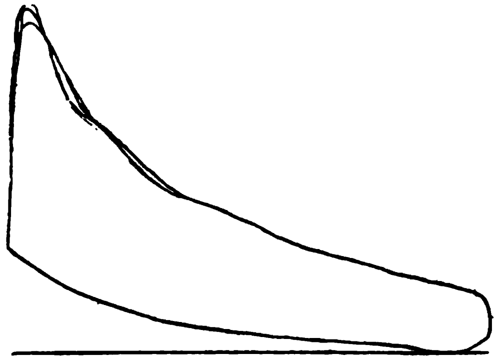


FIG. 17.—Typical compression card. Mean pressure, 76 pounds per square inch.

This card was made when running at full load, and computed at 69 lbs. mean pressure.

Fig. 16 represents a card from the same engine at half load and lessened combustion charge. It shows the same characteristics as to irregularity, and also a lag in firing and a fitful after-com-

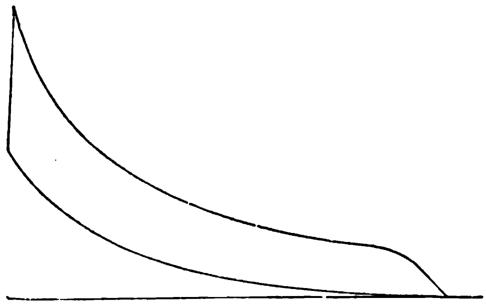


FIG. 18.—Kerosene motor card. Mietz & Weiss.

bustion; but from weak mixture and interrupted firing the cooling influence of the cylinder walls has prolonged the combustion with ignition pressure. Mean pressure, about 68 lbs. per square inch. Fig. 17 represents a typical card of our best compression-engines,

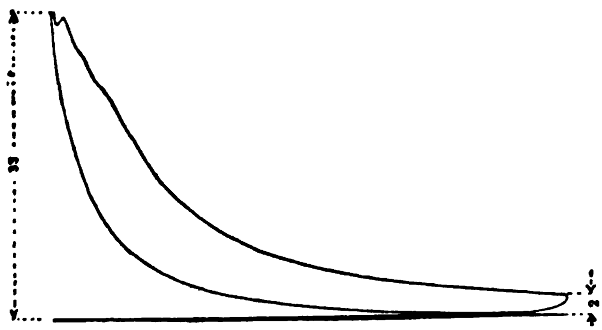


FIG. 19.—Diesel motor card.

with time igniter, at full load and uninterrupted firing. The kerosene-motor card of the Mietz & Weiss engine (Fig. 18) taken from a 20 horse-power actual, motor with cylinder 12 inches × 12 inches, at 300 revolutions per minute, shows a compression of nearly one-half the explosive force. Its efficiency is very high, and by test gave 21½ horse-power from 16½ pints of oil per hour.

A most unique card is that of the Diesel motor (Fig. 19), which involves a distinct principle in the design and operation of internal-combustion motors, in that instead of taking a mixed charge for instantaneous explosion, its charge primarily is of air and its compression to a pressure at which a temperature is attained above the igniting point of the fuel, then injecting the fuel under a still higher pressure by which spontaneous combustion takes place gradually with increasing volume over the compression for part of the stroke or until the fuel charge is consumed. The motor thus operating between the pressures of 500 and 35 lbs. per square inch, with a clearance of about 7 per cent., has given an efficiency of 36 per cent. of the total heat value of kerosene oil.

ADVANCED IGNITION

The governing of an explosive motor, by changing the time of ignition, may be done by advancing or retarding the ignition spark from the dead centre of the stroke. In Fig. 20 is shown the effect of pre-ignition for regulating speed. The relative areas of the combined card show the change in mean pressure and also the increased compression before the crank arrives at its dead centre. This may be carried so far that a reversal of the motor may take place. In automobile practice both the advance and retardation of ignition is employed generally: but is not recommended in lieu of variable-fuel charge, though usually combined with this most effective method of control. The value of an indicator card for ascertaining the true condition of the internal activities within

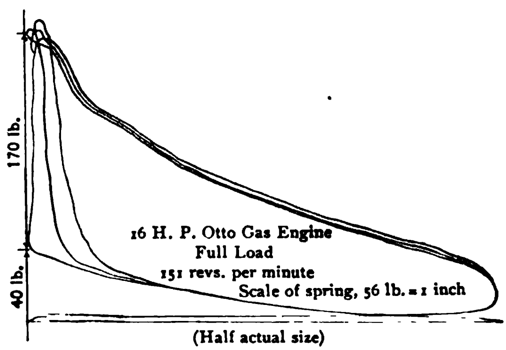


FIG. 20.—Effect of advanced ignition.

the cylinder of an explosive motor is most apparent, and it should always be made the means for finding the cause of trouble that cannot be traced to the outside mechanism. An indicator card, or a series of them, will always show by its lines the normal or defective condition of the inlet valve and passages; the actual line of compression; the firing moment; the pressure of explosion; the velocity of combustion; the normal or defective line of expansion, as measured by the adiabatic curve, and the normal or defective operation of the exhaust valve, exhaust passages, and exhaust pipe. In fact, all the cycles of an explosive motor may be made a practical study from a close investigation of the lines of an indicator card.

CHAPTER V

COMPRESSION IN EXPLOSIVE MOTORS, AND ITS VALUE

THAT the compression in a gas, gasoline, or oil-engine has a direct relation to the power obtained, has been long known to experienced builders, having been suggested by M. Beau de Rocha, in 1862, and afterward brought into practical use in the four-cycle or Otto type about 1880. The degree of compression has had a growth from zero, in the early engines, to the highest available due to the varying ignition temperatures of the different gases and vapors used for explosive fuel, in order to avoid premature explosion from the heat of compression. Much of the increased power for equal-cylinder capacity is due to compression of the charge from the fact that the most powerful explosion of gases, or of any form of explosive material, takes place when the particles are in the closest contact or cohesion with one another, less energy in this form being consumed by the ingredients themselves to bring about their chemical combination, and consequently more energy is given out in useful or available work. This is best shown by the ignition of gunpowder, which, when ignited in the open air, burns rapidly, but without explosion, an explosion only taking place if the powder be confined or compressed into a small space.

In a gas or gasoline-motor with a small clearance or compression space — with high compression — the surface with which the burning gases come into contact is much smaller in comparison with the compression space in a low-compression motor. Another advantage of a high-compression motor is that on account of the smaller clearance of combustion space less cooling water is required than with a low-compression motor, as the temperature, and consequently the pressure, falls more rapidly. The loss of heat through the water-jacket is thus less in the case of a high-compression than in that of a low-compression motor. In the non-compression type of motor the best results were obtained with a charge of 16 to 18 parts of gas and 100 parts of air, while in the compression type the best results are obtained with an explosive mixture of 7 to 10 parts of gas and 100 parts of air, thus showing that by the utilization of compression a weaker charge with a greater thermal efficiency is permissible.

It has been found that the explosive pressure resulting from the ignition of the charge of gas or gasoline-vapor and air in the gas-engine cylinder is about 4½ times the pressure prior to ignition. The difficulty about getting high compression is that if the pressure is too high the charge is likely to ignite prematurely, as compression always results in increased temperature. The cylinder may become too hot, a deposit of carbon, a projecting bolt, nut, or fin in the cylinder may become incandescent and ignite the charge which has been excessively heated by the high compression and mixture of the hot gases of the previous explosion.

With gasoline-vapor and air the compression cannot be raised above about 90 to 95 pounds to the square inch, many manufacturers not going above 55 or 60 pounds. For natural gas the compression pressure may easily be raised to from 85 to 100 pounds per square inch. For gases of low calorific value, such as blast-furnace or producer-gas, the compression may be increased to from 140 to 190 pounds. In fact the ability to raise the compression to a high point with these gases is one of the principal reasons for their successful adoption for gas-engine use. In kerosene injection engines the compression of 250 pounds per square inch has been used with marked economy. Many troubles in regard to loss of power and increase of fuel have occurred and will no doubt continue, owing to the wear of valves, piston, and cylinder, which produces a loss in compression and explosive pressure and a waste of fuel by leakage. Faulty adjustment of valve movement is also a cause of loss of power; which may be from tardy closing of the inlet-valve or a too early opening of the exhaust-valve.

The explosive pressure varies to a considerable amount in proportion to the compression pressure by the difference in fuel value and the proportions of air mixtures, so that for good illuminating gas the explosive pressure may be from 2.5 to 4 times the compression pressure. For natural gas 3 to 4.5, for gasoline 3 to 5, for producer-gas 2 to 3, and for kerosene by injection 3 to 6. For obtaining the compression clearance we have the equations: $(p\,v)^{1.35} = (p_1\,v_1)^{1.35}$. Then $p_1 = p\left(\frac{v}{v_1}\right)^{1.35}$ and $v_1 = v\left(\frac{p}{p_1}\right)^{1.35}$

and substituting values for p, and p_1, we have values for the volume of the clearance, say for 100 pounds gauge pressure of compression, in which v and p represent absolute volumes and pressures. Then using the expression for pressure, say for 100 pounds, in which p = normal absolute pressure and p_1 = absolute compression pressure, the expression becomes for clearance plus stroke, $1 - \left(\frac{14.7}{114.7}\right)^{1.35}$ which worked out by logarithms = .1281 log. $\bar{1}.107549 \times 1.35 = \bar{1}.14519115$ index of which is .1397, the adiabatic ratio of compression for the stroke + clearance, and 1 — .1397 = .8603 the ratio for obtaining the clearance. Then by dividing the stroke in inches by this ratio and subtracting from the quotient the length of the stroke gives the clearance length also in inches.

For example, for 10-inch stroke, $\frac{10}{.8603} = 11.623 - 10 = 1.623$ inches clearance in the length of a plain cylindrical space for 100 pounds compression. If the clearance space is of other form than the plain extension of the cylinders the volumes will have the same relation. For example, for 100 pounds compression, a motor with an 8-inch cylinder and 10-inch stroke, the stroke volume will be 502.6 cubic inches, and $\frac{502.6}{.8603} = 584.2$ cubic inches, and 584.2 — 502.6 = 81.6 cubic inches clearance. From this formula the following table of compression pressures and their clearance ratio in parts of the stroke has been computed:

TABLE VII.—COMPRESSION AND CLEARANCE.

Compression in pounds per square inch.	Ratio. $\frac{\text{Stroke}}{\text{Ratio}}$ — Stroke = Clearance.
100	.8603
90	.8419
80	.8189
70	.7896
60	.7508
50	.6972
40	.6201
30	.5058

The compression temperatures, although well known and easily computed from a known normal temperature of the explosive mixture, are subject to the effect of the uncertain temperature of the gases of the previous explosion remaining in the cylinder, the temperature of its walls, and the relative volume of the charge, whether full or scant; which are terms too variable to make any computations reliable or available.

For the theoretical compression temperatures from a known normal temperature, we append a table of the rise in temperature for the compression pressures in the foregoing Table VII:

TABLE VIII.—COMPRESSION TEMPERATURES FROM A NORMAL TEMPERATURE OF 60 DEGREES FAHRENHEIT.

100 lbs. gauge	484°	60 lbs. gauge	373°
90 lbs. gauge	459°	50 lbs. gauge	339°
80 lbs. gauge	433°	40 lbs. gauge	301°
70 lbs. gauge	404°	30 lbs. gauge	258°

To these values must be added the assumed temperature of the contents of the cylinder above 60° at the moment that compression

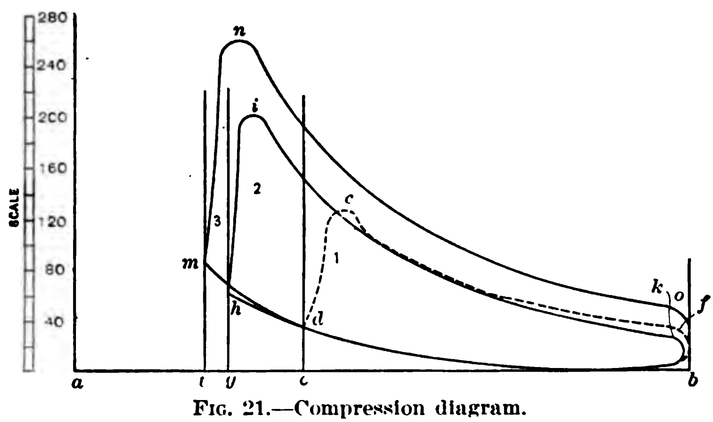


FIG. 21.—Compression diagram.

begins. For example, for obtaining the assumed temperature at the moment that compression begins for 100 pounds compression and for an observed temperature of the exhaust of 750° F. we have the compression clearance of .1397 × 750° = 104.7° and piston volume of .8603 × 60° = 51.6°, making the charged temperature 156.3° to which may be added 10° for increase from the walls of the cylinder = 166° + 484° for compression rise = 650° the probable compression temperature for 100 pounds per square inch compression. This is, no doubt, a crude method, but we find nothing better.

The effect of compression on fuel economy is well shown in trials of a four-cycle gas-engine and given in the following table:

TABLE IX.—COMPARISON OF THE THEORETICAL AND ACTUAL EFFICIENCIES OF A FOUR-CYCLE GAS-ENGINE AND FUEL ECONOMY WITH VARYING COMPRESSION.

Compression pressure, pounds.	Ratio of compression.	Computed efficiency from compression volume.	Actual indicated efficiency by card and fuel.	Gas burned per I. H. P. C. Ft.	Ratio of actual to computed efficiency.
38	.6	.33	.17	24.	$\frac{.17}{.33} = .51$
61	.4	.40	.21	20.5	$\frac{.21}{.40} = .53$
87	.34	.428	.25	14.8	$\frac{.25}{.428} = .58$

From considerations shown in the table it is evident that there is economy in compression and it is claimed that still higher compression may be used to advantage; but from reasons given in the foregoing discussion of this subject, the practical limit of compression may be stated to be at 100 pounds.

The diagram (Fig. 21), drawn to scale from trials with compressions at 38, 61, and 87 pounds, gives an ideal conception of the value of the power of the same engine under various compressions, in which *a, b,* represents the piston and clearance space and *b, c; b, g,* and *b. l,* the relative piston strokes and clearance for the compressions of 38, 61, and 87 pounds. The relative areas show at a glance and the above table shows the relative value of the fuel consumed per indicated horse-power.

A very useful chart for determining compression pressures in gasoline-engine cylinders for various ratios of compression space to total cylinder volume is given by P. S. Tice, and described in the Chilton Automobile Directory by the originator as follows:

It is many times desirable to have at hand a convenient means for at once determining with accuracy what the compression pressure will be in a gasoline-engine cylinder, the relationship between the volume of the compression space and the total cylinder volume or that swept by the piston being known. The curve at Fig. 22 is offered as such a means. It is based on empirical data gathered from upward of two dozen modern automobile engines and represents what may be taken to be the results as found in practice. It is usual for the designer to find compression pressure values, knowing the volumes from the equation

$$P_2 = P_1 \left(\frac{V_1}{V_2}\right)^{1.4} \qquad \ldots\ldots\ldots\ldots\ldots\ldots \quad 1$$

which is for adiabatic compression of air. Equation (1) is right enough in general form but gives results which are entirely too high, as almost all designers know from experience. The trouble lies in the interchange of heat between the compressed gases and the cylinder walls, in the diminution of the exponent (1.4 in the above) due to the lesser ratio of specific heat of gasoline vapor and in the transfer of heat from the gases which are being com-

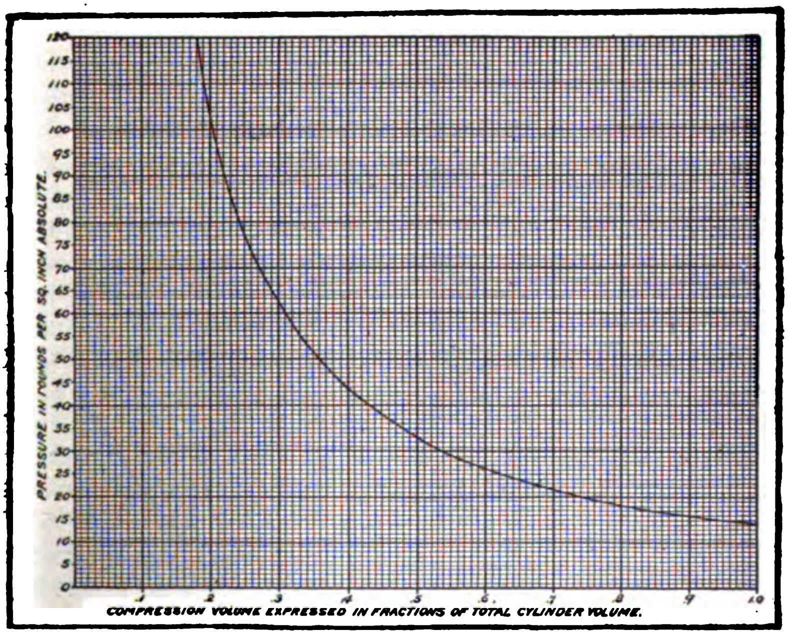


FIG. 22.—Chart showing relation between compression volume and pressure.

pressed to whatever fuel may enter the cylinder in an unvaporized condition. Also, there is always some piston leakage, and, if the form of the equation (1) is to be retained, this also tends to lower the value of the exponent. From experience with many engines, it appears that compression reaches its highest value in the cylinder for but a short range of motor speeds, usually during the mid-range. Also, it appears that, at those speeds at which compression shows its highest values, the initial pressure at the start of the compression stroke is from .5 to .9 lbs. below atmospheric. Taking this latter loss value, which shows more often than those of lesser value, the compression is seen to start from an initial pressure of 13.9 lbs. per sq. in. absolute.

Also, experiment shows that if the exponent be given the value 1.26, instead of 1.4, the equation will embrace all heat losses in the compressed gas, and compensate for the changed ratio of specific heats for the mixture and also for all piston leakage, in the average engine with rings in good condition and tight. In the light of the foregoing, and in view of results obtained from its use, the above curve is offered — values of P_2 being found from the equation

$$P_2 = 13.8 \left(\frac{V_1}{V_2}\right)^{1.26}$$

In using this curve it must be remembered that pressures are absolute. Thus: suppose it is desired to know the volumetric relationships of the cylinder for a compression pressure of 75 lbs. gauge. Add atmospheric pressure to the desired gauge pressure $14.7 + 75 = 89.7$ lbs. absolute. Locate this pressure on the scale of ordinates and follow horizontally across to the curve and then vertically downward to the scale of abscissas, where the ratio of the combustion chamber volume to the total cylinder volume, which latter is equal to the sum of the combustion chamber volume and that of the piston sweep. In the above case it is found that the combustion space for a compression pressure of 75 lbs. gauge will be .225 of the total cylinder volume, or $.225 \div 775 = .2905$ of the piston sweep volume. Conversely, knowing the volumetric ratios, compression pressure can be read directly by proceeding from the scale of abscissas vertically to the curve and thence horizontally to the scale of ordinates.

CAUSES OF LOSS AND INEFFICIENCY IN EXPLOSIVE MOTORS

The difference realized in the practical operation of an internal combustion heat engine from the computed effect derived from the values of the explosive elements is probably the most serious difficulty that engineers have encountered in their endeavors to arrive at a rational conclusion as to where the losses were located, and the ways and means of design that would eliminate the causes of loss and raise the efficiency step by step to a reasonable percentage of the total efficiency of a perfect cycle.

An authority on the relative condition of the chemical elements under combustion in closed cylinders, attributes the variation of temperature shown in the fall of the expansion curve, and the suppression or retarded evolution of heat, entirely to the cooling action of the cylinder walls, and to this nearly all the phenomena hitherto obscure in the cylinder of a gas-engine. Others attribute the great difference between the theoretical temperature of combustion and the actual temperature realized in the practical operation of the gas-engine, a loss of more than one-half of the total heat energy of the combustibles, partly to the dissociation of the elements of combustion at extremely high temperatures and their reassociation by expansion in the cylinder, to account for the supposed continued combustion and extra adiabatic curve of the expansion line on the indicator card.

The loss of heat to the walls of the cylinder, piston, and clearance space, as regards the proportion of wall surface to the volume, has gradually brought this point to its smallest ratio in the concave piston-head and globular cylinder-head, with the smallest possible space in the inlet and exhaust passage. The wall surface of a cylindrical clearance space or combustion chamber of one-half its unit diameter in length is equal to 3.1416 square units, its volume but 0.3927 of a cubic unit; while the same wall surface in a spherical form has a volume of 0.5236 of a cubic unit. It will be readily seen that the volume is increased 33⅓ per cent. in a spherical over a cylindrical form for equal wall surfaces at the moment of explosion, when it is desirable that the greatest amount of heat is generated, and carrying with it the greatest possible pressure from which the expansion takes place by the movement of the piston.

The spherical form cannot continue during the stroke for mechanical reasons; therefore some proportion of piston stroke of cylinder volume must be found to correspond with a spherical form of the combustion chamber to produce the least loss of heat through the walls during the combustion and expansion part of the stroke. This idea is illustrated in Figs. 23 and 24, showing how the relative volumes of cylinder stroke and combustion chamber may be varied to suit the requirements due to the quality of the elements of combustion. In Fig. 21 the ratio may also be decreased by extending the stroke.

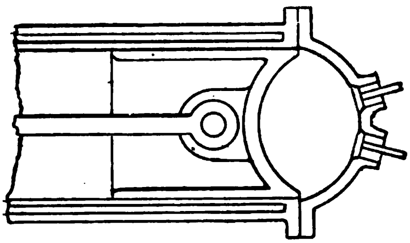


Fig. 23.—Spherical combustion chamber.

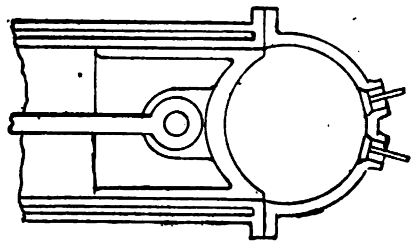


Fig. 24.—Enlarged combustion chamber.

Although the concave piston-head shows economy in regard to the relation of the clearance volume to the wall area at the moment of explosive combustion, it may be clearly seen that its concavity increases its surface area and its capacity for absorbing heat, for which there is no provision for cooling the piston, save its contact with the walls of the cylinder and the slight air cooling of its back by its reciprocal motion. For this reason the concave piston-head has not been generally adopted and the concave cylinder-head, as shown in Fig. 23, with a flat piston-head is the latest and best practice in explosive-engine construction.

The mean temperature of the wall surface of the combustion chamber and cylinder, as indicated by the temperatures of the circulating water, has been found to be an important item in the economy of the gas-engine. Dugald Clerk, in England, a high authority in practical work with the gas-engine, found that 10 per cent. of the gas for a stated amount of power was saved by using water at a temperature in which the ejected water from the cylinder-jacket was near the boiling point, and ventures the opinion that a still higher temperature for the circulating water may be used as a source of economy. This could be made practical in the case of stationary engines by elevating the water-tank and adjusting the air-cooling surface so as to maintain the inlet water at just below the boiling point, and by the rapid circulation induced by the height of the tank above the engine and the pressure, to return the water from the cylinder-jacket a few degrees above the boiling-point. The thermal displacement system of cooling employed in automobiles and the hopper cooled portable engines are working under more favorable temperature conditions than those engines in which cooling is more energetic.

For a given amount of heat taken from the cylinder by the largest volume of circulating water, the difference in temperature between inlet and outlet of the water-jacket should be the least possible, and this condition of the water circulation gives a more even temperature to all parts of the cylinder; while, on the contrary, a cold-water supply, say at 60° F., so slow as to allow the ejected water to flow off at a temperature near the boiling-point, must make a great difference in temperature between the bottom and top of the cylinder, with a loss in economy in gas and other fuels, as well as in water, if it is obtained by measurement.

In regard to the actual consumption of water per horse-power, and the amount of heat carried off by it, the study of English trials of an Atkinson, Crossley, and Griffin engine showed 62 pounds water per indicated horse-power per hour, with a rise in temperature of 50° F., or 3,100 heat units were carried off in the water out of 12,027 theoretical heat units that were fed to the motor through the 19 cubic feet of gas at 633 heat units per cubic foot per hour. Theoretically, 2,564 heat units per hour are equal to 1 horse-power. Then 0.257 of the total was given to the jacket water, 0.213 to the indicated power, and the balance. 53 per cent., went to the exhaust, radiation, and the reheating of the previous charge in the clearance and in expanding the nitrogen of the air. Other and mysterious losses, due to the unknown condition of the gases entering into and passing through the heat cycle, which have been claimed and mathematically discussed by authors, have failed to satisfy the practical side of the question, which is the main object of this work.

From the foregoing considerations of losses and inefficiencies, we find that the practice in motor design and construction has not yet reached the desired perfection in its cycular operation. Step by step improvements have been made with many changes in design though many have been without merit as an improvement, further than to gratify the longings of designers for something different from the other thing, and to establish a special construction of their own. These efforts may in time produce a motor of normal or standard design for each kind of fuel that will give the highest possible efficiency for all conditions of service.

CHAPTER VI

THE MATERIAL OF POWER IN EXPLOSIVE ENGINES

THE composition of illuminating and producer-gases, alcohol, acetylene, gasoline, kerosene and crude-petroleum oil, and air, as elements of combustion and force in explosive engines, is of great importance in comparison, of heat and motor efficiencies. By reported experiments with 20-candle coal-gas in the United States, by the evaporation of water at 212° F., a cubic foot of gas was credited with 1,236 heat units; while reliable authorities range the value of our best illuminating gases at from 675 to 810 heat units per cubic foot. The specific heat of illuminating gas is much higher than for air, being for coal-gas at constant pressure 0.6844, and at constant volume 0.5196, with a ratio of 1.315; while the specific heat for air at constant pressure is 0.2377, and at constant volume is 0.1688, and their ratio 1.408. The mixtures of gas and air accordingly vary in their specific heat with ratios relative to the volumes in the mixture. The products of combustion also have a higher specific heat than air, ranging from 0.250 at constant pressure and 0.182 at constant volume, to 0.260 and 0.190 with ratios of 1.37 and 1.36. A cubic foot of ordinary coal-gas burned in air produces about one ounce of water-vapor and 0.57 of a cubic foot of carbonic-acid gas (CO_2). Its calorific value will average about 675 heat units per cubic foot. A cubic foot of ordinary coal-gas requires 1.21 cubic feet of oxygen, more or less, due to variation in the constituents of different grades of illuminating gases in various localities, for complete combustion. Allowing for an available supply of 20 per cent. of oxygen in air for complete combustion, then $1.21 \times 5 = 6.05$ cubic feet of air which is required per cubic foot of gas in a gas-engine for its best work; but in actual practice the presence in the engine cylinder of the products of a previous combustion, and the fact that a sudden mixture of gas and air may not make a homogeneous combination for perfect combustion, require a larger proportion of air to completely oxidize the gas charge.

It will be seen by inspection of Table II that the above proportion, without the presence of contaminating elements, produces the quickest firing and approximately the highest pressure at constant volume, and that any greater or less proportion of air will reduce the pressure and the apparent efficiency of an explosive motor. There are other considerations affecting the governing of explosive engines, in which the gas element only is controlled by the governor, requiring an excess of air at the normal speed, so that an economical adjustment of gas consumption may be obtained at both above and below the normal speed.

In Table X the materials of power in use in explosive motors are given with their heat-unit and foot-pound values.

TABLE X.—MATERIAL OF POWER IN EXPLOSIVE ENGINES.

Gases, Vapors, and Other Combustibles.	Heat Units per Pound.	Heat Units per Cubic Foot.	Foot-Pounds per Cubic Foot.
Hydrogen, H	61,560	293.5	228,343
Carbon, C	14,540	....	....
Crude Petroleum, sp. gr. 0.873	18,324	....	....
Crude Petroleum, Penn., sp. gr. 0.841	18,401	....	....
Kerosene, $C_{10}H_{22}$	22,000	....	....
Benzine, C_6H_6	18,448	....	....
Gasoline, C_6H_{14}	18,000	....	....
Denatured Methyl Alcohol	13,000	....	....
Acetylene, C_2H_2	21,492	868	675,304
19 candle-power Illuminating Gas	.....	800	622,400
16 candle-power Illuminating Gas	.....	665	517,370
15 candle-power Illuminating Gas	.....	620	482,360
Gasoline Vapor, C_2H_{14}	18,000	692	538,376
Natural Gas Leechburg, Pa	.....	1,051	817,678
Natural Gas Pittsburg, Pa	.....	892	693,976
Water-Gas, average	.....	290	225,620
Producer-Gas, 100 to	.....	150	116,700
Suction-Gas, average	.....	135	105,030
Marsh-Gas, Methane, CH_4	23,594	1,051	817,678
Olefiant Gas, Ethylene, C_2H_4	21,430	1,677	1,304,716

The various other gases than coal-gas used in explosive engines are NATURAL GAS, ACETYLENE, liberated by the action of water on calcium cabide; PRODUCER-GAS, made by the limited action of air alone upon incandescent fuel; WATER-GAS, made by the action of steam alone upon incandescent fuel; SEMI-WATER-GAS, made by the action of both air and steam upon incandescent fuel — also named DOWSON GAS in England — and SUCTION-GAS. Alcohol is also coming into use in Europe.

NATURAL GAS

The constituents of natural gas vary to a considerable extent in different localities. The following is the analysis of some of the Pennsylvania wells:

TABLE XI.—NATURAL GAS CONSTITUENTS, BY VOLUME.

Constituents.	Olean, N. Y.	Pitts-burg, Pa.	Leech-burg, Pa.	Harvey Well, Butler County.	Burns Well, Butler County.
Hydrogen, H	......	22.00	4.79	13.50	6.10
Marsh-Gas, CH_4	96.50	67.00	89.65	80.11	75.44
Ethane, C_2H_4	......	5.00	4.39	5.72	18.12
Heavy Hydrocarbons	1.00	1.00	.56	......	......
Carbonic Oxide, CO	.50	.60	.26	trace	trace
Carbonic Acid, CO_2	......	.60	.35	.66	.34
Nitrogen, N	......	3.00	......	......	......
Oxygen, O	2.00	.80	......	......	......
	100.00	100.00	100.00	100.00	100.00
Heat Units, cubic foot	1,200	892	1,051	959	1,151

Density, 0.5 to 0.55 (air 1).

The calorific value of natural gas in much of the Western gas fields is below these figures.

In experiments recorded by Brannt, "Petroleum and Its Products," with the *oil-gas* as made for town lighting in many parts of the United States, of specific gravity about 0.68 (air 1), mixtures of oil-gas with air had the following explosive properties:

Oil-Gas, Volumes.	Air, Volumes.	Explosive Effect.
1	4.9	None.
1	5.6 to 5.8	Slight.
1	6 to 6.5	Heavy.
1	7 to 9	Very heavy.
1	10 to 13	Heavy.
1	14 to 16	Slight.
1	17 to 17.7	Very slight.
1	18 to 22	None.

It will be seen that mixtures varying from 1 of gas to 6 of air, and all the way to 1 of gas to 13 of air, are available for use in gas-engines for the varying conditions of speed and power regulation; and that 1 of gas to from 7 to 9 of air produces the best working effect. Its calorific value varies in different localities from 600 to 700 heat units per cubic foot. Ordinary oil illuminating gas varies somewhat in its constituents, and may average: Hydrogen, 39.5; marsh-gas, 37.3; nitrogen, 8.2; heavy hydrocarbons, 6.6; carbonic oxide, 4.3; oxygen (free), 1.4; water-vapor and impurities 2.7; total, 100; and is equal to 617 heat units per cubic foot.

PRODUCER-GAS

The constituents of producer-gas vary largely in the different methods by which it is made; in fact, all of the following described gases are made in producers, so-called. The constituents of the low grade of this name are

Carbonic Oxide, CO	22.8	per cent.
Nitrogen, N	63.5	"
Carbonic Acid, CO_2	3.6	"
Hydrogen, H	2.2	"
Marsh-Gas (Methane), CH_4	7.4	"
Free Oxygen, O	.5	"
	100.0	"

The average heating power of this variety of producer-gas is about 111 heat units per cubic foot.

Another producer-gas called

WATER-GAS

has an average composition of

Carbonic Oxide, CO	41	per cent.
Hydrogen, H	48	"
Carbonic Acid, CO_2	6	"
Nitrogen, N	5	"
	100	"

and has an average calorific value of 291 heat units per cubic foot.

SEMI-WATER-GAS

or, as designated in England, *Dowson gas,* from the name of the inventor of a water-gas making plant, has the following average composition:

Hydrogen, H	18.73	per cent.
Marsh-Gas, (Methane), CH_4	.31	"
Olefiant Gas, C_2H_4	.31	"
Carbonic Oxide, CO	25.07	"
Carbonic Acid, CO_2	6.57	"
Oxygen, O	.03	"
Nitrogen, N	48.98	"
	100.00	"

It has a calorific value of about 150 heat units per cubic foot.

PETROLEUM PRODUCTS USED IN EXPLOSIVE ENGINES

The principal products derived from crude petroleum for power purposes may commercially come under the names of gasoline, naphtha (three grades, B, C, and A), kerosene, gas-oil, and crude oil, in the proportions shown at Fig. 25. The first distillate: Rhigoline, boiling at 113° F., specific gravity 0.59 to 0.60; chimogene, boiling at from 122° to 138° F., specific gravity 0.625; gasoline, boiling at from 140° to 158° F., specific gravity 0.636 to 0.657; naphtha "C" (by some also called benzine), boiling

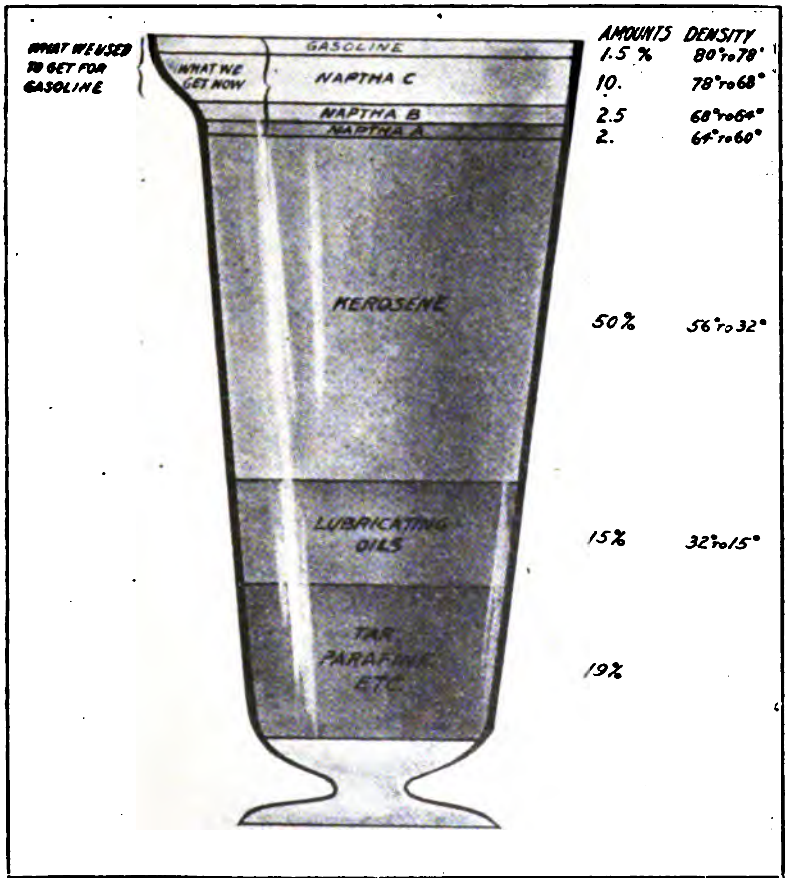


FIG. 25.—Graduate showing proportions of various liquids and their specific gravity that are distilled or obtained from crude petroleum.

from 160° to 216° F., specific gravity 0.66 to 0.70; naphtha "B" (ligroine), boiling at from 200° to 240° F., specific gravity 0.71 to 0.74; naphtha "A" (putzoel), boiling at from 250° to 300° F. The commercial gasoline of the American trade is a combination of the above fractional distillates, boiling at from 125° to 200° F., specific gravity 0.63 to 0.74. Kerosene boils at from 300° to 500° F., specific gravity 0.76 to 0.80. Gas-oil boils at above 500° F., specific gravity above 0.80. Crude petroleum, boiling point uncertain from its mixed constituents, specific gravity about 0.80.

The vapor of commercial gasoline at 60° F. is equal to 1,200 volumes of the liquid, sustains a water pressure of from 6 to 8 inches, and will maintain a working pressure of 2 inches, or equal to any gas service when the temperature is maintained at 60° F., and with an evaporating surface equal to 5¼ square feet per required horse-power, using proportions of 6 volumes of air to 1 volume of gasoline-vapor. Commercial kerosene requires a temperature of 95° F. to maintain a vapor pressure of from ¼ to ½-inch water pressure, requiring a much larger evaporating surface than for gasoline. It may be vaporized by heat from the exhaust, and is so used in several types of oil-engines.

TABLE XII.—PERCENTAGE, SPECIFIC GRAVITY, AND FLASHING-POINT OF THE PRODUCTS OF PETROLEUM.

Products.	Per Cent. of Each.	Specific Gravity.	Flashing-Point, F.
Rhigolene and Chimogene	trace	.....	....
Gasoline } Commercial gasoline	.02	0.650	10°
Benzine naphtha } Commercial gasoline	.10	0.700	14°
Kerosene, light } Commercial gasoline	.10	0.730	50°
Kerosene, medium	.35	0.800	150°
Kerosene, heavy	.10	0.890	270°
Lubricating oil	.10	0.905	315°
Cylinder oil	.05	0.915	360°
Vaseline	.02	0.925	....
Residuum and loss	.16	.....	....
	1.00		

WIDE RANGE OF FUELS FOR HEAVY OIL ENGINES

The following table, prepared from Technical Paper 37, issued by the U. S. Bureau of Mines, shows the wide range of fuel oils that can be utilized in engines using the fuel injection system of combustible gas supply such as the Diesel and various other hot head and high compression ignited forms.

TABLE XIIA.—LIQUID FUELS FOR HEAVY-OIL-ENGINES.

	Specific Gravity.
Petroleum Products:	
Benzine (Pennsylvania)	0.700 to 0.710
Benzine (India)	0.715 to 0.725
Benzine (Roumania)	0.745 to 0.755
Lamp oils and naphthas	0.850 to 0.950
"Solar oil"	
"Gas oil"	0.800 to 0.730
Lima fuel oils, as Eagle oils, gas oil, tar distillates, and Austrian and Russian fuel-oil residues	0.943 to 0.952
Paraffin-oil residues	0.860 to 0.890
Fuel-oil distillates from California and Texas	0.840 to 0.900
Asphaltum oils containing as high as 21 per cent. asphaltum	0.878
Mexican oils	
"Steinkohle" Oils:	
"Steinkohle" oils	1.04
Anthracene-oil distillates	1.1
Tar-oil mixed distillates	1 to 1.1
Bituminous Oils:	
Tar oils (vertical oven)	1.10 to 1.18
Tar oils (horizontal oven)	1.16 to 1.25
Lignite Oils:	
Light benzine	0.800 to 0.820
Solar-oil	0.820 to 0.850
Tar-oil (light)	0.845 to 0.870
Tar-oil (heavy)	0.875 to 0.900
Paraffin-oil distillates	0.898
Cresote-oil distillates	0.957
Turf (peat) Oils:	
Tar-oil distillates as low as	0.8533
Shale Oils:	
Shale oils (Scotland)	0.740 to 0.980
Vegetable Oils:	
Peanut oil	0.916 to 0.920
Cocoanut oil	0.925
Castor oil	0.960 to 0.967
Cottonseed oil	0.913 to 0.930
Palm oil	0.850 to 0.860
Animal Oils:	
Lard	0.913 to 0.919
Alcohol:	
Mixture of 80 per cent. alcohol and 20 per cent. benzine	....
Wood Oils:	
Wood oils or creosote distillates	0.841 to 0.877

All the above oils, and mixtures of them, have been used successfully in heavy-oil-engines, provided they were mobile, free from free carbon, grit and water, and were low in sulphur.

GASOLINE

The gasoline of the American trade varies somewhat in specific gravity from 0.70 to 0.74 as measured by the Baumé scale. Seventy is a light grade and 0.74 was termed stove gasoline from its general use for heating. The analysis of 71 gravity gives carbon, 838; hydrogen, 155; impurities, 007 in 1,000 parts, with a heating value of above 18,000 thermal units per pound. The variation in gravity of gasoline is due to the percentage of hydrogen. The vapor of gasoline is equal to 160 cubic feet per gallon or about 1,200 times its liquid bulk. A saturated "air-gas" of equal parts air and vapor equals 320 cubic feet per gallon of liquid. It is non-explosive and much used as an illuminating gas.

Seventy-four gravity gasoline weighs 6.16 pounds per gallon; its pure vapor is 26 cubic feet per pound and $\frac{18,000}{26} = 692$ heat units per cubic foot. The evaporation of gasoline at atmospheric pressure varies approximately as the relative squares of the temperature; so that in summer, with a temperature of 80° F., the evaporation may be four times greater than in winter at a temperature of 40°. Hence a carburetor may do four times as much work in evaporation, without artificial heat, at one time as at another.

Under the varying temperatures to which carburetors are subject from atmospheric and surface conditions, the more evaporating surface the generator presents, the stronger and more uniform will be the quality of the gas furnished. The boiling-point of gasoline, such as is usually in use for explosive engines, ranges from 150° to 180° F., and the flashing-point of the liquid ranges from 10° to 14° F. The complete combustion of the vapor of gasoline from one pound of the liquid requires 189 cubic feet of air, and as one pound is equal to 26 cubic feet of vapor, $\frac{189}{26} = 7.3$, so that 1 part gasoline-vapor to 7.3 parts air may be said to produce a perfect combustion of the mixture, so that less parts of air may leave a residuum of unconsumed vapor in the exhaust, while an excess of air may add to the fuel efficiency up to a possible limit of 1 part vapor to 10 parts air.

KEROSENE AND CRUDE OIL

Kerosene oil is now taking a front rank among the fuels for explosive power, and crude petroleum is growing in favor as the most economical explosive-power fuel in use. Kerosene-oil motors are largely in the market and a number of concerns are building motors for crude-oil fuel. A "fuel-oil" (distillate) obtained from the residue after the kerosene has passed over from the still, and a grade cheaper than kerosene, is becoming available as an explosive-power fuel. Kerosene has a variable specific gravity from 0.78 to 0.82, a vapor flashing-point at 120° to 125° F., and the oil ignites when heated to about 135° F., and boils at about 400° F. Its vapor is five times heavier than air and requires about 190 cubic feet of air per pound for its complete combustion, or 76 cubic feet of air per cubic foot of its vapor. Its heat of combustion varies slightly from 22,000 B.T.U. per pound. Fuel-oil (distillate) has an average specific gravity of 0.82 and weighs 7.3 pounds per gallon. Its vapor-flashing temperature is at 218° F., and temperature of distillation above 400° F., and it has a heat-unit value of about 18,000 per pound.

Crude petroleum varies considerably in the various parts of the United States in its chemical composition and specific gravity, with an average of 85, C. 14 H, 1.0 in 100 parts, and 0.88 to 0.90 sp. gr. Its heating value is about 20,500 B.T.U.

Crude petroleum and kerosene are available also by injection in a class of oil-engines of the Diesel, Hornsby-Akroyd and Weiss type, in which the oil can be so atomized and vaporized as to make its entire volume available as an explosive combustible, in order that the accumulation of refuse shall be at a minimum. Crude oil is also used in the "Best" oil-vapor and other crude-oil engines by vaporizing the oil in chambers heated by the exhaust of the motor.

Attempts have been made to utilize kerosene with the ordinary spray carburetors of the types widely used in automobile and marine practice, and considerable success has been attained where the vaporizers have been properly constructed with the requirements of this fuel in mind. A complete discussion of the various practical methods of vaporizing kerosene are fully described in the following chapter.

ACETYLENE GAS

Much interest formerly obtained and some experiments were made in regard to the availability of carbide of calcium for generating acetylene gas as a fuel in the motive power of the horseless carriage and launches. Liquid acetylene has been also suggested as the acme of concentrated fuel for power. The gas liquefies at — 116° F. at atmospheric pressure, and at 68° F. at 597 pounds per square inch. Its liquid volume is about 62 cubic inches per pound. The specific gravity of pure gaseous acetylene (C_2H_2) is 0.91 (air 1), and its percentage of carbon 0.923, and of hydrogen 0.077. Its great density as compared with other illuminating gases and the large percentage of carbon is probably the source of its wonderful light-giving power. It is credited by hydrocarbon-heat values with 18,260 thermal units per pound of the gas (14½ cubic feet) and 1,259 thermal units per cubic foot. These figures vary in published statements.

One volume of the gas requires 2½ volumes of oxygen for perfect combustion, which is equivalent to 12½ volumes of air, provided that all the oxygen of the air can be utilized in the operation of a gas-engine; probably the best and most economical effect can be had from the proportion of 1 of acetylene to 14 or 15 of air. This proportion has been used in Italian motors with the best effect. One pound of calcium carbide will yield 5¾ cubic feet of acetylene gas, and requires a little over a half pound of water to completely liberate the gas, so that where weight is a factor, as with motor vehicles, the output of gas will be but 3.83 cubic feet per pound of generating material. The large proportion of air required for perfect combustion makes a favorable compensation for the necessity for carrying water for generating the gas, as compared with gasoline, which yields 26 cubic feet of vapor per liquid pound with its best explosive effect of 9 volumes of air to 1 volume of vapor.

In liberating the gas from carbide in a closed vessel the pressure may rise to a dangerous point, depending upon the clearance space in the vessel, say from 300 to 800 pounds per square inch. In this manner a few accidents have occurred. One pound of liquid acetylene, when evaporated at 64° F., will produce 14½ cubic feet of gas at atmospheric pressure, or a volume 400 times larger than that of the liquid. Its critical point of liquefaction is stated to be 98° F.; above this temperature it does not liquefy, but continues under the gaseous state at great pressures. The heat-unit value of acetylene gas from its peculiar hydrocarbon elements, it will be seen, is far greater than that of gasoline-vapor per cubic foot, but experiments seem to have cast a doubt upon its theoretical value, and assigned a much less amount, or about 868 heat units per cubic foot.

As the comparative volume of explosive mixtures of gas or vapor and air is largely in favor of acetylene over gasoline, and as the weight of material for a given horse-power per hour also favors the use of acetylene, it will no doubt become a useful and economical element of explosive power for vehicles and launches; provided that the commercial production of carbide of calcium becomes available as a merchandise factor in cities and towns. The explosive mixture of acetylene and air spontaneously fires at lower temperatures than illuminating-gas mixtures; it varies from 509° to 515° F., while illuminating-gas mixtures range from 750° to 800° F. Claims of a higher temperature have been made. It is of doubtful availability for high-compression motors.

In the use of liquid acetylene, the cost of liquefying the gas may be a bar to its ordinary use, but for special purposes there are possibilities that only future experiments and trials may develop into useful work from this unique element. In trials of acetylene for power in gas-engines, made in Paris, France, it was found that a much less volume of acetylene was required for equal work with illuminating gas and that it was a practical explosive fuel. The only change required was found to be a more perfect regulation of the valve movement, or a smaller valve to meet the smaller volume of acetylene. In these experiments the explosive mixture was approximately 10 parts air to 1 part acetylene; and using from 4 to 7 cubic feet of gas per horse-power per hour.

From another account of trials in France, it appears, as the result of experiments made by M. Ravel, that 6.35 cubic feet of acetylene gas generates 1 horse-power per hour, which is equivalent to a reduction of two-thirds as compared with petroleum. As to the explosiveness of mixtures of air and acetylene, it was found that 1.35 parts of this gas mixed with 1 part of air began to be explosive, the explosive force of such mixture rising rapidly as the dilution with air increases, attaining finally a maximum when there are 12 volumes of air with 1 volume of acetylene; then as the proportion of air is increased beyond this limit, the explosive force subsides, until at 20 to 1 it becomes entirely extinct. The flashing-point approximates 900° F., whereas in the case of most other gases used to generate power the requisite ignition temperature is about 1,100° F. The temperature of combustion is very much higher than that of the other gases with which it can be compared. The special characteristics of this gas, therefore, are great rapidity of the transmission of flame, low-ignition temperature, high-combustion temperature, and extraordinary energy evolved in the explosion.

For the comparison of gasoline and acetylene, a series of tests were made with mixtures of air and vaporized gasoline in the ratio 4 to 1, which gave the greatest explosive pressure, 165 pounds, at initial pressure of 20 pounds. At the same initial pressure the 9 to 1 mixture of air and acetylene produced a pressure $\frac{273}{165}$ greater than that by the gasoline, so that the volume of acetylene to give the same pressure need only be $\frac{1}{2} \times \frac{165}{273} = 0.304$ of the gasoline.

Taking the theoretical indicator diagrams for the explosion of these two mixtures, the area of the acetylene diagram measured 4.91 square inches, and that of gasoline 1.79 square inches, giving a ratio of power nearly 3 to 1. Indicator diagrams show that the time rate of the actylene explosion is five times faster than that of the mixture of gasoline and air. As vaporized gasoline acts more slowly than acetylene, the practical test makes acetylene (mixture 9 to 1) 3.28 times more powerful than gasoline (ratio of 4 to 1), whereas theoretically it should be only 3 times as great. The calorific value of the acetylene used was 1,350 thermal units and that of gasoline 700 heat units per cubic foot. A cubic foot of each of the above mixtures at initial atmospheric pressure would give 90 pounds and 43 pounds per square inch respectively. Allowed to expand adiabatically to 10 cubic feet, the calculated external work,

$$W = \frac{p_1 v_1}{K-1} \left\{ 1 - \left(\frac{v_1}{v_2} \right)^{K-1} \right\}, \text{ (where } K = 1.405),$$

would be for acetylene 22,403 foot-pounds, and for gasoline 12,132 foot-pounds. But only 0.0625 cubic foot of acetylene was used, while 0.20 cubic foot of gasoline-vapor was needed, or 3.2 times as much. With the given ratios of mixtures only 0.0312 cubic foot of acetylene is required to do the same work that 0.20 cubic foot of vaporized gasoline will do. Or comparing equal quantities of the two gases, acetylene has about 6.5 times the intrinsic energy of vaporized gasoline at the given ratios of air and gas.

Assuming an engine of total efficiency from fuel to useful work of 15 per cent., and a consumption of 22 cubic feet of gasoline-vapor per horse-power per hour, the cost of 1-horse-power hour would be 1.3 cents, at 58 cents per 1,000 cubic feet of vaporized gasoline. The cost per horse-power per hour for acetylene in an engine of equal efficiency would be 2.6 cents, with acetylene $8 per 1,000 cubic feet, or 4 cents per pound. To do the same work with acetylene in place of vaporized gasoline, therefore, would be about twice as expensive. For this reason acetylene would only be of practical use to produce power where gasoline would not be available. In the event of a 50 per cent. reduction in the price of calcium carbide, however, it might probably come into more general use for gas-engines.

ALCOHOL AS FUEL

For some time past the French public has been studying a question interesting from the standpoint of the engineer, important from an economical point of view; the question of alcohol in its domestic and industrial applications. Among the latter the utilization of this combustible in explosive motors is the most interesting, and this is why the experiment has been tried of substituting for imported gasoline a national product resulting from French or colonial crops. One of the unquestioned advantages of alcohol over gasoline is that alcohol is a fixed product, whatever may be its use. The same alcohol for motive purposes can therefore be produced in any part of the globe, and its origin is revealed only by special aromas, which are of no consequence when it is used as a motive force. If the consumption of alcohol-motors is compared with that of gasoline it is seen at once that the former consumes considerably more than the latter; and as the alcohol is the more costly of the two combustibles, the problem would seem *à priori* insoluble from an economic point of view.

Since denatured alcohol contains 4,172 heat units per pound, while gasoline contains 18,000, it has been found necessary to raise the calorific power of the former and at the same time lower its price, and so it has been mixed with high-grade gasoline of 70° gravity, which contains about 18,000 heat units per pound, and which can be produced under good conditions at a low net cost. Mixtures containing from 50 per cent. to 75 per cent. of alcohol have been used; but it is the 50 per cent. mixture, which has a calorific power of 11,086 heat units per pound, which seems to be the most advantageous at the present state of development. From the result of numerous trials made in France it has been found that the consumption of 50 per cent. carburetted alcohol is nearly the same as that of gasoline for a given power, and this notwithstanding the difference in the theoretical calorific powers of the two combustibles, from which it follows that the efficiency of the alcohol-motor is greater than that of the gasoline. Some very exact experiments made by Prof. Musil at Berlin have shown the efficiency of various kinds of motors to be as follows: Motors run on city gas (according to the type), 18 to 31 per cent.; portable steam-motors, 13; kerosene-motors, 13; gasoline-motors, 16; alcohol-motors (mean figure), 23.8 per cent.

The high efficiency is evidently due to the great elasticity derived from the expansion of the water-vapor that is contained or produced by the alcohol at the moment of its combustion, this expansion tending to make the explosions in the cylinders less violent than when gasoline is used, and thus giving a longer life to the wearing parts of the motor. So much has this been found to be the case that in order to increase the beneficial action of the water-vapor the German Motor Construction Company, of Marienfeld, recommends a mixture containing 20 per cent. of water, and it has built motors to run on such a mixture that consume only .17 pound per horse-power hour. The fact must not be overlooked that in order to secure good efficiency with either pure or carburetted alcohol recourse must be had to specially constructed motors having the following characteristics: the stroke nearly double the bore, high compression, and a good spark. Finally, the result of the latest experiments recently made in France on the "Economic" motor, which was specially constructed for use with alcohol, has been a lowering of the consumption to .124 pound per horse-power hour for medium-sized motors, employing a 50 per cent. mixture of carburetted alcohol. For stationary motors the problem is therefore solved.

When it has to do with automobiles the substitution of alcohol carburetted with gasoline is a matter of great interest, for it is evident from statistics that if a liquid containing 50 per cent. denatured alcohol could be used, a large industry would be induced. As the results of late trials in France, the thermal efficiency of the following fuels of power are given: for gasoline, 14 to 18 per cent.; kerosene, 13 per cent.; gas, 18 to 31 per cent., and for alcohol, 24 to 28 per cent. The efficiency of gasoline and kerosene has been greatly improved in the United States in the last few years.

With the use of alcohol, an oxidizing effect has been noticed on valves and seats by the action of acetic acid derived from the occasional incomplete combustion of the alcohol and contained in the large amount of water-vapor from the hydrogen element in the alcohol. This will, no doubt, be overcome by the use of non-corrosive valves and seats made from alloys that resist the action of acetic acid.

THE TAYLOR-WHITE PROCESS OF USING ALCOHOL

While alcohol can be used successfully in engines designed with the requirements in view, owing to its low thermal value it cannot be applied with any degree of economy in motors designed for use with gasoline. An engine designed for the latter fuel will use twice as much alcohol to develop the same amount of energy. A process of recent development that permits one to use alcohol in motors of present design involves the use of a special form of vaporizer. In this, the alcohol vapor passes through calcium carbide before it enters the cylinder. The water which is present in commercial alcohol and which lowers its efficiency as fuel is absorbed by the carbide and the resulting chemical action liberates acetylene gas. As we have seen, this is very explosive and increases the explosive power of the alcohol vapor. When the alcohol-acetylene combination is used it is necessary to add water to the alcohol until a solution containing 17% water and

83% alcohol is obtained. This is not a great disadvantage as water costs nothing to speak of and the increase in bulk of fuel by its addition nearly pays for the carbide. It is estimated that one pound of carbide crystals are used for a gallon of liquid. Alcohol and acetylene vapors combined have proven efficient on motors employing compression pressure as low as 60 lbs. and running over 2000 revolutions per minute. When used alone the slow burning qualities of alcohol vapor has made it most satisfactory on slow speed, high compression engines.

BENZOL AND NAPHTHALINE

Gasoline sells for several times the price that obtains in America at the present time, in England and other foreign countries and therefore the European engineers are experimenting with other fuels. One of these is Benzol, which is said to be adaptable to motors of conventional construction and which also is said to give as much power as gasoline. This material is a by-product incidental to the manufacture of illuminating gas and coke, and while formerly obtained only in small quantities because it was distilled from coal tar, improved methods make it possible to produce about three gallons from every ton of coal changed into coke or gas. The crude product is a foul smelling yellow liquid but when subjected to a refining process it becomes white and loses some of the disagreeable odor.

Repeated attempts have been made in France to use naphthaline or camphor in the shape of moth balls as a fuel for internal combustion engines. The vaporizer employed is very similar to that used with gasoline. In fact two spray nozzles are provided and two float chambers. One is intended to supply the gasoline vapor for starting while the other furnishes camphor gas when the motor becomes heated. The camphor is vaporized by heat derived from the exhaust gases.

The camphor is melted by contact with a separate coil heated by the exhaust gases and after being liquefied it flows to the float chamber through a gauze strainer and the usual pipe connections from the liquefier to that member. After being sprayed from the spray nozzle the liquid further comes in contact with the heated wall of the mixing chamber and is thereby gasified. It is claimed that this material costs about $0.50 per hundred pounds in France when bought in large quantities. A series of tests carried on there has demonstrated that an engine of 15 H. P. will run two miles on a pound of camphor. At that rate the cost

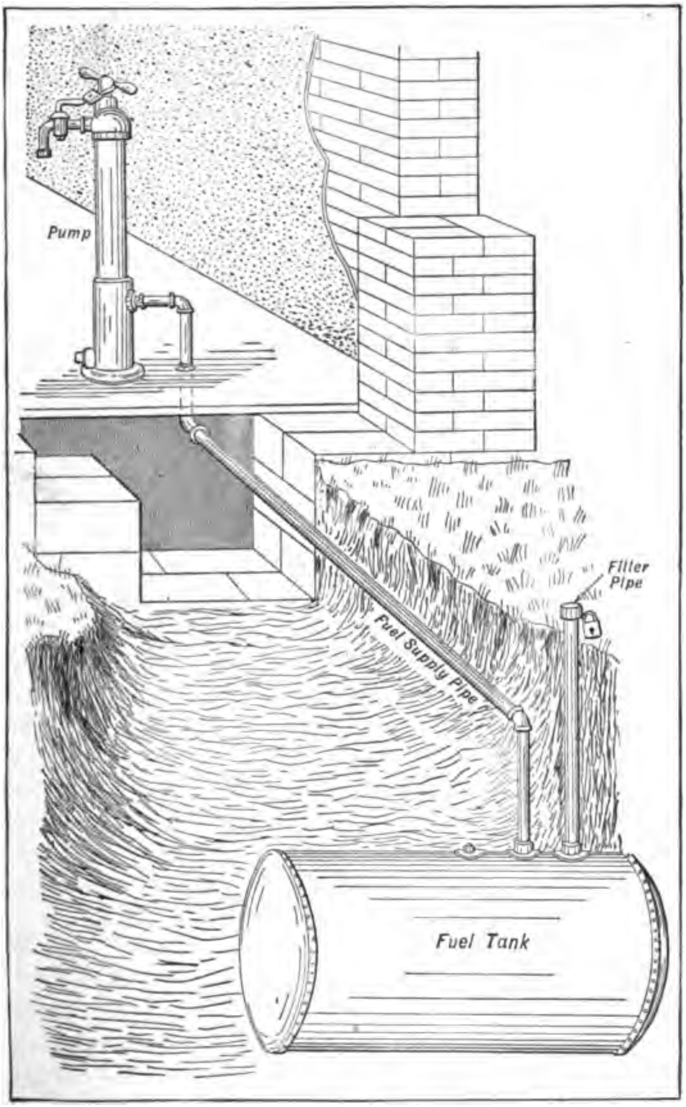


FIG. 26.—Arrangement of parts of approved and convenient fuel storage system

of fuel would be $0.56 per hundred miles as compared with $1.60 for gasoline at present French prices. Thirteen and one-half minutes suffices to heat the engine up enough so that it will run satisfactorily on camphor vapor, and when hot the power plant can be restarted directly on camphor after a stop not exceeding fifteen minutes. Stops of longer duration make it necessary to restart on gasoline.

LIQUID FUEL STORAGE METHODS

The rapidly increasing use of internal combustion motors depending on gasoline and other liquid fuel for the propulsion of all forms of motor vehicles has made it imperative to provide means of storing the large quantities used daily in all of our cities and towns in a way that will reduce the fire risk to a minimum. The regulations of the board of fire underwriters relative to the storage of gasoline and other inflammable liquids are very strict, as only very small quantities are permitted under cover, except in fire-proof buildings. The accepted methods are shown at Figs. 26 and 27. In the former the fuel tank is buried under ground outside of the building while the pump is carried inside where it is easily accessible. In the outfit shown at Fig. 27 both fuel tank and pump are carried outside the building. The pump is an automatic measuring type that may be set to furnish any desired quantity by stops which limit the stroke of the plunger. The containers are of heavy sheet iron or steel, well galvanized or tinned inside and out to reduce corrosion in case of the smaller tanks and thoroughly protected by acid and gasoline-proof paints in the larger sizes. In all cases the filler pipe extends from the tank to make easy refilling possible. The construction of such fuel storage systems, which have the approval of the Underwriters, are so well shown in the illustrations as to make further description unnecessary.

IMPORTANCE OF PROPER FUEL VAPOR AND AIR MIXTURES

It is the consensus of opinion, and so far verified by practical work, that the regulation of the power of the explosive motor has its most economical working condition, first, in the variation of the quantity of fuel injected within certain limits for its highest explosive force with certain mixtures of air: and second, beyond

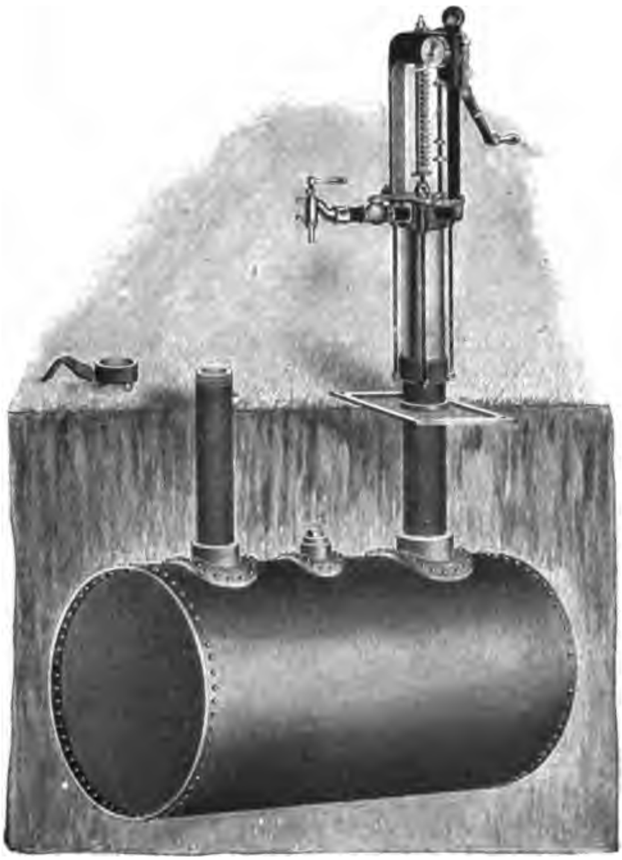


FIG. 27.—Showing construction of Bowser underground gasoline storage tank, with attached measuring pump.

this limit by the regulation of the quantity of the fuel and air mixture in their best proportions for highest effect. It has been stated that mixtures of good illuminating gas, one part to between five and six parts air, give the highest constant volume pressure and the highest temperature by explosive combustion. Also that the time of combustion is quickest under the above proportion. But for all kinds of fuel there is a proportion of air mixture that gives the highest explosive pressure per unit of fuel quantity, and for economic work. This proportion should be retained by the governing mechanism for economic power.

There may be occasions when the over-riding of economical fuel conditions is done for imaginary conveniences in handling high-speed automobiles and launches, which are mostly through misguided judgment in regard to the best conditions of running, or from the ignorance of drivers in regard to the nature of the black clouds of partially consumed gasoline-vapor seen following the track of their vehicles. From the fact that it requires 7.3 parts of air to 1 part of gasoline-vapor for perfect combustion, it is obvious that the feeding of an excess of this fuel is not only a waste, but is also a loss of power, due to decrease of explosive pressure as the proportions are decreased in the charge mixture. The control by the fuel inlet alone should be confined to within the limits of 7.3 of air to 1 of vapor, and 12 of air to 1 of vapor; beyond these limits the control should include both air and fuel for economy.

CHAPTER VII

VAPORIZERS, MIXING VALVES AND CARBURETORS

THE use of the vapor of gasoline, naphtha, and petroleum oil for operating internal-combustion engines is now general in all parts of the civilized world, and will be no doubt the cheapest medium for generating power so long as petroleum and its products do not materially increase in price. In gas-engine running, air saturated with the vapor of gasoline and naphtha is in general use, and when so used is produced by passing air through the liquid or over a surface largely extended by capillary attraction of the fluid by fibrous surfaces dipping into the fluid, by vaporizing the fluid by means of the heat of the exhaust, and by injecting the fluid in small portions into the air-inlet chamber or under its valve, and directly into the clearance space of the cylinder.

Before the development of the very efficient modern spraying mixers and carburetors for gasifying liquid fuels, surface vaporizers were used to furnish gas for stationary engines. These types are practically obsolete at the present time, as the more compact and efficient mixing devices and fuel injection systems render them unnecessary. Such a device is shown at Fig. 28. It is made of wrought iron, has four divisions, in which perforated capillary partitions are set around each division or story of the carburetor, thus greatly enlarging the evaporating surface. The air enters the lower compartment, becomes saturated, and leaves the carburetor from the top. Povision is made for pumping out any residue that may require removal when the carburetor is placed underground. Air thus saturated with gasoline-vapor has a heat value of about 200 heat units per cubic foot.

The evaporation of gasoline of 0.74 specific gravity at a temperature of 60° F. varies somewhat from the form of its elementary constituents, and from the form of the evaporating surface; so that an average of 1,173 grains per square foot of saturated surface per hour in the open air may be assumed as the basis for carbureting surface. When evaporated in a closed vessel, as a carburetor, the vapor may start at about 1,000 grains per square foot of surface per hour; but if the area of evaporating surface is so extended that little or no tension or pressure is produced by its evaporation, due to the draught upon it by the motor, and the temperature of the gasoline is kept near to 60° F., the evaporation may be relied on at about 800 grains per square foot per hour.

This gives a basis for computing the area of carburetted surface at any assumed consumption of gasoline per horse-power per hour. For example, gasoline weighing 6 pounds per gallon, with an assumed requirement of $\frac{1}{10}$ of a gallon per horse-power per hour, and an evaporation of 800 grains per hour per square foot, will require $\frac{\frac{6}{10}\times 7{,}000}{8} = 5\frac{1}{4}$ square feet of evaporating surface in the carburetor per horse-power.

With our present experience there is no doubt in regard to the advantage, economy, and safety in the use of carburetors for gasoline, in which the air becomes thoroughly saturated with the gasoline-vapor before it meets the free air at the charging valve. Air saturated with gasoline-vapor is not explosive, and is considered in practice to be as safe in pipes and gas holders as any other gas used for illuminating purposes. It does not become explosive until further diluted to 5 parts of air to 1 part pure vapor. The mixture of air saturated with vapor of gasoline is in use in some parts of the United States for illuminating purposes, conditioned as to safety and favorable insurance; therefore there is no bar to its use under the same conditions as an explosive element for power. Its safety will always be insured by an excess of evaporating surface in the carburetor.

The grades of gasoline furnished at the present time for fuel purposes are much more difficult to evaporate and are much heavier than the liquids available before the gasoline engine had been as widely applied as at the present time. The more volatile liquids, such as gasoline of .65 to .70 gravity were easily evaporated. The modern fuel, which is about .80 gravity does not evaporate readily and surface carburetors that would be entirely practical with the more volatile grades would not supply gas fast enough or utilize all of the fuel as there would remain some of the heavier constituents that would not be vaporized. The spraying carburetor utilizes practically all of liquid as the heavier elements are sprayed into the mixture with the lighter ones and all are burned.

A point of great value in the economy of fuel has been brought out by German engineers, in trials as to the time of combustion in a cylinder and its relation to the perfection of the mixture of air and vapor. It was demonstrated experimentally that in the ordinary method of mixing a pure gas or vapor with air, ignition at the instant of injection into the cylinder did not produce an instantaneous explosion, but from the first impulse the combustion continued throughout the stroke with a portion of unburned gas in the exhaust. This resulted, as observed, in a reduced initial pressure and consequent reduced efficiency by the indicator card. The continued combustion also increased the heat of the cylinder, as shown by the increase of temperature of a stated quantity of water for cooling a slow-combustion cylinder. It was found experimentally that an injection of equal parts of gas and air into a cylinder required 6 seconds to become fully diffused, and that 1 part of gas to 6 parts of air required from 10 to 12 seconds for perfect diffusion. When, therefore, the time of a single revolution of a gas or gasoline-engine is considered, as compared with the time for charging and compression in a four-cycle cylinder, it will be seen that the mixture cannot become sufficiently intimate to permit the desired instantaneous explosion necessary for the highest fuel efficiency without compressing the charge.

The tendency for augmenting efficiency in gas and gasoline-engine construction is mainly in the line of more perfect mixture of the explosive fuel before injection into the cylinder; and to this we probably owe the possibilities now claimed of from 12 to 14 cubic feet of good illuminating gas, and $\frac{1}{10}$ of a gallon of gasoline per indicated horse-power per hour, and which in most cases has raised the pressure of explosion to 4 or 5 times the pressure of compression in four-cycle engines.

VAPOR-GAS FOR EXPLOSIVE MOTORS

Much of the risk and inconvenience of handling gasoline for motive power may be avoided by using the mixture of air and gasoline-vapor as a gas, and under the same conditions at the motor as with illuminating gas. Many power plants now utilize the vapor of gasoline generated at or in the immediate vicinity of the motor cylinder. This requires the presence of gasoline in quantity within the building, which largely increases the insurance risk, and is always a source of discussion and doubt with underwriters.

The vapor-gas formerly extensively used for lighting dwellings and factories has been brought to such perfection in its generation and application to lighting purposes, as well also to many other applications of heat generated by Bunsen and other forms

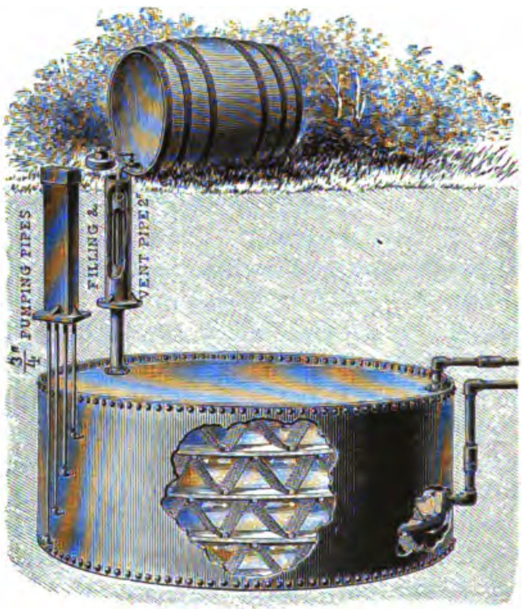


Fig. 28.—Gilbert & Barker surface carburetor.

of gas-burners, that it may now be considered a convenient form for a gas-generating system for isolated places, where an element is required for both lighting and power. The uncertainty of perfect diffusion of vapor and air with the present methods of producing the mixture of vapor and air near or within the cylinder cannot be considered the highest economy in the element of power production, in view of the assumed fact that commercial gasoline of an average of 0.75 gravity, weighing about 6¼ pounds per gallon, is claimed by the builders of the most economical motors to require but ⅛ gallon per actual horse-power per hour. This is equal to 0.78 of a pound, and the pound is credited with 18,000 heat units, or 14,040 heat units per horse-power per hour. This at 778 foot-pounds per heat unit is equal to 10,923,120 foot-pounds per horse-power per hour. The actual or brake horse-power per

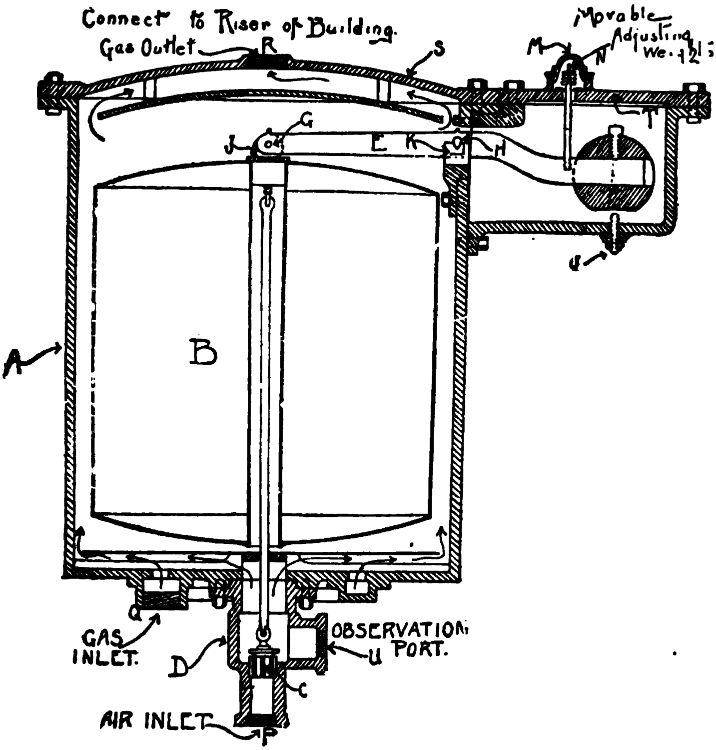


FIG. 29.—The differential gravity regulator.

hour is 1,980,000 foot-pounds or 0.181 per cent. of the theoretical value of gasoline. With more perfect mixtures of vapor of gasoline and air the percentage in efficiency should be increased and a uniformity in the action of the motor obtained by a more perfect diffusion of the elements of combustion.

One of the means for automatically regulating the mixture of vapor and air is illustrated in the Gilbert and Barker combined mixer and regulator shown at Figs. 29 and 30. The mixer and meter air-pump are placed within a building. The carburetor, as shown in Fig. 28, is placed in the ground or a vault outside of the building. The air is forced by the air meter-pump at a low pressure (1 to $1\frac{1}{2}$ inches water pressure) to the carburetor on the outside of the building and returned through another pipe, loaded with the vapor of gasoline, to the regulator, where, by a differential gravity balance, a supplementary valve is opened by

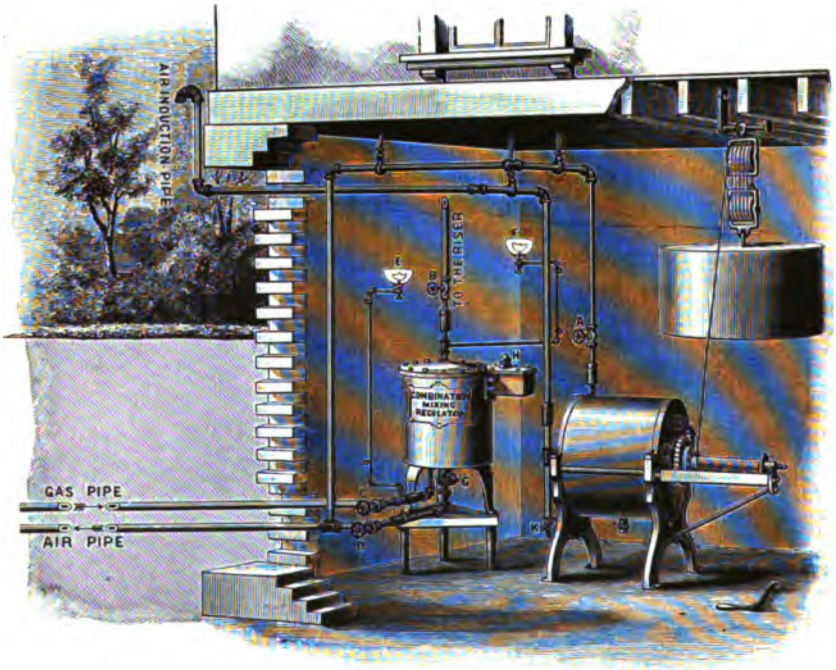


Fig. 30.—The air pump and regulator.

which a direct current of air enters from the pressure-pipe of the air meter-pump and dilutes the direct vapor charge from the carburetor to a uniform mixture, thus producing a constant flow of gas of a gravity for the best effect in lighting, and also, when further diluted at the inlet-valve, for the best explosive effect in a motor.

The pure vapor of gasoline is of a gravity of 2.8 (air 1) and the air-gas vapor as it comes from the carburetor may be of varying gravities from 2.5 to 1.5 (air 1), and it is the difference in the gravity of air and the heavier vapor of gasoline and air as it comes from the carburetor that operates the diluting mechanism of the apparatus to produce a mixture of uniform quality. For this purpose, the float B is a sealed metal can, containing air which with its weight and the air inlet-valve C is exactly balanced

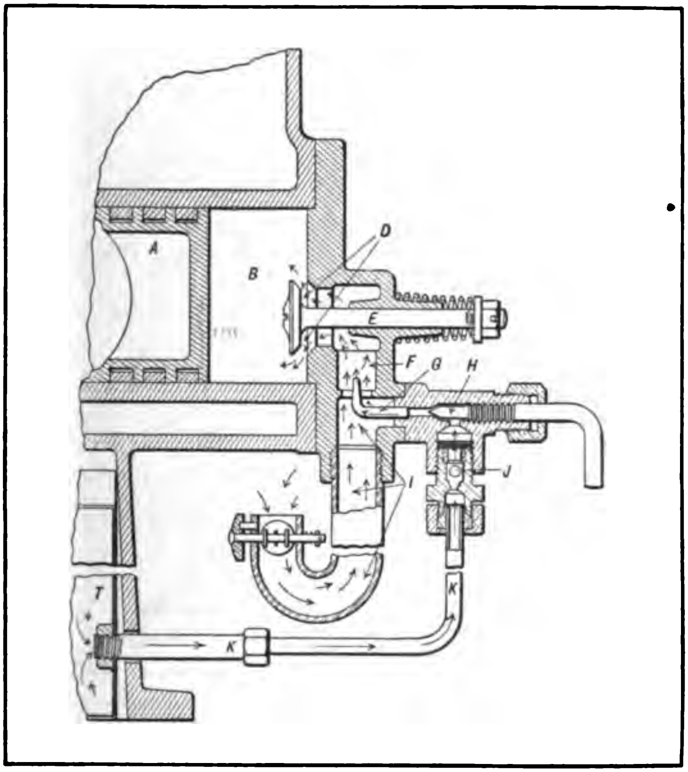


FIG. 31.—Sectional view explaining action of Gray fuel vaporizer.

by an adjustable counterpoise F and enclosed within a cast-iron case. The vapor-gas enters at the bottom through an annular inlet Q from the carburetor and fills the case with a vapor mixture slightly heavier than the balanced can of air, which is thus caused to rise and open the direct air inlet-valve C, admitting air at a slightly increased pressure, due to differential friction, as between the short-air connection with air-pump and the long-pipe connection to the carburetor and back to the regulator.

By the delicate adjustment of the counterpoise weights at M the exact conditions for a uniform gravity gas supply may be obtained for lighting. This is assumed to be also the most economical for combustion in an explosive motor; it then requiring only the regulating admixture of air at the inlet-valve of the motor cylinder for adjusting the force of explosion and for regulating the speed of the motor. Fig. 30 shows the arrangement of setting the air-pump and regulator with the short-circuit of the air-pipe to give a preponderance to the air pressure at the regulating valve C (Fig. 29). For motor service a gas pressure equalizing bag should be used as with other kinds of gas supply. A strong feature of this carburetor, as illustrated at Fig. 28, is the large evaporating surface, it being in fact a compound generator consisting of a number of independent and perfect evaporators, one placed over the other. The effect of cold by evaporation commences at the bottom pan, and the saturation of the air is completed in the next pan, and so on successively, so that deterioration does not commence until the last or top pan is partially exhausted. The air-pump is of the wet-gas meter type with the motion inverted and propelled by a weight as shown in Fig. 30, or by a small overshot water-wheel operated by a jet from any source of water pressure.

SIMPLE MIXING OR GENERATOR VALVES

All devices that will mix gasoline and other more or less volatile liquids and air in proper proportions to produce an explosive vapor are called vaporizers. This general class may be divided into two distinct types of carburetor. The simplest of these are termed "mixing valves" and the principles of operation are very easily understood. The form shown at Fig. 31 combines a suction gasoline feed from a tank in the engine base to the spray nozzle mounted beneath the intake valve. When the suction feed is used the gasoline supplied to the vaporizor stops flowing automatically when the engine stops and starts as soon as the engine is put in action. When the engine piston *A* moves forward a partial vacuum is produced in the firing chamber *B*. This causes the inlet valve *E* to lift from its seat *D* and a current of air is induced into the combustion chamber, through the supply pipe *I* and then past the spray nozzle *G*. The air-pipe is restricted around the spray nozzle so that the air rushing past this point must flow faster on account of the constricted passageway. As a result considerable suction is present at the spray nozzle opening. When the needle valve *H* is opened slightly, gasoline from the tank *T* passes through the fuel feed pipe *K* and is sprayed into the chamber *F* where it mixes with the air current and the resulting vapor is drawn into the combustion chamber *B* past the open inlet valve *E* by the vacuum created by the piston. A check valve is placed in the

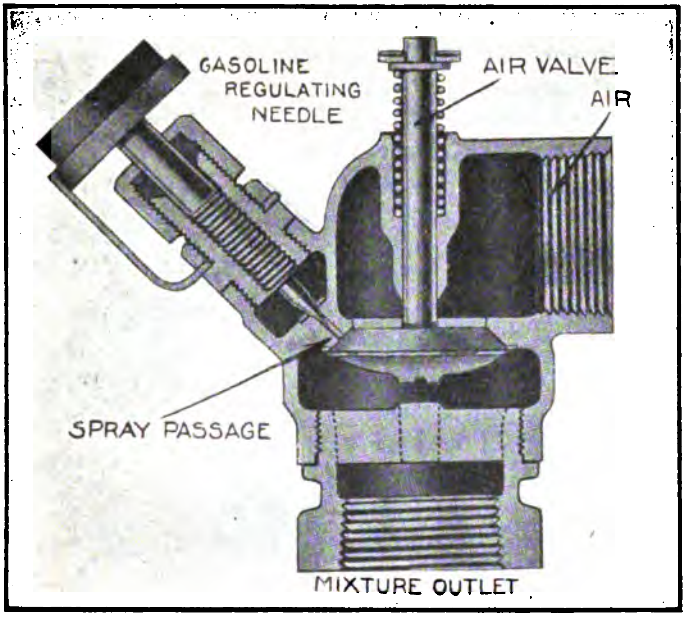


Fig. 32.—Sectional view of simple vaporizer valve.

gasoline line at *J* so the level of liquid will remain at that point while the engine is running. When the piston reaches the end of its suction stroke the inlet valve *E* is closed and the gases in the compression chamber are compacted by the next upward movement of the piston and fired at the proper point by an electric spark.

Another simple form of mixing valve in which the gasoline is supplied to the device by gravity instead of suction is shown at Fig. 32. This consists of a cast bronze body in the form of an elbow having a valve seat machined about half way in its interior. A poppet valve seats against the brass body and separates the device into two parts, as the valve head is held normally in contact with the valve seat by means of a light coil spring. The fuel supply is directed to a small branch member (not shown in illustration) attached to one side of the main body of the mixing valve and having a small passage that communicates with one of the walls of the valve seating member. The area of this passage is regulated by the needle valve which may be turned conveniently by a large knurled head. This passage is normally closed by the head of the mushroom valve. The device is attached to the engine at the connection marked "mixture outlet" and the other open end provides free access for the induced air current. The piston draws in a charge of air on its intake stroke in the manner previously described. This opens the air valve and as that member leaves its seat the gasoline spray passage is uncovered and a stream of gasoline mixes with the incoming air current and is fully amalgamated with it to form a vapor before it reaches the interior of the cylinder. As soon as the inlet valve in the motor closes the air valve in the mixing device it is returned to its seat by the coil spring and both air and gasoline passages are shut off simultaneously. The mixture proportions may be regulated by varying the amount of gasoline supplied, which is done by altering the area of the spray passage with the needle point of the gasoline regulating valve.

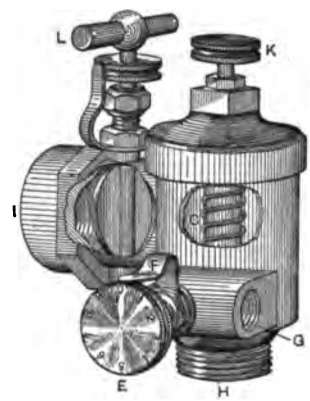


Fig. 33.—Atomizing vaporizer.

A part sectional view of a device constructed on similar lines is shown at Fig. 33. This has an addition not found on ordinary atomizing devices. The charge may be regulated by a throttle valve of the butterfly type which is actuated by a spindle and handle *L*. The air inlet is at *H* while the gasoline enters through the smaller opening at *G*. The needle valve *E* which has a milled index head regulates the opening in the air valve seat and adjustment is retained when the proper point is reached by the spring pointer *F* which registers with a series of figures spaced around the index head. The throttle valve may be held in any set position by the pressure of a spring bearing on the milled disc carried by the spindle.

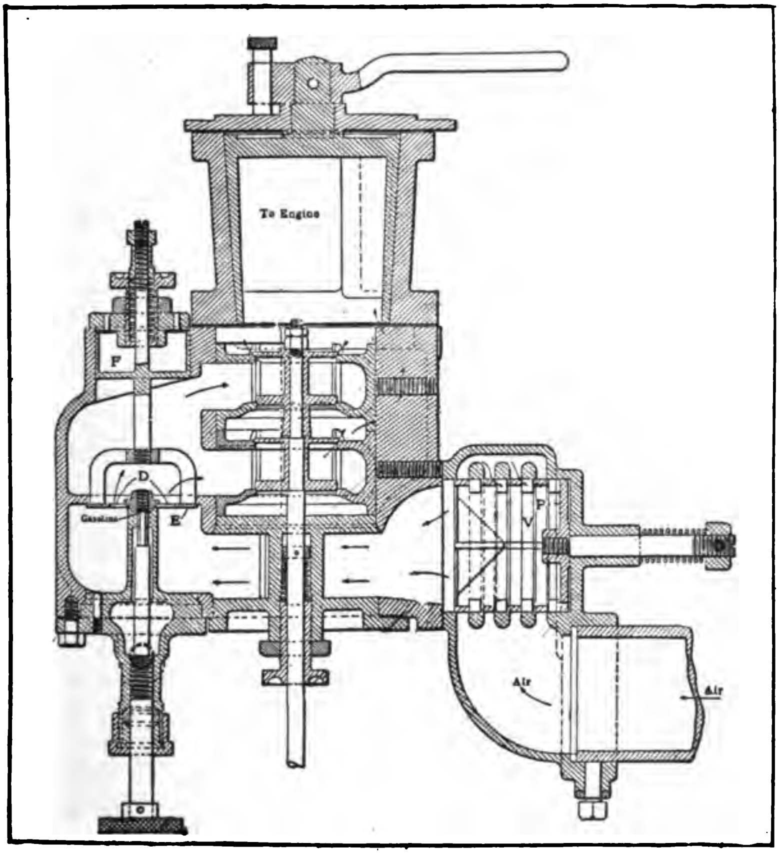


FIG. 34.—Gasoline vaporizer used on I. H. C. gas engines.

The mixing valve supplied on some of the International Harvester Gasoline Engines is shown at Fig. 34. The liquid fuel is introduced into the mixing chamber through a cone-shaped nozzle which reduces it to a fine spray. Air is admitted in direct proportion to the degree of opening to the balanced throttle valve *V*, the movement of which is due to the air piston *P* responding to engine suction. The spring on the horizontal stem draws the valve back to the closed position after each suction stroke. With light loads and when starting, all the air passes through the opening *D* in an auxiliary air valve *E*, the stem of which is attached to the dash pot *F*. All the air then comes in contact with the gasoline vapor and produces a rich mixture that insures regular explosions. With heavy engine loads, however, the increased suction pulls down the dash pot *F* and opens an annular passage around the outer edge of the auxiliary air valve, thus allowing more air to pass through as indicated by the dotted arrows. This arrangement is provided so that, without readjusting the gasoline valves, the amount of gasoline aspirated at different loads will be in proportion to the air admitted and the mixture will not become too rich to fire easily. The amount of gas supplied the engine is regulated by a barrel throttle which changes the quantity by varying the size of the opening leading to the engine.

FLOAT FEED CARBURETORS

While the simple mixing valve forms described are satisfactory for use with engines where no great speed range is desired, such as those employed for marine or stationary work, they are not as reliable and efficient nor do they provide as flexible engine action as the float feed carburetors do on power plants intended for self-propelled vehicles. The main defect of mixing valves is that they are somewhat erratic in action and that the mixture cannot be as well regulated as when float feed carburetors are used. The advantage of the float construction is that the gasoline is maintained at a constant level in an auxiliary chamber at the side of the main mixing chamber and that this level is maintained regardless of engine speed. In the simple forms of generator valves in which the gasoline spray opening is controlled by the poppet-valve a leak in either that member or the valve seat will allow the fuel to flow continuously whether the engine is drawing in a charge or not. During the idle strokes of the piston when there is no suction effect exerted to draw in gasoline vapor, the liquid fuel will collect around the air opening and when the engine does draw in a charge the gas is excessively rich because it is saturated with globules of raw fuel.

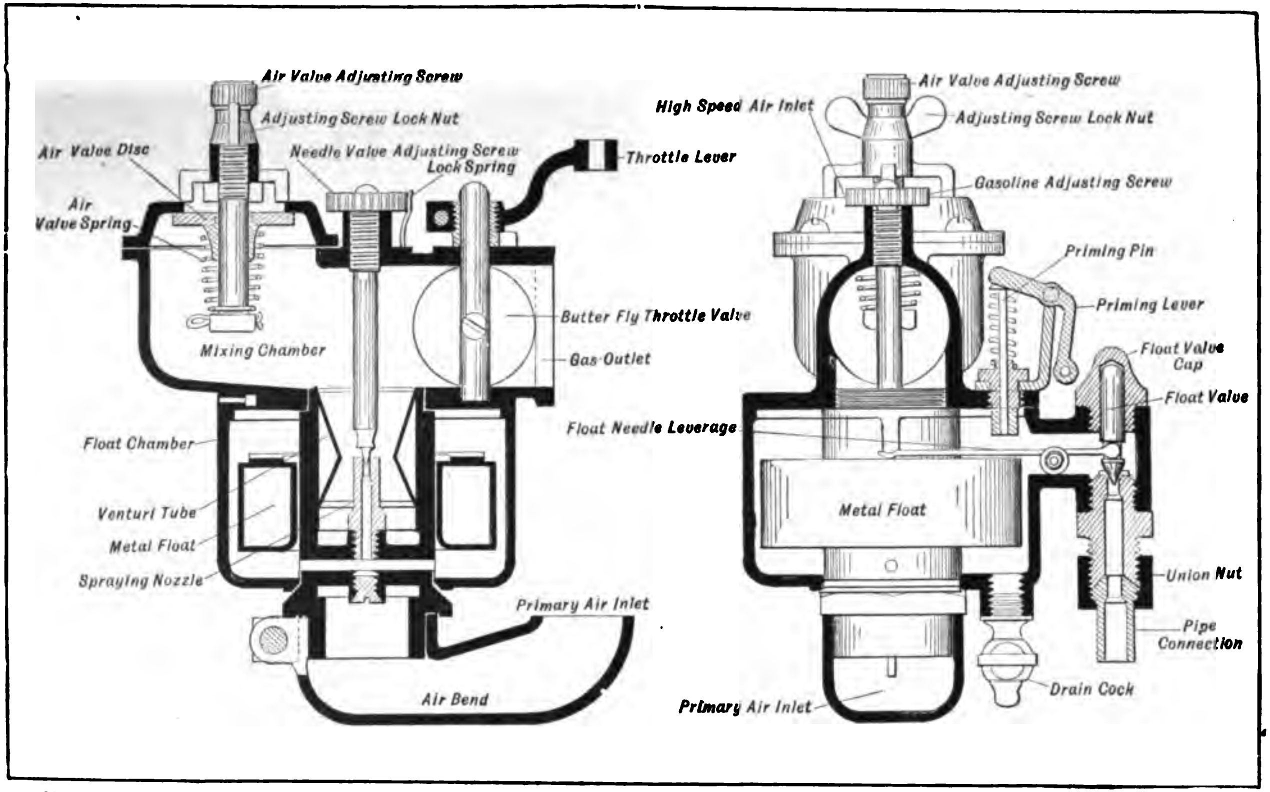


FIG. 35.—Showing details of Breeze carburetor, a simple automatic instrument. Note fuel adjustment needle valve over spray nozzle.

With a float feed construction, a constant level of gasoline is maintained at the right height in the stand pipe and will only be drawn out of the spraying nozzle by the suction effect of the entering air stream. The objection to the simple mixing valves utilizing suction feed is that the tendency is to draw off only the more volatile constituents of the fuel and that after a time the heavier elements comprising gasoline will remain in the container and will not be properly vaporized. With a float controlled spray nozzle the spray is composed of all the elements of the liquid and the lower grade portions that are mixed with those having higher evaporation points are drawn into the cylinder and burnt instead of settling to the bottom of the tank.

A simple form of float feed carburetor having the mixing chamber which is the interior one concentric with the float chamber, or outer member, is shown at Fig. 35. As will be apparent a metal float is carried in the outer compartment which regulates the flow of gasoline through the fuel inlet by a lever operating a needle valve. When the level of gasoline reaches a point about on a line with the upper portion of the spray nozzle the float shuts off the fuel supply by bringing the needle valve against its seat in the supply pipe. The amount of fuel issuing from the spray nozzle is regulated by a needle valve. Extra air is supplied through an auxiliary air valve carried at the top of the mixing chamber. The amount of gas supplied the engine is regulated by a butterfly throttle valve. The action of this device is just the same as that of the simpler forms, as the entering air stream passing through the air bend attains a high velocity through the mixing chamber because of a narrowing of the passage at a point approximately level with the top of the spray nozzle. The flow of air mixes with the atomized gasoline and produces an explosive gas.

Two other forms of concentric float vaporizers in which the gasoline supply is regulated by needle valves are shown at Figs. 36 and 37. The latter differs from the former in the method of admitting the auxiliary air which is through a series of ports controlled by balls instead an opening controlled by a poppet valve. As all parts of the devices are clearly shown, it will not be difficult to understand their principle of action in view of the explanations previously given. The carburetors shown are known as automatic carburetors because an auxiliary air valve is provided to admit air to the mixture at high engine speed at which time the mixture will be too rich in the simple forms of carburetors.

Mixing devices intended for use on motor vehicles or boats utilize the concentric float chamber feature to insure a constant level of fuel at the spray nozzle. If that member is carried at one side of the float chamber in a separate compartment, if the carburetor tilts, as is possible when the vehicle is running on rough

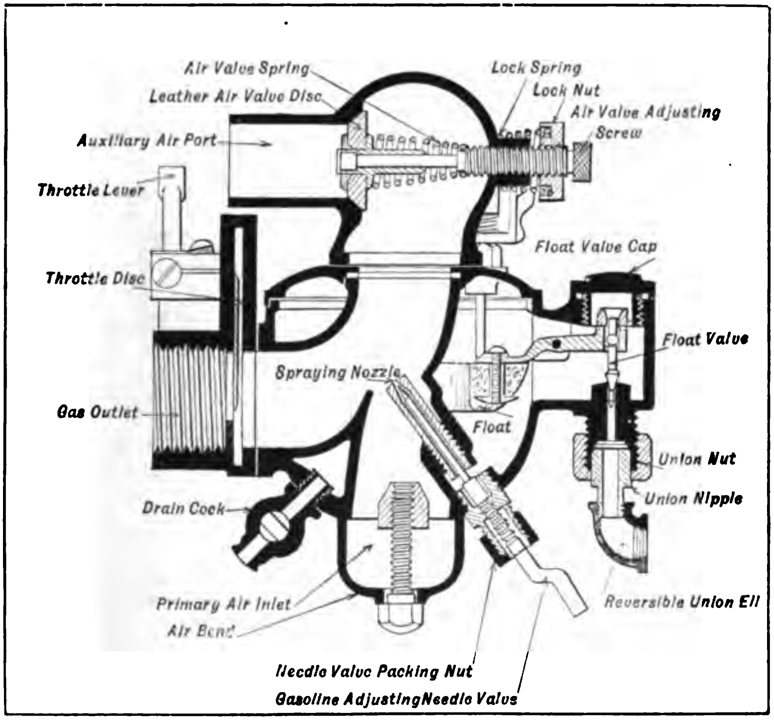


FIG. 36.—Schebler carburetor construction outlined. This is one of the simplest forms that have been used extensively.

ground or climbing hills or when a boat is bobbing around in rough water, the float chamber may be higher than the nozzle of the carburetor. This results in a surplus of gasoline flowing out of the spray nozzle. If conditions are reversed and the spray nozzle is higher than the float chamber the mixture will be too thin because the fuel will not flow readily. When the spray nozzle is placed at the central point of the device no reasonable amount of tilting will change the height of the liquid at that point and a mixture of constant proportions is insured under all operating conditions.

The Krice annular spray carburetor differs from the conventional construction in that the fuel is delivered to the air stream

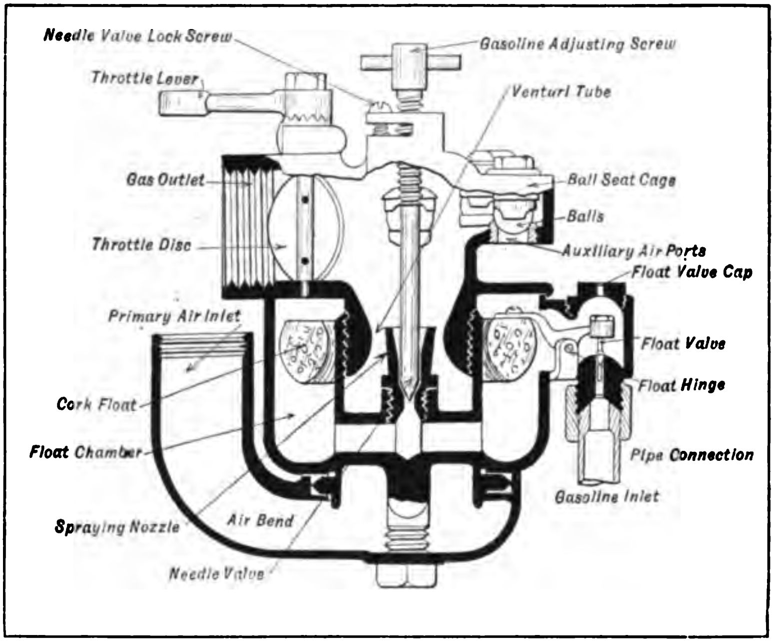


FIG. 37.—Kingston automatic carburetor admits auxiliary air through ball-controlled ports at side of mixing chamber.

by spreading it over the walls of the mixing chamber instead of delivering it in the form of a spray or solid jet controlled by a needle valve. In the sectional view of the device shown at Fig. 38, the application of the annular spraying crevice which is but .008″ to .010″ is clearly depicted. It is claimed that the fuel is spread over the walls and is thus more easily evaporated than if it issued in a solid stream. The internal construction of the mixing chamber is such that the entering air stream must pass over the entire surface of the mixing chamber wall adjacent to the fuel supply crevice and thus become thoroughly saturated with the gasoline vapor. The amount of fuel supplied the mixture is varied by the usual form of needle valve.

Forms of float feed carburetors are produced in which two or more nozzles are used instead of a single spraying member. A two-jet carburetor of simple design is shown at Fig. 39. The small nozzle is called upon to supply fuel only at times when the engine is running slowly. The secondary jet, which provides more gasoline is brought into play when the fuel demand becomes greater due to augmented throttle opening. The Woolsely three-nozzle carburetor is shown at Fig. 40. In this construction, three spraying nozzles are used, one for slow running, one for intermediate speeds and the third for extreme high speeds. The nozzle having the smallest hole is mounted nearest the float chamber and is in the form of a pilot jet that supplies a rich mixture for starting and running the engine at low speed. As the movable throttle barrel is operated, the pilot nozzle is shut off and the gas is drawn from the central mixing chamber. A further movement of the throttle sleeve crosses the central mixing chamber and brings the

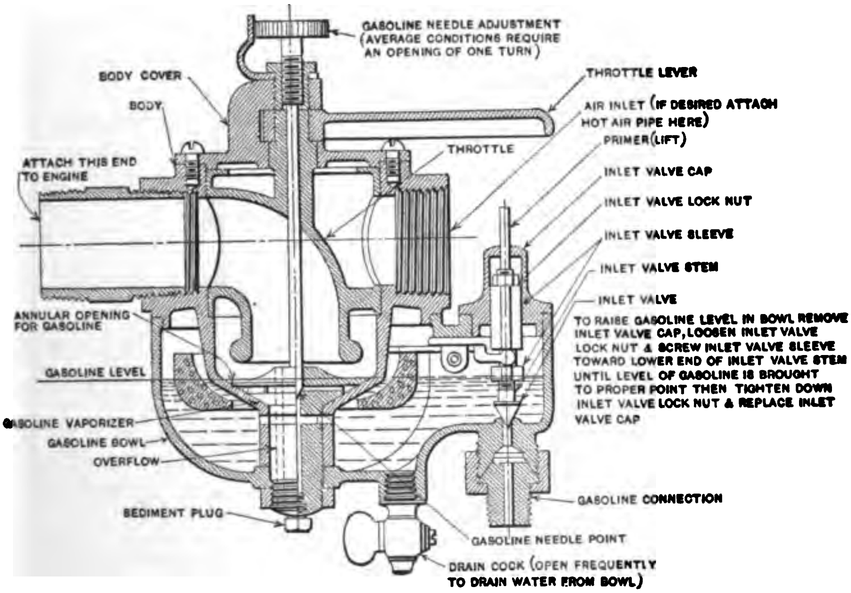


FIG. 38.—Sectional view showing parts of Krice carburetor.

third nozzle into play. An auxiliary air valve is provided at the top of the device to supply extra air to the mixture at such times that it is apt to be too rich. As a rule the multiple jet carburetors are more complicated than the simpler forms and are more apt to give trouble than the forms employing but one spray nozzle. They also require more skill to set them to deliver the proper mixture, though once properly adjusted for a given engine, they will operate very efficiently.

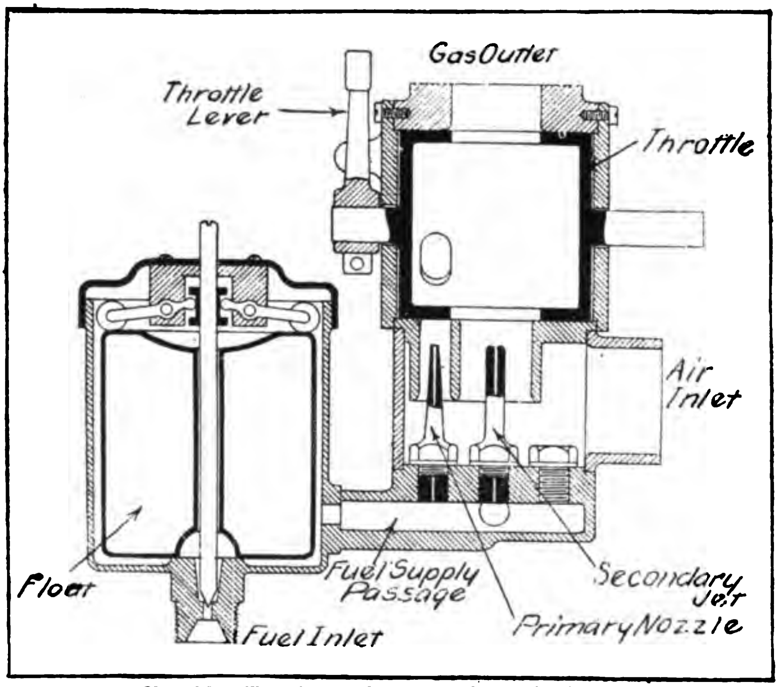


FIG. 39.—Two-jet carburetor of simple design.

KEROSENE CARBURETORS

Kerosene does not evaporate very rapidly and in fact, it will not begin to vaporize at temperatures below 135 degrees to 145 degrees F. Even the poorest grade of gasoline supplied at the present time will start to vaporize under ordinary atmospheric temperatures. The relative non-volatility of kerosene is the property that has proven to be the greatest stumbling block in the development of commercially practical kerosene carburetors. There are a number of methods by which kerosene may be used in engines of conventional design, but practically all of these include attachments to heat the kerosene before it is fed to the carburetor and

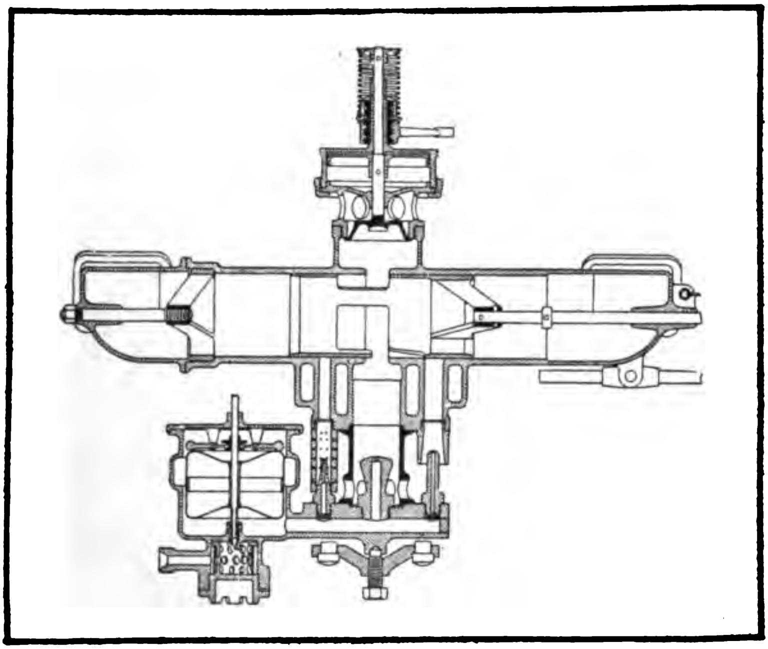


FIG. 40.—Multiple jet gasoline carburetor.

provide heat to the entering air, and lastly to heat the mixture before it goes into the engine cylinder in order to secure a thorough amalgamation of the liquid particles with the air. It is more difficult to apply kerosene to four-cycle engines than to two-cycle forms. The ordinary gasoline motor having a warm air connection running from the exhaust pipe to the carburetor will vaporize kerosene but not efficiently. The motor may be fitted with a large chamber over the exhaust pipe to which an extra carburetor is attached and in which the fuel spray and air are heated before being supplied to the cylinder. This makes it necessary to employ two carburetors, one for gasoline and one for kerosene. The idea is to make the start on the gasoline carburetor and to turn on the kerosene when the engine had become sufficiently heated to vaporize it.

Some engines of the two-cycle type have been operated on what is known as the "balanced" fuel feed in which the kerosene is injected directly against the piston deflector through a jet in the by-pass passage. A typical kerosene mixing device applied to a two-cycle engine is shown at Fig. 41. This is based on the fuel

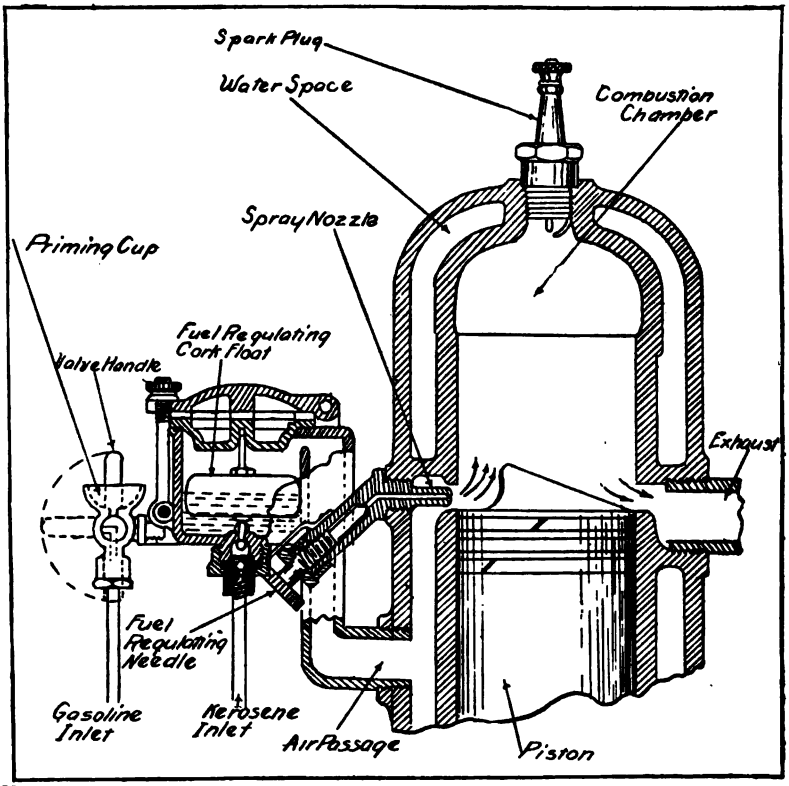


FIG. 41.—Defining method of utilizing kerosene in two-cycle motors by direct injection in cylinder.

injection principle and is very easily understood. The device has two principal parts, a fuel reservoir and the injection nozzle. When the piston in the engine goes up to compress the charge, it produces a suction in the engine base and a condition of partial vacuum in the fuel reservoir which controls the level of fuel. A cork float is used to shut off the fuel supply when the proper height is reached in the reservoir. The float chamber is directly connected with the spray nozzle and as the piston moves down, due to the pressure of the exploded gas against the top, it compresses the air in the crank case and air passage and a portion of this pressure on the surface of the liquid in the fuel reservoir forces liquid to spray through the nozzle and strike the deflector plate on the piston head.

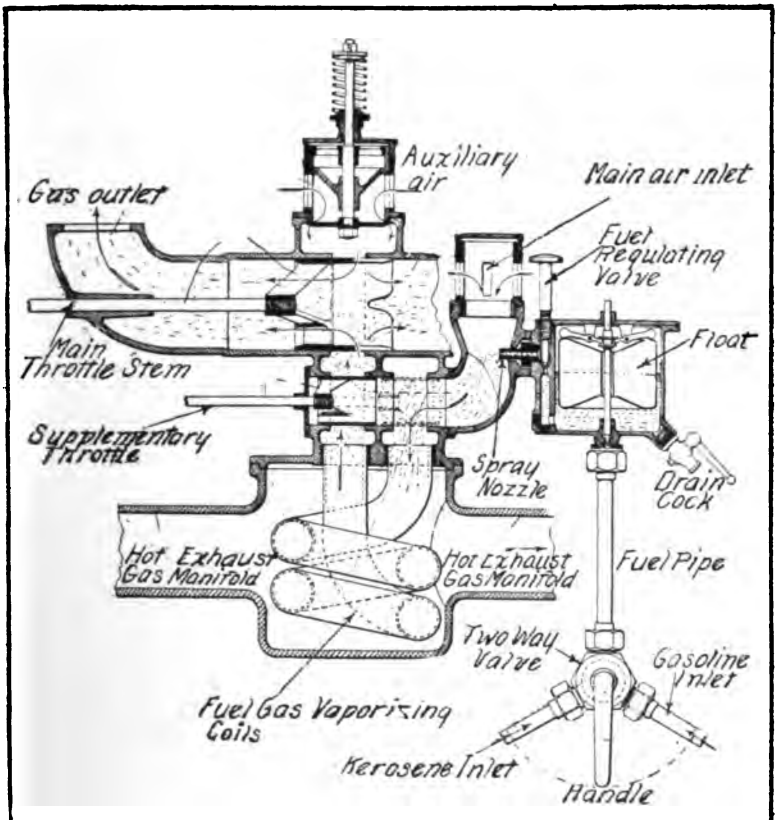


Fig. 42.—Sectional view of Woolsely two-fuel carburetor, which is adapted to vaporize either gasoline or less volatile fuels, such as kerosene.

The amount of fuel supplied is regulated by a needle valve and as the spray strikes the hot deflector it is immediately vaporized and mixes with the air from the engine base to form an explosive gas. It is necessary with this system to provide an auxiliary pipe by which gasoline may be introduced into the fuel reservoir, in order to start the engine and permit it to become sufficiently hot to vaporize the heavier fuel.

The method commonly used of vaporizing kerosene for four-cycle engines is shown at Fig. 42. This is a two-fuel carburetor in which either gasoline or kerosene is supplied the float chamber through separate pipes attached to a two-way valve. When gasoline is used, the supplementary throttle is closed and the gasoline vapor passes directly into the cylinder through the induction manifold controlled by the main throttle. When kerosene is used the supplementary throttle is opened so that no mixture can pass into the

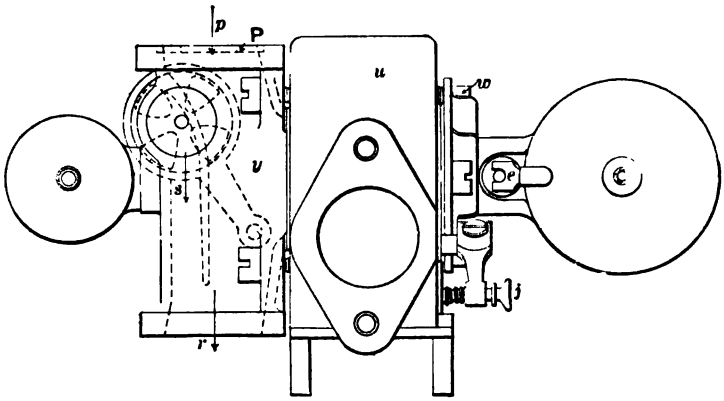


FIG. 43.—Plan and top view of carburetor. (Dotted lines show auxiliary air intake.)

main induction pipe without first passing through the fuel gas vaporizing coils interposed in the path of the hot exhaust gases flowing through the manifold. As in the device previously described, the start is made on gasoline and the kerosene is used only when the engine becomes heated.

THE CLAUDEL OIL-CARBURETOR

A French design by M. Claudel for carbureting air with kerosene or the heavier oils by the heat of the exhaust is shown at Fig. 44. The carburetor is composed of a double heating chamber *u*, in the centre of which is placed the retort *m*. In the annular space included between the retort and the outer walls of the heating chamber, the exhaust from the motor circulates, entering by the pipe *k* and escaping by the pipe *l*. The position of the retort *m* is asymmetric with regard to the centre of the heating chamber, in proportion to the supply and exhaust-pipes *k* and *c*; so that the amount of heat imparted to the retort may be regulated by the movement of the valve *s* in Fig. 44. With the valve in the position shown, the flow of heated gases from the exhaust follows the course of the arrow 2, Fig. 45, being in contact with only a small portion of the circumference of the retort, and imparting but

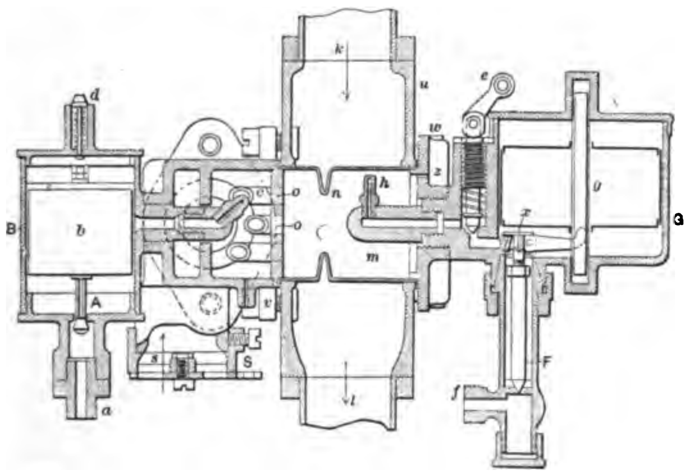


FIG. 44.—Vertical section of Claudel carburetor for heavy oils.

A. Regulator for gasoline supply. B. Gasoline reservoir. F. Stop-valve for oil supply. G. Oil reservoir. P. Damper of main air supply. S. Damper of auxiliary air supply. T. Locking lever of air-damper of exhaust. a. Gasoline supply. b. Gasoline float. c. Gasoline feed-nipple. d. Button for lowering float. e. Independent oil-supply valve. f. Oil-supply pipe. g. Oil float. h. Oil feed-nipple. j. Stop and lever of exhaust-pipe valve. k. Supply pipe from exhaust to carburetor. l. Discharge pipe of exhaust. m. Retort. n. Rib of retort. o, o, o. Mixing pipes from retort. p. Main air supply. r. Pipe from carburetor to motor. s. Auxiliary air supply. u. Heating chamber for retort. v. Drain. x. Adjusting-screw of oil-supply valve. y. Mixing chamber. z. Air-duct to retort.

little heat. With the valve in the position shown by the dotted lines, the current of gas, following the direction of arrow 1, almost completely surrounds the retort.

The difference between the two passages is further increased by a very thin wall on the right of arrow 2, which may be in the form of a screen or damper permitting an ingress of outside air; while the wall on the left of arrow 1 is a part of the casting of considerable thickness, thus retarding the radiation. The air-valve is operated by the lever and spring stop, while the cam lever *T*

(Fig. 45) regulates and locks the cooling damper. By the proper adjustment of these two valves, and the diversion of the exhaust, the retort may be maintained at any desired temperature up to the maximum limit of the exhaust.

The retort is made of drawn tubing, which may be formed with an internal web *n*, increasing the heating surface and breaking the flow of the combustible contents. The retort is connected with the mixing chamber *y* by the tubes *o, o, o,* of such size and form as to act in connection with the web *n* to break up the various elements within the retort and to provide the throttling which is essential to automatic regulation. The mixing chamber is provided with three openings; one for the main air supply, *p;* one for an auxiliary air supply, *s;* and one, *r,* for passage of the mixture to the motor. An internal diaphragm directs the course of the air admitted by *p* and *s,* and regulates the suction according to the speed and other conditions. The opening *s* is fitted with a damper by which the auxiliary supply may be regulated according to the kind of oil used.

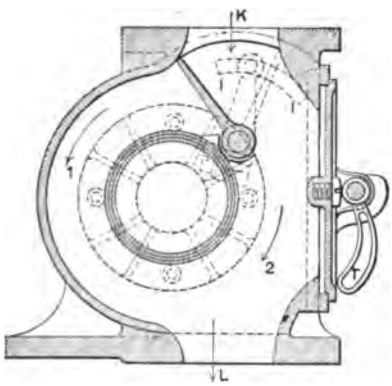


Fig. 45.—Transverse vertical section through retort and exhaust-pipe.

Attached to the mixing chamber is the float chamber B of the ordinary gasoline carburetor, with the float *b,* regulating the level of the gasoline which enters by the tube *a,* and which is discharged into the air of the mixing chamber by the nipple *c,* on first starting the motor. The regulation of the heavy oil supply is through the float chamber G, and float *g,* the oil entering at *f,* under the control of the point F. The float *g* operates a lever, which acts on the upper end of the pointed rod F, the exact adjustment being made through the screw *x* and its nut. Between the discharge-nipple *h,* within the retort, and the float chamber G is a spring valve operated by the lever *e,* by which the passage of the oil may be controlled. A very important detail of the retort is the plate *w,* which connects it with the oil-float chamber, and which is pierced, as shown by a small opening *z*, which admits the necessary amount of free air in proximity to the nipple *h*.

In practical operation, the motor may be started by means of the auxiliary gasoline-carburetor on the left, with a small reservoir for fuel, and when well under way and with the exhaust going, the gasoline may be shut off and the kerosene turned on. The motor may, however, be started directly on the oil, provided a torch is first used to heat the retort until a flow is secured from the exhaust.

The oil supply in the reservoir G is maintained at a constant level by means of the float *g* and its lever acting on the valve F; the rate of feed through the nipple *h* is regulated by the amount of pressure within the retort *m*, which is in turn dependent upon the flow of the gases through the contracted opening of the rib *n* and the indirect passages of the mixing tubes *o, o, o*, which serve to alter the effect of the motor's aspiration and to make it prolonged and regular instead of intermittent. At the lower speeds there is very little resistance to the flow from the retort to the mixing chamber; but as the speed increases and the aspirations of the motor become more powerful, the effect is to throttle the gas in its way through the indirect passages. The result of this apparently contradictory phenomenon is an automatic regulation which is practically perfect. Once set for a given quality of oil, the supplementary air supply *s, s*, may be left without further attention; the air duct *z* of the retort remains unchanged; and the position of the regulating valve *i* in the exhaust-pipe as set by the lever and stop *j* is also unchanged. It has been found in practice that the exhaust supply-pipe *k* should be placed as close as possible to the heads of the cylinders.

KEROSENE CARBURETORS USING WATER SPRAY

A float feed carburetor of conventional design arranged to burn kerosene is shown at Fig. 46. This device is intended for use in connection with the power plant of a gas traction engine and is said to give very good results in practice. It will be noted that the float chamber is surrounded by a jacket through which hot water can enter through the connection at the bottom and pass out through the opening at the side, thus heating all parts of the float chamber and raising the temperature of the liquid to a point where it will evaporate quicker than at ordinary temperature. The kerosene enters the float chamber through the usual pipe connection and the flow is regulated by the form of float control valve standard for this purpose.

In the ordinary automatic carburetor, as the speed increases the tendency is to draw in more fuel and the automatic valve opens to admit more air to the mixture to dilute the rich gas. The carburetor illustrated utilizes the auxiliary valve in a different manner. Instead of the spray nozzle being the conventional form, having but a single opening at the top it is a stand pipe that has numerous fine holes on one side and is called a "spray turret" as

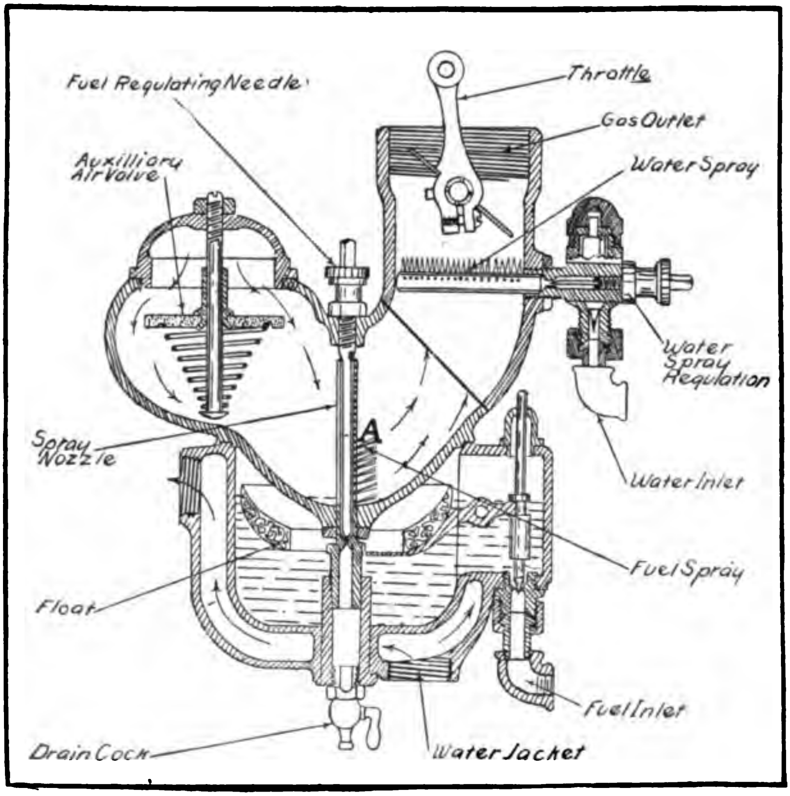
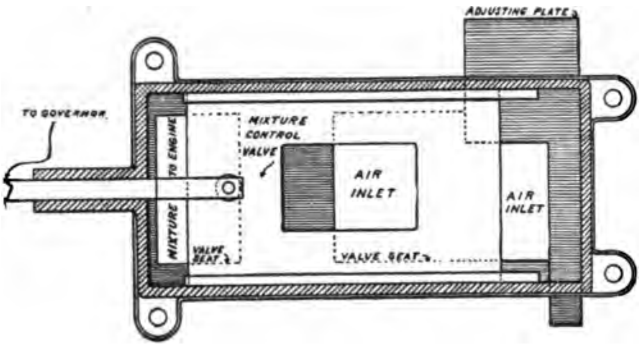


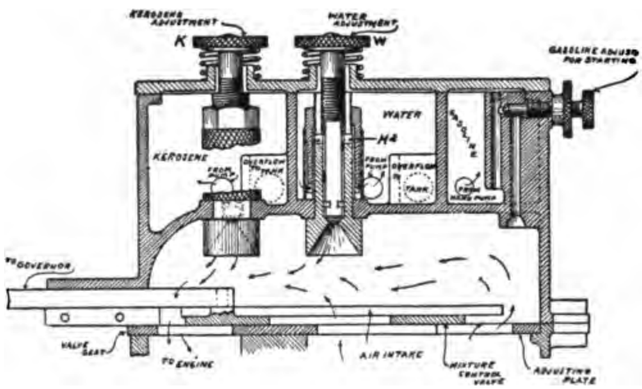
FIG. 46.—Outlining method of utilizing water spray in kerosene vaporizer to facilitate combustion.

shown at *A*. The bottom of the spray member and the upper wall of the float bowl is shaped in such a way that it forms a shallow vessel in which a small pool of liquid fuel may collect. The main air intake is divided and is only sufficient to supply air to run the motor idle. The air that enters the primary intake passes directly over the pool of fuel and a rich mixture is obtained for starting on. The auxiliary air supplied enters through the opening controlled by the air valve and before it enters the cylinders it must pass the spray turret. This lifts the fuel up in the stand pipe where it is drawn through the fine holes in the form of a spray. The greater the amount of air passing through, the higher the fuel rises in the spray turret until when the throttle is wide opened it is sprayed from all the holes in the stand pipe. The amount of liquid sprayed into the mixture is regulated by a needle valve. In order to burn kerosene successfully a certain amount of water is introduced into the mixture through the auxiliary spray nozzle *I*. It is claimed that this supplies an extra amount of oxygen and insures more complete combustion of the kerosene vapor than would be the case if just air was used. To obtain the best results, the induction manifold should be jacketed and a portion of the hot exhaust gases by-passed around it in order to prevent condensation of the vapor and to insure thorough mixing of the air and fuel.

A fuel supply system which is known as the Secor-Higgins is illustrated at Fig. 47. This also is employed on gas traction engines and it is said that it will burn kerosene, distillate and other low grade oils successfully. The system covers an automatic variation in the quantity of fuel mixture in accordance with the slightest variation of speed and load and automatic control of the internal temperature through the admission of water as part of the fuel mixture and thorough and uniform mixture of fuel, water and air charge by mechanical means without the application of additional heat. The inventor of this carburetor claims that the admission of water makes for superior combustion by controlling the temperature of vaporization so there is practically no cracking of the low grade oil with its attendant carbon deposits. The use of water permits a higher compression with kerosene than would be possible without it. It is claimed that the good features are obtained more by the regulation of the temperature and explosion



—Position of sliding valve at light load.



—Longitudinal section of carbureter.

FIG. 47.—Views of Secor-Higgins carburetor, used on Rumely "Oil Pull" tractors. Position of sliding valve at light load at top, side sectional view of device, showing control device, air, and mixture ports, and fuel and water regulation at bottom.

pressure than by the changing of the water into its component gases hydrogen and oxygen. It is not contended that there is no dissociation of water into its component gases as it is believed that the good results obtained by the injection of the water is due in some small measure to the liberation of oxygen. The process of injecting water converts the explosion into a long steady push which is shown by a flatter indicator card instead of a short sharp blow, and careful brake tests have shown an increase of power of at least 15% when water is injected than that of the same engine without it.

The carburetor is mounted above the cylinders and is connected with the inlet valve chambers with a short manifold presenting but little opportunity for the mixture to stratify before it is completely vaporized. It contains constant level chambers for kerosene and water, an overflow being provided for each of these. It also incorporates a chamber for gasoline which is employed for starting and which is connected by a siphon with the mixing chamber. Turning the engine over creates suction enough to draw upon the contents of this chamber but a vent is provided so that if a start is not made immediately the siphon action is interrupted.

The plan view at the top of the illustration shows the position of the valve plate at light load. Two air inlets are opened, providing a large ratio of admission to outlet area and thus greatly reducing the relative vacuum in the mixing chamber.

As the load increases the governor throws the sliding valve forward augmenting the area of the outlet to the cylinder, increasing the air inlet in the middle and decreasing or entirely closing the air opening at the right.

A sectional view from the side at the bottom of the illustration shows the arrangement of the kerosene and water regulating needle valves, the overflow, etc. It will be noted that the water level is lower than the kerosene level. The suction is not great enough to lift the water to point H^2 where it can flow down the tube surrounding the needle valve until the engine reaches about half load. From half to full load the ratio of water to fuel increases rapidly until the amount of fuel and water used are practically equal. The sliding plate is the only moving part in the carburetor and that is positively controlled by the governor. The carburetor is so designed that the fuel needle valve *K* should be adjusted at

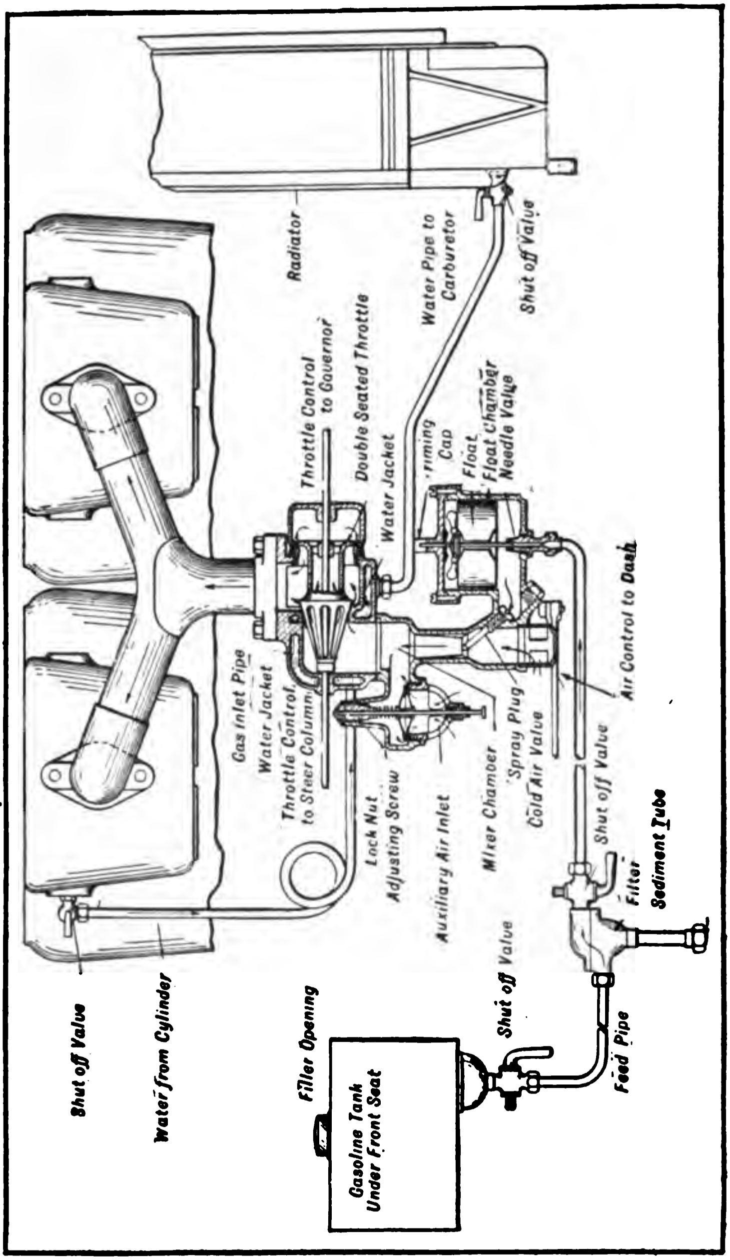


FIG. 48.—Complete fuel system used on some models of Peerless cars, showing method of supplying carburetor with fuel and joining it to cylinders.

the full load position when the plate is furthest to the right. This order of procedure is important because at this position, the adjustable plate has no effect upon the area of the air inlet opening. The adjustment of the air should be made at "no load" position and after once made need never be changed unless the engine reaches a different altitude. This adjustable slide permits each carburetor to be adjusted to suit the engine it is to serve, thus taking care of the slight variations in manufacture.

CARBURETOR INSTALLATION

The usual application of a carburetor and its relation to the complete fuel system in an automobile is shown at Fig. 48. The mixture outlet of the device is connected to the valve chambers

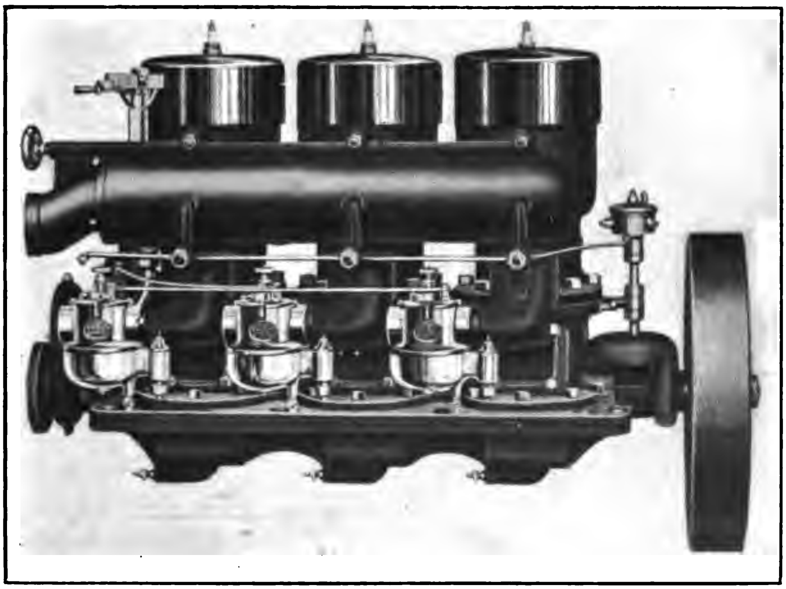


FIG. 49.—Showing application of three carburetors to marine motor.

through a Y shaped manifold having two branches, one going to each valve chamber. A portion of the hot water from the cylinder flows through the water jacket around the mixing chamber and passes to the bottom of the radiator. Shut off valves are interposed at each end of this water pipe in order to interrupt the flow of liquid during warm weather when there is no necessity for heating the carburetor. The fuel supplied from the gasoline tank is first directed to a filter which strains the liquid before it is delivered to the float chamber. Other automobile systems include a tank which is carried lower than the carburetor from which the fuel is drawn by means of pumps or by air pressure forced into the tank against the liquid. In the system shown at Fig. 48, the

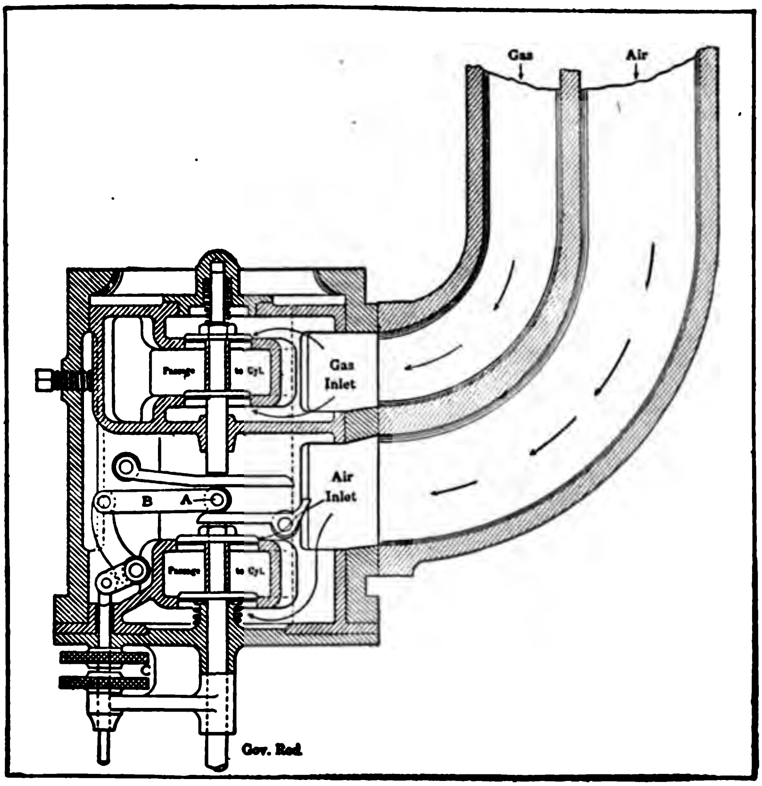


FIG. 50.—Gas and air-mixing valve for use on I. H. C. engines.

gasoline tank must be carried higher than the carburetor and the fuel flows by gravity.

Considerable care is needed in the design and construction of the inlet manifold, especially on multiple cylinder engines having more than four cylinders. Care must be taken to have the manifold so designed that the passages will be of nearly equal length between each cylinder and the carburetor and that these be as direct as possible in order to avoid the condensation of the liquid fuel in the form of globules, which process is promoted by long piping or by pipes having a rough interior or many angles and sharp bends for the mixture to go by in passing from the carburetor to the cylinder.

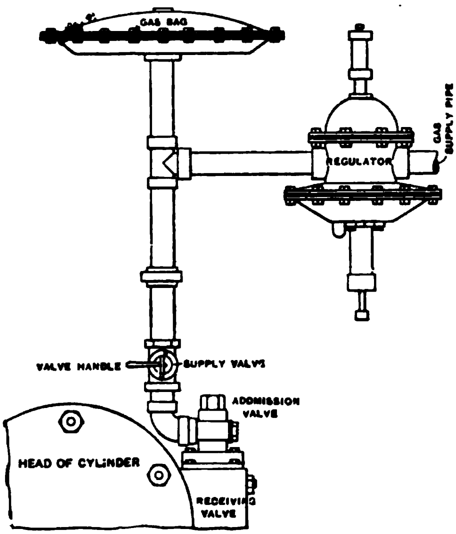


Fig. 51.—Gas-regulator and gas-bag.

The general rule is to use but one carburetor on four-cycle engines ranging from one to six cylinders. Some six cylinder power plants, especially forms designed for marine use or those of the two-cycle type are furnished with carburetors, each device supplying three cylinders. Two carburetors are needed with an eight cylinder engine to obtain the best results and where more than eight cylinders are used it is general practice to provide a separate carburetor for each group of four cylinders. In some marine motors of the two-cycle type it is found advantageous to supply one carburetor for each cylinder, as shown at Fig. 49. The common practice, however, is to use but one mixing device for engines up to four cylinders, in either type. Some difficulty obtains in employing more than one carburetor on the same engine owing to the greater complication and to the difficulty in adjusting each vaporizer to supply a mixture of the same mixture proportions as would be supplied by either one of them individually.

GAS MIXING VALVES AND REGULATORS

Where natural or city gas is used as fuel a combined throttling and mixing valve suffices to produce a correct mixture of the air and gas. When natural gas is used, the compression can be higher than when illuminating gas is employed. The combined throttling and mixing valve used on International Harvester gas-engines is shown at Fig. 50. Both the air and gas valves are of the balanced poppet type and are raised or lowered by the governor reach rod. It will be noticed that a roller *A* mounted on the end of arm *B* is interposed between the gas and air valves and that the position of this is adjustable horizontally through a set of links by the adjusting nuts *C*. When the roller is in the position shown both air and gas valves will have equal lift and when it is shifted over to the right, the lift of the gas valve is less than that of the air valve, and *vice versa*. By means of this mechanism any desired quality of mixture may be obtained within reasonable limits by varying the position of the roller.

In order to eliminate variation in gas charge due to variation in either engine suction or gas pressure a regulator is provided to equalize the pressure of the gas and a gas bag having a flexible wall and with a capacity several times as great as required for a charge of gas in the cylinder is kept filled with the gas. The engine draws from the gas bag and as there is always more gas than is needed in that member a full charge is insured under all conditions where the maximum power effect is desired. A typical regulator and gas bag assembly is shown at Fig. 51. The regulator prevents too rapid gas flow due to its pressure as it is intended to keep the gas flow constant, while the gas bag acts in the nature of reservoir or equalizer to keep the supply from varying as relates to quantity.

CHAPTER VIII

CYLINDER CAPACITY OF STATIONARY GAS AND GASOLINE-ENGINES

THE cylinder volume of gas and gasoline-engines seems to be as variable with the different builders as it is with steam-engines in its relation to the indicated power. The proportion of diameter to stroke varies from equal measures up to 38 per cent. greater stroke than the measure of the cylinder diameter. The extreme volume of cylinder capacity (measured by the stroke) varies from 28 to 56 cubic inches for a 1 horse-power engine and from 48 to 98 cubic inches for a 2 horse-power engine; for a 3 horse-power engine from 77 to 142 cubic inches, while for a 6 horse-power engine it ranges from 182 to 385 cubic inches. This disparity in sizes for equal indicated power may be caused by the different kinds of gas and its air mixtures under which the trials for indicated power may have been made, or it may be partly due to relative clearance and facility for exploding the charge at some fixed time.

It may be readily seen from inspection of the heat value of different kinds of gas — varying as they do from about 950 heat units per cubic foot for the highest quality illuminating gas to from 185 to 66 heat units in the different qualities of producer-gas — that large variations in effective power will result from a given-sized cylinder. It will also be plainly seen that with the extreme dilution of producer-gas with the neutral elements that produce no heat effect, no combination with air that also contains 80 per cent. of non-combustible element can produce even a modicum of power in the same-sized cylinder as is used for a high-power gas. In view of this it seems necessary to build explosive engines with cylinder capacities regulated by the heat-unit power of the combustible to be used, as well as to the method of its application.

In the following tables are given the indicated and actual power, R. P. M. and other data relating to various sizes of gas engines for purposes of comparison.

TABLE XIII.

The Nash.				The Sintz.			
Actual Horse-Power.	Revolutions per Minute.	Diameter of Cylinder. Inch.	Stroke. Inch.	Horse-Power.	Revolutions per Minute.	Diameter of Cylinder. Inch.	Stroke. Inch.
⅓....	350	3	4	1.....	425	3½	3½
½....	350	3½	4	2.....	400	4	4
1......	325	4	4½	3.....	375	4¾	5
2......	300	5	5	4.....	350	5	6
3......	300	5½	6½	6.....	300	5¾	6
4......	300	6	7	8.....	270	6½	7
5......	280	6½	7½	10.....	250	8	8
				15.....	225	9	9

TABLE XIV.

Star.				Dimension Table, Page 110.			
Actual Horse-Power.	Revolutions per Minute.	Diameter of Cylinder. Inch.	Stroke. Inch.	Horse-Power.	Revolutions per Minute.	Diameter of Cylinder. Inch.	Stroke. Inch.
2......	250	4½	6	2.....	350	4½	5½
3......	240	5	6	3.....	350	5	6¼
4......	220	5½	10	4¾...	325	6	7½
6......	220	6½	12	7.....	320	7	8¾
8......	180	7	13	10.....	300	8	10
10......	180	8	14	13.....	275	9	11¼
				17.....	250	10	12½

TABLE XV.—RATING OF SOME ENGLISH ENGINES.

Indicated Horse-Power.	Revolutions.	Diameter. Inches.	Stroke. Inches.	Name.
9.............	164	6	16	Crossley.
9.............	164	8	16	Crossley.
14.............	200	7	15	Crossley.
16.............	160	11½	20	Burt's Otto.
18.............	180	9¼	16	Burt's Otto.
19.............	160	9½	18	Crossley.
20.............	184	9¾	17	Stockport.
20.............	164	12	18	Wells.
24.............	180	10	18	Barker's Otto.
30.............	170	12	20	Barker's Otto.
33.............	210	17	21¼	Crossley.
40.............	160	18	24	Tangye.

The apparent discrepancies in the above table of cylinder capacities, as to their size when compared with their indicated power, are not really so great as may be noticed at first inspection; for the mean pressure varies very much with the various fuels, as well also from the relative variation of the proportion between the volume of the combustion chamber and the volume swept by the piston. The difference in speed between the various engines noted also complicates the direct comparison for cylinder capacities. The whole subject of size and weight of explosive engines for stated powers appears to be still in the experimental stage, which by continued experiment and experience may be brought into an approximate uniformity in practice for specified values of fuel and speed at some later date.

CYLINDER DIAMETER, STROKE, AND MOTOR PARTS

The practice in cylinder proportions in the United States appears to vary considerably among engine builders, from equal diameter and stroke to from 1¼ to 1½ their diameter for length of stroke, while in Europe the smaller-sized engines have strokes of more than twice the diameter, grading to 1½ times in the larger engines. Like the steam-engine cylinder proportions, there seems to be no settled opinion as to the best ratio, except that high speed indicates short stroke. The longer stroke European engines are quoted as low speed and run at from one-half to two-thirds the speed of most American engines of the same power rating.

In the following table of gas and gasoline-engine dimensions we have figured the speed at about the maximum rate and have endeavored to show about the average practice with builders of four-cycle engines in the United States for ordinary power use. The table has been computed for convenient measurement for amateur use and is not intended to meet the exact and decimal values for expert designers. In assigning these values a consideration of 60 pounds M.E.P., with a clearance of from 30 to 35 per cent. of the piston stroke, has been made for the combustion chamber. The tabulated horse-power has been computed on the basis of the M.E.P. of 60 pounds per square inch with an adiabatic compression of $\frac{39}{100}$ of the total volume and a mean back-pressure from the compression stroke of 26 pounds per square inch, which is deducted from the mean of the explosive-pressure stroke of 89 pounds per square inch; which being 63 pounds, from which a deduction of 3 pounds is made for losses from leakage, leaves a net mean pressure of 60 pounds.

Then the cylinder area × mean explosive-pressure — mean compression pressure × impulse stroke travel in feet per minute and product divided by 33,000 = indicated horse-power.

$$\frac{A \times \text{M.E.P.} \times S}{33{,}000} = \text{I.H.P.}$$

To obtain the value of S, multiply the stroke in feet or decimals of a foot by one-half the number of revolutions per minute, which is the impulse travel of the piston per minute. If misfires are made they should be deducted from the half number of revolutions in practice. As an example, considering an 8 × 10 four-cycle engine at 300 revolutions per minute, we have area of cylinder 50.26 square inches and $S = \frac{10}{12} \times \frac{300}{2} = 125$ feet piston travel per minute. Then $\frac{50.26 \times 60 \times 125}{33{,}000} = 11.41$ I.H.P., which we have rated as 10 actual horse-power in the table. In the smaller engines the difference between indicated and actual horse-power increases as the size diminishes.

The thicknesses of cylinder wall, water-jacket, and water space have been assigned with due regard for overcharged explosions and the possibilities in core-making for the water space; they are often made thicker than given in the table. The length of the connecting rod from centre to centre is made from medium practice, or about 2½ times the stroke with the piston-pin at the centre of the piston. The figured dimensions of piston-pins of the same bearing length as the crank-pin, as also the crank-pins and shaft, are derived approximately from formulæ which we find variable with different authorities, as well as variable in size by different builders of explosive motors. The dimensions in the table are a medium suitable to a clearance ratio of 3 to 3.5.

Approximate Dimensions of Four-Cycle Motor Parts.

For M.E.P. 60 lbs. Clearance, 30 to 33 per cent. Compression, 50 to 60 lbs. Explosive Pressure, 160 to 200 lbs.

Table XVI.

Actual Horse-Power.	Revolutions.	Cylinder Diameter.	Stroke.	Clearance.	Thickness Cylinder Wall.	Water Space.	Water Shell.	Length Connecting Rod.	Size Piston-Pin.	Size Crank-Pin.	Width Crank-Pin.	Size Main Journal.	Length Main Journal	Diameter Fly-Wheels.	Weight Fly-Wheels.	Size Inlet-Valve.	Size Exhaust-Valve.
		Ins.	Ins.	Ins	Ins	Ins.	Ins.	Ins.	Ins	Ins	Ins	Ins	Ins.	Ins	Lbs.	Ins	Ins
1/4..	500	2	3 1/2	1	5/16	3/16 x 3/4	Ribs	8	5/16	5/8	5/8	1/2	1	13	66	1/2	5/8
1/2..	450	2 1/2	4	1 1/8	3/8	3/16 x 1	Ribs	9	3/8	3/4	3/4	5/8	1 1/4	15	133	5/8	3/4
3/4..	425	3	4 1/4	1 1/4	3/8	3/8	1/4	9 1/2	7/16	1	1	3/4	1 1/2	17	200	3/4	7/8
1....	400	3 1/2	4 1/2	1 3/8	7/16	1/2	3/8	10 1/4	1/2	1 1/4	1 1/4	1	2	18	270	7/8	1
1 1/2..	350	4	5	1 5/8	1/2	5/8	3/8	11 1/4	5/8	1 3/8	1 1/2	1 1/8	2 1/4	20	475	1	1 1/8
2....	350	4 1/2	5 1/4	1 7/8	1/2	5/8	3/8	12	3/4	1 1/2	1 5/8	1 1/4	2 1/2	23	525	1 1/8	1 1/4
3....	350	5	6 1/4	2 1/8	9/16	3/4	1/2	14 1/4	7/8	1 7/8	2	1 1/2	3	26	575	1 1/4	1 1/2
4 3/4..	325	6	7 1/2	2 3/8	5/8	3/4	1/2	17	1 1/8	2 1/8	2 1/4	1 3/4	3 1/2	32	800	1 1/2	1 3/4
7....	320	7	8 3/4	2 3/4	11/16	3/4	1/2	20	1 1/4	2 1/2	2 3/4	2 1/4	4 1/2	38	900	1 3/4	2
10....	300	8	10	3 1/4	3/4	1	1/2	22 1/2	1 1/2	3	3 1/2	2 5/8	5 1/4	44	1,130	2	2 1/2
13....	275	9	11 1/4	4	3/4	1	1/2	25 1/4	1 5/8	3 1/4	3 3/4	2 7/8	5 3/4	50	1,500	2 1/4	2 3/4
17....	250	10	12 1/2	4 3/8	3/4	1 1/8	5/8	28 1/4	1 3/4	3 3/4	4	3 1/4	6 1/2	64	2,350	2 1/2	3
22....	200	12	15	5	7/8	1 1/8	5/8	34	2	4 1/4	4 1/2	3 7/8	7 3/4	66	3,600	3	3 1/2
30....	175	14	17 1/2	5 7/8	7/8	1 1/4	5/8	39 1/2	2 1/4	4 3/4	5 1/4	4 1/4	8 1/2	72	6,000	3 1/2	4
43....	160	16	20	6 5/8	1	1 1/4	5/8	45	2 1/2	5 3/8	5 3/4	4 5/8	9 1/4	82	9,500	4	5
57....	150	18	22 1/2	7 1/2	1 1/8	1 3/8	3/4	50	2 3/4	5 3/4	6 1/2	5 1/4	10 1/2	96	10,500	5	6

The diameters and weights of fly-wheels vary to a considerable extent among engines by different builders to adapt them to special service where the steadiness of speed is a special factor of design. For electric-lighting purposes, either or both diameter and weight of the fly-wheels may be increased above the tabulated figures, which have been computed for ordinary power service. The sizes of the inlet and exhaust-valves have been figured for a free inlet and discharge at the maximum speed in the second column of the table. For higher speeds of special motors the valve area should be somewhat increased.

Of explosive motors of the larger units now in the market, we detail in the following table some of their most salient features as a study of the progress of this class of prime movers for large power instalments. (See Table XVII.)

Still larger units and installations are built and in use in Europe and in the United States, for the use of blast-furnace gas. The Cockerill type is now built with single-acting cylinders, for blast-furnace gas, up to 600 brake horse-power, and double-acting up to 1,200 horse-power. By doubling up these units any desired power may be obtained in a single installation.

TABLE XVII.

Builders.	Diameter Inches.	Stroke, Inches.	Revolutions per Minute.	Brake Horse-Power.	Clearance, Per Cent.	System of Governing.	Type of Engine.	Weight. Engine Complete, including Fly-Wheels.	Weight. Per Rated Horse-Power.	Fly-Wheel. Weight, Pounds. Total.
Struthers, Wells & Co. (Warren)	21	24	180	300	20	Throttling.	Ver. 2-cyl., 4-cy.	75,000	250	12,000
National Meter Co. (Nash)	13.5	16	225	125	19	Hit or miss.	Ver. 3-cyl., 4-cy.	28,500	228	{ 3,600 / 2,400
The Bessemer Gas Engine Co.	13.5	20	180	100	14	Throttling.	Hor. 2-cyl., 2-cy.	23,000	230	5,800
Marinette Iron Wks. (Walrath)	14	14	250	125	23	Throttling.	Ver. 3-cyl., 4-cy.	23,000	184	6,600
The Alberger Co.	17	19	200	125	21	Auto cut-off.	Hor. 2-cyl., 4-cy.	25,000	200	7,000
Lazier Gas Engine Co.	15	21	160	50	20	Hit or miss.	Hor. 1-cyl., 4-cy.	14,000	280	4,000
National Meter Co. (Nash)	9	11	270	50	22	Hit or miss.	Ver. 3-cyl., 4-cy.	11,000	220	3,600
Westinghouse Machine Co.	18	22	200	300	21	Throttling.	Ver. 3-cyl., 4-cy.	95,000	316	8,600
Westinghouse Machine Co.	8	10	325	38	21	Throttling.	Ver. 3-cyl., 4-cy.	10,500	276	{ 1,750 / 1,150

The double-acting Nurnberg engine is now being built with cylinders of fifty-nine inches in diameter; with duplex tandem double-acting cylinders, in units up to 1,800 horse power. In Germany, blast-furnace gas-engines are in use up to about 2,000 horse-power, in unit combinations of double-acting cylinders of forty-one inches diameter by four and one-quarter feet stroke. The low-explosive pressure of blast-furnace gas has greatly favored large cylinder dimensions, and thus given an impulse to the building of large power-motors with the least number of individual units. Many of the latest developments in the large power units of various types are considered at length in a chapter devoted exclusively to engines of large capacity.

CHAPTER IX

GOVERNORS AND VALVE GEAR

THE regulation of the speed of explosive engines has an important bearing upon their usefulness and freedom from constant personal attention. By experience from trials during the few years of the growth of the new motor, much progress has been made in perfecting the details of this important adjunct of safety and uniformity in speed regulation through the action of a governor. There are four principal methods in use for controlling the speed, viz.: (1) By graduating the supply of the hydrocarbon element; (2) by completely cutting off the supply during one or more revolutions of the crank; (3) by holding the exhaust-valve open or closed during one or more strokes; (4) in electric ignition by arresting the operation of the sparking device.

To vary the quantity of the hydrocarbon fuel by the action of the governor is claimed to be the most economical as well as the most satisfactory method in use, if the variation in the work of the engine does not carry the charge beyond the limit of combustion; otherwise the second method seems to give the best results. In Figs. 52 and 53 are two elevations of the centrifugal ball-governor, as used on the Robey and other engines in Europe, and adopted with many variations on a number of American engines. In this type the bell-crank arm of the governor, by its centrifugal action, raises or depresses a yoke and sleeve which operates a bell-crank lever with a forked end astride a rotating disk which rides on the cam of the secondary shaft. The disk has a lateral motion on the end of the valve lever, so that the action of the governor rides the disk onto or off the cam, and thus makes a hit-or-miss stroke of the inlet-valve.

The centrifugal governor (Fig. 54) is another application of the hit-or-miss principle, by the use of a pick-blade operated from the governor by a balanced bell-crank and connecting rod. The cut fully explains the detail of its construction and operation,

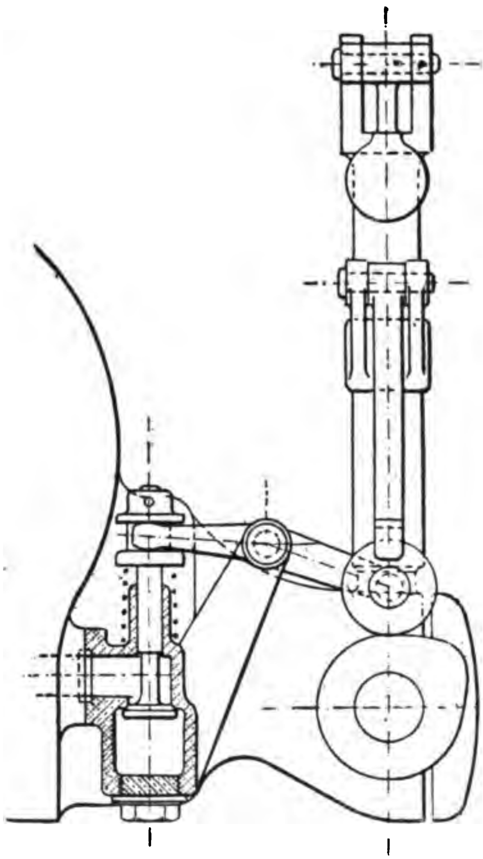


FIG. 52.—The Robey governor.

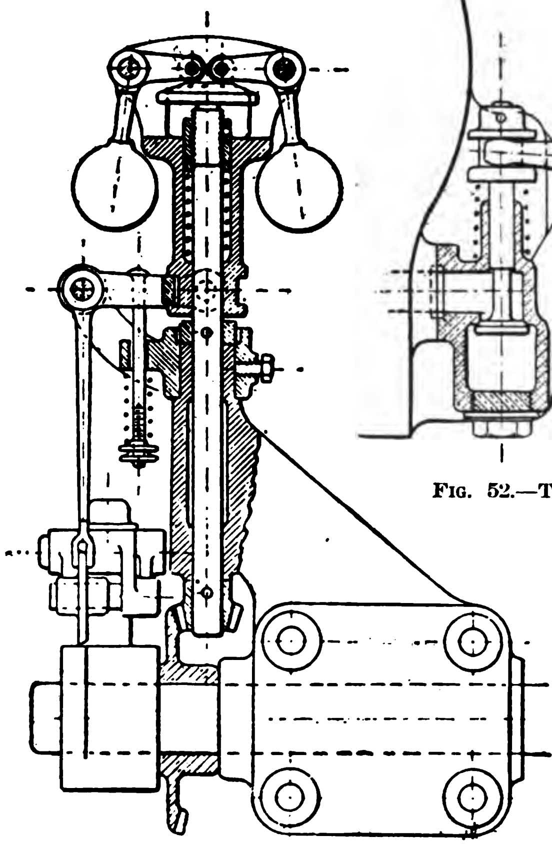


FIG. 53.—The Robey governor.

by which an abnormal speed of the governor pulls the pick-blade away from the gas-valve spindle. In some forms graduated notches are made on the pick-blade or spindle-blade, so that in action the governor gives a varying charge within certain limits and a mischarge when the speed is beyond the limitation.

The inertia governor used on the Crossley engine in England, and with many modifications in use on American engines, is illustrated with plan and elevation in Figs. 55 and 56, in which A is

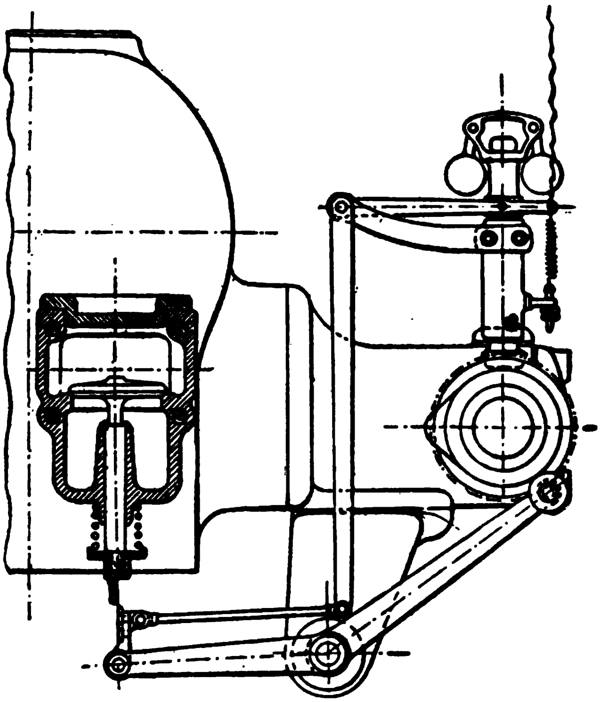


FIG. 54.—The pick-blade governor.

the cam shaft, B the cam, C the roller, D the lever, H the lever-pin, L the spring to hold the roller C to the cam, J the governor weight, K the adjusting spring, G the pick-blade, and F the valve stem. In the action of this governor the initial line of motion of the ball J, in regard to its centre of motion H, is shown by the dotted curved line. By the sudden movement of its pivoted centre L, the ball is retarded in its motion by the regulating spring K,

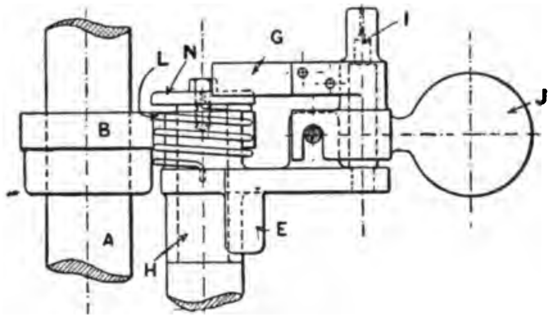


FIG. 55.—Inertia governor, plan. "Crossley."

which tends to throw the pick-blade G off the shoulder of the valve stem F. It will be readily seen that the inertia of the vibrating ball will vary as the speed of vibration, so that by carefully adjusting by the spring K, the action of the ball will vary the

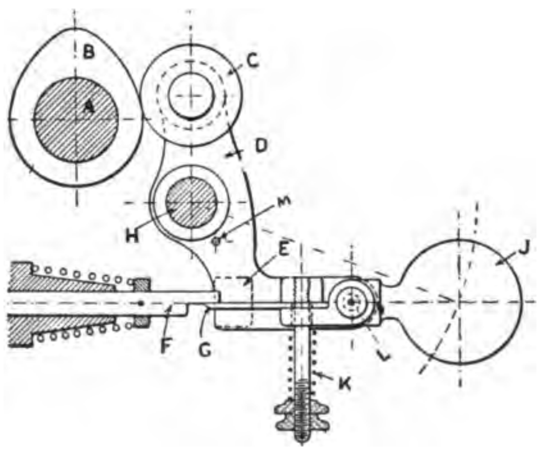


FIG. 56.—Inertia governor, elevation. "Crossley."

disengagement of the pick-blade to correspond with the over-speed of the engine, and make an entire miss at the limit of its variation. The air-valve may also be operated by the spud E.

Another form of governor, involving the same principles of inertia as the last one, is used on the Stockport engine in England, and is illustrated in Figs. 57, 58A, and 58B. It consists of a weight A, balanced on the vibrating arm B. A groove around the weight operates a bell-crank, to which the pick-blade is attached. The balance spring is adjustable for regulating the position of the pick-blade and its contact with the valve spindle. By the

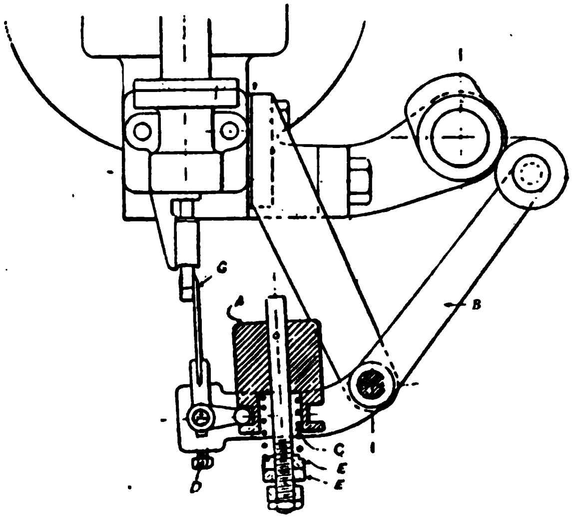


FIG. 57.—The vibrating governor, elevation. "Stockport."

variation in overcoming the inertia of the weight by the spring with different vibrating speeds in the lever, the disengagement of the pick-blade with the spindle-blade is varied or a miss-stroke made.

The pendulum governor (Fig. 59) is also an inertia governor in the principle on which it operates. It is attached to the exhaust-valve push-rod, and vibrates horizontally with the rod. The weight or ball has an extension or neck, with a pivoted eye, a yoke, and a vertical lug. The eye is pivoted in the box, and the yoke embraces the push-blade stem, which is also pivoted horizontally with the eye in the box or frame. The lug bears on an adjusting spring, which is set up by a screw so as to limit the swing of the ball to the normal speed of the engine, so that when the speed rises above the normal the inertia of the ball holds it back in its vibration and lifts the push-blade out of contact with the valve stem. In some engines the position of the ball is reversed, and it stands above the valve push-rod on a finger and is made adjustable in its length of oscillation by its distance from the fulcrum.

Apart from the ordinary methods of operating the valves of explosive motors by reducing spur gear and the reducing screw gear for driving a cam-shaft for four-cycle engines, we illustrate in Fig. 60 and Fig. 61 two very simple methods of operating the charging or exhaust-valve by the direct action of a push-rod from an eccentric on the main shaft.

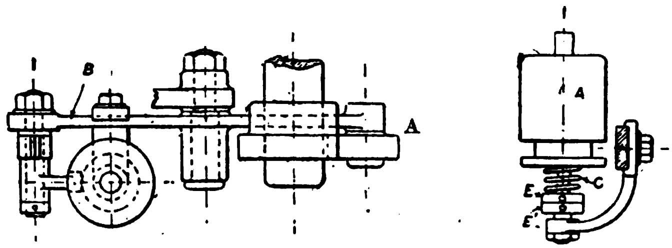


FIG. 58.—A. The vibrating governor, plan. "Stockport." B. End view, elevation. "Stockport."

In Fig. 60 the vertical section shows the form of the cam on the central thread of a two-thread worm on the main shaft with the push-rod and valve. The horizontal diagram shows the worm and intermittent ratchet-wheel pivoted in the fork of the push-rod. At every revolution of the shaft the cam section of the worm falls into a shallow notch of the ratchet and thus gives a push stroke of the valve at every other revolution of the shaft.

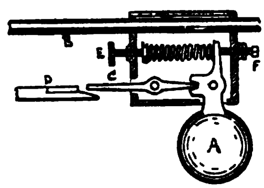


FIG. 59.—The pendulum governor.

Fig. 61 illustrates another form of ratchet push-rod. In this device the ratchet is mounted on a friction-pin which may be adjusted by a thumb-nut and soft washer so as not to turn backward, yet may easily be rotated forward by the motion of the cam-moved push-rod. The upper figure shows the tooth of the push-rod on the shallow notch and missing contact with the valve spindle; at the next revolution of the shaft the tooth catches the deep notch and makes contact with the valve spindle. The throw of the eccentric should be slightly greater than the distance between two consecutive teeth in the ratchet.

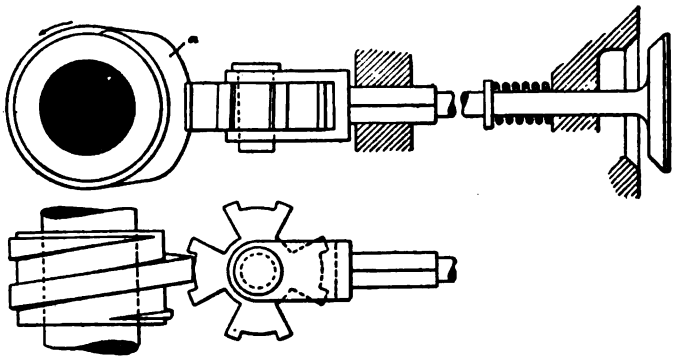


Fig. 60.—The worm cam push-rod.

A governor of the inertia or ball type can be attached to the push-rod with a step contact on the valve spindle, making a very simple valve movement and regulation. The ring-valve gear (Fig. 62A) is another way of operating the exhaust push-rod of a four-cycle engine directly from a cam on the main shaft. The inner-ring gear is swept around within the outer fixed gear, skipping by one tooth at each revolution of the engine-shaft. The outer stationary ring has twice the number of teeth in the ring gear,

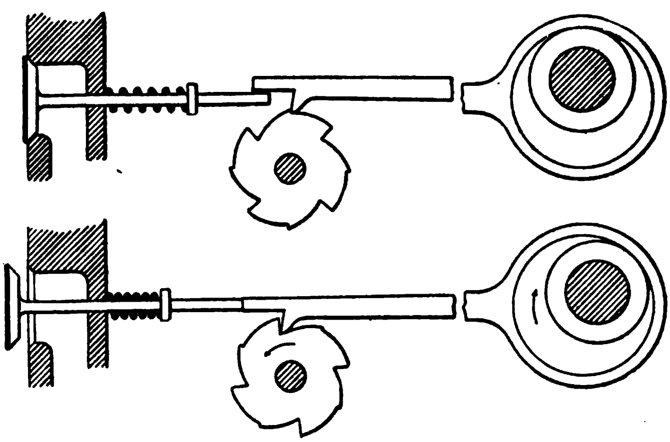


Fig. 61.—The ratchet push-rod.

plus a hunting tooth, which makes a contact of a ring-gear tooth with the exhaust-valve rod at every other revolution.

A double-grooved eccentric (Fig. 62B) is another method of operating the exhaust-valve of a four-cycle engine by traversing the push-rod end, in the grooves which cross each other on one side of the cam; the groove on one section of the cam being enough smaller than the groove on the other section to give the valve its direct proper movement.

The pendulum governor (Fig. 62C) is a simple and unique arrangement derived from the musical beat pendulum. It is hung in a frame that is attached to and vibrates with the push-rod. The swing of the pendulum is adjusted by the distance of the small compensating ball from the centre of motion to vibrate synchronously with the push-rod at the required speed of the engine. Increased speed increases the range of vibration and releases the curved pawl of the push-blade C and catches it again at the next stroke.

The differential cam (Figs. 62D and 62E) is much in use on the Otto engines in Europe and the United States. It is also called the step cam and is made for four grades of valve lift with corresponding differential charge. The centrifugal movement of the governor-balls slides the sleeve on the governor-shaft and through the bell-crank lever the step-cam sleeve *a* on the valve-gear shaft. The disk-roller *b* on an arm of a rock-shaft, rolls upon one or the other cam steps at *c*, thus varying the movement of the inlet-valve, which is connected to another arm of the rock-shaft. The tread of the roller *b* is beveled and the steps of the cam are also beveled to match, so that the roller cannot slip off the cam.

The double-port inlet-valve (Fig. 62F) is one of the methods of mixing the charge of gas or gasoline and air directly into the cylinder. It is made in reverse design and with a groove around one or both the valve disk and valve seat, so that the gas or gasoline may be injected through the seat or from beneath the valve.

In Fig. 62G is shown a gas-engine valve gear in which both valves are operated by an inlet and an exhaust-cam through a bent lever. The form and set of the cams give the proper time and the set-screws in the lever adjust the lift of the valves. E is the inlet-valve; F the exhaust-valve; C, a double cam with groove that rides the sliding roller H alternately on to the inlet and

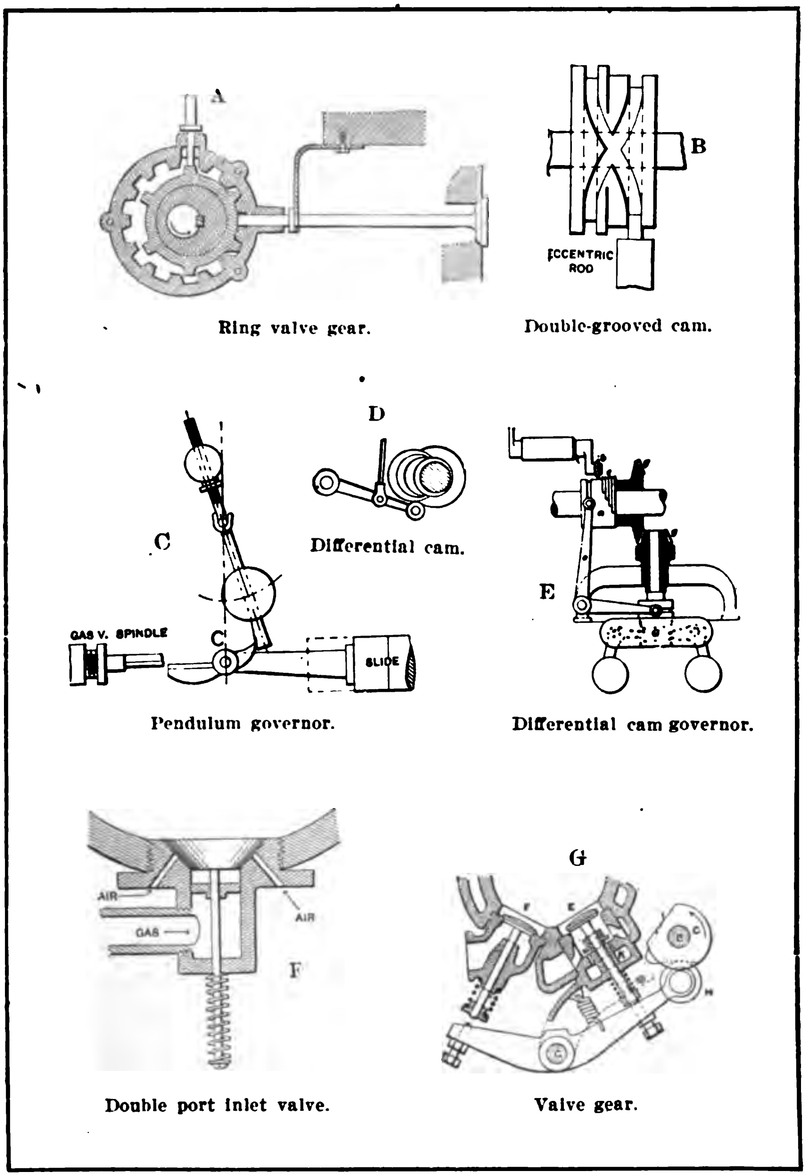


FIG. 62.—A. Ring valve gear. B. Double-grooved cam. C. Pendulum governor. D. Differential cam. E. Differential cam governor. F. Double port inlet valve. G. Valve gear.

exhaust section. The inlet-valve is double seated, the small flat disk covering the gas inlet from the chamber K, the air inlet being between the disks.

The "Union" valve gear (Fig. 63) has a double push-rod. The one for the charge is operated by a cam on the reducing gear with a straight lever to bring the rod in line with the valve. A second cam and lever for the exhaust-rod changes the direction of the push by a bell-crank. The governing device of the Ruger and Olin gas and gasoline engine is of the centrifugal type and consists of two weighted levers L, L (Fig. 64), which operate a small bell-crank and adjustable spindle which rides the push-roller onto or off the exhaust cam, thus holding the exhaust-valve open during excessive speed.

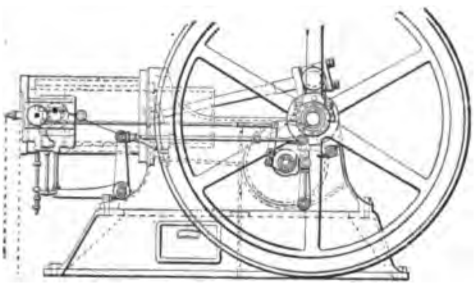


FIG. 63.—"Union" valve gear.

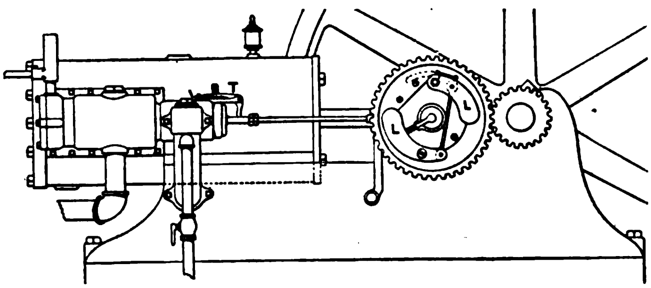


FIG. 64.—Centrifugal governor.

A hydraulic governor of the diaphragm type shown at Fig. 65 is located directly above the water pump of the Packard automobile power plant. It is operated by the pressure of the water in the water circulation system. The governor prevents the motor from racing when the load is removed, as by throwing out the clutch or stopping the car without stopping or shutting down the motor. The governor also tends to maintain a constant speed of the car within the limits of the hand throttle setting, when road conditions

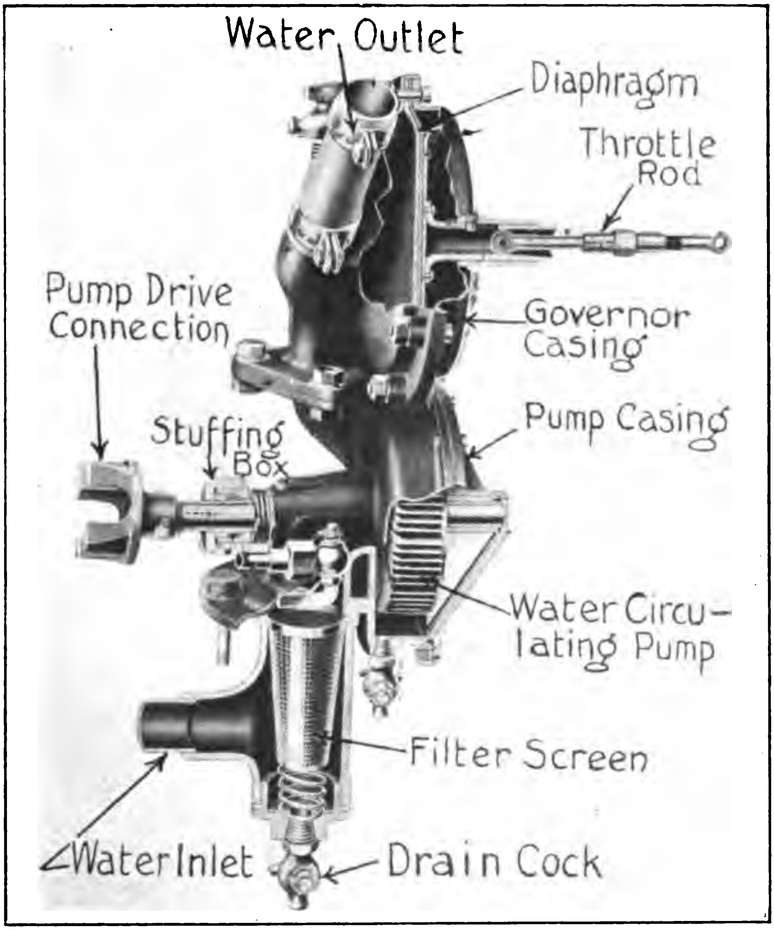


FIG. 65.—Showing construction of hydraulic governor used on Packard motor cars.

vary. It consists of a circular chamber divided by a flexible diaphragm of leather and rubber. On one side of the diaphragm is a water space through which passes the water of the circulating system. On the other side is an air space and a plunger head against which the diaphragm presses. The plunger is directly connected with the throttle valve. If a decrease in the load on the motor causes its speed to increase, the pressure of the water, circulated by the pump, increases and, consequently, the diaphragm exerts more pressure toward the rear, tending to move the plunger and thereby close the throttle. As the motor speed decreases, the

FIG. 66.—Centrifugal governor attached to carburetor.

water pressure against the diaphragm is lessened and the throttle may open. If the load on the motor increases, the opposite action of the governor will result.

CHAPTER X

EXPLOSIVE-MOTOR IGNITION

THE devices for firing the charges in explosive motors have been of many types and designs through the decades of their development; but the early forms using outside flames and sliding ports having been generally abandoned in favor of newer devices, we have therefore omitted their illustration in this edition. The successful operation of the explosive motor depends very much on the perfection of the ignition outfit.

The outside flame gave way to the hot-tube system, which we represent but do not recommend, as it seems to be fast fading in favor of the methods of electric ignition, which seem to fulfil all the requirements for rapid and accurate ignition, as well as for the time adjustment so essential in high-speed motors. For stationary motors many manufacturers still supply both hot-tube and electric combination for gas-engines and a few for gasoline-engines. The various ignition methods are shown in simplified form at Figs. 67 and 68.

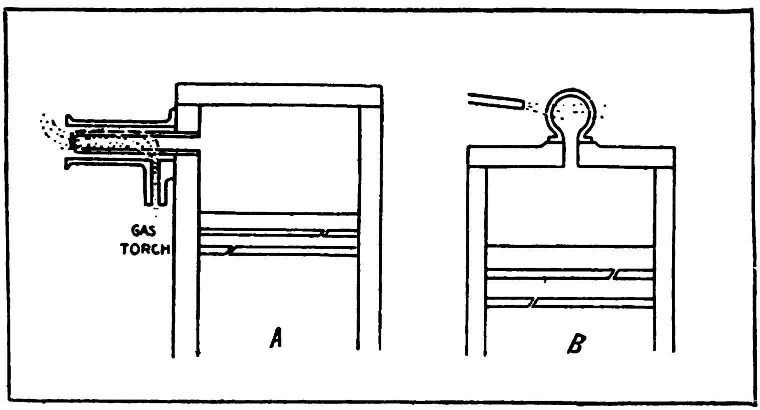


FIG. 67.—Diagrams outlining principal methods of exploding charge by heat. A. Hot tube system. B. Hot bulb method.

HOT-TUBE IGNITERS

Much of the difficulty in maintaining a constant and uniform explosive effect from the hot tubes used in the early or experimental period of the explosive motor was due to the inability to know or see what was the exact condition of the progress of combustion which was taking place within the tube and passage to the combustion chamber of the cylinder.

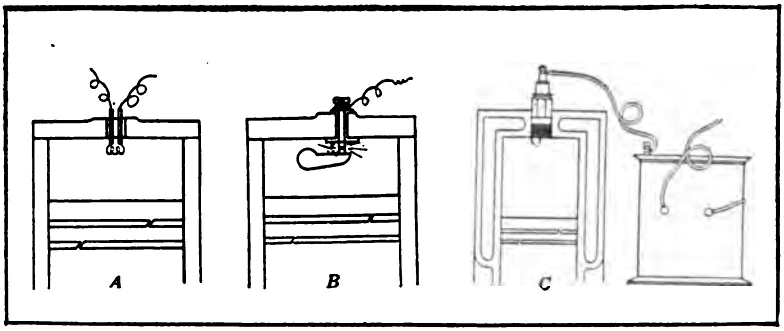


FIG. 68.—Diagrams outlining possible methods of electrical ignition. A. By incandescent platinum wire. B. By make and break or low tension spark. C. By high-tension or jump spark.

The want of a durable and inexpensive material for the ignition-tubes was an unsatisfactory experience in the early days of the explosive motor. The use of iron, with its uncertain and perishable nature, under the intermittent high pressure and at the continual high temperature of the Bunsen burner, oxidized the tubes on the outside, making them thin, so as to burst in a month, a week, or a day; but only occasionally a tube would last a month, although by the use of extra-strong iron pipe their life has somewhat lengthened. One of the principal causes for the short life of the iron tube may be found in the management of the Bunsen burner. A tube of iron or any other metal should not be used at a white heat even at any one spot. A uniform band at a full red heat all around the central or other part of the tube suitable for timing the ignition is the most desirable temperature for ignition, and for the lasting quality of the tube. In the construction and setting of the Bunsen burners, the point of greatest heat in the flame is too often made to impinge directly against the tube, heating it to a white heat at one spot. This causes a change in its molecular condition, weakening it by crystallization and oxidation, when, in a short time, the constantly repeated hammering of the explosions bursts the weakened metal.

Porcelain tubes are free from the oxidizing properties of metals, but require considerable care in fastening them in place. When once properly set their wear is imperceptible, and if not broken by accident, they seem to stand the pressure well and have a life of a year or more at the trifling cost of from 20 to 30 cents for the sizes ordinarily used, and in quantity at a much lower price.

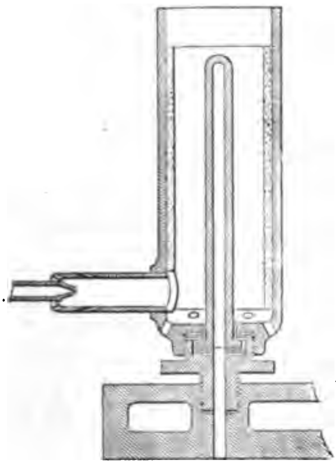


FIG. 69.—Porcelain-tube setting.

The best metallic tubes now on the market are made from nickel-alloy rods. The rods are furnished in about 6-foot lengths, of sizes $\frac{3}{8}$, $\frac{1}{2}$, $\frac{9}{16}$, $\frac{5}{8}$, and $\frac{11}{16}$-inch diameter. This metal was formerly in use by many gas-engine builders in the United States, and its lasting quality has been amply tested by more than a year's wear, and in some cases two years' wear for a single tube. The only trouble or shortening of the running time of the nickel-alloy tubes has been from excessive heating and from sulphurous gas, such as the unpurified producer-gas, and in a few instances from sulphurous natural gas, against which the porcelain tubes seem to be proof. The drilling of the nickel-alloy tubes requires considerable care in order to keep the drill centred in the rod, which is best done by revolving the rod in a dead-rest and feeding the drill by the back centre. Drills should be hard and kept sharp. Use milk for lubricating the drill. The running out of the drill will make a thin side to the tube, which will be liable to overheat, and by expansion and contraction, due to unequal temperature, will cause the thin side to bulge and finally rupture.

Platinum tubes have been used to considerable extent in Germany and a few in the United States; their cost will probably send them out of use in view of the lasting quality and cheapness of the nickel-alloy and porcelain tubes. In Fig. 69 is shown one of several methods for setting the porcelain tube in a socket to be screwed into the cylinder. The packing may be asbestos washers, dry or moistened with wet clay. The application of a new device for hot-tube ignition as used on some of the Mietz & Weiss engines, by which a short and plain porcelain or lava tube, open at both ends and set between sockets with asbestos packing, has made a

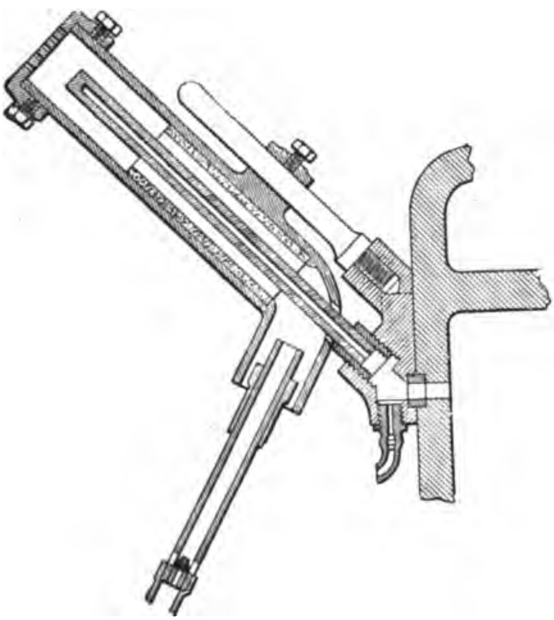


FIG. 70.—Adjustable-tube igniter.

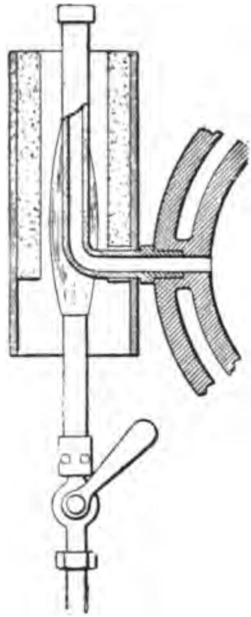


FIG. 71.—Bent-tube

marked progress in simplifying the care and adjustment of tubes and time of firing. A reënforcement of the combustion passage of this device by an iron-pipe extension enlarges the power of the small hot tube by prolonging the burning of the firing charge, and thus making a short tube available to meet the requirement for timing adjustment. Such tubes should last indefinitely; they are cheap, quickly changed, and easily cleaned.

The hot-tube igniter (Fig. 70) shows a view of an ignition-tube used on some of the Robey engines, which is adjustable for the position of the igniting surface of the tube as well as for the position of the Bunsen burner, being combustion chamber, igniter passage, and Bunsen burner pivoted to the chimney frame, which allows the burner to be tilted slightly to regulate the distribution of the flame around the tube. The set-screw in the chimney socket allows of a ready adjustment of the position of the chimney and burner for the time of ignition. Fig. 71 shows a bent-tube igniter of German model.

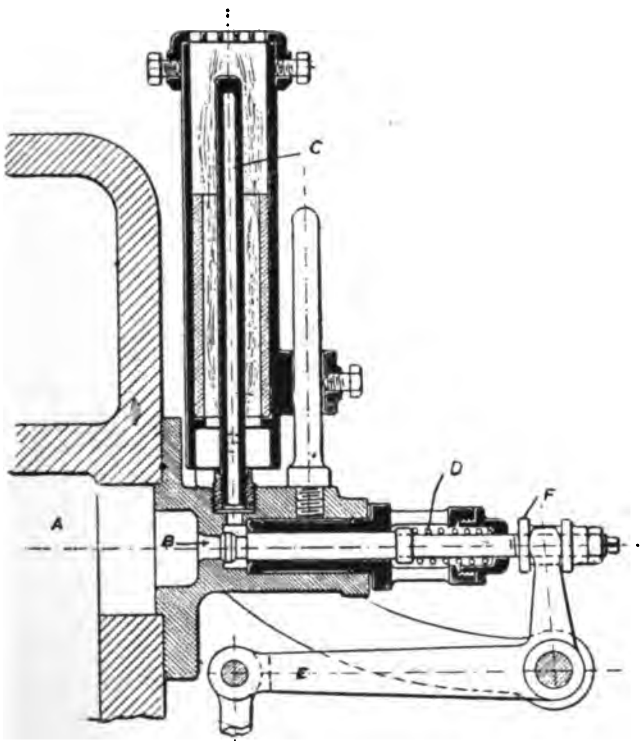


Fig. 72.—Timing valve. "Robey."

IGNITION TIMING VALVES

The value of an exact time of ignition for producing uniformity of speed in explosive engines is attested by the exhaustive experiments of years with the many devices made for the ordinary tube igniters, and the final recourse to electric ignition. A satisfactory result has been obtained in several designs for operating a valve at the mouth of the ignition-tube that admits the compressed charge to the ignition-tube at an exact point in the piston-stroke. In Fig. 72 is illustrated a timing valve used on the Robey engine, in which A is the combustion chamber; B the passage leading to the hot tube, a double-seated valve and spindle held to its front seat by the spring D; E a lever operated from the cam shaft; F adjusting spool with set-nuts. In action the valve is opened

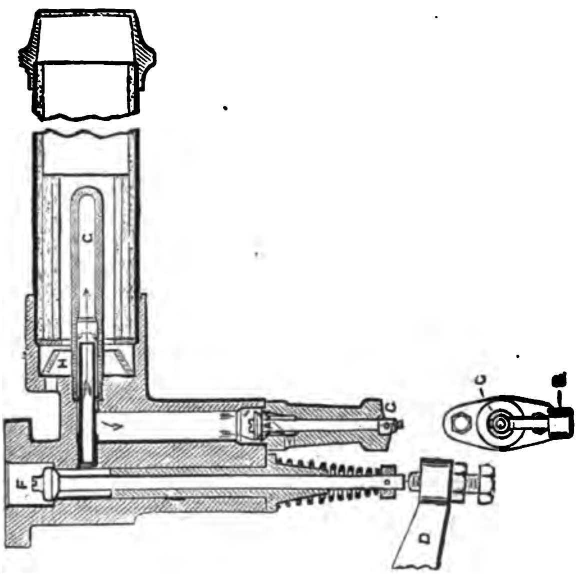


FIG. 73.—Timing valve and starter. "Stockport."

at or about the end of the compression-stroke and kept open during the exhaust-stroke, thus clearing the ignition-tube uniformly and insuring exact time of ignition.

In Fig. 73 is illustrated a combined timing-valve igniter and starter, as used on the Stockport engines. In this arrangement a double tube is used, with an annular space between the inner tube and the hot tube, through which the products of combustion may be blown out, followed by the explosive mixture, into the hot tube, by compressing the timing valve and the starting valve at the same moment. Referring to the cut, F is the timing valve, operated by the lever D; A the starting valve, with its waste outlet at V; H is a mantle to draw the flame closer to the igniting tube. There are many variations in form and attachments for timing valves in use in Europe and the United States.

ELECTRICAL IGNITION DEVICES

The ignition devices have been a puzzle to motor builders and operators during the decades of explosive-motor development, and so-called improvements are still in vogue. For gas-engines, tube ignition has had its day for want of a better way and is still in use to a considerable extent, probably because it is simple and cheap to make; but the short life of the tubes when made of iron has made this material unreliable and the resort to a nickel alloy and porcelain has bettered a condition which still has its annoyances. Electric ignition has become the most reliable and is easily managed and adjusted to meet the requirements for timing. In its best designs it has been largely adopted by motor builders, and has become a favorite with motor engineers. Notwithstanding the troubles with early designs of electric igniters, from unseen causes due to the hidden position of their vital parts, the later improvements have brought their action to almost a positive condition.

Of the types of electric igniters in use, the break-contact or hammer type involves the motion of a spindle arranged as a rock-shaft with a contact-arm or hammer acting upon a stationary electrode, or a vibrating spindle passing through the walls of the cylinder to make contact with the same hammering force. The hammer type, whether it involves the action of a spring to cause a draw break-contact or by a direct-face contact, is subject to wear that either changes the adjustment for timing or prevents ignition by enlarging the contact-faces to such an extent as to allow the spark to occur before the charge can pass in between the faces. Many igniters of this type are made with broad-faced hammers, which become fouled or are so tightly faced by the hammer action that the spark passes before the gas charge can reach the spark between the faces, causing misfires. This has been remedied by reducing the size of the contact-faces and rounding their surface, which serves to give free access of the explosive charge to the spark at the moment of break of contact. The single wire-wound sparking coil and battery seems to be the cheapest means for producing electric current for the internal break-contact igniter.

The jump-spark igniter is increasing in favor among engineers and operators, owing to the simplicity and fixedness of its cylinder terminals, which places the intermittent action on the outside of the cylinder, thereby allowing of ready observation and adjustment without stopping the motor. In the early form of the jump-spark igniter with both terminals passing through a single insulation in the plug, the space on the insulated face of the plug was made so short that by the fouling of the surface the electric current was short-circuited and no spark was produced; this gave much trouble from the necessity of frequently removing the plug for cleaning the insulating surface. Its construction has been modified so as to increase the distance between the terminals by an extension of one of the terminals from the body of the plug, which is an improvement, but still defective. A later improvement has been made by extending the porcelain insulator beyond the face of the plug from a half to three-quarters of an inch and extending the opposite terminal from the face of the plug with a hooked end and clearing the insulator by a quarter inch, thus giving more than three-quarters of an inch of insulating surface between the electrodes. In some motors the plug terminal is a single positive electrode, while the negative electrode is fixed to the cylinder-head away from the plug, making a greater distance over which short-circuiting has to pass, but this is a mistake, for the insulated part of the plug is the limitation of short-circuit possibilities.

The jump-spark system of ignition requires a secondary or induction-coil, and, for further efficiency, a condenser, and a breaking device operated from the valve-gear shaft to close the otherwise open primary coil from which the secondary or jump-spark is generated by induction in the secondary winding at the moment of closure for timing the spark. There are two methods of operating the jump-spark ignition; in one a magnetic vibrator is employed which makes and breaks the primary circuit many times during the contact of the time switch by the secondary shaft, during which moment a series of sparks is sent across the terminal electrodes in the combustion chamber, thus insuring ignition by repeated sparking.

In the use of the induction-coil without the vibrator, but a single weak spark is produced at the opening and a single strong spark again at the closing of the timing switch, thus giving two sparks; but the first is not considered available, except from a more powerful induction-coil than needed for the vibrating attachment. The distance or opening between the terminals of a sparking plug is of greater importance than generally considered, as much hidden trouble has arisen from the form and spacing of this important adjunct in the operation of explosive motors.

For a satisfactory effect a six-element battery in series and an induction-coil for sure ignition should give a spark of maximum range from three-eighths to half an inch, for which the terminals of the sparking plug should be set at from two to three sixty-fourths of an inch apart. The voltage for a reliable spark need not exceed one and a quarter volts in each of a six-battery series, equal to seven volts, acting through an induction coil consisting of a soft iron wire-core five-eighths of an inch diameter, No. 12 gauge, insulated by a paper-tube spool five inches in length between the shoulders, on which is wound two layers of cotton-covered copper wire, No. 12, B. & S. gauge, well insulated with paper and shellac varnish. For the secondary coil, wind 10 ounces of No. 36 B. & S. gauge cotton-covered copper wire, shellacing and covering each winding with a layer of uncallendered writing paper. See details of induction-coil further on. A vibrating hammer and condenser is believed to add to the efficiency of the jump-spark igniter.

ELECTRIC IGNITION

Of the two forms of ignition by the electric spark, it has been shown in practice that both the break-spark and jump-spark are equally applicable and efficient for all speeds and on single or multiple-cylinder motors. The jump-spark method possesses the advantage of mechanical simplicity and the disadvantage of electrical complication, while the break-spark possesses electrical simplicity and mechanical complication. Either method can be successfully used with any of the regular apparatus for furnishing the electric current — that is, the battery, dynamo, or magneto, or combination of dynamo or magneto and battery, providing the complete apparatus is consistently designed. It is noticed that the jump-spark with battery or magneto is meeting with probably the greater favor by American manufacturers, while the European builders are using the break-spark more extensively with the alternating-current magneto, a few with the alternating magneto and jump-spark.

Batteries possess the advantage, over other forms of current generators, that their maximum strength can be used for starting the engine, but the disadvantage that, after the engine is running, they grow weaker, until they are exhausted. Some kinds can be recharged to advantage; others must be replaced with a new battery when exhausted. The first cost of batteries is low, and their care is fairly well understood by the average operator. The facts that it is impossible to determine in any practicable way just when a battery will become exhausted, and the cost of maintenance, are probably its most objectionable features.

PRIMARY IGNITION-BATTERIES

Much of the success of explosive-motor running depends on the efficiency of the ignition outfit. The usual primary battery and spark-coil do not always give uniform results. The life of the battery depends on the chemicals of which it is composed; or, in other words, on its ampere-hour capacity; on the number and voltage of cells connected in series; on the internal resistance of the cells; on the speed of the engine and number of hours which it runs per day; on the design of the igniting mechanism — that is, on whether or not the sparking points make contact every, or every other, revolution or only at times when fuel is admitted; on the length of time points are in contact; on the resistance and efficiency of the spark-coil; on the insulation of the sparking plug, and on the resistance of the external circuit.

By ampere-hour capacity of a cell is meant the quantity of current, measured in amperes, which a cell will furnish for a definite number of hours. Thus, a 300-ampere-hour cell is supposed to be capable of furnishing a current of one ampere for 300 continuous hours. Dry cells are not regularly given an ampere-hour rating for the reason that individual cells vary greatly and, moreover, it is difficult to determine their capacity since, on account of rapid polarization on discharge, it is impossible to take a constant, continuous current from them. The dry battery, which is used most extensively, is reliable and cleanly, but of short life, making it

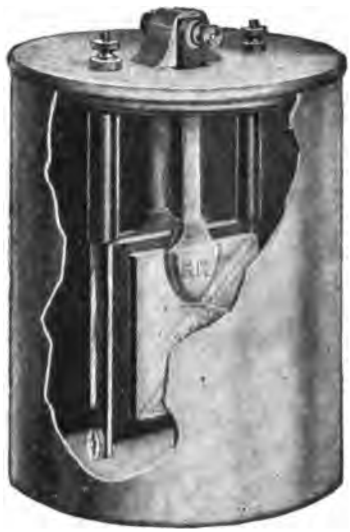


FIG. 74.—Type R R, 7¼ x 10½ inches. Edison primary battery.

expensive to maintain. It will regain part of its original strength, if allowed to rest after being exhausted; but, when once partially exhausted, a new battery should be considered a necessity, and installed at the earliest opportunity.

The Edison Primary Battery, formerly known as the Edison-Lalande battery, is one of the leading types for efficiency and lasting quality for primary battery ignition for all types of stationary explosive motors. The batteries are made in varying sizes to meet the requirements for stationary, portable, and launch service. In the construction of these batteries, a double zinc plate forms the negative element and a single plate of compressed oxide of copper forms the positive element of the battery. The fluid is a solution of caustic soda, which is sealed by a layer of paraffine oil to prevent evaporation and creeping of the solution. The plates are all suspended from the cover of the battery, as shown in Fig. 74, which is the largest (or R R) size contained in a porcelain jar, of which five cells, having a capacity of 300 ampere-hours, is the usual outfit of a large motor plant.

For launch motors, the size V is in general use, having a liquid-tight cell of enamelled steel, which will stand hard usage, and of which six cells are sufficient for single or double-cylinder two-cycle or four-cycle motors. On three- or four-cylinder motors two batteries of six cells each are recommended, which have a capacity of 150 ampere-hours each.

The sparking coil used with this form of igniter is shown in Fig. 75. It consists of a bundle of iron wire, insulated and wrapped with insulated copper wire. It is a simpler device than the induction or Ruhmkorff coil, but will not project a strong spark or at a great distance between the electrodes, as may be obtained from a Ruhmkorff coil — the breaking device being necessary in either case. A simple primary sparking coil may be made with a core of iron wire (No. 16) ten inches long and one inch in diameter. Fasten heads for the spool on this, and cover the core with a few turns of brown paper shellaced to make a tube. Wind No. 14 single cotton-covered magnet wire on this to a depth of about ⅝ inch, insulating each layer from the next by a layer of paper. Give each layer a

FIG. 75.—Construction of low-tension sparking coil.

coat of shellac also. The coil is used in series with a battery, and the spark is obtained when the circuit is broken. With six or eight strong cells a thick spark will be given. This coil is illustrated in Fig. 75, only instead of four windings make six to eight windings. This spark coil is the result of large experience in an effort to produce the largest spark from the least battery current. Its short length and large number of wire turns make the magnetic changes instantaneous, producing a hot and powerful spark, so necessary in high-speed motors.

DRY BATTERY CONSTRUCTION AND WIRING

In the dry battery the electrolyte is in the form of a paste instead of a liquid and is therefore much more suitable for portable work than any form employing a liquid excitant. Sectional views of standard dry cells of European and American construction are shown at Fig. 76. Considerably more care is exercised in making the foreign type and it is more enduring and has greater capacity than the cheaper American constructions. The American dry cell consists of a zinc can lined with blotting paper or similar substance which is saturated with the liquid chemical employed as an electrolyte. A depolarizing substance is packed in the centre surrounding the carbon element or rod. The function of the depolarizor is to keep the cells active for a longer period than if only a simple electrolyte was used. When the cell is in action hydrogen gas bubbles collect on the carbon or neutral element and as this is a non-conductor of electricity, if no means were taken for neutralizing its bad effects the internal resistance of the cell would be so great that the current generated would be inadequate to overcome it. The depolarizing medium is some substance rich

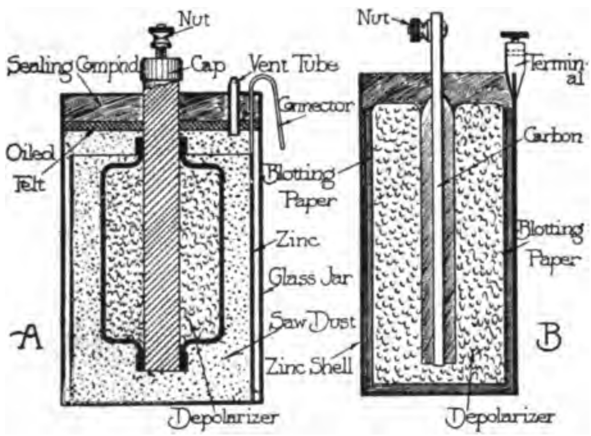


FIG. 76.—Sectional views of standard dry cells.

A. European construction. B. American design.

in oxygen, and as this element has a great affinity for hydrogen it unites with the bubbles of gas on the carbon element and forms water, which is a good conductor of electricity, especially if slightly acid or alkaline, and also serves the useful purpose of keeping the interior of the cell moist. The zinc serves as the active member that is acted on by the electrolyte while the carbon is the neutral or collecting member. The terminal attached to the zinc can is known as the "negative" and is commonly indicated by a minus sign, while the terminal attached to the carbon is known as the positive and is indicated by a plus sign. The terminals form the connections for the outer circuit. The internal circuit for the passage of the current is completed by the electrolyte and the depolarizor.

The cell will only generate electricity when a complete external circuit is established. A single dry cell does not have enough power, unaided, to produce a spark that will have sufficient strength to ignite the gas. A number of dry cells are joined together to form a battery, three different combinations being shown at Fig. 77. The usual method is in series in which the positive terminal

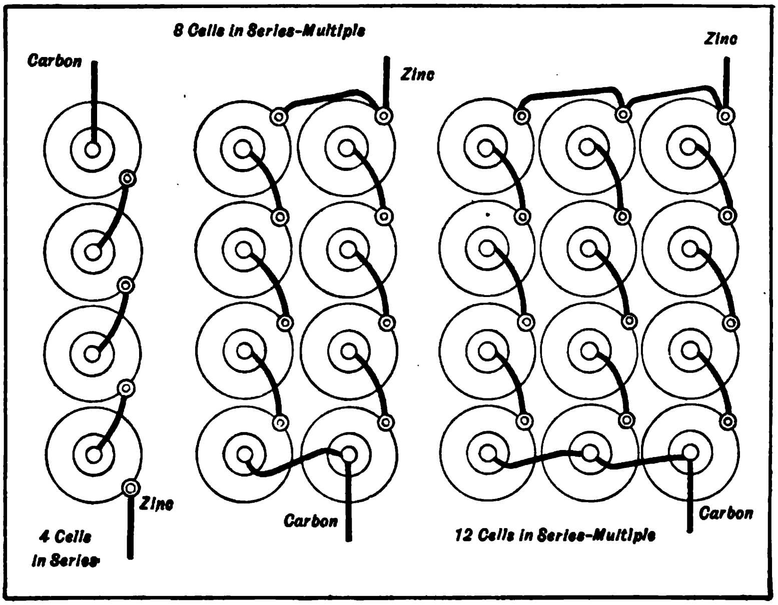


FIG. 77.—Methods of joining dry cells to form batteries of varying value.

of one cell is always attached to the negative member of its neighbor. Under these conditions the battery has a voltage equal to that of one cell times the number of cells joined together. Sometimes two or more sets of four cells are wired in multiple, which increases the capacity. When wiring in multiple or parallel connection similar terminals are joined together instead of different ones. The standard dry cell is six inches high and two and one-half inches in diameter and it is known generally throughout the trade as size No. 6.

STORAGE BATTERY CONSTRUCTION

The storage battery is a chemical current producer capable of being recharged when it is exhausted by passing a current of electricity through it in a reverse direction to that of the current given out. The interior construction of a typical storage battery adapted for ignition purposes is shown at Fig. 78. This is a three cell form in which three sets of elements or plates are carried in distinct compartments forming part of a common carrying case. Storage batteries are composed of elements of practically the same material when discharged and can only become active after the nature of the plates have been changed by passing a current of electricity through them. The materials generally used are grids of lead filled with a paste composed of lead oxides. In passing a current of electricity through these plates, they become enough different in nature so a difference of electrical condition or potential exists between them. When the battery is fully charged it simply indicates that the plates have become changed in nature as much as possible. As the current is drawn from the battery the plates gradually return to a condition where they are almost chemically identical and at such times only a very feeble current will be delivered. The plates of a given assembly are separated from each other by ribbed wood and perforated hard rubber separators. The top of the storage battery is provided with vents to liberate the gas evolved in charging and yet to retain the liquid electrolyte and prevent it from splashing out. A storage battery is termed a "secondary" source of electricity because it must first be energized before it will do useful work. The connection between the cells comprising the assembly is made by plates of lead which are burned or welded to the elements leaving but two terminals free, one of which is the negative member and one positive. The average ignition battery will deliver a current having a pressure of 6.6 volts when fully charged and a capacity of 60 to 80 ampere hours.

The storage-battery, in connection with the dynamo or direct-current magneto, forms an ignition system which is almost ideal theoretically. The storage-battery is of great strength and is reliable until exhausted, providing proper care is taken of it; but unless it is given this attention it will prove a failure. For instance, if it be charged above a certain maximum rate, it will not receive a normal charge, and will therefore become exhausted earlier than it would naturally do. If it be discharged above a certain maximum rate, the battery will not only fall short on its present charge, but on all subsequent ones; and the time of its ultimate destruction is hastened by the excessive discharge rate. If the battery has been allowed to discharge after the voltage has

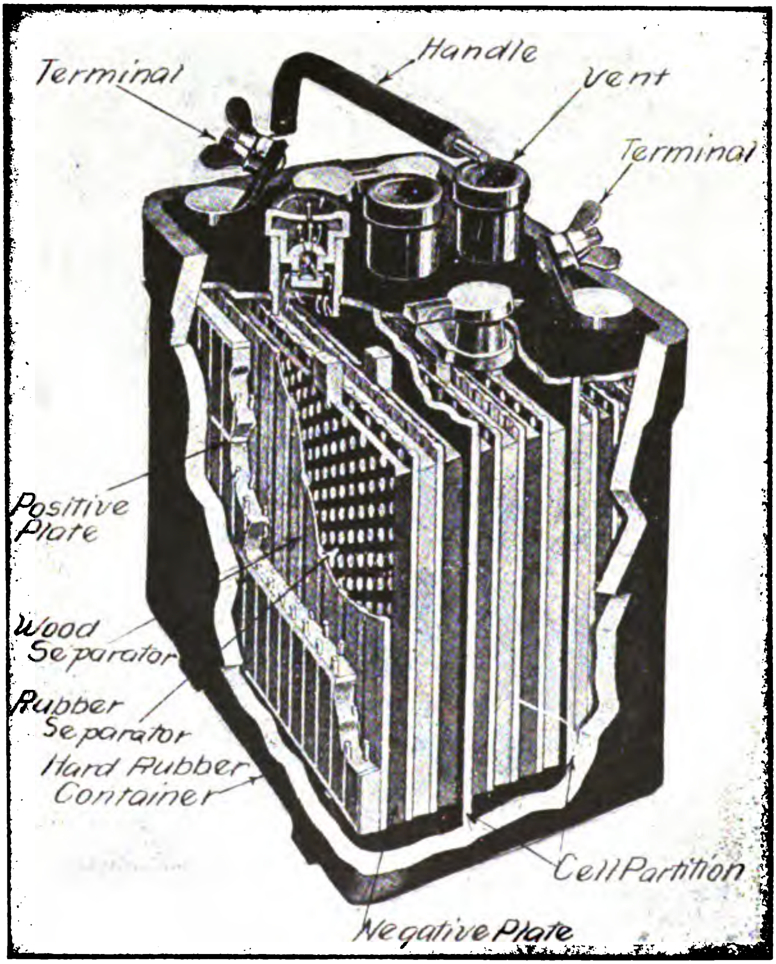


Fig. 78.—Sectional view of Geiszler storage battery for ignition service.

reached a certain minimum indicated by the makers of the battery, generally about one and eight-tenths volts per cell, sulphating and its consequent troubles result. Owing to the nature of automobile work, this last abuse is probably responsible for the bad reputation that storage-batteries have acquired with those experienced with them. The storage-battery should be charged through an ammeter; and the discharge should be checked with a voltmeter, not to mention tests with hydrometer for specific-gravity.

LOW TENSION MAGNETOS AND DYNAMOS

Drawing electricity from any chemical producer is comparable to drawing liquid from a reservoir filled with a definite supply. As the demands upon the reservoir increase, the amount of liquid it contains becomes less in direct proportion. Batteries cannot maintain a constant output of electricity for an indefinite period and their strength is reduced according to the amount of service they give. A mechanical generator of electricity produces current with but minor depreciation. There is some wear in the mechanical parts but this is so small compared to the service it will give that its effect is practically negligible as regards current output. The first forms of mechanical generators were patterned after dynamo electric machines used for various industrial purposes. This was later simplified by the elimination of one set of windings and made lighter and more effective. A generator with two sets of windings is termed a "dynamo" while that utilizing permanent magnets to produce the magnetic field is termed a "magneto." The dynamo form of generator is used where current of considerable value is needed for ignition purposes. A magneto may be made so it will be a complete ignition system in itself, whereas a dynamo generally furnishes low voltage current and serves to merely replace a chemical current producer.

A typical low tension magneto having an oscillating armature is shown at Fig. 79. In the conventional construction, the armature is a revolving member that turns between the field pieces. In this device it does not make a complete revolution. The mechanism outlined forms a complete ignition system because it is attached to and forms a part of the igniter plug which is adapted to be bolted to the side of the cylinder in such a way that while the igniter points project into the combustion chamber the magneto or current generator is in a position close to the cylinder head where it may be easily operated by a push rod and trip from the engine cam shaft. When a spark is desired between the igniter

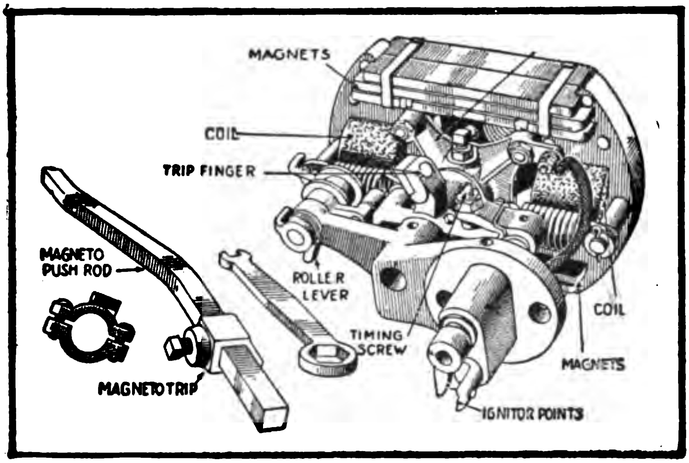


FIG. 79.—Oscillating armature low-tension magneto and integral igniter plate a complete ignition system.

points the inductor is given a quick movement which produces a flow of electrical energy through the winding and at the same time that the current is produced the points separate and a spark takes place in the combustion chamber.

The simple device shown at Fig. 80 is a revolving armature type that is used for supplying current whenever a constant flow is desired. The magnetic field is produced by a pair of permanent magnets attached to the pole pieces forming the armature tunnel. The electricity produced in the coil is collected by means of brushes which form terminals corresponding to the positive and negative poles of the battery. An automatic governor is usually provided so the armature will not be driven by the friction drive pulley if the safe speed of magneto rotation is exceeded. The friction drive pulley is intended to receive motion by virtue of adhesion between its surface and that of the flywheel.

The dynamo system for ignition, with the speed-governing pulley, is theoretically a fine ignition system; and, if operated by one familiar with caring for electrical apparatus, it is a very satisfactory method. This system, however, possesses two very great disadvantages: first, the dynamo generates a direct current of low voltage, necessitating care and attention to be given the dynamo; second, the dynamo must run at a constant speed, necessitating the use of a speed-governing device, which, for the service required, has not proven altogether reliable.

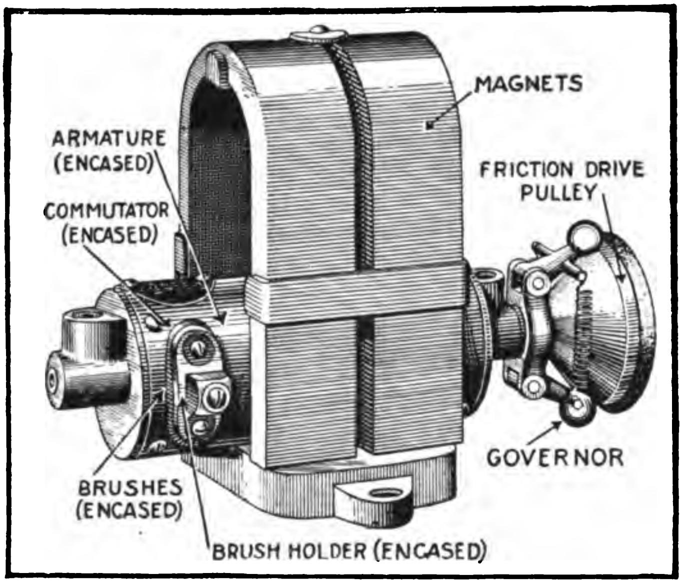


Fig. 80.—Revolving armature magneto delivering low-tension current. Used in connection with induction coil, timer and spark plug to form complete ignition system.

In Fig. 81 a neat and compact ignition-dynamo known as the Apple is shown. It is entirely enclosed in a case, practically water and dust proof. The pulley has a friction-clutch governor acting on the rim of the pulley and attached to the spindle of the armature. The clutch shoes of the governor are closed on the rim by the springs, while the centrifugal force of overspeed releases them, and between the action of the two forces, the dynamo runs steadily with a variable speed of the motor.

In the sectional detail of the parts of the Apple dynamo, A is the cast-iron case; B, the hinged cover; C, one of the cast-iron pole pieces of the field magnets, which are fixed to the case by screws as shown; D, the armature, the core of which is built up from thin-toothed disks of soft sheet-iron; E, the coil of one of the

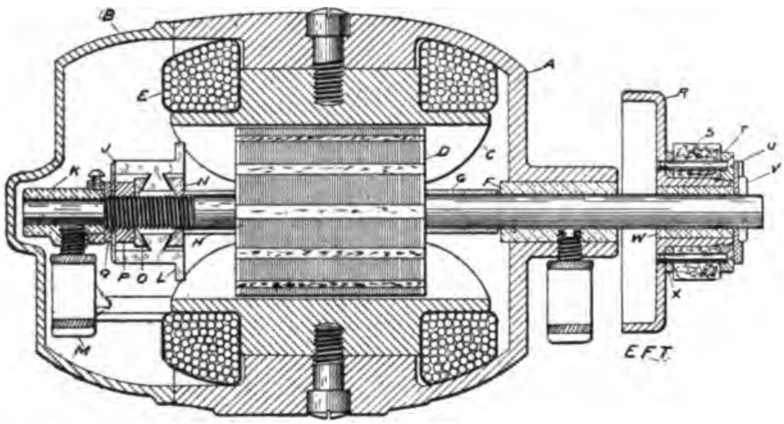


Fig. 81.—Sectional diagram of the Apple dynamo.

field magnets; F, brass bearing; G and H, hard-fibre tubes covering the spindle; K, brass spider and spindle bearing; L, commutator with mica insulation; M, wick-feed oil-cup; N and P, bevelled nuts clamping the commutator bars; R, driving disk and rim containing the centrifugal clutch cover; S, pinion fixed to driving disk R and revolving freely on the spindle.

A very efficient type of governed dynamo evolved for use in connection with an electric cranking system for automobile motors is shown at Fig. 82. This operates on exactly the same principle as those previously described as the current generation is by the inductive effective of the magnetic energy in the armature winding. A governor is used to keep the speed of rotation to the same point and is of a centrifugal type. As the armature speed increases the arms fly out and draw the clutch plates apart. When the speed decreases the spring tension overcomes the tangential force of the governor weights and forces the clutch plates together. The device is thoroughly covered, protected from dirt and dust and the driving clutch is provided with a series of fan blades so it tends to draw a current of air through the device and keep the coils cool.

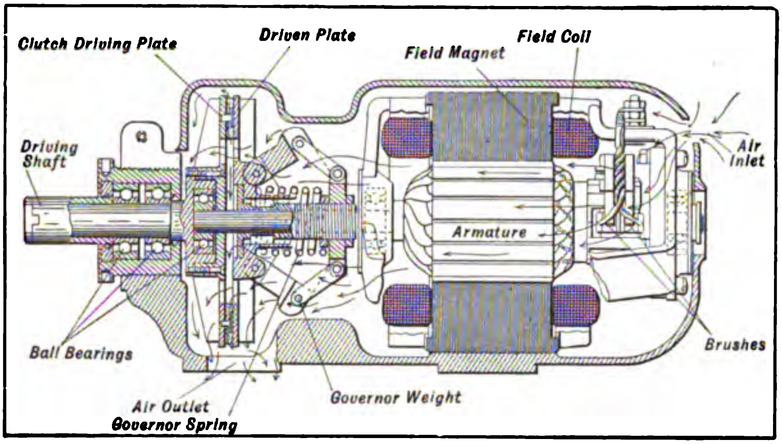


Fig. 82.—Gray & Davis governed dynamo, an appliance for producing electricity by mechanical means.

PRODUCING THE SPARK IN CYLINDER

At Fig. 83 the two common methods of producing an electric spark in the combustion chamber are shown. The low tension igniter plate at A has a movable electrode or hammer that strikes a sparking point attached to a fixed member or anvil. The anvil is insulated from the plate forming the main portion of the device. The movable member is actuated by a bell crank and is always in contact with the anvil except when a spark is desired at which time the points separate. The other construction shown at B involves a high tension current which produces a spark by leaping the air space between the spark points. The low tension igniter plate has a disadvantage of wearing on account of the constant movement of the hammer member, which is continually exposed to scale and carbon deposits promoted by the heat of combustion. While formerly very popular on all types of engines, the igniter plate, or make and break spark producer, is used only in large stationary engines running at low speed where a hot, fat spark is desired. In the high tension spark plugs there are no moving parts and it is not difficult to make the device gas tight and reasonably heat proof. The construction is extremely simple, consisting of a main member or shell which is screwed into the

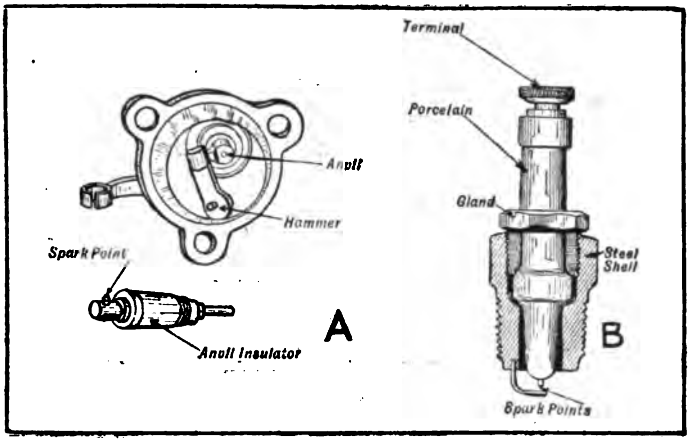


Fig. 83.—Devices for producing spark in engine cylinder.

A. Low-tension igniter plate. B. High-tension spark plug.

cylinder, an insulator and electrode assembly and a gland to pack the joint between the porcelain insulator and the plug body. One of the electrodes is attached to the main shell while the other spark point, which is usually in the form of a light rod or wire, is carried to the centre of an insulating medium which prevents the current from leaking to the steel shell and which forces it to leap the air gap between the points. While spark plug construction varies widely, as may be determined by comparing the device

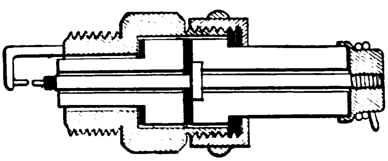


Fig. 84.—French ignition plug.

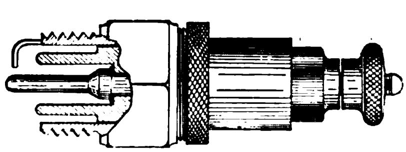


Fig. 85.—Soot-proof sparking plug.

shown at Fig. 83 with those at Figs. 84 and 85, the principle of operation is the same in all types. The differences are in matters of minor detail, such as the type of insulator used which may be porcelain, mica, steatite, glass or combinations of these materials and the construction and arrangement of the electrode and spark points. There is practically no part of this device that will wear in service, and even should a plug fail to work because of a cracked porcelain or other defect the entire device may be renewed for 50 cents or 75 cents and replaced by any one who is able to handle a wrench. In most forms the insulator is easily removed and a spare insulator and electrode assembly may be readily substituted for the imperfect one with ease.

In all forms of gasoline internal-combustion engines, the spark plug must resist 350 pounds pressure per square inch, must stand a high temperature (it is exposed to flame under pressure at a temperature of 3,000°), and in addition it must perfectly insulate a high-pressure electric current of from 10,000 to 25,000 volts. It is also exposed to deposits of carbon which tend to allow the spark to escape by providing a path for it to go where the combustible gas cannot get to it, thus causing misfires or total stoppage of motor. When they fail to work properly it is always because of some easily remedied fault which should be sought intelligently and removed. In case of failure to ignite at all, the first thing to inspect is your coil; see that the vibrator works when circuit is on; next, remove wire from top of plug, hold it ¼ of an inch from metal parts and observe if spark will jump the gap. It must be capable of jumping at least six times the space of gap between spark points inside, as the resistance of hot gas under pressure is much greater than free air. If spark is weak, a new battery or coil is required; but if this cannot be supplied at once, a plug having shorter spark gap may be made to work, or the one in hand may have gap shortened.

The best distance for most circumstances is $\frac{1}{32}$ of an inch, but with weak battery better results may be secured by a shorter gap. While with strong spark, capable of jumping greater resistance, a more certain ignition is secured by having a somewhat wider gap, it all depends on the power of coil and battery or of the magneto what width is best, and changes should not be made unless sure that extra plugs are available. If the spark is good, the plug should next be removed and inspected for carbon deposit, or cracks in insulation. Carbon deposit will not take place unless you are feeding too much oil, or burning more gasoline than can be completely consumed. If carbonized, the deposit may be washed out with gasoline or kerosene and a small sliver of wood. If an insulating tube is cracked or broken, a new one must be inserted. If sparking end of plug appears all right, the next thing is to remove gland from top of plug, and see if it is wet or coated with carbon on inside. If wet, it must be wiped dry, and replaced; if black, it must be cleaned, and a new packing washer inserted inside of steel shell under shoulder of porcelain.

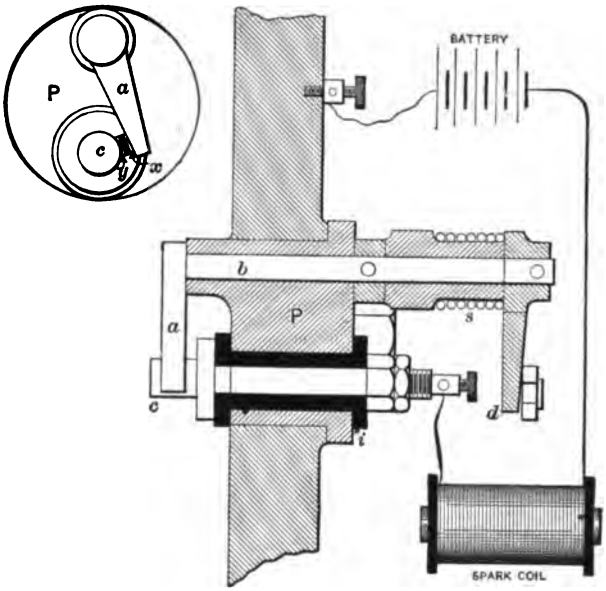


Fig. 86.—Hammer spark igniter.

LOW TENSION IGNITION SYSTEMS

The application of a low tension ignition system in diagram form is shown at Fig. 86. The source of current is a battery one end of which is grounded to the engine, while the other is attached to the spark coil. The other lead of the spark coil, which is in series with the battery, is connected to the insulated terminal of the anvil. This contact point (c) is kept out of contact with the main body of the device P by the bushing of insulating material (i). The hammer member A is attached to the oscillating stem B which is actuated by the lever (d). The contact points are shown at x y. With this system the battery is short circuited through the spark coil all the time the points x and y are in contact. When these points separate a brilliant spark is produced.

The method of operating an igniter plate used on high speed engines of the automobile type is illustrated at Fig. 87. The plate 3 is fastened to the side of the cylinder so that the points project

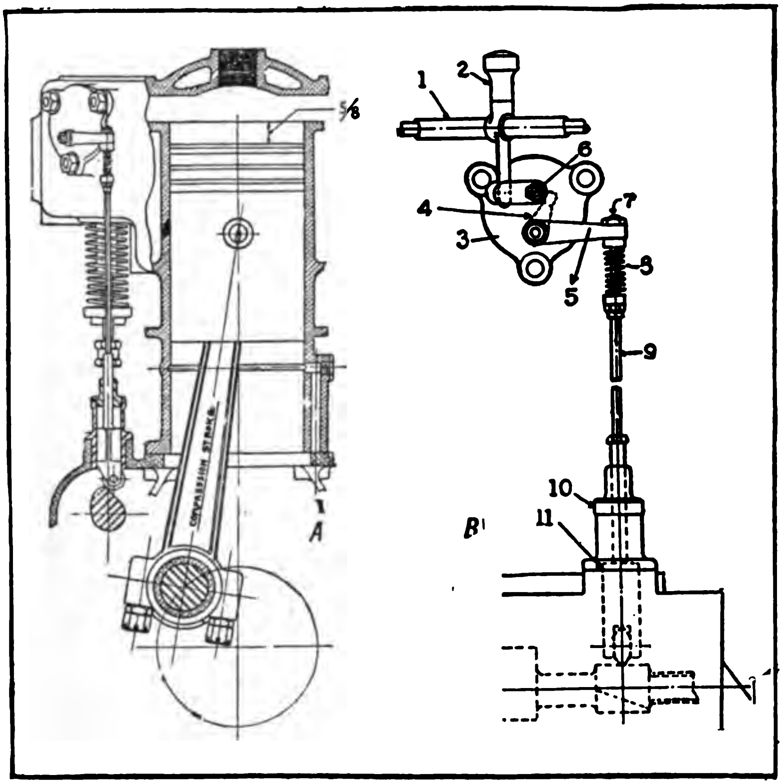


FIG. 87.—Showing action of Locomobile low-tension igniter plate.

into the combustion chamber. The hammer 4 is inside while the hammer operating arm 5 is on the exterior of the plate. A tappet rod 9 is guided by the bushing 10 and carries a cam roll assembly 11 which is in contact with the actuating cam employed to separate the points. The view at A shows the timing of this low tension igniter plate.

A complete ignition system applied to a four-cylinder engine is depicted at Fig. 88. Two sources of current are provided, one a mechanical generator of the magneto form for regular service while the other or emergency source is a battery of ten dry cells in series. A two-way switch is provided so either battery or magneto may be used as desired. The four igniters are connected to a common bus bar which is insulated from the metal portions of the engine and which is connected from the top of the switch by a single wire. A direct wire from the collector brush on the magneto is attached to one side of the switch while a conductor

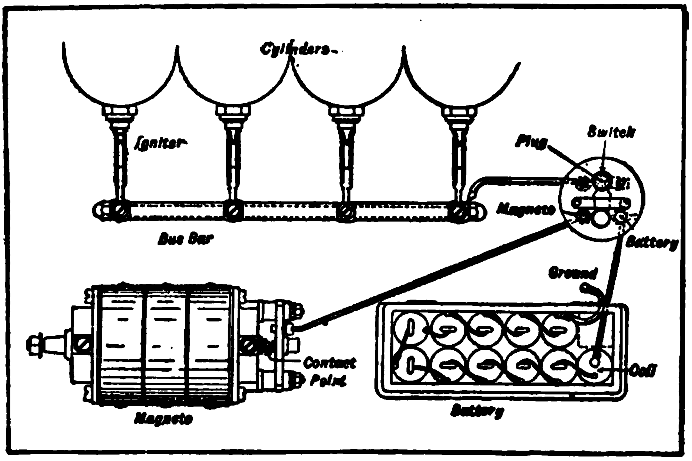


FIG. 88.—Typical four-cylinder, low-tension ignition system.

from the spark coil, which is in series with the batteries, connects to the other side of the switch. One of the battery terminals is attached to the frame or base of the engine and one of the magneto armature leads is also grounded. With the switch lever in the position indicated, neither the magneto or battery is in service, and there can be no spark in the cylinder. If the switch lever is moved to the right the current will be derived from the batteries, while moving it to the bottom on the left will disconnect the battery and put the magneto in circuit with the igniters. Owing to the character of the current produced by the average magneto no spark coil is necessary to increase the intensity of the spark. In the diagram shown at Fig. 89 a low tension dynamo is used, and both dynamo and battery current must be passed through the intensifying coil before going to the ignitor.

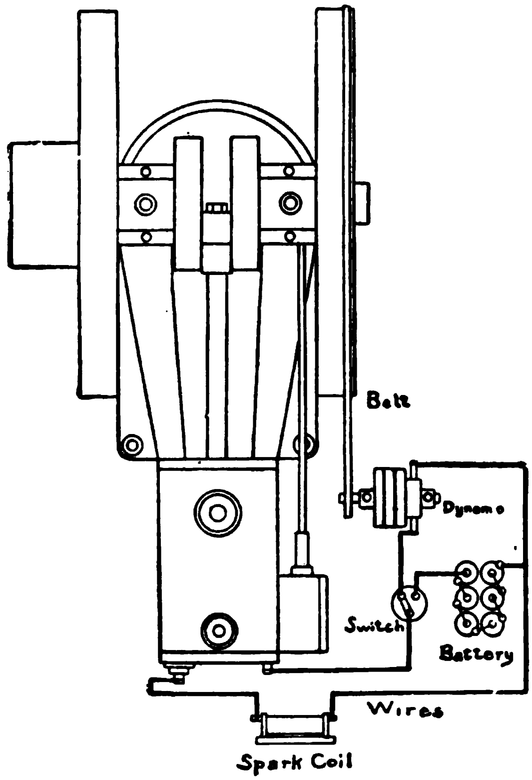


FIG. 89.—Dynamo wiring.

SIMPLE BATTERY IGNITION METHODS

The components of a simple battery ignition system of the high tension type are shown in outline form at Fig. 90 and in very clear diagrammatic form at Fig. 91. The source of current in the simple diagram is a storage battery which is wired to the induction coil by a direct conductor from the positive terminal and through a mechanically operated switch or circuit breaker connected to the negative terminal. The induction coil, which is shown in very simple form, is a most important element. It is composed of two windings, one of which is connected into the primary or battery circuit while the secondary winding is attached to the spark plug. Every time the rotary cam at the timer establishes an electrical contact with the insulated segment carried by the timer case a current of electricity will flow through the

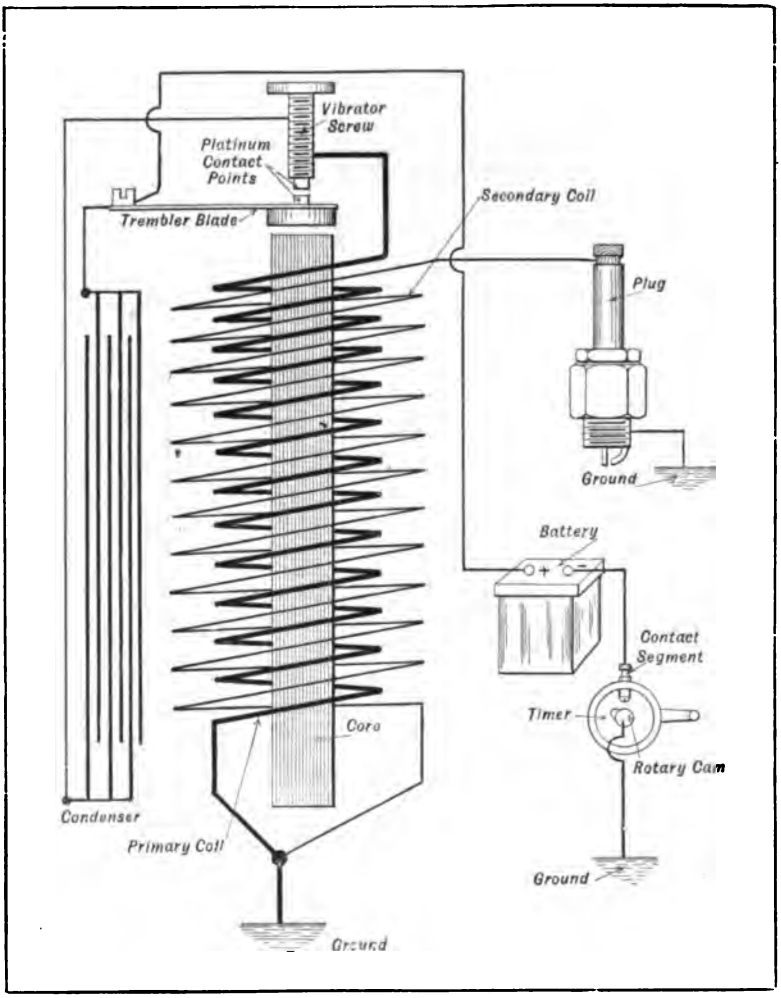


Fig. 90.—Simple ignition system for one-cylinder motor showing important components and their relation to each other.

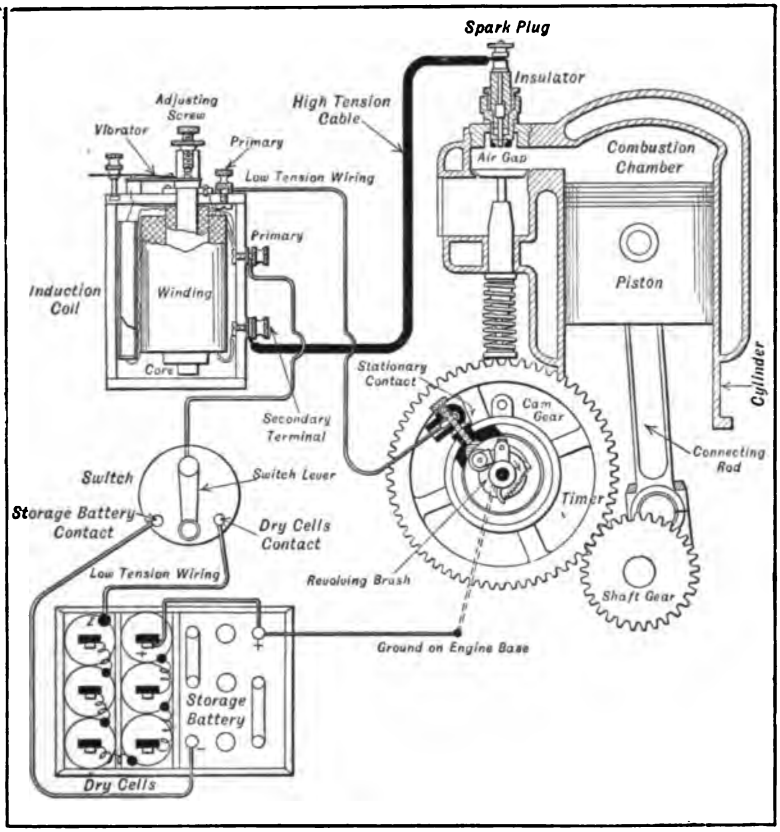


Fig. 91.—Simple high-tension ignition system for one-cylinder motor, showing arrangement and wiring of principal parts.

primary winding of the induction coil and this will produce a current of high voltage in the secondary winding that will have sufficient potential to overcome the air gap between the spark points and ignite the mixture. As the rotary cam is timed so it makes contact with the insulated segment only at the time when the piston reaches the end of its compression stroke, a spark is produced in the cylinder only when the compressed gas must be ignited. In an engine of the four-cycle type, which is shown in outline at Fig. 91, the revolving brush of the timer is driven at half the crankshaft speed. It is usually attached to the camshaft, which is driven by a cam gear having twice the number of teeth of the gear on the engine crankshaft. In the system shown at Fig. 91, two separate sources of ignition current are shown, one a battery of dry cells, the other a storage battery.

The practical application of the various simple ignition methods to a single cylinder stationary engine are clearly shown at Fig. 92. At A we have the simple battery ignition system, comprising a dry cell battery and induction coil, a suitable timer and the spark plug. At B a high tension magneto suffices to supply the ignition current. The combination system shown at C consists of an induction coil which may derive its energy from either low tension magneto or dry-cell battery, either of which may be brought into circuit by the two point switch attached to the side of the battery and coil box.

INDUCTION COIL CONSTRUCTION

For a better understanding of the detail of construction of an induction-coil of suitable size for the ignition of the explosive charge of a gas, gasoline, or oil-engine, we illustrate in Fig. 93 the details of such a coil without a vibrator, and in Fig. 94 the same coil with the vibrator. A coil of the size here given and detailed should give a full and hot spark for any ordinary engine across a $\frac{1}{32}$ to $\frac{3}{64}$-inch space between the electrodes. Its full-length spark should be equal to a jump of from $\frac{1}{2}$ to $\frac{5}{8}$ of an inch between wire terminals. The iron core, H, H, is made up of annealed wire, No. 20 wire gauge, 6 inches long, as many pieces as can be pushed into a $\frac{5}{8}$-inch paper tube, $5\frac{3}{4}$ inches long, made by wrapping paper on a $\frac{5}{8}$ rod with shellac varnish between the layers, say a half-dozen layers, and shellac the outside. Push onto each end of the paper tube a square wooden flange, $\frac{1}{2}$ inch thick, 4 inches diameter, even with the end of the paper tube and square with it. Fasten the wood ends strongly with shellac and shellac their entire surface.

This will then make a spool $4\frac{3}{4}$ inches long for winding the coils. Bore a hole in one of the heads close to the paper tube to pass one end of the primary coil through and another a little farther around to receive the other end. Wind on the spool two layers of No. 16 double cotton or silk-covered copper wire with the ends passed through the holes in the spool flange. Give the

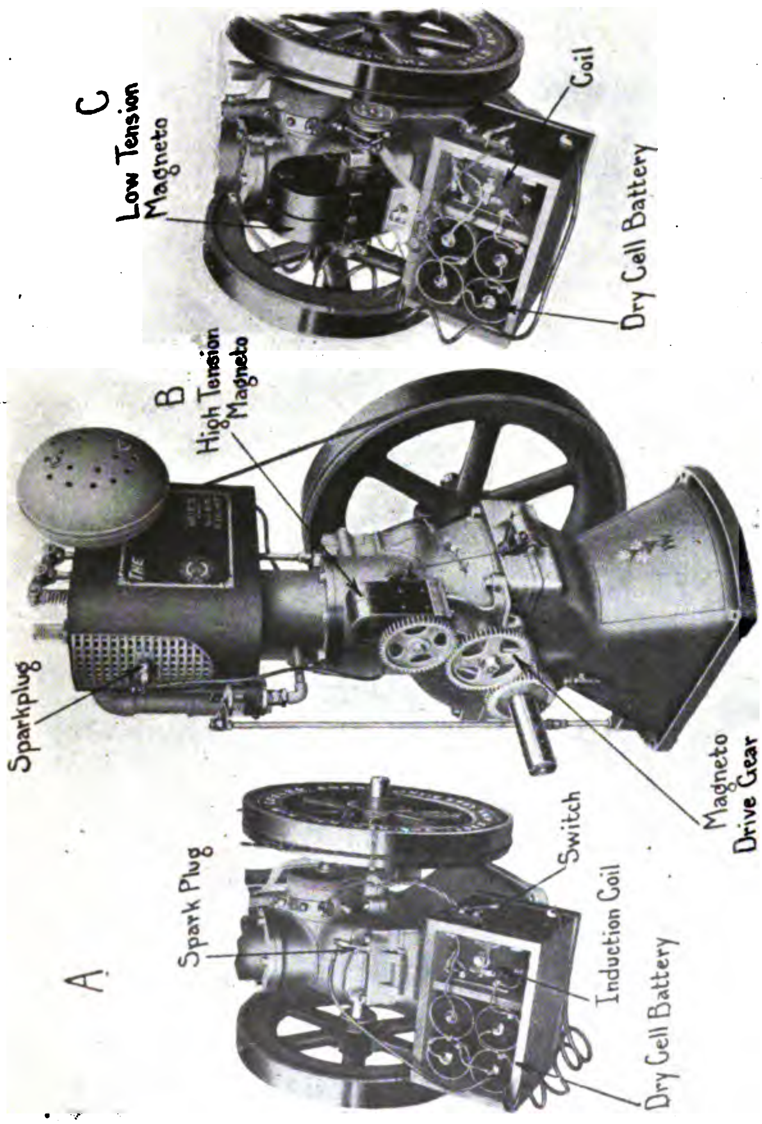


FIG. 92.—Outlining three standard methods of ignition applied to stationary motor.
A. Simple battery and coil system. B. High-tension magneto. C. Combined low-tension magneto and battery system.

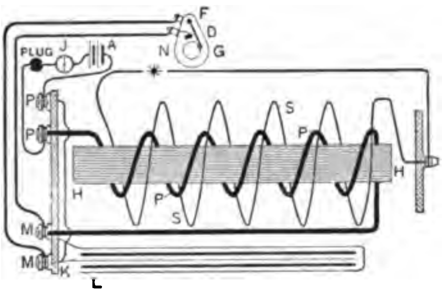


FIG. 93.—Jump-spark coil, without vibrator.

coil a coat of shellac varnish and dry. Then wrap the primary coil with three thicknesses of paper with shellac varnish between each wrapping with a perfect closure at the flanges and over the exit wires of the primary. Dry and shellac the outside.

The secondary coil may be made of 8 ounces of double silk-covered copper wire, No. 36 gauge; commencing by passing one end through the hole in the opposite flange from the primary terminals and winding closely but not tight, one layer, shellac and cover with two layers of paper, shellaced, and a third layer at each end to make a sure closure against a spark passing across the layers at the ends of the spool. Continue this back and forward method of winding for the whole amount of wire, covering each layer as the first, and terminate through a hole in spool flange at the same end as it commenced. This should not be a hurried job; give each layer time to dry. The perfection of the whole coil depends upon its thorough insulation, especially at the ends of the layers, where the difference in potential is greatest, with a liability of sparking from layer to layer of the coil and the ruin of the work. The coil assembled in its wooden case is shown at Fig. 95.

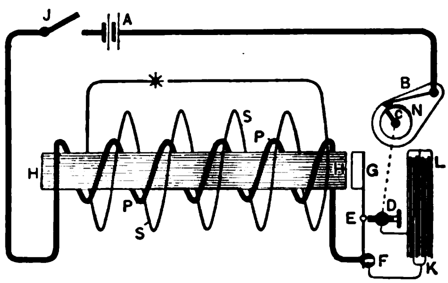


FIG. 94.—Jump-spark coil, with vibrator.

Such a coil may be used without a vibrator, and referring to Fig. 93, in which the leading principles of construction are shown, P, P, M, M are the primary binding posts. The upper posts, P and P, are connected through the battery and switch. The lower posts, M and M, are connected through the breaker on the reducing gear from the crank-shaft represented at N, F, D, G. The upper post P, and the lower post M, are directly connected, making a complete primary circuit from the battery A, through the switch J and post P around the core and post M to the breaker at D, and through the lower post M and across by the upper post P to the battery. The condenser L is composed of strips of tin-foil separated by paraffined paper in series and connected at M M as a shunt across the contact-breaker for the purpose of absorbing an extra current induced in the primary coil by the breaking of the circuit, which would tend to prolong the magnetization of the core beyond the desired limit in a high-speed engine.

The condenser may be made of a size to be enclosed in the hollow base upon which the coil is to be fixed, and made up of about 71 sheets of plain uncalendered writing paper, say 5 by 8 inches, dipped in melted paraffine or varnished with shellac on each side; interleaved with 70 sheets of tin-foil, cut 4 by 7 inches, with an ear at one corner of each sheet to project beyond the paper sufficient to allow of the alternate sheets to be connected together on opposite corners. The pile may then be clamped together with 2 pieces of board well shellaced. The ears of each set of 35 sheets may then be pressed together and clamped for connecting to the binding posts M M. Condensers are not absolutely necessary and many jump-spark coils are in use without them. The theory is that the electro-magnetic force of self-induction in the primary, which is principally instrumental in causing the spark at break contact, will expend most of its energy in charging the condenser, causing the break-spark of the primary to be less and the current to become zero with greater rapidity. The practical effect of the condenser on the spark volume of the secondary is very great, producing what is commonly called a fat spark.

The vibrating coil (Fig. 94) is of the same general construction as described, with the addition of a spring vibrator shown at F G. The steel spring G F may be 1½ inches in length and ½ inch in width, fastened to a post at F and fixed to a small armature of soft iron at G with a platinum or, what is better, an alloy of platinum and iridium contact piece at E. D is a brass post with a platinum-iridium-point adjusting screw, and connected to the breaker N and to the condenser K L, completing the primary circuit through the post F, the switch J, and the breaker B. The

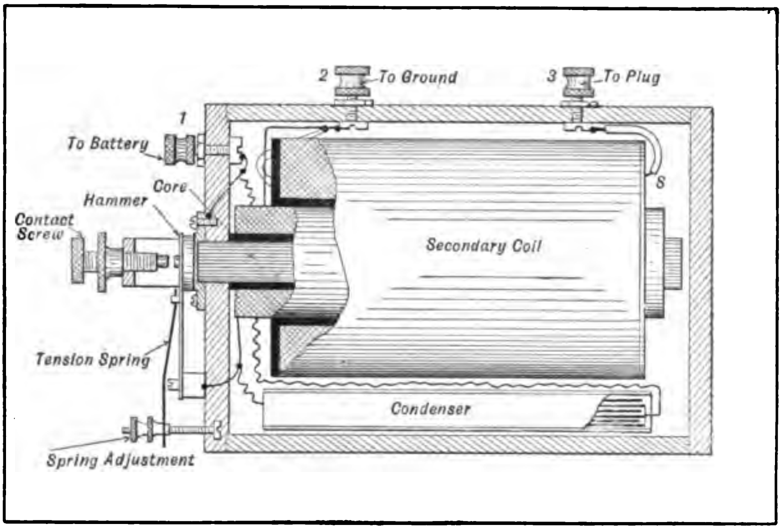


Fig. 95.—Part sectional view of simple induction coil, an important component of all battery ignition groups, and sometimes used with magnetos.

office of the vibrator is to give a rapid intermission of the primary current while the commutator bar C is in contact with the spring B. By this means the induced secondary current also becomes intermittent and so secures a succession of sparks at the electrodes that insures a positive ignition. A coil of this nature may be purchased more cheaply than the cost of making.

The length and stiffness of a vibrator spring on a jump-spark coil causes considerable variation in its time beat and in this way, by varying the time of ignition, may influence a motor's running not easily observed and this source of trouble may become a cause of anxious search in the action of very high-speed motors. A vibrator may have a possible variation of from 15 to 150 strokes per second, and the sparking time may therefore vary from $\frac{1}{15}$ to $\frac{1}{150}$ of a second.

With a motor running 1,800 revolutions per minute, a revolution is $\frac{1}{30}$ of a second, so that the strokes of the vibrator at 15, 30, 45, 60, and 120, may coincide with the strokes of the motor and their synchronism will produce exact and uniform time sparks. Any variation in the running time of the motor and the time vibration of the armature will advance or retard the sparking moment; so that for the most uniform sparking effect under the varying speed of a motor, the highest effective speed of the vibrator will give the best results.

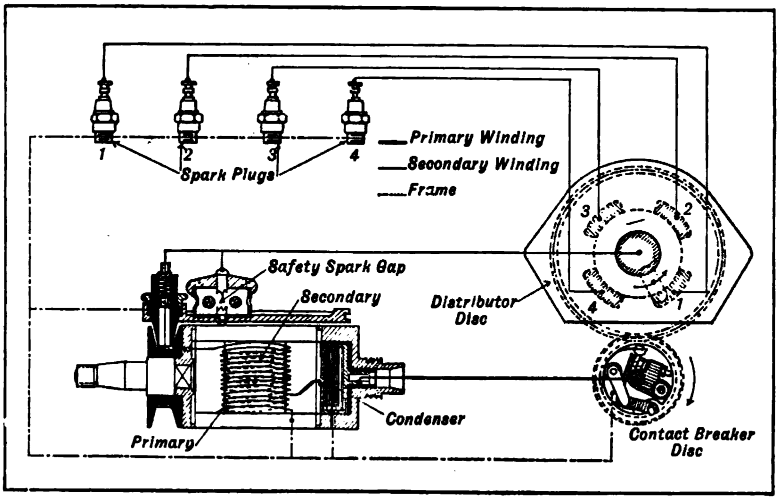


FIG. 96.—Simple wiring scheme when four-cylinder magneto is utilized for gas-engine ignition. Magneto members shown separate to facilitate explanation of principles of operation.

HIGH TENSION MAGNETO TYPES

The high tension magneto is in the form generally used in automobile ignition systems and its popularity is increasing among other gas-engine operators and manufacturers as well. The advantages of the true high tension magneto are that it comprises in one device all the elements of the current generating and intensifying appliances and all that is needed in connection with a high tension magneto to complete the ignition system are the spark plugs and the wires by which they are connected to the instrument. A marked advantage of the high tension magneto is that forms adapted for four-cylinder engines are but little more complicated than those utilized in connection with the simpler power plants. The only difference is in the number of contacts in the distributor and the speed at which the device is driven. The wiring diagram of a typical high tension magneto employed for four-cylinder ignition is clearly shown at Fig. 96. The various parts of a true high tension magneto, separated in order that their construction may be readily ascertained, are shown in some detail at Fig. 97.

The armature of a high tension magneto is of the two-pole type having an approximately H section wound with two coils of wire. One of these is a comparatively coarse one, that corresponds to the primary of an induction coil while the other, which is of finer wire, takes the place of the secondary coil. The armature is usually mounted on ball bearings to insure its easy rotation and to guard against untimely bearing depreciation. The magnetic field in the device shown at Fig. 97 is composed of two pairs of compound horse-shoe magnets which are attached to pole pieces forming the armature tunnel. A condenser is mounted on and turns with the armature and it is in shunt with the contact points in the magneto breaker box. A true high tension magneto is always driven by positive chain or gear drive and is invariably timed in such a way that the contact points on the contact breaker will separate only when a spark is desired in the engine.

On a four-cylinder engine the magneto is driven at crankshaft speed and the arrangement of the contact breaker is such that the points separate twice during each revolution of the armature. Every time the points are separated a current of electricity leaves the armature by a high tension collecting brush which bears on an insulated contact ring carried at one end of the armature shaft and is led to a distributing brush at the centre of the secondary current distributing member. The spark plugs are attached to wires which lead to the segments in the distributor, there being one segment for each spark plug. The distributor shaft is revolved at half armature speed by means of suitable reduction gears and as the revolving contact brush closes the circuit with one of the segments each time that the spark points separate, a current of electricity is directed to the plug which is in the cylinder about to fire. It will be apparent that a high tension magneto includes current generating and commutating means as well as the timing mechanism. A magneto for a two-cylinder four-cycle engine would have but two distributor segments and would be driven at camshaft speed or half that of the crankshaft. A magneto for a three-cylinder engine would have the segments spaced 120 degrees

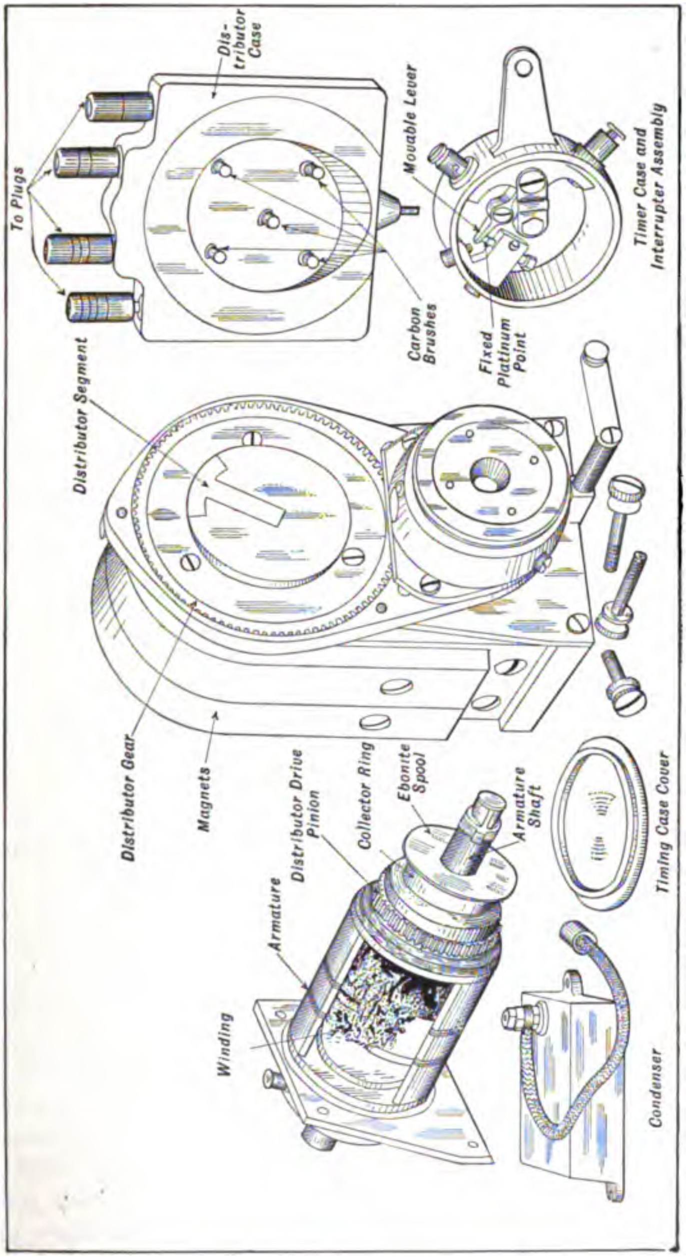


Fig. 97.—Partially dismantled four-cylinder magneto, showing important parts of current-producing and distributing elements.

apart in the distributor and would turn at three-quarters crankshaft speed. On a six-cylinder motor, the distributor would be provided with six segments spaced 60 degrees apart and the armature would turn at one and one-half times crankshaft speed. When used on two-cycle engines the speed would be doubled.

The contact breaker corresponds to the timer of a battery ignition system. In its most popular form it consists of a fixed member which carries one of the platinum contact screws while the movable bell crank carries the other platinum contact. The condenser is used for the same purpose as in the induction coil previously described, as it absorbs the surplus current due to self induction between the various windings of the armature. The safety spark gap is interposed between the high tension brush and the ground in such a way that any excess current that might injure the winding if allowed to go through the instrument in the regular manner will be permitted to flow to the ground without passing through the external circuit. This device performs the same function for the magneto as the safety valve does for a steam boiler.

TRANSFORMER COIL MAGNETO SYSTEMS

Another form of magneto ignition system that has received some application, also delivers a current of high potential to the spark plugs. But this current is not derived directly from the magneto armature as is the case in the true high tension type. The armature is wound with a single coil of wire and delivers a current of low potential which must be intensified in value before it will have sufficient power to overcome the resistance of the air gap at the spark plug. The accepted method of increasing the voltage of the current before it is delivered to the distributor of the magneto is to pass it through an induction coil or transformer which may be incorporated in the device as at Fig. 98 or carried as a separate fitting as in Fig. 99. Where the coil is housed under the arch of the magnets, as in the Connecticut magneto, one obtains practically as compact an ignition device as a true high tension form and the wiring is no more complicated. With the transformer coil as a separate fitting a certain amount of primary wire is necessary in connection with the secondary wiring which complicates the ignition system to some extent. Those who favor the

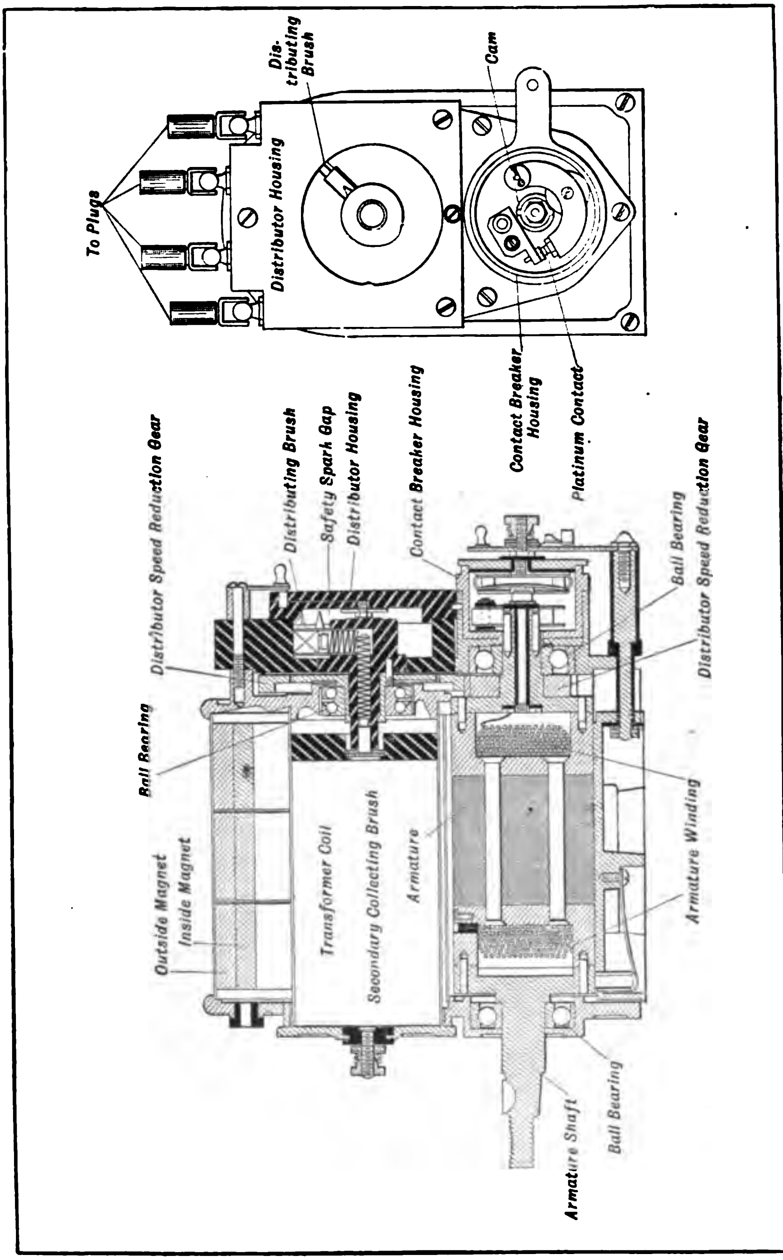


FIG. 98.—Defining construction of Connecticut magneto, a form in which transformer coil is placed between magnets above armature tunnel.

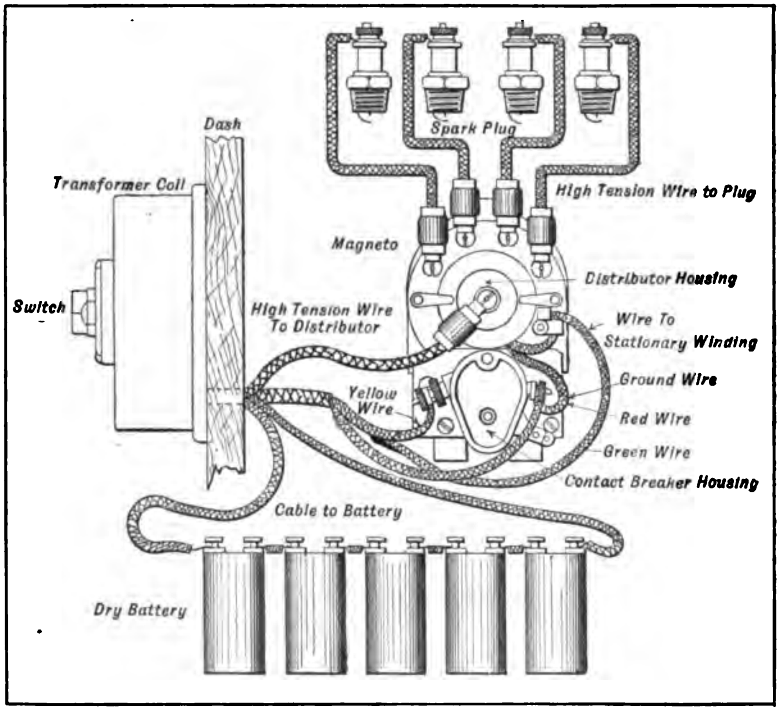


FIG. 99.—Wiring diagram outlining method of combining magneto and transformer coil to form device for four-cylinder ignition.

transformer coil system contend that by transforming the current in a separate coil, not only more efficient windings can be secured but better opportunities are provided for housing a large condenser and obtaining positive insulation of the high tension windings. Both forms have been used successfully, and as the high tension type has demonstrated that it is capable of igniting any gas charge that the transformer coil type will explode, in view of the marked simplicity of the true high tension device it is the most generally applied where energetic ignition is desired with minimum complication.

SOME TYPICAL IGNITION SYSTEMS

Mention has been previously made of the vibrator lag and its effect on the ignition of a multiple cylinder engine. In an attempt to synchronize the explosions, i. e., to have them all occur at the same relative point in the cycle and to have a spark of practically the same value, high tension distributor systems are employed in which a single unit coil having but one vibrator is used to serve all cylinders as at Fig. 100. The form shown, which is a straight battery system, utilizes a storage battery for regular ignition service and a dry cell battery for emergency use. The high tension wire from the coil is led to the central point of the distributor, which is practically the same in construction as that of a magneto and the high tension current is commutated to each of the four plugs in succession. A four-point contact timer is employed to interrupt the primary energy and is carried at the lower part of the distributor housing. The distributor consists of two distinct members which are electrically insulated from each other, the lower one or timer for interrupting the primary current while the upper one distributes the current of high potential or voltage.

A very complete double ignition system in which both a magneto

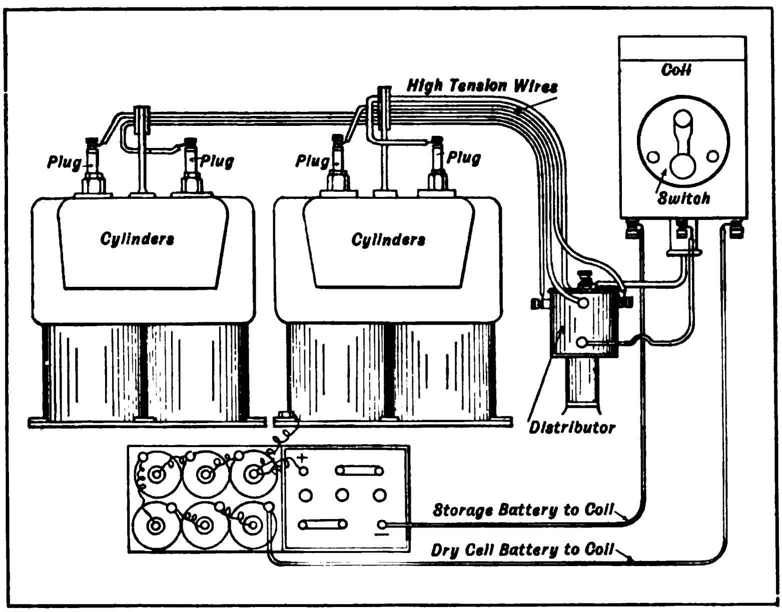


FIG. 100.—Methods of employing single coil to fire four cylinders when secondary current is distributed instead of battery energy.

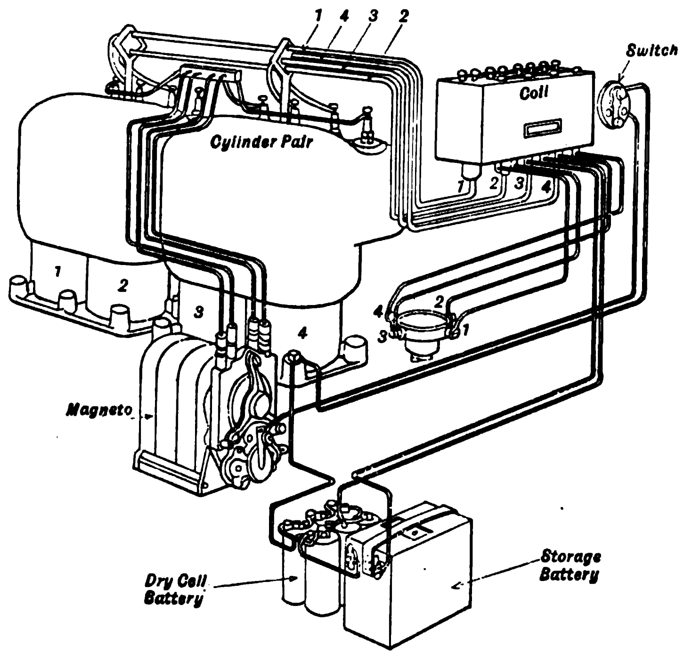


FIG. 101.—Practical application of double ignition system to four-cylinder power plant.

and battery are depended on and which consists of two separate and independent ignition groups is shown at Fig. 101. Each ignition system has its own set of spark plugs and the engine may be run on either as desired. This form of ignition has been widely applied on the automobile and motor boat power plants. Another double ignition system in which a battery or magneto may be used as desired is illustrated at Fig. 102. This system demands a magneto construction incorporating two secondary distributors, one for each set of spark plugs. As the details of wiring of this and the other systems are so clearly shown it will be unnecessary to describe them further.

It has been found possible on large engines of the high speed types to materially increase the power by using two-spark ignition. This is especially true in engines of the T head form having valves at opposite sides of the cylinders. In order to apply the system successfully, a double pole spark plug such as shown at Fig. 103A must be used in connection with a regular single pole plug as shown at B. As will be apparent the double pole plug is a member that is completely insulated from the engine and in which a complete circuit is obtained without any chance of leakage to the steel body. The current enters the top terminal and passes down

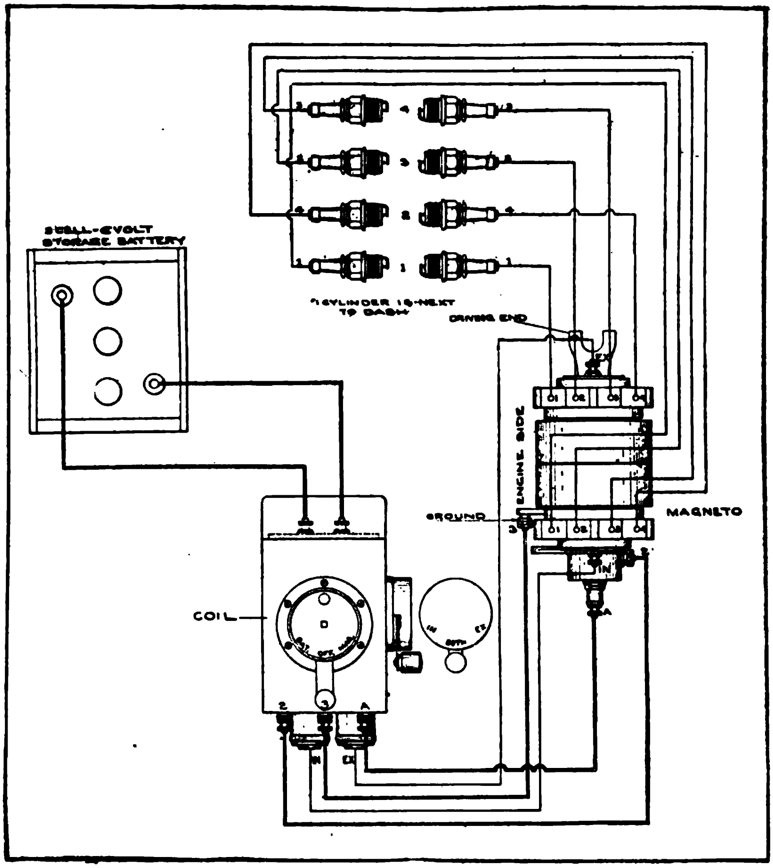


FIG. 102.—Double ignition system, showing wiring for both battery and magneto, and all parts to be inspected in event of ignition trouble.

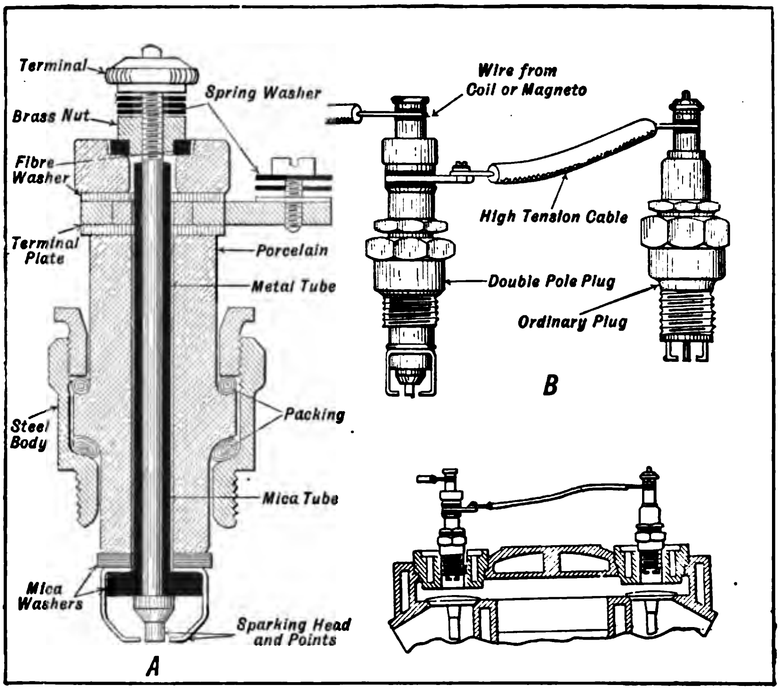


FIG. 103.—Double pole spark plug and method of applying it to obtain two sparks in the cylinder.

through the central electrode to the sparking head. This central electrode is thoroughly insulated by a mica tube and washer from the metal tube carrying the spark points and in electrical connection with the terminal at the side of the spark plug. After the current overcomes the resistance of the air gap, instead of passing to the ground as in the single point plugs, it flows along the metal tube and out through the side terminal and the short wire to the insulated terminal of the regular patterned plug. The spark, after leaping the gap at the conventional form of plug, is grounded in the usual way.

The double plug system outlined is more suitable for battery ignition groups than where magneto ignition is used. When a double system is desired with a magneto, the best method is to use a double distributor form with two sets of spark plugs, each set being served by one distributor. The contact breaker is arranged so that it times the spark for both distributors. When the two spark system is used it is claimed that the increase of power is noticed only at speeds over 1000 R. P. M. In some cases the power output has increased 20% with the two spark system over the best record previously made with the single spark method. The reason for the increase of power is said to be more rapid ignition of the compacted charge, which is kindled at two points instead of at but one portion of the combustion chamber.

CHAPTER XI

CYLINDER LUBRICATION, COOLING AND EXHAUST SILENCERS

THE lubrication of cylinders of explosive motors is a matter of great importance, as the intensely hot gases in immediate contact with the lubricating oil, although the oil is in contact with a comparatively cool metallic surface, have an evaporative effect, tending to thicken the oil into a gummy lining on the surface of the cylinder. To avoid this and keep a perfect lubrication, an oil that is adapted to this severe heat trial should be used and fed to the cylinder walls and piston in constant flow, and not too much or too little, but just enough so that the oil cannot be pushed into the combustion chamber in excess, so as to be blown through the exhaust-valve to clog the passages with oily soot. The sight-feed and capillary drop-oil feeders have been perfected to such an extent in the United States that they are widely used. Yet on some engines with revolving valve-cam shafts, the facility for obtaining easily the motion for a mechanical lubricator has made this form most popular.

In Fig. 104 is illustrated a mechanical lubricator used on the Crossley engines in England, and with some variations on other European and American engines. A small belt from the valve-cam shaft to the pulley A gives the required motion to the spindle and crank C C, to which is loosely attached a wire D, that dips into the oil and carries a minute portion to the wiper E, from which the oil drops into the passage to the cylinder. In Figs. 105 and 106 are shown a section and plan of a lubricator used on the Robey engines, which is an improvement over the previous one, in that it has a small receptacle above the level of the main oil cistern, which is fed by a revolving shaft and crank arm with drop wire reaching to the bottom of the cistern and wiping the oil on a fixed wiper over the receptacle, from which a second crank arm and drop wire lifts the oil to the wiper that feeds the passage to the cylinder. By this arrangement the oil for the cylinder is drawn from a fixed level, and the feed is therefore perfectly uniform at any level of the oil in the cistern.

Strict attention should be given to the quality of the oil used in the cylinder. Such oil is now made and sold as *gas-engine cylinder oil* of a less density and viscosity than the ordinary cylinder oil, and more fluid, so that it flows readily over the surface of the piston. Such oil does not readily gum in the cylinder and on the piston. It evaporates more readily than heavy oil and in a measure mixes with the explosive charge, and is burned and discharged with the

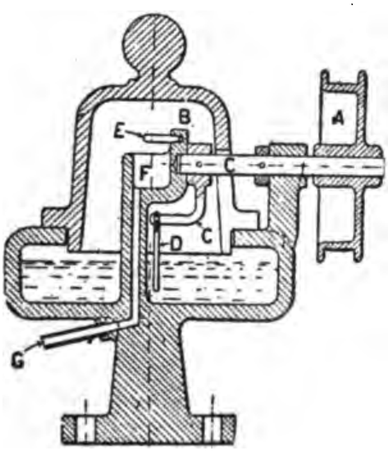


Fig. 104.—The mechanical lubricator. Crossley.

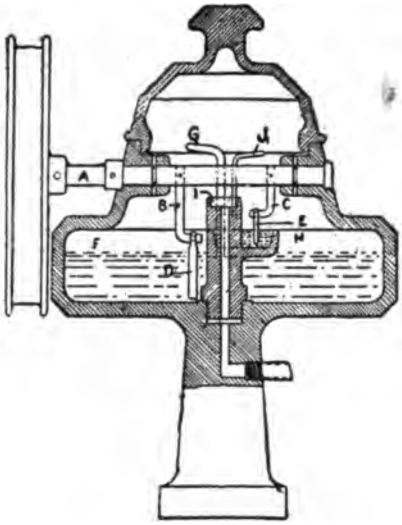


Fig. 105.—The Robey oil-feeder, section.

gases of the exhaust, thus avoiding the sooty oil that lodges in the muffler and exhaust-pipe from the heavier oils. A very small quantity of finely pulverized graphite used with this oil occasionally is said to give good results as a cylinder lubricant and imparts a smooth and glossy surface to both cylinder and piston. For all other parts of the engine the best engine oil is none too good. The poorer grades of machinery oil are not economical at any price. The oil feed to the main journals of a motor is of importance as to its constancy, and has suggested some ingenious devices for this purpose in the form of chain belts and rings running over the journals and dipping into an oil bath. In Fig. 107 we illustrate the ring feed as used on the Mietz & Weiss and other oil-engines. A cavity at the outer end of the journal box returns the excess of oil to the oil-well, as shown in the illustration.

LUBRICATING OILS, THEIR ACTION, PROPERTIES AND TESTS

Oil used for lubricating purposes must form a definite film between the running surfaces of greater or lesser thickness, depending upon the amount of pressure tending to keep the bearings in contact and upon the body of the oil. To secure perfect lubrication, the film must always be of sufficient thickness to

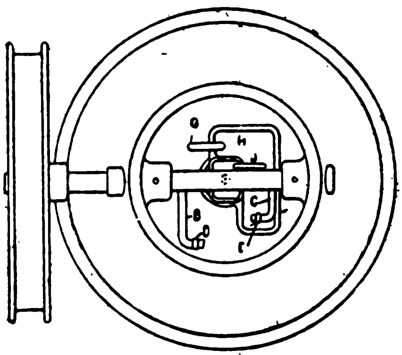


FIG. 106.—The Robey oil-feeder, plan.

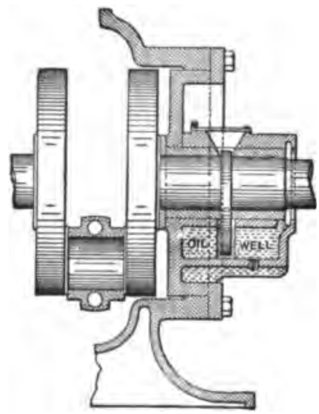


FIG. 107.—The constant oil-feeder.

prevent actual metallic contact between the bearings points or surfaces. The oil should not only have sufficient body to fill the microscopic cavities or pores in the metal but it must also have sufficient elasticity to yield to the rougher points in such a manner that the high spots on the shaft are prevented from coming in contact with the bearing body and low spots are sustained without a perceptible change in the position of the rotating member. The oil should not permanently identify itself with either member of the bearing but should remain an intermediate film having portions that are in constant motion. Those portions of the film which are closest to the fixed bearing member remain nearly at rest while the particles constituting those portions which are practically in contact with the journal tend to take this motion. Between these two surfaces are particles whose velocity is a mean between the two extremes of rest and motion. When rubbing surfaces are lubricated with oil there is a constant movement of the particles upon one another and the ease with which this movement of oil occurs determines the frictional resistance of the bearing. The use of a heavy viscous oil which flows with difficulty produces more friction than an oil of lesser body. Thus any oil lubricated surfaces float on a film of fluid which yields to their inequalities

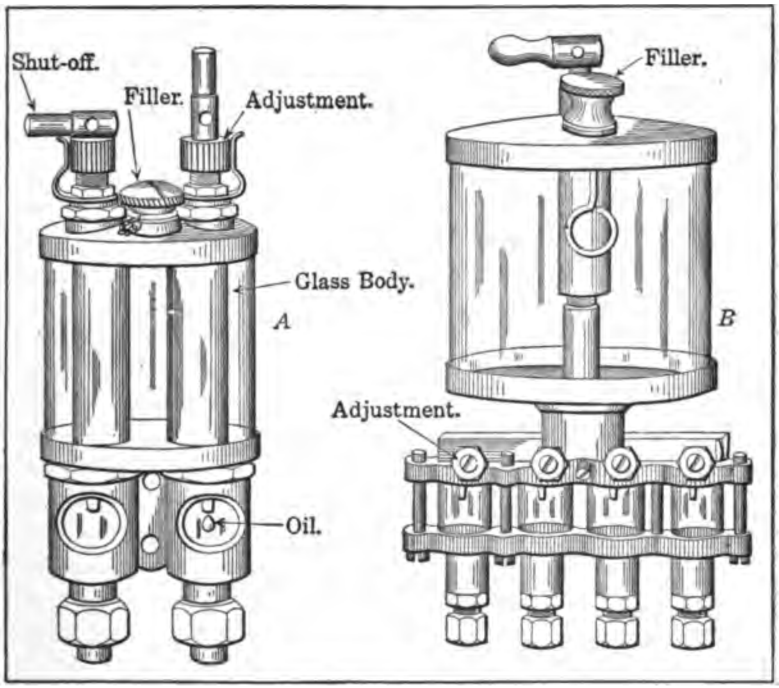


Fig. 108.—Simple gravity-feed oil cups with glass body to show height of lubricant in container, and sight gauges to give visible evidence of amount of oil supplied.

and makes up for their low spots. For best results, there must be a constant renewal of the supply of oil and a corresponding waste or discharge of used lubricant from between the rubbing surfaces.

We present herewith the specifications of the U. S. War Department for engine oil to be used in gas or gasoline engines having water-cooled cylinders. All the specifications outlined give in some detail the different tests the oil must pass in order to be used. The general rule that the oil must be a pure filtered mineral oil and that it must be free from acid, alkali and suspended matter applies to all lubricants specified except a compounded oil used in kerosene-engine cylinders. The oil for general use must pass the following test: Specific Gravity — Must not be less than .874 nor more than .877 at 60 degrees F. Flash — Must not flash below 385 degrees F. Fire — Must not burn below 430 degrees F. Viscosity — Must not be less than 4.61 at 50 degrees C. Cold Test — Oil must flow at temperature of 33 degrees F. Acid — Must not give an acid reaction on polished copper in 24 hours. *Alkali* — Ash must not show an alkaline reaction. *Water* — Must not froth or bump when heated in flash cups, as this shows the presence of H^2O. *Saponification* — Must be unaffected by an alcoholic solution of caustic potash.

Medium duty engine oil must conform to the following: Specific Gravity — Not less than .875 nor more than .879 at 60 degrees F. Flash — Must not flash below 380 degrees F. Fire — Must not burn below 420 degrees F. Viscosity — Must not be less than 2.98 (Engler) at 50 degrees C. Cold Test — Must flow at a temperature of 22 degrees F. In other respects, it is the same as the oil previously described.

Engine oil for severe duty should have the following properties: Specific Gravity — Must not be less than .877 nor more than .880 at 60 degrees F. Flash — Must not flash below 340 degrees F. Fire — Must not burn below 390 degrees F. Viscosity — Not to be less than 4.50 (Engler) at 50 degrees C. Cold Test — Oil must flow at 32 degrees F.

The engine oil for kerosene engine cylinders is not a pure mineral oil but is compounded of 20% acidless tallow oil and 80% filtered mineral oil. It should be free from acid, alkali and suspended matter and must also be free of water. As regards saponification, if the oil is treated with alcoholic solution of caustic potash, it should show 20% tallow oil in the form of soap. It must conform to the following tests: Specific Gravity — Not less than .884 and not more than .888 at 60 degrees F. Flash — Must not flash below 395 degrees F. Fire — Must not burn below 430 degrees F. Viscosity — Not less than 4.06 at 50 degrees C. Cold Test — Must flow at 30 degrees F.

The S. A. E. specification for light lubricating oils suitable for automobile engines is as follows: Oil should be derived from a mineral base, should be free of animal fillers, acid, alkali, or suspended matter. Should also be free from water and should show no traces of fatty oils when subjected to the saponification test. The following properties are desired: Specific Gravity — Not less than 32 degrees Baumé which is equivalent to .864 and not more than 38 degrees B. which is equivalent to .886 at 60 degrees F. Flash Point — Not less than 400 degrees F. Fire Test — Not less than 450 degrees F. Viscosity at 100 degrees F. with Saybolt Viscosimeter, 300 seconds. At 210 degrees F. 40 to 50 seconds. Carbon Residue — Not over 0.50%.

Instructions are appended for making a number of simple tests to determine the presence of carbon and acid, and also the accepted methods of making flash and cold tests.

Carbon in Oils: All oils for lubrication should be pure minerals, without any admixtures of animal or vegetable oils. The heavier the oil, the higher the fire test and viscosity test, but also in an oil of this character the darker will be the color which necessarily means a greater amount of carbon. It is therefore advisable to get as light an oil as possible, as color denotes very largely the amount of carbon contained in the oil. Buyers should be cautioned, however, against purchasing oils that have been bleached with acid. Oils that have their color lightened in this manner usually contain traces of acids which are destructive to the metal of the working parts of the engine, and oils of this character contain all of their original carbon.

Free carbon can only be removed from an oil by actual filtration through bone black, fullers earth, etc., thus lightening the oil without deteriorating its lubricating properties.

Acid Test: A very simple test to detect acid in an oil is with Blue Litmus Paper, which will show a pinkish color if there is any acid present. Another sensitive test, and a very practical one, is to partly cover a polished steel or copper plate with a strip of flannel or lamp-wick saturated with the lubricant to be tested. Expose this to the sunlight for about twenty-four hours. When the plate is wiped dry if the lubricant is free from acid the metal will have retained its gloss. If dull spots have developed on the surface covered it is the sign of the presence of acid.

Flash Test: It is a comparatively simple matter to take a flash test of an oil by heating a small quantity in a porcelain vessel over a Bunsen Flame, stirring the oil with a thermometer and applying a lighted match to the surface of the oil occasionally. When a bluish flame spreads over the liquid and dies out quickly the temperature should be noted, for this is the flash point. From about 410 to 420 degrees Fahrenheit is a proper flash point.

No cylinder oil should have a flash point lower than 400 degrees Fahrenheit, nor higher than 450 degrees Fahrenheit. The viscosity should not be lower than 200 at 70 degrees Fahrenheit nor higher than 500.

Cold Test: Another point that is of interest, especially in winter time, is to obtain an oil with a proper cold test. It is practically impossible to give a mineral oil a very low cold test; and one that shows from fifteen to twenty degrees below freezing point may be considered all right for the purpose. It is a very small matter to test this by taking a small quantity of the oil in a test tube and surrounding the test tube with cracked ice, over which is sprinkled a handful of salt. The thermometer should then be introduced into the oil and the temperature noted at which it begins to solidify, for this is its cold test. It should be remembered that the darker the oil the poorer the cold test, as the heaviest oils solidify most rapidly.

GRAVITY OIL CUPS WITH SIGHT FEED

The simplest devices for supplying lubricating oil to the gas-engine cylinder are the various forms of sight feed lubricators or oil cups which have been widely employed in general steam engineering practice for some time. Two devices of this nature are shown at Fig. 108. That at A is adapted to supply lubricant to two bearings points, each being controlled by its independent adjustment, while that at B discharges the lubricant into a manifold from which it goes to the various bearings connected to the four leads. They consist essentially of an oil container, usually composed of a glass body which is held at the top and bottom between metal flanges. This permits one to ascertain the amount of oil available at a glance. In the form shown at A two central supply tubes terminate in small compartments under the main body which also have a glass wall or inset so the drops of oil may be seen dripping from the end of the supply tube. The amount of oil is regulated by an adjustable needle valve which makes it possible to compensate for differences in temperature and viscosity by regulating the size of the oil supply orifice. The needle valves are normally spring-retained against the seat and may be lifted to allow the oil to flow by a small fulcrumed lever on top of the knurled adjusting nut. In the device shown at A the oil is allowed to feed in the cup to the right because the needle valve is raised from its seat while no oil can flow in the member at the left on account of the valve being seated. While the sight feed oiler has been very practical on stationary engines, where they could be

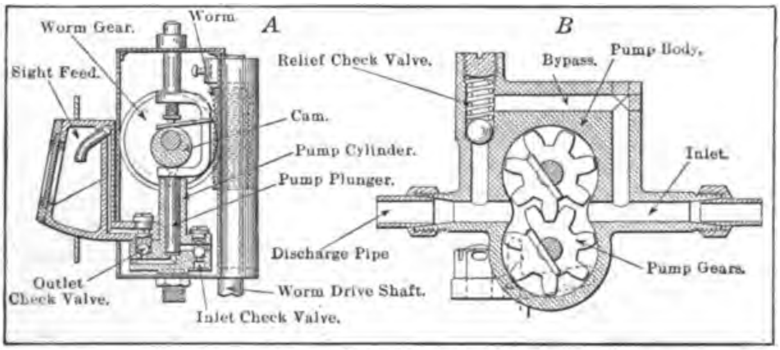


Fig. 109.—Positive mechanical methods of supplying lubricant. A. Worm gear driven plunger pump oiler. B. Gear pump with high-pressure relief valve.

given the attention they need, they are not so well adapted for severe service such as obtains in motor vehicle and marine practice, or the mechanical lubrication systems in which the oil is forced positively to the bearing points by pressure produced by some form of pump.

FORMS OF OIL PUMPS

When positively driven oil pumps are used one is sure that the oil will be delivered to all bearing points, whereas in the sight feed or gravity oiler, if the lubricant thickens, or if the opening in the supply pipe becomes clogged with some small piece of foreign matter, such as a bit of lint or wax, the oil feed stops; and unless there happens to be a considerable amount in the crank case the engine may be injured by failure of the lubricant to reach the bearing point. Two forms of oil pumps are shown at Fig. 109. That at A is a simple plunger type operated by an eccentric cam driven by the engine which delivers a definite quantity of oil (which may be regulated by altering the pump stroke) to a sight feed device from which the oil is forced to the bearing point. The pump shown at B is a gear pump designed to maintain a constant circulation of lubricant in engines that are lubricated by the constant level splash system.

Various forms of individual pump lubricators are shown at Fig. 110. The sectional view at A shows the method of drive by a belt pulley on the exterior of the device which operates worm

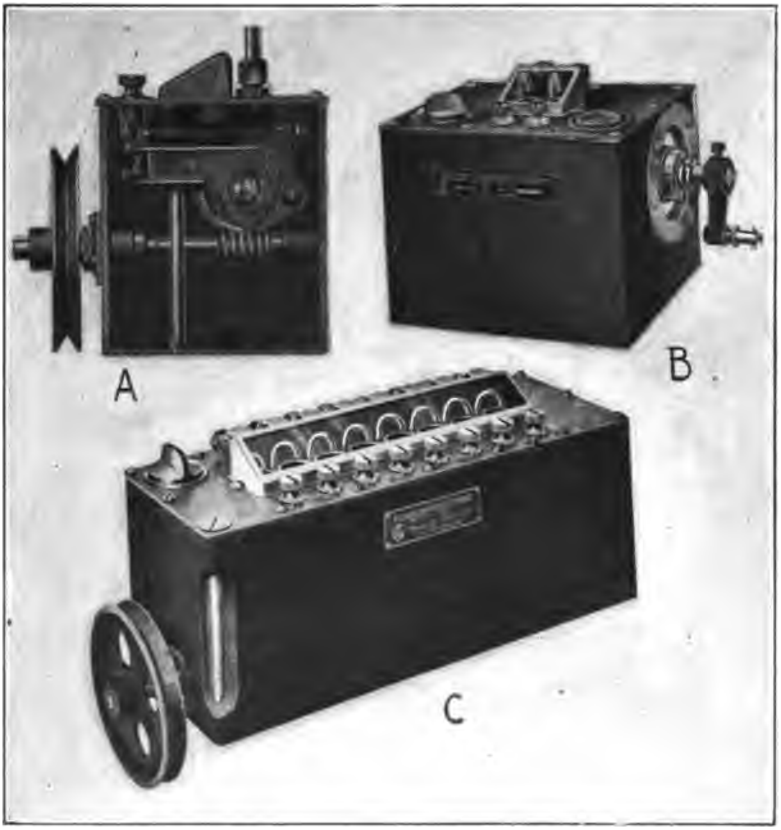


Fig. 110.—Types of mechanical oilers having individual leads to bearing points.

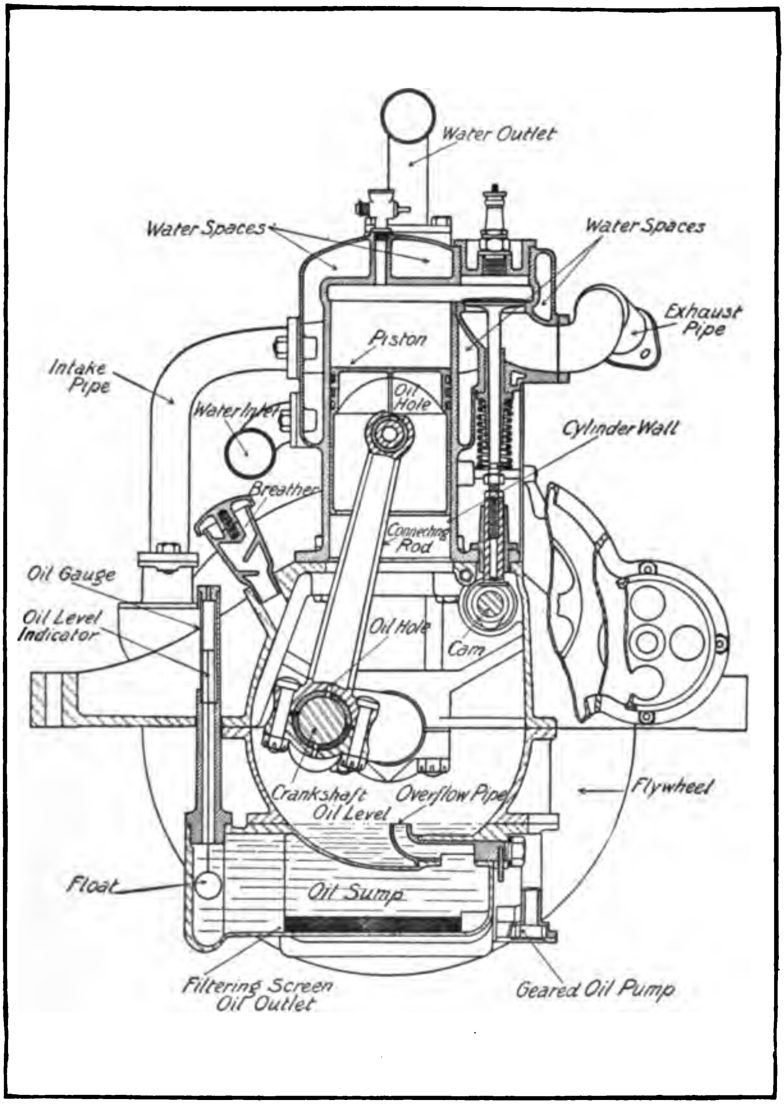


FIG. 111.—Sectional view of typical motor, showing parts needing lubrication and method of applying oil by constant level splash system.
Note also water-jacket and spaces for water circulation.

gearing to actuate the pump plungers in the interior of the case. The form shown at B is provided with two leads and is driven by a ratchet which is adapted to feed oil by a reciprocating movement instead of the continual rotary movement. The device shown at C contains eight individual oil pumps and will deliver oil to eight bearing points. The constant level splash system is a popular method of supplying lubricant to automobile engines.

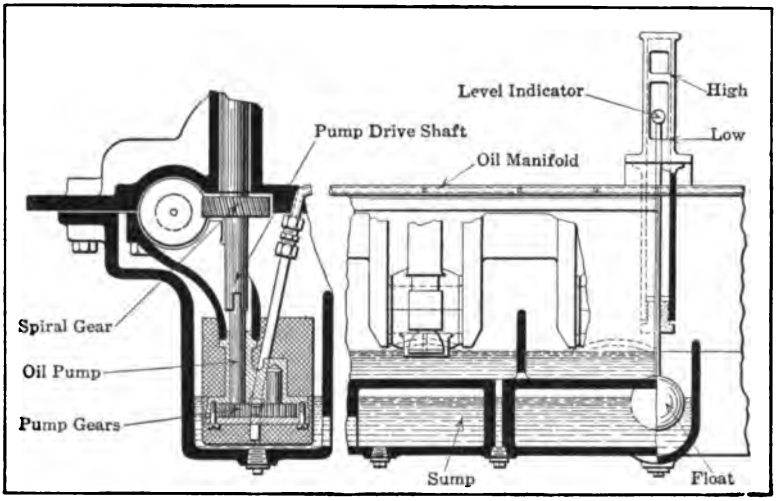


FIG. 112.—Sectional view of part of Rutenber engine, depicting method of driving oil pump and distribution to bearing points.

This is shown in some detail at Figs. 111 and 112. The oil supply is carried in the sump or container integral with the crank case and is drawn from this member by a gear pump which forces it into the engine interior through a suitable outlet pipe. After the oil spray lubricates the interior of the engine thoroughly it collects at the bottom of the crank case, where it is picked up by suitable oil scoops on the bottom of the connecting rod and thrown around the engine interior. In the form shown at Fig. 111 an adjustable overflow pipe is provided so that the oil level in the crank case may be regulated. The height of oil is determined by the position of the overflow pipe and surplus lubricant flows back into the sump through this passage.

The mechanical oiler shown at Fig. 113 involves the use of a separate oil pump for each bearing point. A sight feed fitting is placed on the dash of the car provided with a number of regulating screws and this manifold is supplied with oil from a main pump. After the flow is adjusted by the regulating screw, the oil that collects in the sight feed classes is drawn from these members by individual pumps and forced to the bearing point through suitable conductors.

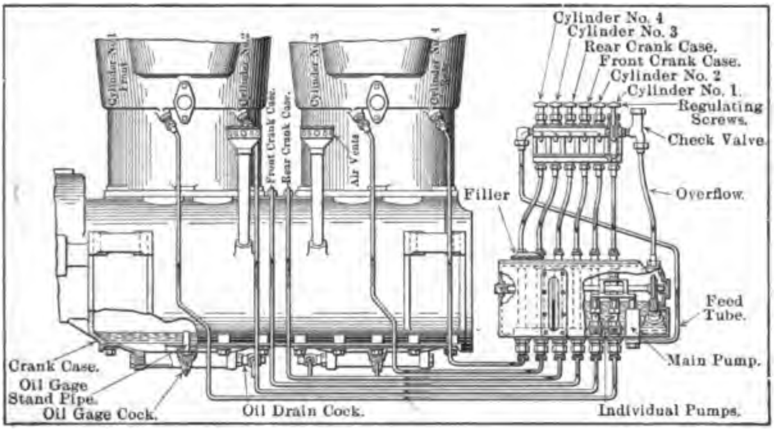


FIG. 113.—Showing application of mechanical oiler having individual pumps and leads to bearing points in connection with sight-feed gauge on dash.

A very simple lubricating system that has given good satisfaction on marine motors is shown at Fig. 114. In this no oil pumps are used, as the lubricant is introduced into the engine interior by being mixed with the gasoline. This is not a new idea as the same principle has been used with considerable success in lubricating steam-engines, where in many cases, the oil is introduced by feeding it drop by drop into the steam pipe from which point it is carried by the vapor into the cylinder. The oil is mixed with the gasoline or other fuel in proportions of one pint of lubricating oil to every five gallons of fuel. It is said that the easiest way to mix the oil is to pour a pint into a gallon of gasoline and after stirring it well to pour that into the fuel tank and then to pour in four more gallons of gasoline which has no oil mixed with it. It is claimed that the oil will stay in solution and will not settle in the tank after standing.

The cross-section of the engine at Fig. 114 on the left shows the piston on the upward position. A charge of gas has been drawn through the carburetor. In this operation the oil and gasoline separate. Here is the reason: Good lubricating oil is a heavy liquid and evaporates very slowly. Gasoline on the contrary evaporates quickly so that when the charge of air is drawn through

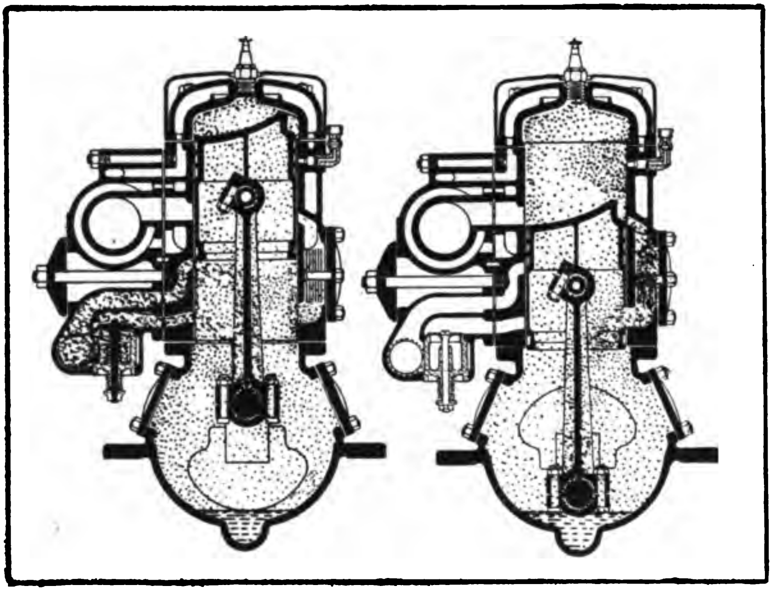


FIG. 114.—Lubricating system for two-cycle engine, in which oil is fed in with fuel.

the carburetor the gasoline mixes with the air and becomes a gas. The oil, which does not evaporate, collects in small globules which float in this gas and are carried into the crank chamber by the suction of the motor. The dots represent the passage of the oil globules.

The connecting rod being in the direct path of the incoming gas receives a coating or film of oil, which collects and flows through a special oil duct to the crankpin. The latter becoming coated with a film of oil as represented by the ring in the cut. On the down stroke of the piston as shown in cut to the right the piston pressure forces the oil laden gas through the by-pass into the cylinder, where the oil is instantaneously deposited on the cylinder wall. This operation is repeated every engine revolution. A new film of oil is supplied each time.

Where oil in the old-fashioned oil cups or oiling devices was directed to one bearing point it took time for it to spread all over the surface. In gasoline lubrication, through the rapid travel of the gas, oil is deposited in every accessible portion almost instantaneously so that with this system the piston, cylinder and crank are lubricated all over. No one side is favored as with earlier lubricating systems.

ENGINE COOLING METHODS

As power is produced in the cylinders of the explosive motors by a rapidly occurring series of explosions, and as the temperature of each of these may be over 3000 degrees F. in some cases, it will be apparent that this continued series of rapid combustion effects would soon heat the metal parts adjacent to the combustion chamber to a red heat if no method of cooling was provided. Under these conditions of extreme high temperature it would be difficult to lubricate the cylinder because even the best quality of lubricating oil would be burnt. Some trouble might also be experienced through the valves warping or the piston expanding sufficiently to become tightly bound in the cylinder. The fact that the ratio of engine efficiency depends upon the amount of useful work delivered by the heat generated from the explosion, makes it important that the engines be cooled only to a point where the cylinder will not be robbed of too much heat. While it is important that the engine should not get too hot it is equally desirable that it is not cooled too much.

The usual method of cooling small stationary and gas-engines when water is employed is shown at Fig. 115. That at A is the simplest, as it consists merely of surrounding the cylinder with a large water jacket in the form of a hopper which is open at the top and which must be filled from time to time as the water evaporates due to the heat it absorbs. In the form shown at B the water is cooled after it is discharged from the water jacket surrounding the cylinder and may be used over and over again. The water supply is carried in a water cooler from which it is drawn by a pump of the plunger type and forced to the lower portion of the water jacket. It passes out through another opening

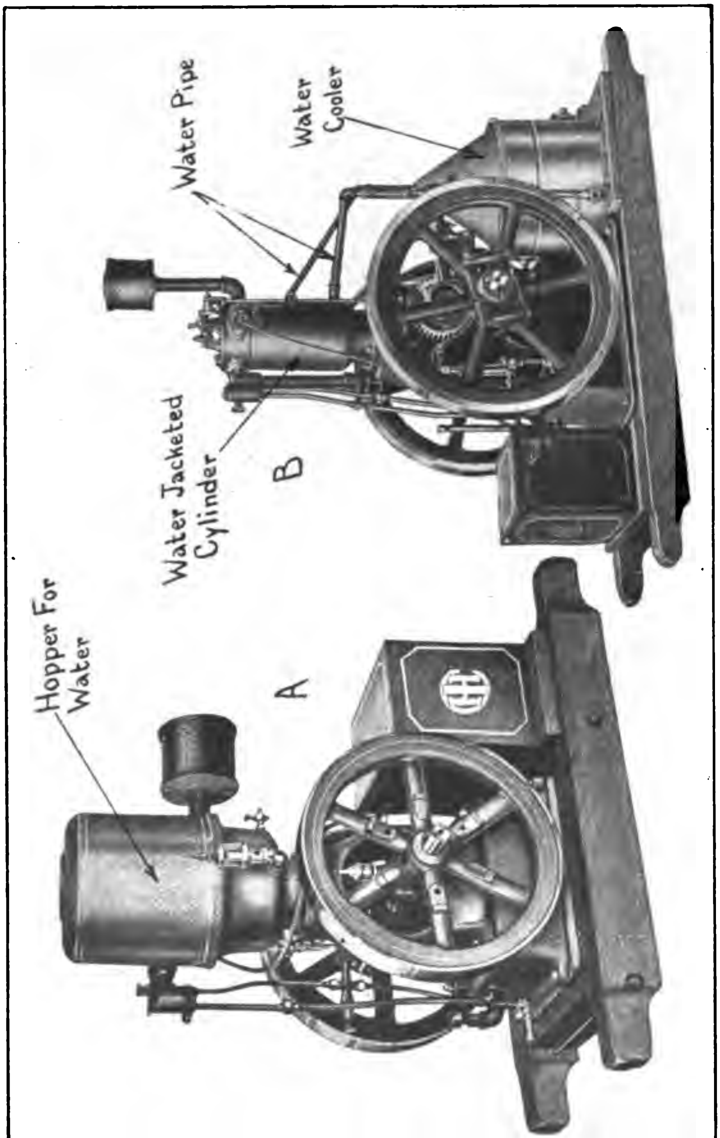


Fig. 115.—Types of water cooling systems used with vertical cylinder stationary engines.
A. Hopper cooled. B. Cooling by water circulation.

after it is heated and flows on the top of a cooling cone composed of wire gauze which exposes the hot water to the air and permits it to give off or radiate some of its heat.

Engines employed in motor vehicles cannot have the unlimited supply of water available on stationary engines so a combined water container and cooler in the form of a radiator is usually placed at the front of the vehicle where it will be subjected to an air blast. There are two methods of keeping the water in circulation through the radiator. The simplest of these, shown at

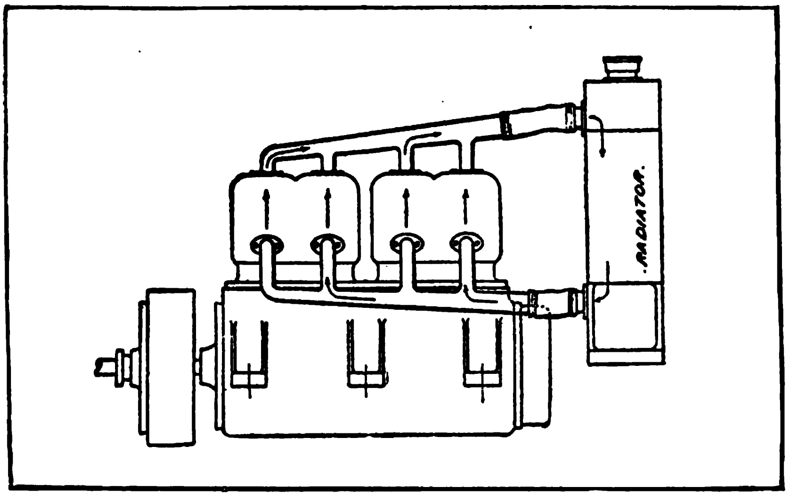


FIG. 116.—Illustrating action of simple thermo-syphon water-cooling system.

Fig. 116, utilizes the natural principle that as a hot liquid is lighter than a cold liquid, it will tend to rise, while the cool fluid will take its place. The cool water from the bottom of the radiator is directed to the bottom of the water jackets and after it becomes heated it passes out of the top of the water jackets to the top of the radiator. As it becomes cooler in that member it flows to the bottom of the device and back again to the bottom of the water jacket. The system generally favored, if an engine is subjected to heavy duty, is to use a forced circulation such as outlined at Fig. 117 in which some form of mechanically driven pump is included in the cold water line and which draws the water from the bottom of the radiator and forces it through the water jackets. As the water is kept continually in motion, independently of its temperature, it is not necessary to carry as large a supply as is needed in the thermo-syphon or natural system.

AIR COOLING METHODS

The earliest known method of cooling the cylinder of an internal combustion motor, which was employed by Daimler, was to utilize

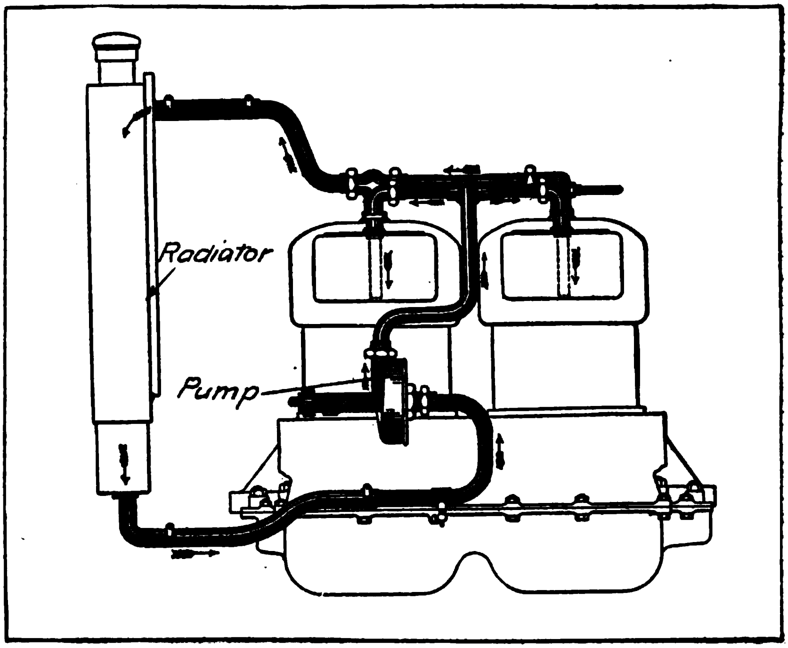


Fig. 117.—Conventional forced circulation system employing mechanically driven pump to insure movement of water.

the flywheel as a fan and to direct the current of air from that member through a jacket surrounding the cylinder. As the gasoline-engine at that early period was not as efficient as the later forms, other conditions materialized and made it desirable to cool the engine by water. Even to-day when air cooling has demonstrated that it is thoroughly practical in numerous applications, there exists a prejudice against it and it is used only to a limited extent.

The simplest method of cooling by air is as outlined at Fig.

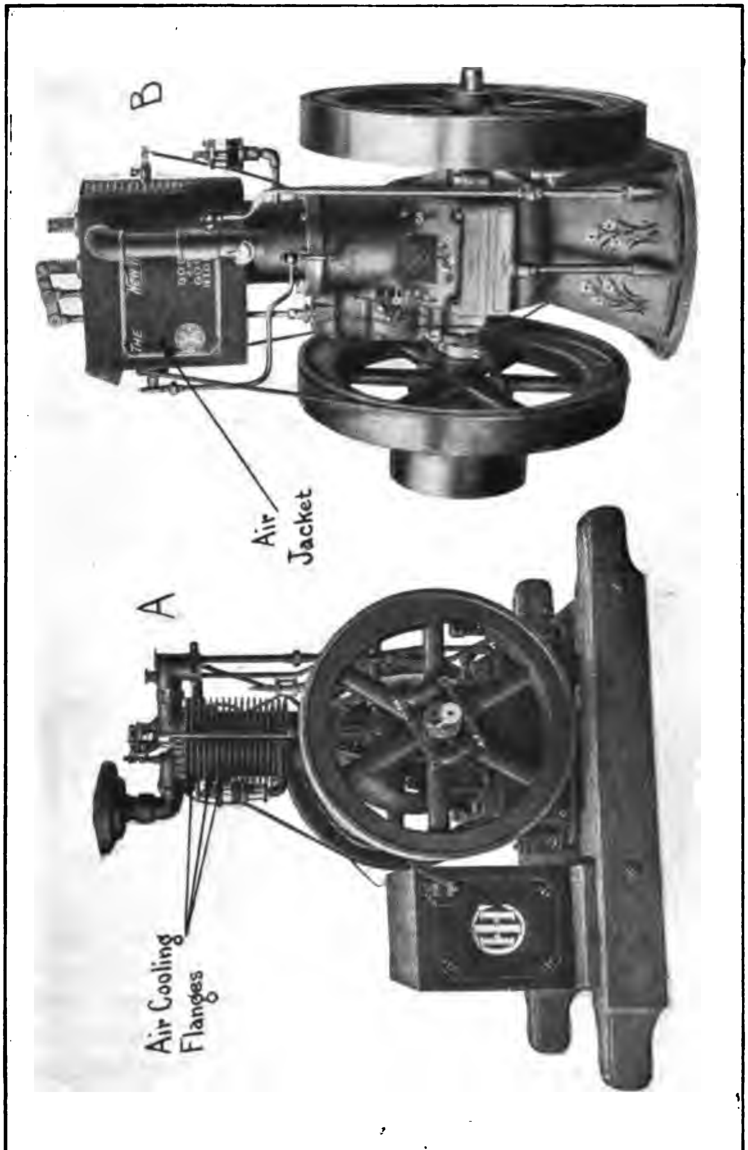


Fig. 118.—Two systems of air-cooling used on stationary engines. A. Heat radiating flanges on cylinder. B. Air blast in cylinder enclosing jacket.

118A. This depicts a small stationary power plant in which the cylinder is provided with a series of ribs or flanges which increase the available radiating surface. In the engine shown cooling is purely by radiation, as the heated air rises from the cylinders and its place is taken by cooler, heavier air. In the air cooling system outlined at B a practical application of the air jacket method is exemplified. A fan is mounted at one side of the cylinder (as shown in the sectional view at Fig. 120 in which the air jacket is removed) and as the air current is confined against the cylinder flanges and must pass over them before being exhausted it will be apparent that cooling is by air in motion, the air flow being produced by the fan. The fan rotates at a high rate of speed, as it is driven by a leather belt that encircles the large flywheel. The air cooling system by means of radiating flanges has also been used with some degree of success on two-cycle engines, a typical example of which is shown at Fig. 119.

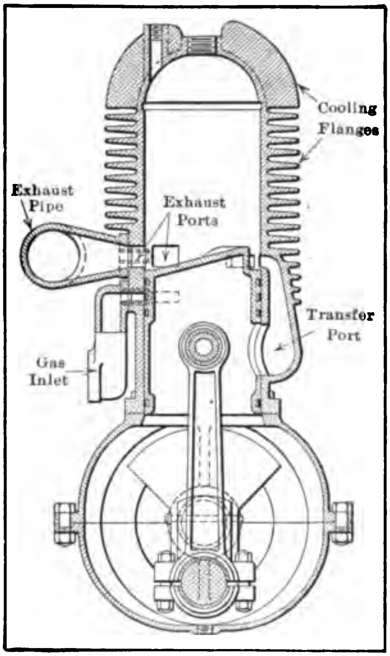


Fig. 119.—Sectional view of Chase two-cycle engine, a two-stroke form successfully cooled by air flanges cast integral with cylinder.

Air cooling systems are based on a law formulated by Newton which is "The rate for cooling for a body in a uniform current of air is directly proportional to the speed of the air current and the amount of radiating surface exposed to the cooling effect." There are certain considerations which must be taken into account in designing successful air-cooled engines. In the first place, large valves must be provided to insure rapid expulsion of the flaming exhaust gas and also to admit the fresh, cool mixture from the carburetor promptly. The most successful air-cooled engines have the valves placed directly in the cylinder head except on very

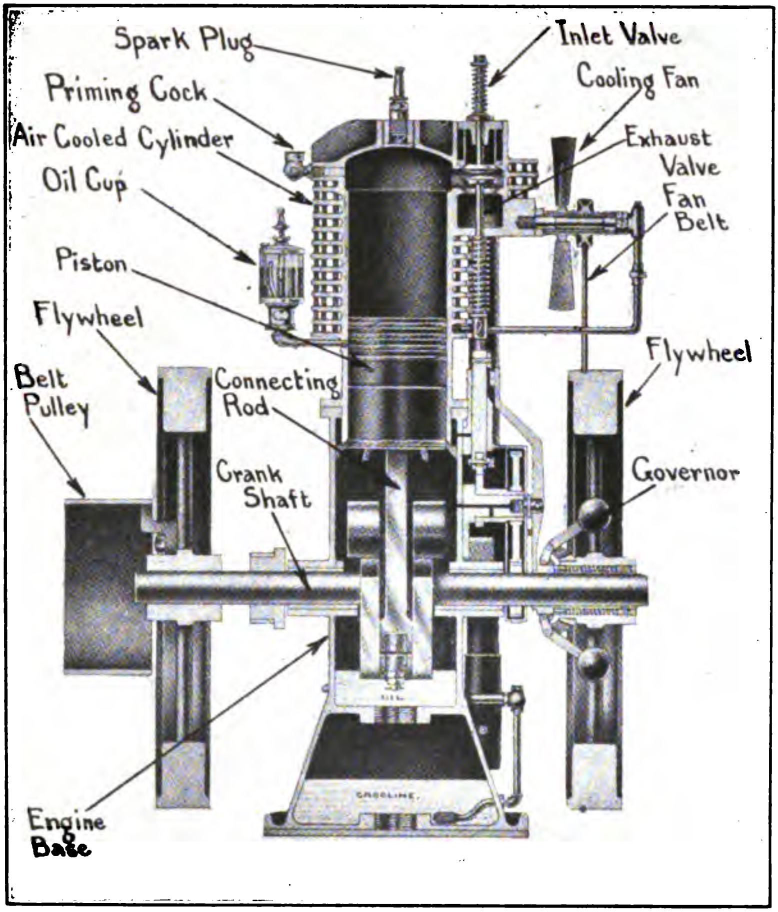


FIG. 120.—Sectional view of simple air cooled engine, showing internal construction.

small engines used for motorcycle work. If the cylinders of the L or T form are used considerable difficulty will be experienced, if these members are of large size, from unequal expansion of the heated metal due to the projecting and unsymmetrical valve chambers. When the valves are mounted in the head of the cylinder, the dimensions of that member are uniform and as it is a purely cylindrical ribbed member with the metal uniformly disposed, it will expand equally at all points.

When high power is desired, or when it is imperative that engines develop high speed, as in motor vehicle service, much better results will be obtained by the use of multiple cylinder engines because there is a certain limit to the size of a successful air cooled cylinder. While it is possible to build slow speed motors intended for stationary service with cylinders of seven and eight inches bore it is not considered practical to use cylinders much larger than four inches bore for motor vehicle service. All successful air cooled engines of the automobile type have been of the multiple

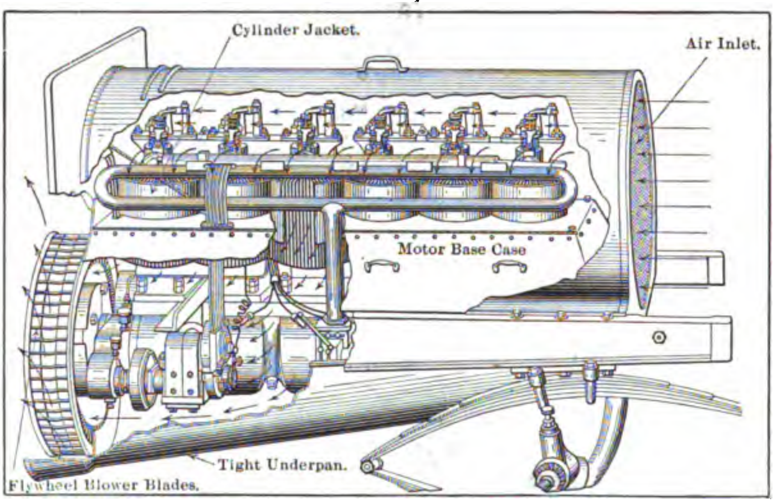


FIG. 121.—Positive cooling method used on Franklin automobiles, in which all currents are drawn through cylinder jackets by flywheel fan suction.

cylinder form. A refinement that is said to make for more effective cooling is the use of an auxiliary exhaust valve which is set to open as soon as the full force of the explosion has been spent. This is located at a central point in the cylinder so that an auxiliary exhaust port will just be above the top of the piston, when that member reaches the end of its power stroke. It is claimed that over 70% of the flaming gas will leave the cylinder through the auxiliary exhaust valve and this leaves but 30% of the exhaust gases to be discharged through the regular exhaust member in the cylinder head.

The engine shown at Fig. 121 is an automobile power plant which is cooled successfully by an ingenious system of air circulation. The cylinders are provided with vertical ribs or flanges and are encased by jackets which form part of a sheet metal casing that covers the entire lower portion of the power plant. The flywheel is provided with a series of curved blades so that it is practically a Sirocco blower and as it turns it creates a partial vacuum in the compartment formed by the motor base casing and the air tight under pan. Owing to the strong suction created, air is drawn in through the front end of the hood and down through

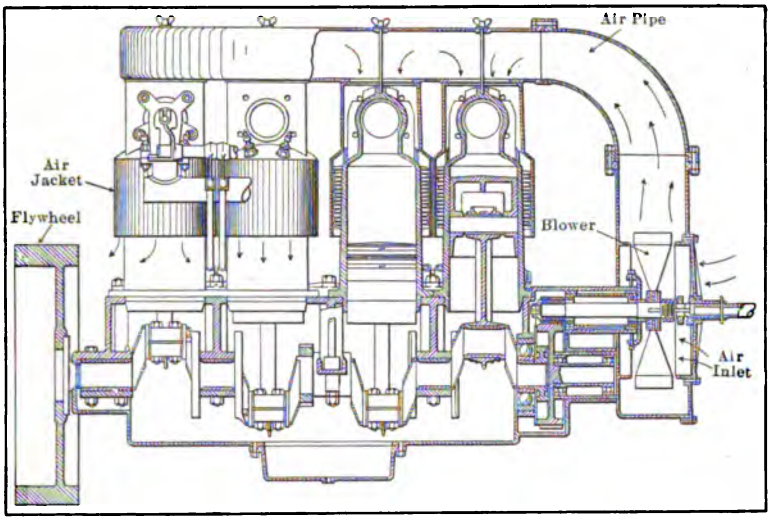


FIG. 122.—Air-jacketed Frayer-Miller engine used in Kelly trucks. Cooled air currents directed over cylinders by positive air-blower system.

the cylinder jackets. As the air currents pass over the flanges at high velocity and as there is a large amount of exposed surface, the excess heat is promptly disposed of and absorbed by the air passing around the cylinder. As this becomes heated it is ejected from the motor base compartment by the action of the blower flywheel. As the fan is part of the flywheel and is driven directly from the engine crankshaft there can be no failure of the driving means and a positive draft of air must be induced around the cylinder head and over the cylinder walls as soon as the motor is started. As the velocity of the air currents increase directly as the motor speed augments, positive cooling is obtained at all engine speeds.

Another air cooling system on somewhat the same plan as that previously described is shown at Fig. 122. In this the blower is mounted at the front end of the motor and supplies the cooling air to the cylinder jacket through a large air pipe joining the blower casing with extensions from the top of the air jacket.

The fans used in air cooling systems are of two types and are shown at Fig. 123. The three-blade fan driven by positive gearing

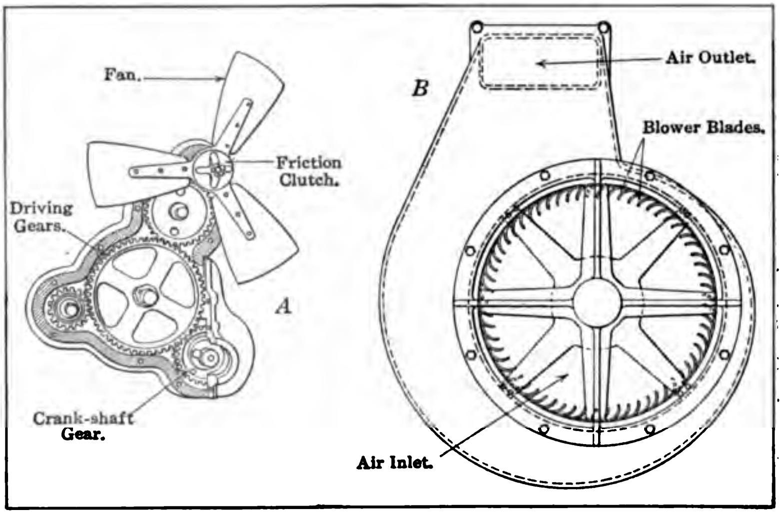


FIG. 123.—Two forms of positive air fans used in automobile cooling system.

A. Gear-driven, three-blade fan utilized to draw air through Winton radiator. B. Blower member used on Kelly air-jacketed cylinder motor.

is shown at A while the blower type which must be connected to an air jacket is shown at B. Air cooling has a number of important advantages, especially when applied to small stationary engines and to power plants of automobiles and motorcycles. The greatest advantage gained by its use is the elimination of cooling water, which feature is valuable in extreme cold weather when the conditions would militate against the successful use of water cooled systems without the addition of chemicals to the water to prevent it from freezing. Air cooled engines as a rule, use less fuel than those cooled by water because the higher temperatures of the cylinder does not permit of taking in as dense a fuel charge on the intake stroke. Obviously the power is reduced proportionately, though as more of the heat units liberated by the explosion are turned into power an air cooled engine of given dimensions will prove somewhat more economical of fuel than a water cooled power plant of the same size. One of the disadvantages of air cooling methods as stated by those who do not favor this system is that engines cooled by air, especially if the simple radiation system is employed, cannot be operated for extended periods under overloads or at very high speed without heating up to such a point that premature ignition of the charge may result. When used properly, air cooled engines give very good results, especially in the light stationary forms and in the types adapted for motorcycle use and aerial navigation.

MUFFLERS FOR EXPLOSIVE MOTORS

The method of muffling the sound of the exhaust, as well as the sound or clack of the valves, was a puzzling problem to the early builders of gas-engines. The matter has finally sifted down to a plain cast-iron box of from 1 to 3 cubic feet capacity, set near the engine, and into which the exhaust-pipe is connected, and continued by a separate connection to the outside of a building. Connection of the exhaust with a chimney should not be made under any circumstances, as there are unknown elements of explosion liable to be accumulated in the line of the exhaust that might do damage to a chimney; and for the same reason the muffler-box should be made strong enough for a pressure equal to the explosive power of the gas and air mixture, or say 175 pounds per square inch. This insures safety from any explosion that may accidentally occur in the exhaust by missed explosions in the cylinder or otherwise. The muffler-pot is also a water-catch, in which part of the water-vapor formed by the union of the hydrogen and oxygen is condensed. It should have a draw-off cock a few inches above the bottom, so that the muffler may always have a little water in the bottom, the water having been found to have a deadening effect on the exhaust. A second muffler-pot has been found to still further deaden the exhaust, and is preferable to throttling the exhaust by mufflers with perforated diaphragms, as used on vehicles and boats. In all cases an enlargement of the exhaust-pipe from the muffler to the roof by one or two sizes

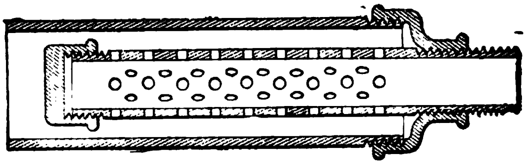


FIG. 124.—Gas-pipe muffler.

larger than the engine exhaust, will modify the intensity of the exhaust at the roof, and often abate a nuisance.

In Fig. 124 is shown a simple muffler easily made from ordinary gas-pipe and fittings, consisting of a perforated exhaust-nozzle within an open-end pipe of larger size. Its construction is shown in the cut. The outside or shell of all mufflers should be felted with asbestos to deaden the vibration and sound.

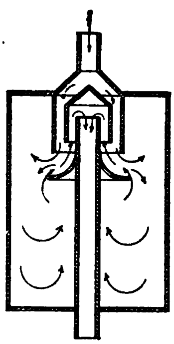


FIG. 125.—Thompson muffler.

Fig. 125 shows a novel muffler of the Thompson type, which has a cylindrical chamber with a hooded spreading inlet-pipe; and a deflector on the exit pipe, by which the exhaust puffs are expanded in the cylinder and issue in a nearly constant stream. Other types of mufflers have strong wire-gauze cylinders within the drum so arranged as to break the impact and disperse the exhaust before it leaves the outer shell.

Mufflers for automobiles and launches have been the subject of much designing in order to have them meet the requirement of almost absolute silence, so much to be desired. The method of perforated tubes with wire-cloth casings of large area for cutting the exhaust into infinitesimal streams, and of so large an area that the back-pressure may be reduced to an imperceptible amount, seems to be in the right direction for vehicles, and an extension of the terminal under water at the stern of launches with a small vent above water has given good results. The vent prevents water drawing back to the muffler when the motor stops. For large stationary motors a variety of designs for the internal space of a muffler-box have been made, all seeming to tend to obtain the desired conditions. A series of perforated plates, both flat and circular; small stones filling the muffler-box, through which the exhaust passes; a spiral case within the muffler-box; in fact, almost any device which tends to stop the sudden impact of the exhaust and its expansion are the means that modify and in a measure prevent the noisy propensities of the explosive motor. To prevent nuisance to neighbors by open-air exhaust, the turning down of the exhaust-pipe into a barrel or second muffler-pot with a few

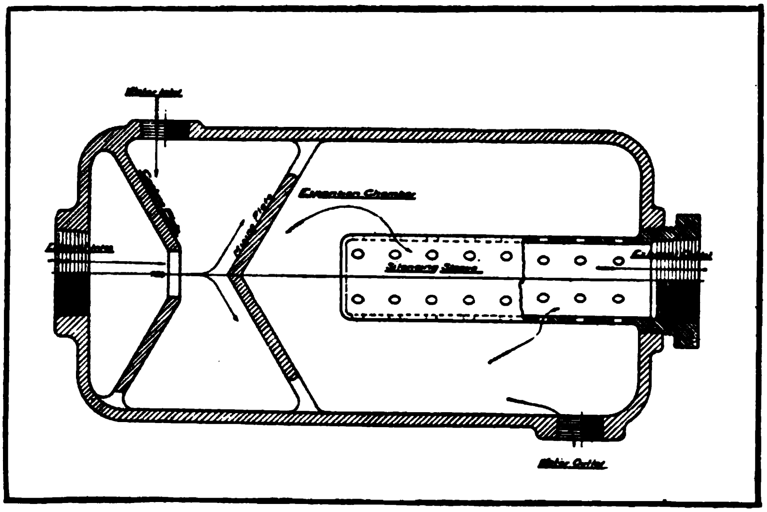


Fig. 126.—Simple muffler for marine engines.

inches of water, has given satisfaction in many cases. It prevents the spread of oil-vapor into neighboring windows.

An effective marine silencer is shown at Fig. 126. In this device the cooling water enters through an inlet at the top of the muffler-box and flows down a cooling plate which is struck by the entering stream of gas coming in through the inlet opening. As this exhaust gas is saturated with particles of water the stream of hot gas and liquid will strike the mixing plate and flow into the expansion chamber through the openings at the edge of the mixing plate. As the gas has expended some of its heat in vaporizing a portion of the cooling water injected into the muffler it is comparatively cool by the time it reaches the expansion chamber. To pass out from the expansion chamber to the air it is necessary for the gas to pass through a series of perforations in a tube extending from the exhaust outlet to the interior of the expansion chamber. The water that has not been vaporized collects at the bottom of the expansion chamber and is discharged through a suitable outlet on the bottom of the silencer.

It is necessary to provide mufflers or silencers of maximum efficiency on automobiles because it is not only important to muffle the gases so that but little noise is present, but this must be done without producing back or negative pressure in the muffling device that will cause a serious loss of power in the engine. A number of muffler forms adapted especially for motor vehicle service are shown at Fig. 127. The simplest of these, outlined at A, consists of a sheet metal shell having its ends closed by cast metal pieces. This has several times the volume of the cylinder and the gases expand to about atmospheric pressure before they are discharged through the series of small holes in the bottom. The form at B consists of a number of concentric chambers which afford an excellent opportunity for the gas to expand to atmospheric pressure and which break the entering gas stream into a series of smaller streams, which issue from the exhaust opening in the bottom of the muffler shell in a very silent manner. The form shown at C is a modification of that outlined at B and is provided with a large number of vertical baffle plates to break up the gas instead of being separated into concentric expansion chambers. The form at D consists of a central pipe member around which are placed a number of pairs of light stamped sheet metal discs, each pair forming an expansion chamber. The gas issues from the centre of the pipe into the expansion chamber and as the discs spring away slightly at their edges under the pressure of the issuing gas stream, an annular discharge passage is formed at the point of contact of each pair of cupped discs which insures a thorough breaking up of the exhaust gas. It is also claimed as an advantage of this method of construction that the thin sheet metal cups are easily cooled and that a material silencing of the gas is produced by cooling it as well as by allowing it to increase in volume. The form shown at E is built on the ejector principle and is claimed

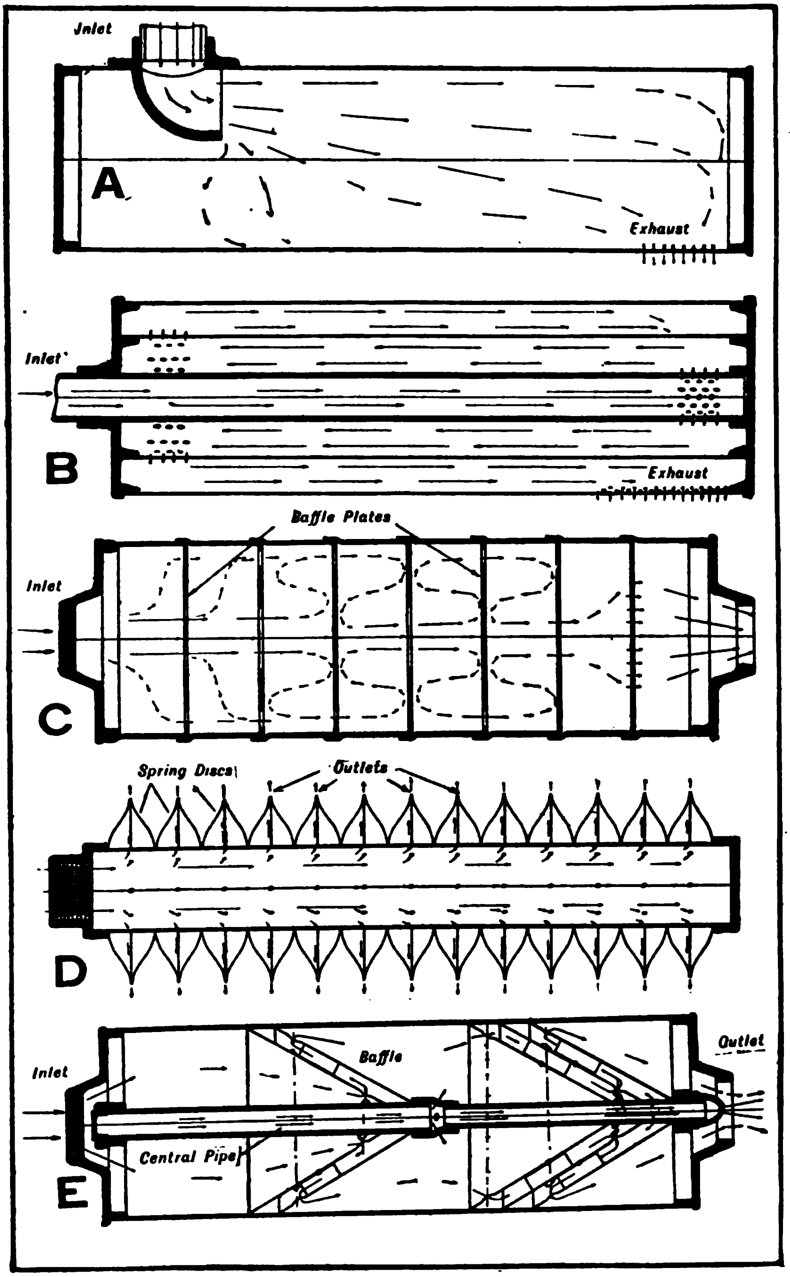


FIG. 127.—Muffler forms adapted to reduce pressure of exhaust gases before discharging them.

to be particularly efficient, which is due to its design. The internal construction is such that part of the gases which pass through the central pipe do so with such speed as to tend to produce a condition of partial vacuum in the interior of the device which in turn promotes a more ready expulsion of gas by drawing the main portion through the muffler. This device is of the baffle plate type having conical or funnel shape partitions instead of the usual vertical or horizontal dividing walls because the angular form lends itself to the ejector principle better than other types.

In order to take advantage of the gain in power, which results when the gases are discharged directly into the air instead of being passed through a muffling device many automobile and marine engine builders provide a simple relief or cut-out valve between the exhaust manifold and the muffler. This is arranged so that when opened the gases are free to pass directly to the air instead of through the muffling device and the elimination of back pressure tends to produce more power from the motor. Even with the most efficient form of muffler it is said that a gain of about 5% in power is possible if the gases are exhausted directly to the air. A cut-out valve is also valuable because it permits one familiar with gasoline motors to detect irregularities in engine operation by the sound of the exhaust.

CHAPTER XII

CONSTRUCTION DETAILS AND PARTS OF LARGE ENGINES

The design of an explosive motor should start from some assigned dimension of the cylinder, based upon the assumed number of revolutions, its required horse-power, and the quality of the fuel to be used. Compression is also a factor to be considered in a nice adjustment of the details for the required power. In Chapter VIII we have given a few samples of practice among builders of engines as to size, power, and speed, and a table of sizes of the essential parts for a clearance of 33 per cent. and compression of 50 to 60 pounds per square inch. The table

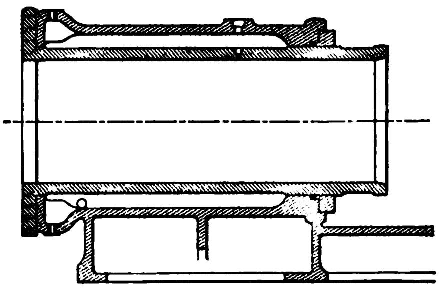


Fig. 128.—The cylinder.

represents the actual or brake horse-power, and the sizes of the cylinders and speed are a mean, as in ordinary practice for stationary engines. High-speed motors are a specialty and require some experience for successful designing. The diameter and stroke of a proposed design must be derived from some assumed mean pressure and speed for the relative conditions of impulse for either of the cycles contemplated. The factors of fuel power and compression are also essential elements of design in construction that need primary consideration. From these data the indicated horse-power may be computed and the actual or brake horse-power obtained from some known mechanical efficiency of this class of motors. From the many sectional and detailed illustrations throughout this work, the general constructive design of the various models of the two types of motors of the horizontal and vertical styles, and in the stationary motor vehicles, and marine classes, are sufficiently shown as a guide for the draughtsman and amateur

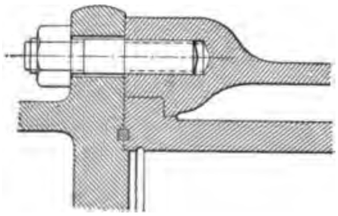


Fig. 129.—Gasket-joint.

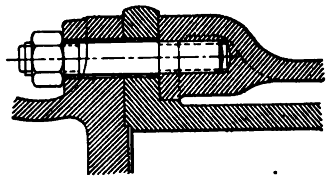


Fig. 130.—Plain joint.

of constructive ability; and together with the computed sizes of parts formulated, should enable any draughtsman of ordinary experience to make a creditable design of an explosive motor.

In Fig. 128 is shown the German method of making the cylinder and water-jacket in separate castings; the jacket being made an integral part of the bed-frame and bored with aligned bearings to fit their counterparts on the cylinder. The two designs for bolting

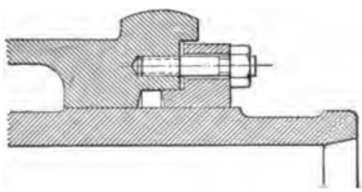


Fig. 131.—Stuffing-box joint.

the cylinder and water-jacketed head separately to the jacket are shown in Figs. 129 and 130. In one a groove is made to hold a metallic packing, while the other may be a ground-joint or plain gasket. In Fig. 131 are given the details of the stuffing-box. By this arrangement the cylinder is allowed a movement due to difference of temperature between the cylinder and jacket, and yet makes a rigid connection between the cylinder and bed-frame through the jacket.

In Fig. 132 is illustrated a section of a piston of German type, nearly two and a quarter times its diameter in length, showing the German practice in regard to the number of rings and their disposition. In Fig. 133 is given another piston of twice its diameter in length, and in Fig. 136 a bushed piston of one and a half times its diameter in length, one and a half diameters for the length of the piston being the average of American practice. The length of the cylinder must include the assumed length for clearance, less an allowance for protrusion of the piston at the end of the outward stroke, which may be studied from an examination of many sectional views of engine details in the pages of this work. The short piston in Fig. 134 is nearly the proportion in general use in the United States, with the number of rings varying with different builders.

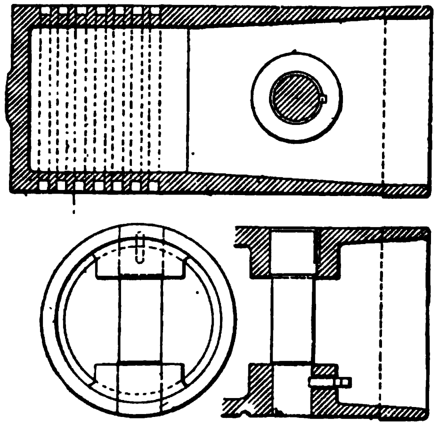


FIG. 132.—The long piston.

The following plan has been suggested by Mr. E. W. Roberts for easily entering a piston and rings into a cylinder: Take half

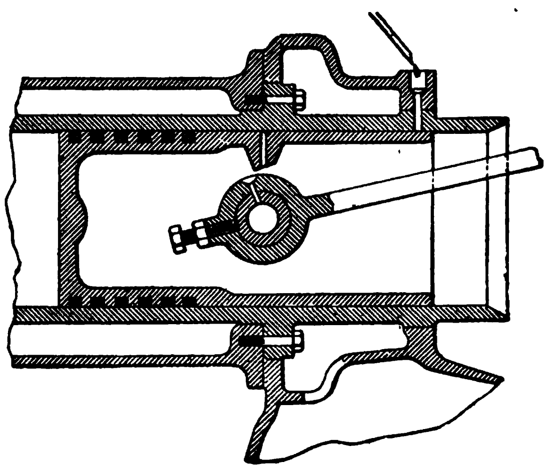


FIG. 133.—Medium-length piston and oiling device.

a dozen or more strips or bands (S, S, Fig. 135), the thickness of which is equal to one-half the difference in the diameter between the bore of the cylinder and that of the counterbore. Slip the

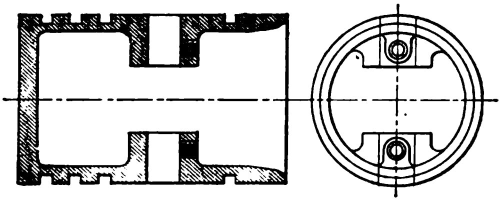


Fig. 134.—Section of short piston.

piston in part way and then put in the strips. Bend the strips outward, as shown in the sketch, forming a tapered guide which will gradually close the rings as the piston is pushed in. In case there is a port leading into the counterbore these strips will also

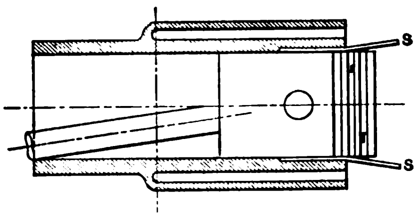


Fig. 135.—Replacing a piston.

prevent the rings from jumping into the port. Almost any machinist will realize that this is a very sure and efficient method, and it does not shove the edge of the rings against the end of the counterbore, which is quite often an abrupt shoulder and likely to require much pressure to push the rings past the shoulder of

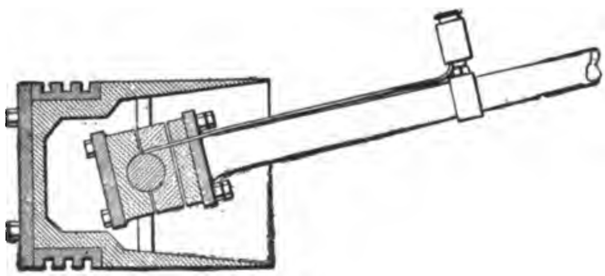


Fig. 136.—Bushed piston and oiling device.

the counterbore. The number of piston-rings varies with different builders, the Germans using the larger number. For small engines, three rings are sufficient, while four are used on medium-sized pistons, with sometimes an extra ring toward the open end of the piston.

The connecting-rod should always have an adjustable box at the crank end and in medium and large engines also at the

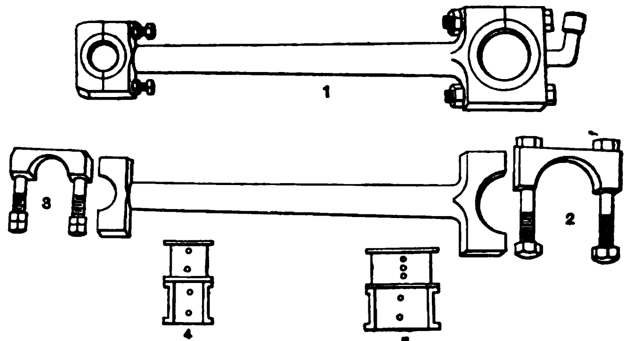


FIG. 137.—Bushed piston-rod.

piston end. Very small engines need only have a solid eye at the piston end, bushed or not as judged best. In Fig. 137 are shown the details of a bushed piston-rod much in use, and in Fig. 138 a box-rod with a strap take-up and keys for the piston end. A

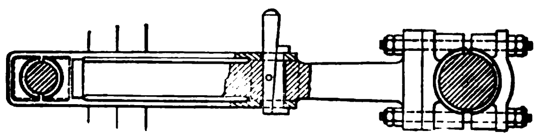


FIG. 138.—Strap take-up piston-rod.

novelty in the make-up of large vertical motors has been adopted in the Warren motor. The connecting-rods are made in two parts, as shown in Fig. 139, joined by a heavy bolted flange near the centre of the rod, which allows the piston to be taken down through the bottom of the cylinder for inspection and repairs without disturbing the cylinder-head and valve gear, which is attached to the cylinder-head.

In Fig. 140 is shown a simple piston-pin oiling device. A small tube B, extending from the oil-cup C, and attached to the oil-port in the piston, conveys the oil to a recess in the connecting-rod box at D. The recess is long enough to receive the oil in all positions of the connecting-rod, the sight-feed oil-cup at O, feeding both the piston and its pin.

Fig. 141 details the balanced crankshaft, with a novel method for oiling the crankpin, consisting of a disk with a cavity to

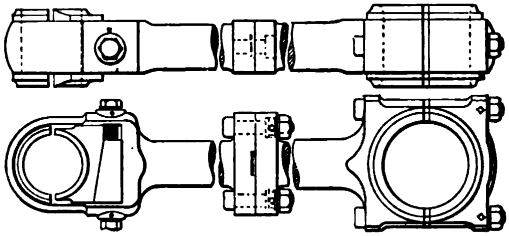


FIG. 139.—The two-part connecting rod.

receive the oil which is spread to the outer side by the centrifugal force of revolution and through the drilled passages to the crankpin box. The proportions are to a scale in parts of the crankpin diameter. The base frame as usually made with flange-bolted cylinders is shown in Fig. 142, but its design is illustrated, with many variations to suit special conditions, in the general views in the following pages.

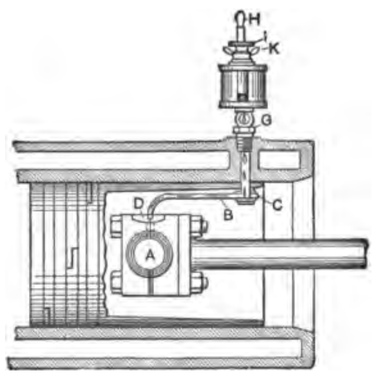


FIG. 140.—Piston Pin Lubrication.

In Fig. 143 is delineated the crankshaft of the larger German motors with an outboard-bearing and enlarged shaft diameter for the safer keying of the flywheel. It will be seen that the left-hand end of the shaft has its size reduced to accommodate the desired small size of the spiral gear. All parts of tnis cut are made to a scale derived from the diameter of the main journal as a unit.

It will be noticed that the shoulders of the journals are lipped in order to divert the excess of oil into the ring oil-reservoirs.

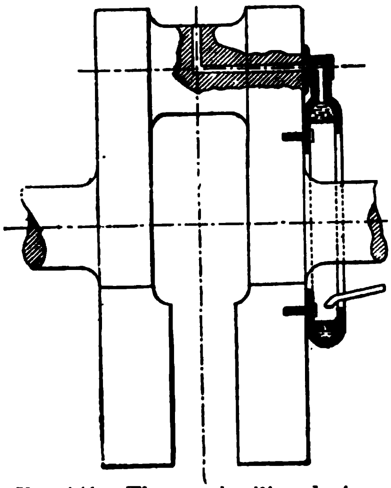


FIG. 141.—The crank oiling device.

In Fig. 144 is shown the method of fastening the counterbalance to the crank by a short stud-bolt, with the nut in a mortise in the side of the counterbalance. The centrifugal strain is countered by the diagonal keys in the side-bearing. Fig. 145 shows the ordinary method of fastening the counterbalance weights to the crank: a close fit and two strong tap-bolts or stud-bolts and nuts for each weight.

In Fig. 146 is shown the design of a flywheel of approved form. The curved side of the spokes should turn forward, as shown by the arrows, which produces compression of the spokes at the moment of impulse and thus avoids possibility of fracture. This form is also safest in casting, as it avoids fracture by shrinkage. The models of straight-arm flywheels are illustrated further on; for flywheel dimensions see Chapter VIII.

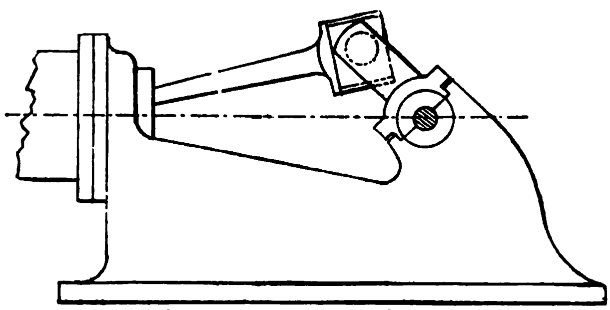


FIG. 142.—The base frame.

In Fig. 147 is shown the model of the plain single-crankshaft in general use. In Fig. 148 is shown the three-throw crankshaft of the Westinghouse Machine Company, with their method of balancing by screwing the balance-blocks to the crank-arms. In Fig. 149 are represented a German type and hori-

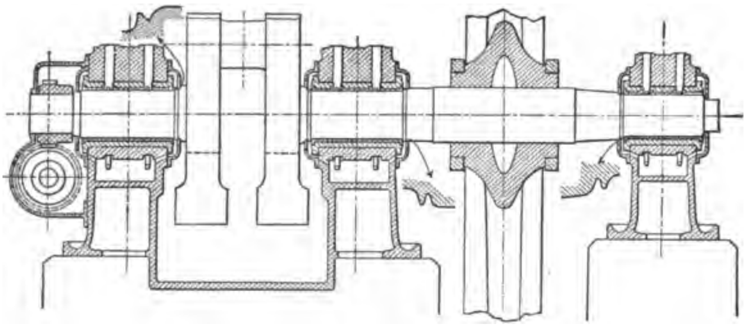


Fig. 143.—German shaft and bearings.

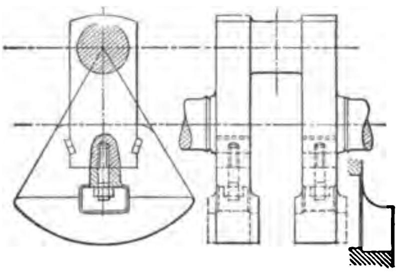


Fig. 144.—Fastenings of the crank counter-balance.

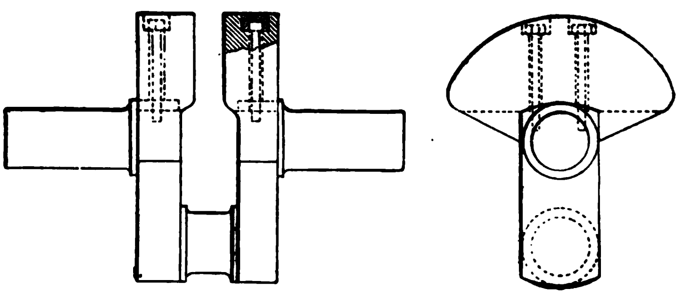


Fig. 145.—Counterbalanced crank, bolts or stud-bolts and nuts for each weight.

zontal housings, with the method of keying the crank-counterweight in addition to the usual stud-bolts and nuts. In Fig. 150 are illustrated a longitudinal and a cross section of a German journal-bearing with a double-ring self-oiler. The cuts represent nearly the exact proportions, using the journal-shaft diameter as a unit. In Fig. 151 is the sectional design of a single-ring oiling device of German design.

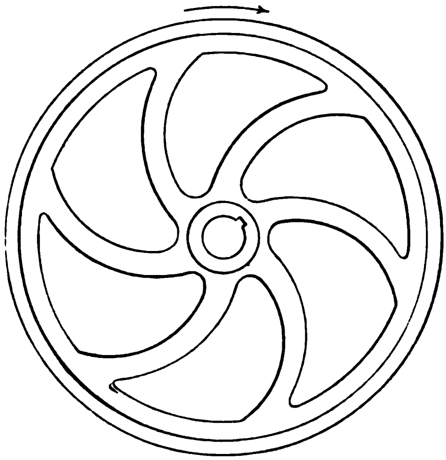


Fig. 146.—Flywheel of approved design.

The pillow-block of an explosive motor is deserving of special care in its design, in order to withstand the shock of explosion without injury to itself or the crankshaft. A perfect journal fit will often save the breaking of a crank. In Fig. 152 is detailed a half-section of a main journal-box of approved design. The composition box has a stop-rib to keep it from turning. The length of the journal-bearing should be twice the diameter of the crank-

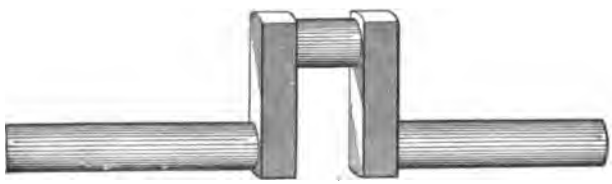


Fig. 147.—The plain single crank.

Fig. 148.—Westinghouse three-throw crank.

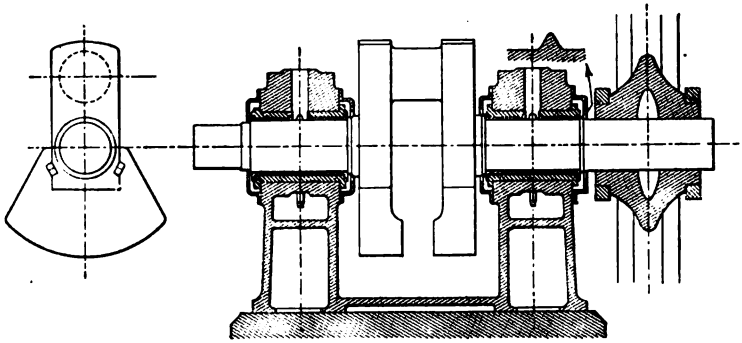


FIG. 149.—Crank-shaft and housings.

pin. The proportions in the cut are a fair representation with the journal-shaft diameter as a unit. Also see illustrations of motor details further on. We illustrate both the horizontal and angular style of journal-box housings, as both are in general use. It is claimed that the angular housing is the least complex and most reliable for strength and wear to sustain the one-direction shock of explosion.

One of the fine points in fitting the main journal-boxes for perfect work is to give the ends a perfect bearing, so that they may not sag at the inner end by the explosive blows and elasticity of the shaft, and thus extend the length of the shaft between its actual bearings; this condition being too often neglected, resulting in the mystery of a broken shaft. Boring the housings and turning the bored boxes on the outside with keys to hold them in place is probably the best practice.

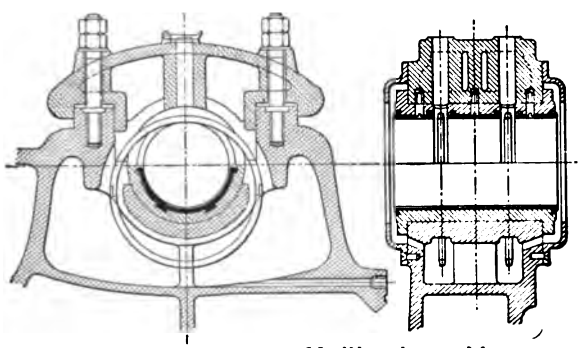


FIG. 150.—Horizontal self-oiling journal-box.

There is a difference of opinion among designers and builders of explosive motors in regard to the kind of metal or alloys for the journal-boxes, each advocating some special composition as the best: phosphor bronze, Tobin bronze, aluminum bronze, tin-copper

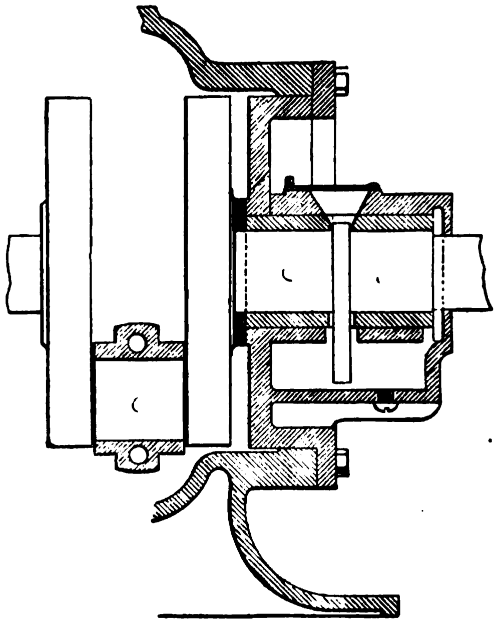


FIG. 151.—Single-ring self-oiling journal-box.

bronze, and Babbitt metal being in use. For low-compression motors the phosphor and Tobin bronzes give good results. Babbitt metal is a cheap substitute in fitting; but the hard alloy is weak and liable to crack under the heavy blows of explosion, and the soft alloys are still weaker and liable to spread. For high-compression motors the ten per cent. alloy of aluminum and copper (aluminum bronze) and those of tin and copper are tough and resisting, wearing well. Probably there is nothing better than aluminum bronze for hard work.

EXPLOSIVE MOTOR DIMENSIONS

The diameter of the cylinder of an explosive motor and its initial pressure are the safest bases from which to compute the dimensions of all the parts subject to strain by the action of the motor. As compression of the explosive charge has a great controlling influence on the initial explosive pressure, it should be

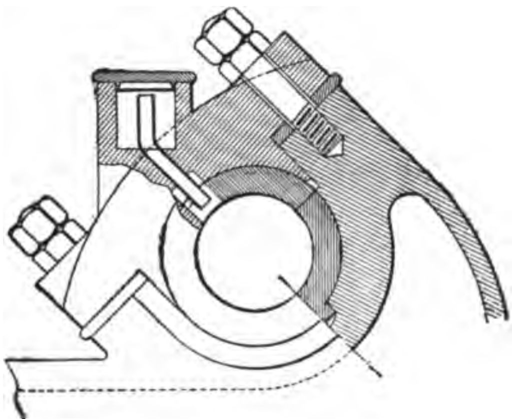


FIG. 152.—Main journal-bearing.

made an exponent in every formula for strength against the strains of explosive pressure. In a cylinder, as well as in other parts, the dimensions given in the formulas are for finished sizes; for cylinders, ample allowance should be made in the casting for boring. Any simple proportion of diameter to thickness of cylinder wall, while giving the relative strain on different-sized cylinders, does not satisfy the practical condition of manufacture; which, to be safe and practicable on a basis of five times the extreme pressure, would be practically too thin for small cylinders and too thick for the larger size. The tendency of constructive design at the present time is toward economy of material in general terms and the special requirement of lightness for marine and automobile service.

The strains on the various parts of a motor most to be considered are derived from the explosive moment, which are the pressures and strains due to the most intense part of the motor's work. The ultimate or breaking resistance of the material of construction of the quality suitable for such work, is for cast iron suitable for cylinders — from 18,000 to 20,000 pounds per square inch, for which one-sixth, 3,000 pounds, is a safe factor or margin for computing the thickness of cylinder walls subject to an extreme pressure of 500 pounds per square inch. Then for obtaining the least safe thickness of cylinder wall, under the consideration of strength alone, the safe-resisting thickness will be derived from the extreme or maximum pressure in pounds per square inch multiplied by one-half the diameter of the cylinder in inches.

$$P \times \frac{D}{2} = \text{stress, and } \frac{\text{stress}}{\text{factor of strength}} = \text{thickness in inches}$$

or decimals.

For example, a 10-inch cylinder and maximum pressure of 500 pounds with a safe factor of 3,000 pounds. $500 \times \frac{10}{2} = 2{,}500$, and $\frac{2{,}500}{3{,}000} = .833$ inch thick, to which should be added enough to meet the contingencies of unequal thickness in setting the core and for boring in the making of the pattern.

The next vital point from which trouble may arise, is the compression-strain on the piston-rod, boxes, pin, and crankshaft at the moment of explosion. The torsional strain on the crankshaft does not reach its maximum effect until the piston pressure has fallen to one-half the initial pressure, and on this only depends the diameter of the main journals to resist torsion due to the flywheel resistance. The dimensions of these parts have been developed both theoretically and by practice, from which these formulas have been derived. The author finds that the square root of the diameter in inches, divided by 5, $\frac{\sqrt{D}}{5}$, gives a much more satisfactory thickness of cylinder wall for low compression, say 40 pounds and under. For higher compression, say up to 100 pounds, a compression exponent should be added to the above formula, for which we propose $\frac{\sqrt{D}}{5} + \left(\frac{\sqrt{D}}{5} \times \frac{\text{comp.}}{250}\right)$ as giving a satisfactory safe thickness for high-compression cylinder walls at the clearance end of the cylinder. The crank end may be made thinner when the cylinder is supported by the jacket casting, or should have its thickness uniform when it is to be bolted to the frame with a flange. By this formula a low-compression 4-inch cylinder wall may be .4 inches thick, and for high compression .56 inches. The gradation will give a 10-inch cylinder .63-inch and .87-inch wall, and for a 16-inch cylinder .89 and 1.27 inches respectively for low and high compression.

For the water space, the thickness is a matter of expertness in making cores that will stand the strain of moulding and casting; but on general principles the thickness of the water space should equal the thickness of the cylinder wall; except when the jacket is made in a separate piece, when the water space may be made to suit the convenience of construction. The thickness of the water-jacket wall with a cored water space may be one-half the thickness of the cylinder wall, depending upon the method of fastening the cylinder to the bed-frame; whether flanged on the head or with side-flanges on the jacket. These are matters of study, shown in the detail illustrations throughout this work.

The sizes of valve-aperture are a ratio of the volume and piston speed for the best effect and we find that the square root of the cylinder diameter in inches, multiplied by the piston speed in feet per minute, the product divided by 600 — $\frac{\sqrt{DS}}{600} = d$, gives a very satisfactory size for the inlet-valve aperture. The exhaust-valve should be one-fifth larger in diameter. This is suitable for motors at ordinary speeds, to have the valves fitted in the head of the cylinder; but for high-speed motors, up to 1,000 or more revolutions per minute, side-chambers may be made available for larger valves. The form of valve seats, their angle and width, with the variations in practice, are fully shown by the detailed illustrations throughout this work, and in the section on valves and their design.

The dimension design of pistons varies considerably in European and American practice; but on general principles lightness, with due regard to resistance to the impact of explosion on the piston-head, and to lessen the balancing weight, is most desirable. For pistons of 8 inches diameter and under, there need be no bracing ribs at the back of the head, while for larger sizes the ribs strengthen a comparatively thin head and increase the cooling effect from air circulation within the piston. For the cylindrical shells of all sizes up to 20 inches diameter, the thickness of the metal under the ring-grooves and beyond the pin-bosses may conform to the formula $\frac{\sqrt{D}}{6}$ for shell-thickness and $\frac{\sqrt{D}}{4}$ for the heads. The pin-bosses should have a proportion for the strain on the forward side with a sub-boss for the set-screws. The number of rings varies somewhat among builders of motors; but good practice seems to indicate three rings on pistons up to 6 inches diameter and four to five on the larger diameters. A supplementary ring near the open end of the piston is not recommended as of any value.

The bearing length of piston-pins varies somewhat among builders in Europe and the United States from 1¼ to twice their diameter. One and a half diameters for the bearing length is a good proportion, and for this proportion the formula for the diameter may be $\sqrt{D} \times \frac{\text{comp.}}{150}$ makes a fair ratio for different cylinder-diameters in inches to meet the difference in extreme explosive pressures due to difference in compression.

The length of the connecting rod of an explosive motor varies from two to three times the length of the stroke; the longer rods being better adapted to the horizontal model. The diameter of a round connecting rod should be at its largest part a slight swell from the crank end for one-third its length and with a gradual taper to the piston end, to four-fifths of the largest diameter. For the largest diameter the formula $\frac{\sqrt{D}}{2} \times \sqrt{\frac{\text{comp.}}{75}} = d$ gives a safe size for explosive pressure.

The crankshaft requires much consideration from the great strain that it sustains at the moment of explosion, when the shaft and crankpin are on the center line and at that moment subject to the greatest strain. The strain is at first a bending one, changing to a torsional one as the crank angle increases. The basis of a formula is from the cube root of the square of the diameter multiplied by the compression and their product divided by 100 gives good proportions for steel shafts with strong fuel-pressure in inches of diameter. D = diameter of Cylinder. $\sqrt[3]{\frac{D \times \text{comp.}}{100}}$

The journals should be twice their diameter in length and the diameter of the crankpin should be from 12 to 15 per cent larger than the main journals for equivalent strength to resist the initial blow of explosion. The width of the crank-arm should be 1.33 times the diameter of the crankpin, and its thickness .7 the crank pin diameter. The form of the frame or engine-base is so varied among builders that we can only advise following the designs illustrated throughout this work, with a main view to a safe margin of strength due to the assumed pressures on the piston in the top member of the frame; the other parts to conform to lightness and constructive effect.

The method of counterbalancing the reciprocal and revolving parts of a motor that contribute to its vibration is still a mooted point among designers without arriving at a possible balance system for both motions. As these conditions of reciprocating combined with circular motion cannot be made to agree, a mean equalization of the two forces seems the only possible solution.

The following formula for the weight of a counterbalance of the form in Fig. 145, bolted to the crank, is an approximation for equalizing the reciprocating and revolving parts $\frac{P+C}{2} \times \frac{R}{r} = W$, in which P = weight of piston and rod; C, weight of crank and $\frac{1}{3}$ of rod, crank-end weight; R, radius of crank in inches; r, radius of centre of gravity of counterweight.

The flywheel of an explosive motor is a matter of much consideration in regard to its weight and diameter for the many conditions for its application to the speed-control of the motor-impulse. On general principles, a four-cycle motor requires more flywheel control than the two-cycle type. A single cylinder of either type more than motors of two, three, or four cylinders. Again, slow-speed motors of any type or number of cylinders require more flywheel control than high-speed motors. A high-compression motor more than one of low compression; so that the problem becomes a complex one in order to exactly meet every condition of motor service for stationary, marine, and vehicle propulsion.

For stationary power, a flywheel diameter of four times the stroke of the piston is the usual practice. For marine and automobile service the flywheel diameter should be much smaller to meet the conditions of boat and vehicle construction with their weight increased to the motor requirement.

The formula $\frac{\text{I. H. P.}}{\text{rev. pr M}} \times 34{,}000$ gives a good average weight of the flywheel rim for diameters of four times the piston-stroke.

The diameter of a flywheel hub should be $2\frac{1}{2}$ times the diameter of the shaft; the spoke-web, $3\frac{1}{2}$ times the shaft diameter. The spokes should taper slightly from web to rim, and each have a mean area of $\frac{3}{8}$ the shaft area at the web. A study of details illustrated in this work will suggest the best forms of rims and other parts from the practice of builders.

The reducing gear of the worm-gear type may be made an exact relation for difference of speed, which for the four-cycle explosive motor valve gear should be two revolutions of the crankshaft to one revolution of the valve-shaft. As the relative pitch diameters of the gears cannot always be made the same, some fixed relative diameter must be made and the spiral angle of their teeth cut to meet the required speed relation; or with a fixed angle of the teeth, the pitch diameters must be made to meet the required speed relation. Thus if the spiral angles of two matched gears are the same the velocity ratio will be inversely as the pitch diameters; but if the spiral angles are not equal, as in the usual gas-engine gears, the number of teeth per inch of pitch diameter will vary as the cosine of their angles. In any case the velocity ratio will depend upon the number of teeth and their spiral angle, as expressed in the following proportion: v, the velocity of the small gear, is to V, the velocity of the large gear, as D, the pitch diameter of the larger, multiplied by the cosine of its spiral angle, is to d, the pitch diameter of the smaller, multiplied by the cosine of its spiral angle.

Then, for example, a shaft spiral gear of twice the pitch diameter of the cam-shaft gear and running at twice its speed, their relative teeth spiral angles will be $2 \times 2 = 4$, and for the proper meshing of their teeth, requires that any $\frac{\text{cosine}}{4}$ that will equal its sine, will represent the proper angle of the teeth of the driving gear with the plane of its motion; while the angle of the driven gear-teeth will be the cosine of the plane of motion of the driven gear. By comparison of sines and cosines as tabulated, we find that a $\frac{\text{cosine}}{4}$ is equal to the sine of 14° 2′, and the cosine 75° 58′, which represents the relative angles of the teeth of the driver and driven gear with their planes of motion in the above case.

For spiral gears of equal diameter for velocities of 2 to 1 to match, with the shafts at right angles, the engine-shaft gear should have the lesser angle and the gear on the reducing or secondary shaft should have the greater angle as referred to their planes of motion respectively. The cosines of these angles must bear the same relation to each other on the pitch line as their velocities, and by inspection of a table of sines and cosines this relation is easily found; for example, in following along the columns of sines and cosines we find .44724 is as 2 to 1 to .89448, which agrees nearly to 26° 34′ and 63° 26′, the respective angles of the teeth with their planes of motion for equal-sized gears; their sum being equal to 90°.

The details of valve gears may be made variable to meet the fancy of designers or their judgment of fitness; but there are a few points in their operating principle which must be made to meet the requirements not only of each form of explosive element to be used, but also of the varied values of gases in gas-engines, from acetylene to producer and blast-furnace gas, and of the

FIG. 153.—The worm-gear.

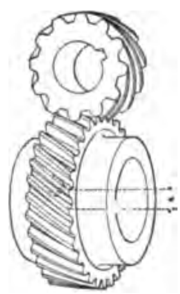


FIG. 154.—Spiral or worm-gear.

volatility of the variable grades of gasoline, kerosene, and the cruder oils, and which dominate the sizes and relative proportions of the inlet and exhaust-valves. The forms of the faces and seats of valves seem to have been varied to meet the fancy of designers in a great measure, and even the crudity of a spindle riveted to the valve disk has been used and published as a desirable makeshift. The flat-faced valve is also in use, but from the author's experience is unreliable for large engines and makes an imperfect seat by use. Conical-seated valves with faces at from thirty-five to forty-five degrees from the axis of the spindle are giving good service. A flatter cone of from fifty to sixty degrees is in use with apparent wearable properties and with slightly less lift for its full area than with the deeper-seated valves. A fifty-degree angle is recommended for high-speed motors. Spindle-valves with stems one-fifth to one-quarter the outside diameter of the valves, well filleted under the disks, give general satisfaction for ordinary speeds; but for very high-speed motors the valve stems should be somewhat larger. The general valve arrangements are well shown in their various modifications as illustrated in this work.

The relative size of these valves has been a subject of inquiry and discussion, with so far no fixed general rule applicable to the required conditions of each element. Some designated speed should first be assigned for any given-sized cylinder volume, from which the size of the valves may be computed for the full flow of the inlet charge and for the discharge of the exhaust without undue back-pressure during the times of the inlet and exhaust-strokes. This means larger valves for high-speed than for low-speed motors — a practice too often ignored, to the detriment of motor efficiency, by making these valves too small for the motor's best work; while if made to meet the requirements for highest speed capacity their efficiency action will be best for all lower speeds. The present practice with builders of large motors in regard to the size of the valves seems to vary the extreme diameter of the exhaust-valve from a quarter to four-tenths of the diameter of the cylinder, and the charging valve a little less, sometimes but one-fifth of the diameter of the cylinder.

Indicator cards taken from motors with small valves, if properly done, plainly show the effect of back-pressure from both the exhaust and charging strokes. Good practice suggests the larger valves with full lift of one-quarter their diameter for developing the full power of the motor. The width of the valve contact-seat has been the cause of much trouble with valve action by the mistaken judgment of designers — that great width of contact adds to tightness and wear of the valve and seat. Practically this is an error that should only be tolerated with inlet-valves having fuel feed through holes or channels in the seats. The width of bearing on inlet and exhaust-valves should have no more than one-eighth of their diameter. The conical bearings should also be the limit of inside and outside diameter for valve and seat.

A satisfactory material for low speed engines is solid valves of mild cast steel, "machinery-steel" grade; of which the drop-forgings (Fig. 155) are good examples; the tips to be cut off in finishing. For high speed motors, nickel and tungsten steel alloys have been developed that have many points of advantage, especially in air-cooled motors or types subjected to continuous severe duty.

There are differences of opinion in regard to the methods of opening the inlet-valve, the "suction or vacuum," and the "mechanical-lift," of which both are in use, the principal difference visible turning on the point of simplicity and complexity in valve-gear construction. Theory, as well as practice, places the percentage of efficiency in favor of the "mechanical-lift." With

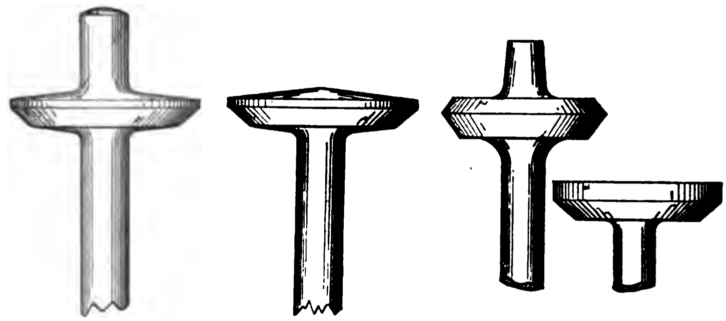


FIG. 155.—Steel drop-forgings.

the suction-lift the piston must travel a certain distance in the cylinder to create a vacuum strong enough to act upon the surface of the valve to lift it, and overcome the tension of the light spring that is acting against it to cause it to return to its seat quickly. The tendency of the suction-valve is always to return and remain on its seat, and it is only opposed from doing so as long as the vacuum in the cylinder is strong enough to hold it therefrom. Thus the valve chatters as it remains in space trying to respond to the summons of both agencies, the spring and the vacuum. While so doing it retards the inflow of mixture to the cylinder. If the spring has too great a tension the vacuum cannot properly lift it, and the cylinder is deprived of a sufficient amount of mixture. If the tension is too weak then the valve does not seat quickly enough, and part of the charge drawn in is forced back again through the inlet until the valve has made a proper seating, with the possibility of back-fire. Thus can be seen the value of a spring possessing the proper tension. Another thing that can be looked for is that a spring, when new and possessing the proper tension, will, in the course of constant use, lose some of its tension and change the results. The mechanically operated valve possesses a superiority over the suction type in several ways, and the additional expense and complication of operating an intake-valve is not worthy of mention. With a mechanically operated valve the necessity of having the spring tension to a certain point is obviated. But the spring should be strong enough in tension so as to always ride the cam that lifts it, but not too strong, to make working on the mechanical parts too severe. A motor with a mechanically operated valve will start more easily and is more sure of starting than the suction-lift, for the simple reason that the cam, being timed properly, will open the valve immediately as the piston starts on its suction-stroke and the vacuum immediately acts on the vapor without any extra duty to perform or obstructions in the way to give free access to a full and uniform charge.

The cyclical succession of operations, crank angles, and piston positions for the crank angle of each phase of the action of a four-cycle motor is shown in Fig. 156. Commencing with the inner circle, it will be seen that the charging may commence just before the crank reaches the dead centre owing to the momentum of the exhaust just before the piston stops; resulting in an extension of the charging to a point beyond the outward dead centre. The momentum of the charge through the inlet-valve and the compression through the balance of the return-stroke are shown on the diagram; then ignition at any designated point just before, at, or just after the dead point of the stroke. The explosive impulse in the outward stroke to a designated point for the exhaust-valve to open and exhausting to near the end of the return-stroke at which point the exhaust-valve closes by its spring pressure, just before the crank reaches the dead centre, are also shown in the outer circle. The crank should move in the direction of the arrow and by withholding the closure of the exhaust-valve mechanically, a scavenging effect may be had by the momentum of the exhaust in its pipe-passage. The diagram is an example that may be changed to suit any required conditions, so as to show at a glance the piston positions and relative crank angles.

The designing of explosive-motor cams, by many considered a difficult problem, can be worked out on the drawing-board with accuracy when the conditions of the opening and closing time are given: For an exhaust-valve cam for a high-speed motor, assuming to open at 40° crank motion above the terminal of the impulse-stroke and closing at 10° past the rear centre, as shown in the motion diagram (Fig. 156).

Thus the valve is held open through 230° of the crank's revolution and therefore through 115° of the cam-shaft's revolution. The cam proper is made up of two parts—one portion, B M A

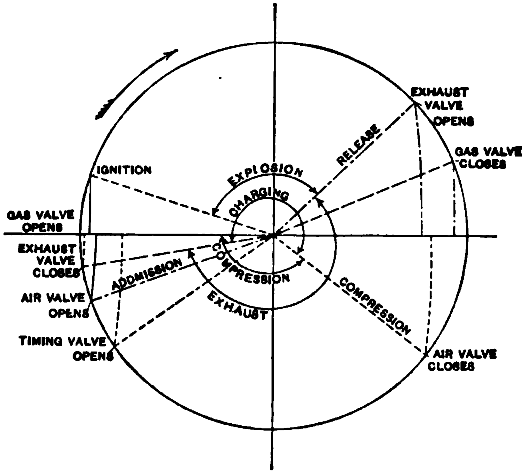
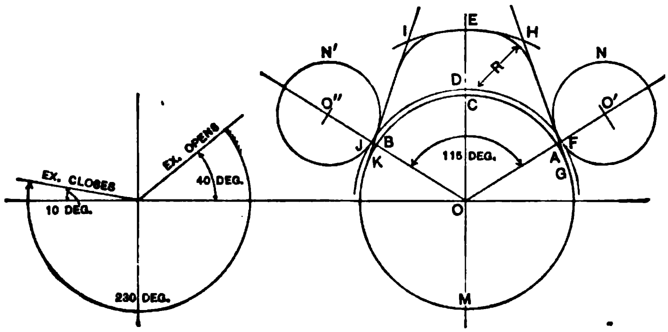


FIG. 156.—Cyclic phases of a four-cylinder motor.

(Fig. 158), concentric, and another portion, G E K, eccentric to the shaft. For convenience we will consider the cam to be standing still and the cam-roller to travel around the cam-counter clockwise — i. e., from A toward B.

From centre O, lay off a circle A B M equal in diameter to the concentric portion of the cam. Then from O lay off O A and O B 115° apart. O A is the line on which the valve begins to open, and O B the line on which it is just closed. Lay off C D equal to the amount allowed for lost motion before the valve begins to open, and D E equal to the amount of the opening of the valve. With the centre O draw arcs of circles through D and E, respectively; E will be on the outer extremity of the cam. On O A and O B, produced, lay off circles O′ F N and O″ J N′ equal in diameter to the cam-roller and tangent to the arc F D J. Draw G H tangent to both circles A B M and O′ F N; similarly K I tangent to both A B M and O″ J N′. This gives us the bounding lines of the eccentric portion, G E K, of the cam. The corners at H and I should be rounded off with radius R to suit the judgment of the designer.

For medium-speed motors the crank-angle opening of the exhaust may be made much less than the extreme figures above named and so varied for assumed speeds to as low as 25° crank-angle open-



FIGS. 157 and 158.—Exhaust-cam design.

ing and 5° for closing. These angles are also applicable where piston-ports are used. A similar method applies to the inlet-cam as well, although the angle of opening is somewhat less than that of the exhaust-cam.

The various forms of rotary and sleeve valves used in some modern internal combustion automobile engines are discussed in a special chapter, as is also valve timing for engines of this type.

CHAPTER XIII

THE MEASUREMENT OF POWER

THE methods of measuring power are of but two general forms or principles, although the individual machines or instruments for accomplishing the measurement are of many kinds and of a variety of construction. The most popular form is especially adapted for the measurement of the available power of prime movers under the various conditions of the application of their elementary power constituents, by the absorption of their whole output of power at the point of delivery and there record the value of its force and velocity. Its representative is the brake-dynamometer, or Prony's brake, in the various details of construction that it has assumed as designed and applied to meet the views or fancies of mechanical engineers.

The second form is a marked departure from the structural form of the first, and with the principle in view of placing as little obstruction as possible to the transmission of power from the prime mover to the receiver of power, to measure the actual net or differential tension of a belt or gear, and with its velocity indicate the exact amount of power delivered to a line of shafting or a machine. These are called transmitting dynamometers in distinction from the absorption dynamometers of the Prony type. They are of two kinds: one with a dial and index-pointer, by which the hand on the dial must be constantly watched and recorded for a length of time and a mean pressure obtained from the varying record; the other carries a self-marking register moved by clockwork, by which the actual pressure is a constant record for any desired time, or a full day's work, the only personal observation required being the speed of the pulley or belt or its average throughout the time or day.

In Fig. 159 we illustrate the first form, a simple absorption dynamometer or Prony's brake, named after its inventor, in which A is the radius of the pulley-drum or shaft to which resistance may be applied; B, the length of the lever from the centre of the shaft

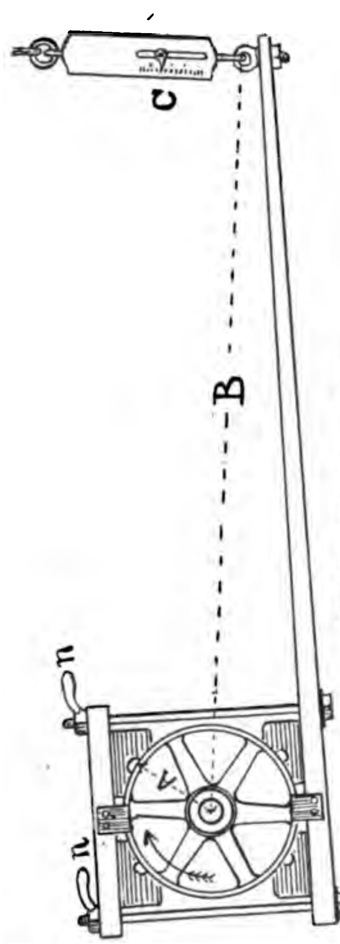


Fig. 159.—The Prony brake.

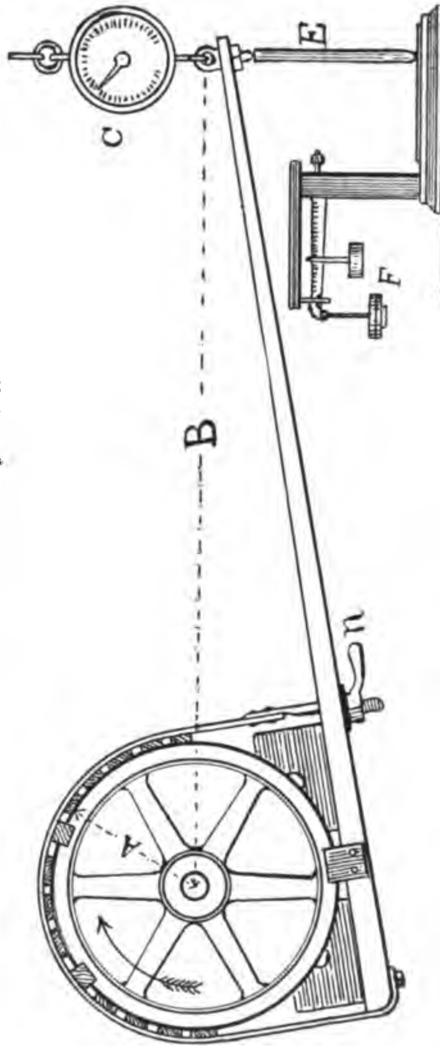


Fig. 160.—The Prony strap-brake.

to the point of attachment of the spring scale or other means of measuring the tension of the lever; C, a spring scale, which is preferable for light work within its range; and N N, lever-nuts for quick control of pressure. In Fig. 160 is presented a simple and inexpensive arrangement of a power-absorbing brake for a large driving pulley or finished flywheel in which a belt is lined with blocks of wood spaced and fastened to the belt with screws or nails, a few of the blocks projecting over the edge with shoulders to prevent the belt from running off the pulley.

Spring scales may be purchased of the straight and dial pattern up to one or two hundred pounds capacity at reasonable figures, and are a source of satisfaction in showing the amount of vibration due to irregular pulsations of the motive element and crank motion. Where the measurement of power beyond the range of a spring balance is required, the use of a platform scale or any other weighing device may be made available. With a platform scale the light wooden strut E (Fig. 160) may be adjusted to any length of lever, vertically reaching from the platform to the horizon line B, from the centre of the shaft; lanyards or any convenient means being used to keep the end of the lever from swaying. Water from a squirt-can is the best lubricant for this class of dynamometers, as it can be easily thrown upon the face of the pulley at the interstices of the blocks and lagging, and by its quick evaporation carries off the heat generated by friction. Soapy water has been used with good effect in preventing irregular pressure or stickiness of the friction surfaces.

It matters not in what direction the brake-lever is placed to suit the convenience of observation, so long as the pull of the scale is made at right angles to the radial line from the shaft centre Its weight, as indicated on the scale, with the friction-blocks or strap loosened in any position that it may be set, should be noted and a record made of the amount, which must be deducted from the total observed weight of the trial. If it is necessary to reverse the position of the lever or the relative direction of the motion of the pulley (as shown in Figs. 159 and 160), then the weight of the lever must be added to the weight shown by the scale under trial. When the platform scale is used the weight of the lever must necessarily be downward and should be deducted from the weight shown by the scale under trial. Making D equal the diameter of the face of the pulley, flywheel, or shaft upon which friction is applied, in feet or decimals of a foot, B the length of the lever from the centre of the shaft to the point of the scale suspension, A the radius of the pulley flywheel, or shaft, also in feet or decimals of a foot, and R the number of revolutions of the shaft per minute: the weight used in the formula must be the net weight of the power stress, or the gross observed weight less the weight of the lever. Then

$$\frac{D \times 3.1416 \times R \times \frac{B}{A} \times \text{weight}}{33{,}000} = \text{horse-power},$$

$$\text{or } \frac{B \times 6.2832 \times R \times W}{33{,}000} = \text{horse-power}.$$

$\frac{B}{A} \times$ weight = the stress or pull at the face of the pulley, and $D \times 3.1416 \times R$ = the velocity of the face of the pulley or of the belt that it is to carry.

In Fig. 161 is represented a simple and easily arranged differential strap-brake or dynamometer for small motors of less than two horse-power. It consists of a piece of belting held in place on the pulley by clips or only strings fastened parallel with the shaft to keep the belt from slipping off; two spring scales, one of which is anchored and the other attached to a hand-lever to regulate the compression of the belt upon the surface of the pulley, when the differential weight B — C on the scales may be noted simultaneously with the revolutions of the pulley. The simple formula

$$\frac{D \times 3.1416 \times R \times \text{differential weight}}{33{,}000} = \text{horse-power}.$$

Fig. 162 illustrates a rope-absorption dynamometer or brake with a complete wrap on the surface of the pulley, very suitable for grooved pulleys or flywheels used for rope-transmission. In this form the friction tension may be regulated with a lever as at A. The weight W in the formula is the differential of the opposite tensions of the two scales, or $B - C = W$ (Fig. 162), and the formula will then be: $\frac{D \times 3.1416 \times R \times W}{33{,}000}$ = horse-power, as in the notation (Fig. 161).

Thus it may readily be seen that the difference of the pull in a rope or belt on the two sides of a pulley, multiplied by the velocity of the rim in feet per minute, and the product divided by 33,000, gives the horse-power either absorbed or transmitted by the rope.

Many readers of inquiring mind may have tried to understand the methods of rating internal combustion power plants the vogue and in making inquiries to satisfy a natural curiosity, or looking up authorities on mechanical subjects, have been confronted with such a mass of technical data that their efforts to enlighten themselves have been in vain. Yet the question of power determination is a vital one to every gas-engine user, as it is the capacity of the engine that determines the speed and hill climbing ability when used in an automobile, or its capability for operating machinery when used in various stationary applications.

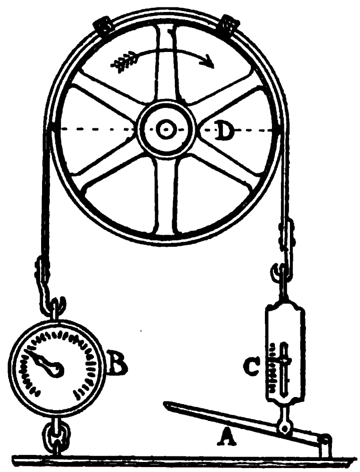


FIG. 161.—Differential strap-brake.

Power is not hard to define in simple language, and the various dynamometers and other testing machines used to determine its value are not hard to understand as they are based on principles that are easily explained.

Power is always defined as the rate of doing work, which in turn is the product of a force acting through a certain distance in a certain time. If a man raises a weight of 50 pounds one foot, he has done work equivalent to 50 foot pounds, but unless some unit of time is stated, it is very difficult to make any comparison with any standard unit. It may have taken the man one minute to lift the load that distance; obviously he did not exert himself as much as though the weight had been raised one foot in one second. The standard unit of measurement is one horse-power, which is the ability of lifting 33,000 pounds one foot in one minute or 550 pounds one foot in one second, or any combination of weight, distance and time that will produce the same resulting product.

If the work performed is more than 33,000 foot pounds per minute, more than one horse-power has been expended, if less than that amount of work has been done, it has not required the expenditure of a full horse-power.

Why the capacity of a piece of mechanism such as a steam or gas-engine or electric motor should be compared to that of a horse has often puzzled the layman. The apparent inconsistency is really founded on the good judgment of an early mechanic and inventor. In 1765, James Watt perfected the practical form of steam engine, and about the same time he believed it desirable to have some basis of comparing its power to that of some known force so the important factors of fuel cost, operating expenses and efficiency could be gauged with some degree of accuracy. At that time the horse was used for power purposes as well as a draft animal, and was very generally utilized to work pumps for relieving mines of water. As this was the work the early steam engine was first applied to, Watt believed that he would be able to interest prospective customers for his engines sooner if he could give them some idea of the work they could perform by comparing them with the animals they were to replace and thus demonstrate that it would be economical to utilize steam instead of equine power.

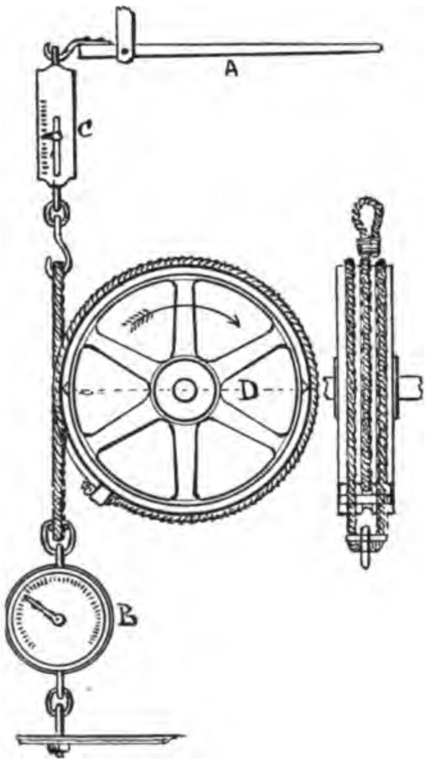


FIG. 162.—Differential rope-brake.

The question that then confronted Watt was to determine the real power of a horse as no such unit of measurement then existed. A long series of tests was made until he determined that the average work horse could lift a weight of 150 pounds 2.5 miles per hour for ten hours a day, or exert sufficient force to lift 33,000 pounds one foot per minute. While at that time, this unit, termed horse-power, conveyed some idea of the working ability of a horse, its significance has been lost at this time, but because of its general

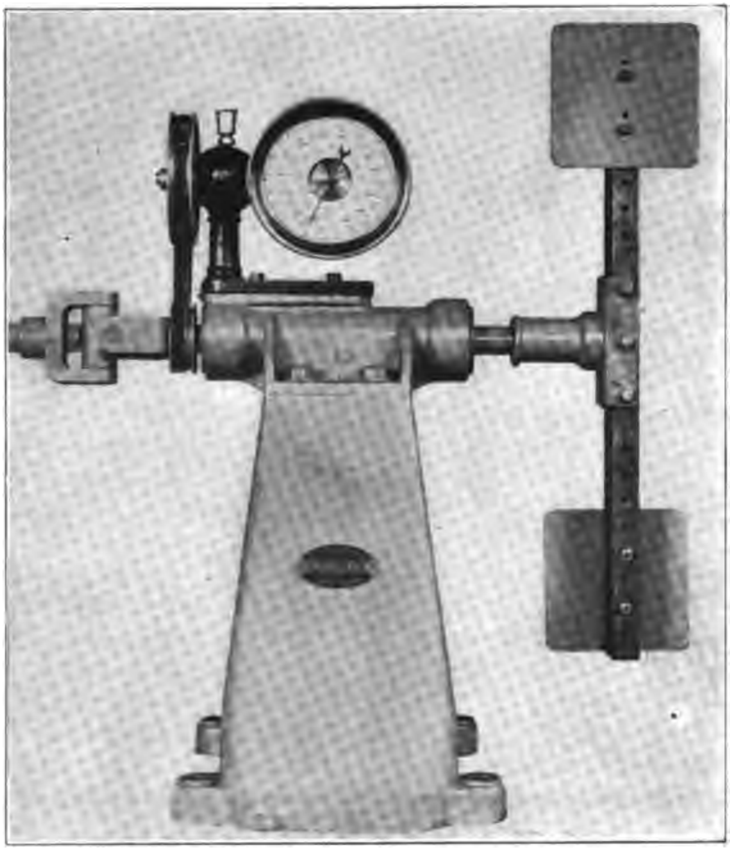


FIG. 163.—Typical fan dynamometer for making horse-power determinations.

use as a unit of work for over a century it is universally recognized by engineers the world over as a means of measuring power production or energy consumption by various forms of machines and prime movers.

In a gas, gasoline or crude oil motor, which is the popular power plant of the twentieth century, power is obtained by the rapid combustion of an inflammable gas in the cylinders. The pressure resulting from expansion tends to force a movable member, called the piston (which works inside the cylinder) down a certain distance, called the stroke. The piston is coupled to a crank by a connecting-rod as its reciprocating movement must be converted to a rotary motion in order to utilize it more effectively in driving the traction wheels of the automobile or the rotating shafting of machinery. Thus the amount of the expansion gives a force, the length of the stroke gives the distance through which the force is applied and the number of strokes per minute adds the time factor. If one knows the amount of the explosive force or pressure acting on the piston top in pounds, the piston travel in inches or feet and the number of strokes during which power is applied to the piston per minute the horse-power that should be theoretically obtained from a gasoline-engine cylinder can be readily approximated.

Thus in determining indicated horse-power, one must consider the following cardinal points. First, the mean effective pressure against the piston top, which is an average of the number of pounds per square inch acting during the entire stroke. Second, the area of the piston top in square inches. Third, the length of the stroke in feet. Fourth, the number of explosions per minute. The product of these factors divided by 33,000 gives the approximate horsepower, and may be expressed by the following simple formula:

$$\text{I. H. P.} = \frac{P \times A \times S \times \text{E. P. M.}}{33{,}000}$$

In which P is mean effective pressure in pounds per square inch.
A is area of piston top in square inches.
S is length of stroke in feet.
E. P. M. is number of explosions per minute.

To show how easily power computations are made by this formula, let us consider a typical case in point. The motor is a 4 cylinder type, having a cylinder bore of 5 inches and a stroke of 6 inches, and running normally at 1,200 revolutions per minute. If it is a four-cycle type, such as is generally used, there will be but one explosion per cylinder every two revolutions, or two power impulses per revolution. This means that there will be 2,400 explosions per minute and that each of these will move the piston 6 inches or half a foot. The mean effective pressure is usually taken as 90 pounds per square inch in considering motors of good design, and the area of a 5 inch piston is 19.635 square inches. Substituting known factors for the symbols given in formula, we have the following:

$$\frac{90 \times 19.635 \times .5 \times 2400}{33,000} = \frac{2,120,568}{33,000} = 64.5 \text{ I. H. P.}$$

While the method of estimating horse-power given above is widely used for preliminary determinations there are many factors which make the actual horse-power less than that one would expect from the results obtained from an empirical formula such as the one used. Heat and friction losses obtain that are not taken into account, so the best method of ascertaining the power actually delivered is to test the finished motor by some form of power absorbing device with suitable indicating mechanism to ascertain the amount of energy produced. All of the power delivered by the cylinder of the motor is not available for driving the car, as it takes a certain amount to open the valves, overcome the thrust of the pistons against the cylinder walls, turn the crankshaft and connecting-rods against the resistance of their bearings, drive the magneto, water pump, oiling mechanism and cooling fan and in other ways keep all moving parts in motion. The amount of power lost in this way depends upon the care taken in construction, the fits of the various parts and the proportions of the components. For motors in fair running order and of conventional design, the loss is a fairly constant percentage.

Thus, to determine the actual power output of the motor, various methods are used, all of which depend upon the absorption of energy and the measurement of the power thus taken by the testing apparatus. The most common appliances used are the friction brake, the electric generator and the fan brake. The earliest form of dynamometer which has been previously described was invented by G. de Prony, a French engineer, early in the nineteenth century, the first reference to the Prony brake having been made in 1821, though it may have been evolved at an earlier date.

The Prony brake has a great disadvantage that prevents its being used for continuous tests. As the power is absorbed by friction of the band upon the drum, excessive heat is developed and it is very difficult to keep the various parts cool. For this reason, the drums are sometimes made hollow if these are a separate part driven by the engine and filled with water and sometimes a spray of water is allowed to play directly upon the surface of the heated parts when the brake band is applied to the engine flywheel. Even with these refinements it is practically impossible to hold a given load produced by engines in excess of 8 or 10 horse-power for any length of time, and with an air cooled brake, as is usually rigged up, reliable readings cannot be obtained when the drum has become highly heated.

Many manufacturers who are producing engines in large quantities regard a running-in test as one that has sufficient value for all practical purposes and as a simple fan dynamometer may be easily attached to the power plant, it is not difficult to form an opinion of the capacity of the motor by comparing its action with that of others known to be efficient. The fan dynamometer is very simple and economical. A typical appliance of this kind is shown at Fig. 163, it consisting essentially of a through shaft supported on ball bearings mounted in a heavy cast iron standard or base, having a wooden arm carrying two aluminum plates at one end and a universal joint or other coupling for attaching to the engine to be tested on the other. In this instance, a tachometer or revolution indicator is fitted, this being driven by the shaft carrying the resistance arm and plates so the speed of the fan may be easily determined. Horse-power determinations are made by calibrating the fan from some standard motor and checking the results by some other form of dynamometer to determine if the fan is correct at the start. When once set, however, the power delivered by any motor may be noted by ascertaining the number of revolutions per unit time and comparing the results with a plotted curve or chart determined by a former calibration.

The fan dynamometer is usually employed to note the effect of a maximum load as it must be run sufficiently fast so the resistance will impose a steady load upon the motor to be tested. It is not practical to run such a device at low speeds because the load cannot be varied as easily as in a Prony brake or electric dynamometer. To vary the resistance offered by such an appliance, the vanes must be shifted on the arm to which they are attached. The nearer the centre they are placed, the less the resistance offered and the higher the speed possible with the same motive force. When one desires to test the output of a motor under varying speeds, such as might be caused by varying the mixture quality or quantity or spark time, and also to determine the capacity of the engine at low speed, the fan dynamometer has disadvantages which militate against its use.

The principle of air resistance is made use of in apparatus of this type. It has been determined by Kempe that the power required to move a plane surface through the air varies with the velocity, it increasing as the cube of the speed, as shown by the following table:

Speed.		Horse-power Per Sq. Ft. Effective Surface.	Speed.		Horse-power Per Sq. Ft. Effective Surface.
Miles Per Hour.	Feet Per Minute.		Miles Per Hour.	Feet Per Minute.	
10	880	0.013	50	4,400	1.64
15	1,320	0.044	75	6,600	5.54
20	1,760	0.105	100	8,800	13.13
25	2,200	0.205	150	13,200	44.29
30	2,640	0.354	200	17,600	105.00

It will be evident that by proper proportions of vane area, the distance they are carried from the centre and the number of revolutions per unit time, the load imposed upon the motor may be varied within wide limits. Suppose that the apparatus had vanes 1 square foot in area, and that these described a circle five feet in diameter. The distance travelled per revolution would be 15.70 feet and if the fan was turned at 500 revolutions per minute by a certain motor we could easily deduce the horse-power developed by simple computation. Two vanes are turned through the air at a speed of 7,850 feet per minute, and each vane has one square foot area. We find that it requires about 7 horse-power to turn one square foot plane member at that speed, therefore it would take approximately 14 horse-power to overcome the resistance imposed by two vanes with an aggregate area of 2 square feet.

FAN DYNAMOMETER FOR AUTOMOBILE CHASSIS TESTS

A fan dynamometer may be easily rigged up for testing the capacity of the entire automobile, that is, the amount of horse-power exerted at the rear wheels if it is driven by the rear axle instead of the motor. It may also be used to determine the power loss in the transmission by interposing the change speed and driving elements between the engine and the brake, and taking readings when on the direct drive or high speed and when driving through the reduction gears. The simple fan test stand shown at Fig. 164 has been devised to determine the power delivered by the engine

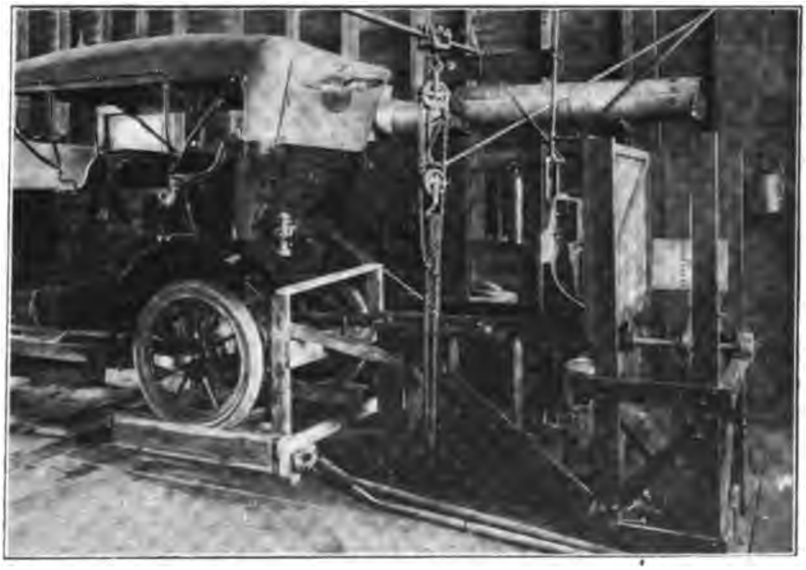


FIG. 164.—Method of testing rear wheel horse-power of motor car with fan dynamometer.

when installed in a motor car, and is easily attached to the motor to be tested by merely unfastening the bevel gear driving shaft at the transmission and attaching the dynamometer coupling in its place.

Motors may be tested separately by mounting them in the heavy stand just forward of the dash carrying the coil box when desired, a chain falls being rigged over the stand to lift the motor in place. Such a stand is very useful and can be easily rigged up in any shop at small cost, and it would seem to the writer that many automobile and gas-engine repair shops could install such mechanism as shown at Fig. 164 to advantage to test either the motor alone, the efficiency of the change speed gear or the power losses in the entire transmission system after overhauling a car. Positive indication would then be obtained, and the owner could be shown that the work had been properly performed by making actual tests in his presence.

ELECTRICAL DYNAMOMETERS FOR MOTOR TESTING

When one demands a flexible dynamometer, it is difficult to conceive any apparatus that is superior to those in which electricity is utilized. Electric testing apparatus may be either of two classes: that in which a standard dynamo electric machine is driven by the motor to be tested and its output determined by suitable resistance and measuring meters, and the other form having an oscillating field member restrained from undue movement by a lever arm and weights. An installation of the first class is shown at Fig. 165. Here the dynamo is attached to the engine shaft by a flexible coupling and its output measured by suitable instruments on the switchboard. The current developed is absorbed by the resistance coils shown at the right of the switchboard. A tachometer is attached to the wall just above the dynamo and is driven by a pulley on the end of the dynamo shaft. As the amount of current delivered will vary with the speed at which the dynamo armature revolves, and this in turn is dependent upon the power of the motor, it will be evident that the power can be read directly from the current recording instruments. Electrical horsepower is measured in watts, it taking a little over one mechanical horse-power to furnish 746 watts of current. This unit is obtained by multiplying the amount of current as expressed in amperes by its pressure in volts; thus, if the voltage was 100, it would take 74.6 amperes of current to do work equivalent to an electrical horsepower. In other words one may say that 746 watts is equal to 33,000 foot pounds per minute.

Suppose the dynamo coupled to the motor shown at Fig. 165 was delivering a current of 7,460 watts or nearly 7.5 kilowatts, if there were no losses in transforming mechanical into electrical energy, the motor would be developing 10 horse-power. As it is, the power exerted by the power plant would be about 10 per cent greater than this, or 11 horse-power. If the indicating instruments

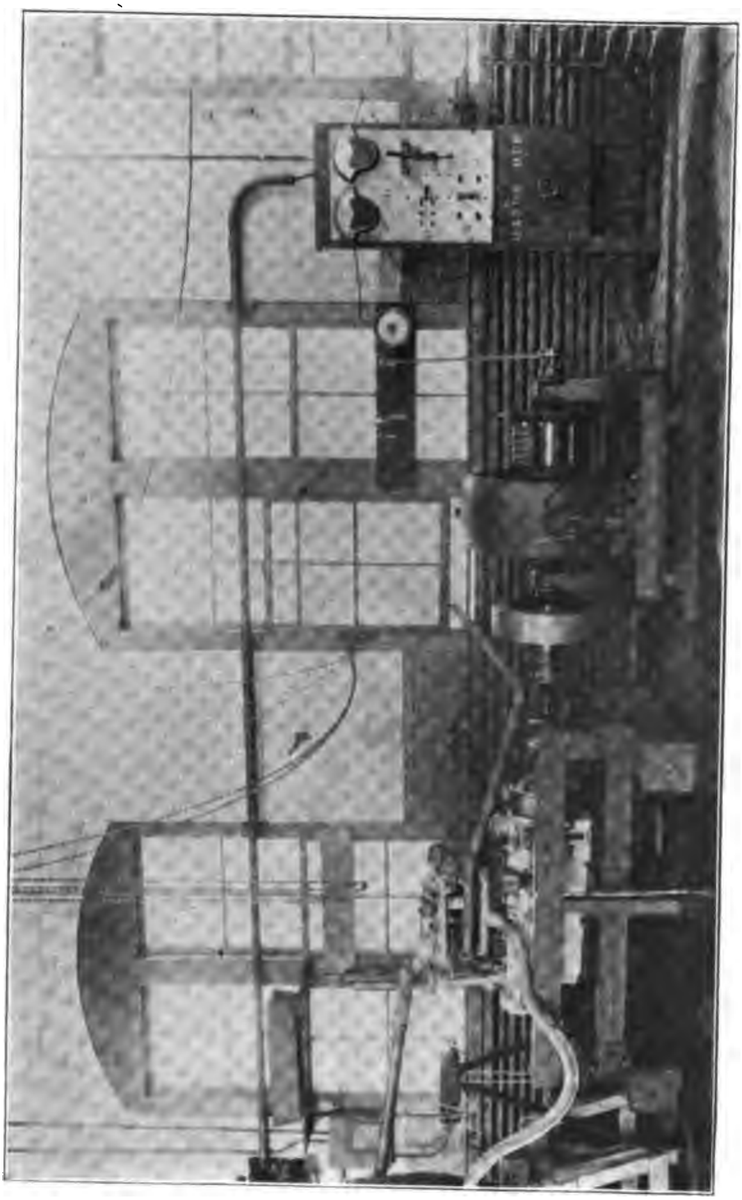


FIG. 165.—Application of dynamo to show horse-power delivered by gasoline engine.

showed that a greater current of 37.5 kilowatts was being absorbed by the resistance coils, the gasoline engine would be delivering in excess of 55 horse-power.

Sometimes a bank of incandescent lamps is utilized instead of the resistance coils for absorbing the current produced, in other installations especially where a large number of motors are being tested regularly, the electrical energy is delivered to motors at various points in the plant and made to do useful work by opera-

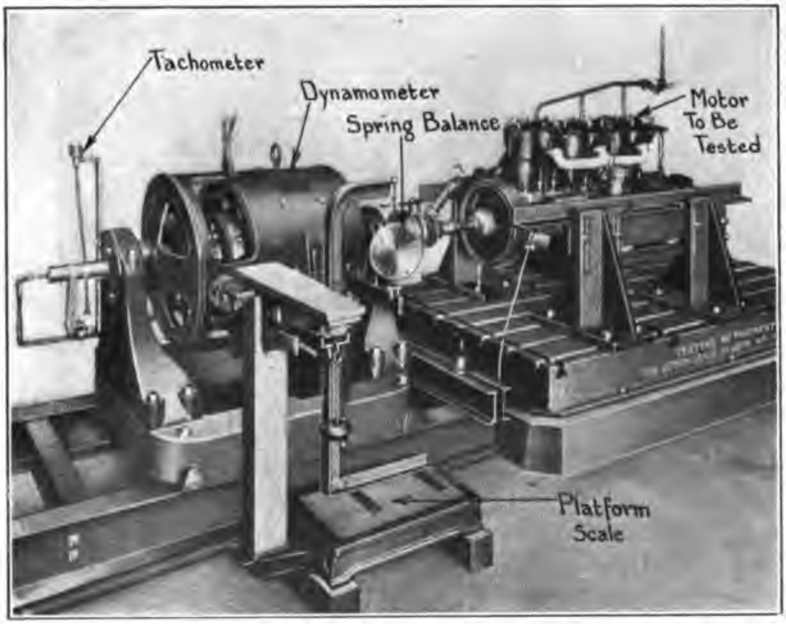


FIG. 166.—Cradle dynamometer for making exact horse-power tests.

ting machinery, or it may be fed direct to lamps and other apparatus. The current is sometimes conserved by means of a storage battery installation.

The second class of electro-dynamometer is shown at Figs. 166 and 167. This differs from the dynamo test previously described in that a lever arm is attached to the field piece of the machine, which is journaled in a frame in such a manner that the tendency of the field is to follow the armature, this pull increasing as the horse-power is augmented just as in

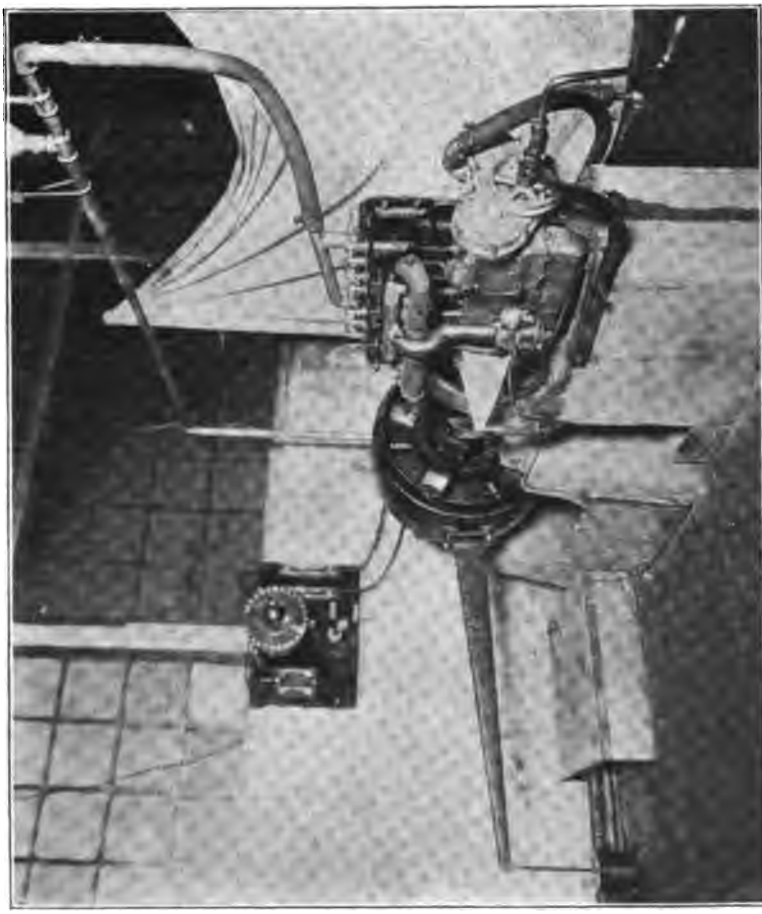


FIG. 167.—Front view, showing test of automobile motor with cradle dynamometer.

Prony brake. The weights at the end of the long lever resist the tendency of the field to follow the armature. Suitable electrical instruments are mounted on the board to control the current, and the horse-power delivered by the motor may be easily computed by a simple formula very similar to that previously mentioned for measurement of torque absorbed by friction brakes.

The cradle dynamometer, as the form shown at Fig. 166 is called, is often used in making engine tests with various auxiliary systems to determine relative efficiency of carburetors, ignition

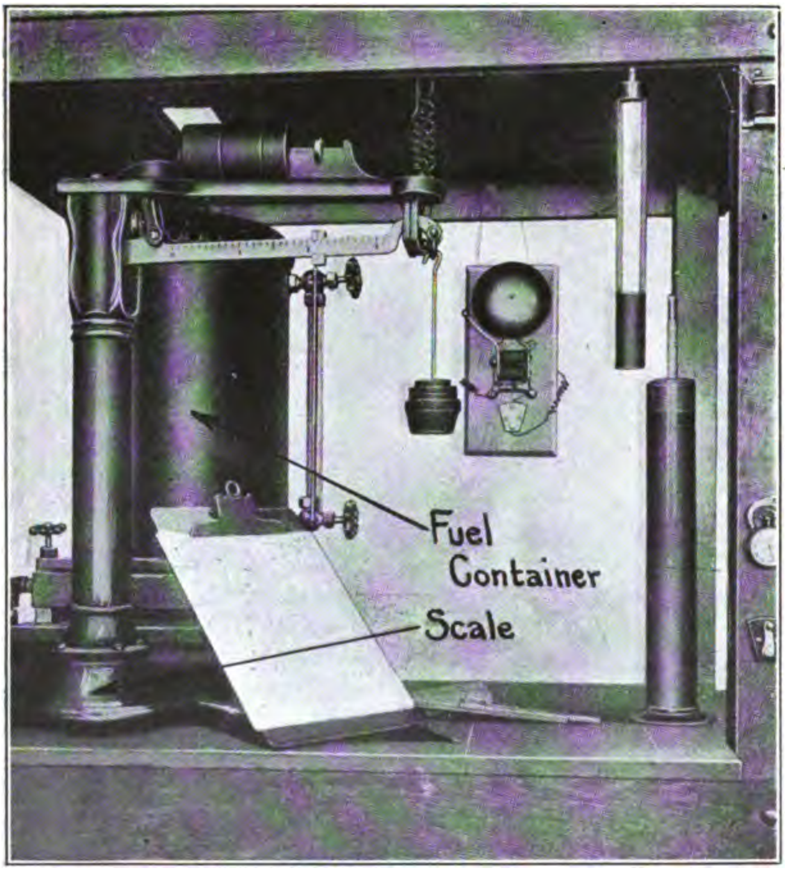


FIG. 168.—Fuel container and weighing outfit for measuring gasoline consumption.

systems etc. This is because of the close graduation possible of the load and accurate determination of power delivered. The apparatus used in determining fuel consumption in connection with electrical or other dynamometers is clearly shown at Fig. 168. The fuel container is placed on a platform scale and the amount consumed for any given period may be easily determined by the diminution in the aggregate weight.

In addition to the common methods outlined, some engineers have devised testing apparatus in which water or other liquids play a part, the power being absorbed by some form of circulating pump or paddles in liquid, and its quantity measured by determining rate of flow or revolutions per minutes of the paddles in the resisting medium. One form, known as the hydro-dynamometer, consists of a large disc or a plurality of such members revolved at high speeds in a water bath contained in a suitable housing. Others are merely large centrifugal pumps, and still others consist of an ordinary large boat propeller revolved in a tank of brine or oil.

All gas-engine factories have some form of testing apparatus to make sure that the engines produced have the proper power before they are shipped, and after receiving a thorough test, they are sent to the purchaser with every assurance that they will function properly and deliver full power as long as operating conditions are normal. The length of time that a motor will retain its efficiency after leaving the factory depends upon many factors, one of the most important being the amount of care it receives at the hands of the owner, or those entrusted with its maintenance. The best of motors will fail in service if neglected or abused, while an indifferent design may give very satisfactory service if properly cared for.

THE MEASUREMENT OF SPEED

The revolutions of a motor may be readily obtained by an ordinary hand-counter, with watch in hand to mark the time; but for accurate work and to show the variations in the flywheel speed by the intervals of revolution between impulses, and especially the effect of mischarges or impulses due to governing the speed, there is no more accurate method than by the use of the centrifugal counter or tachometer. These instruments are designed to show at a glance a continuous indication of the actual speed and its variation within

2 per cent. by careful handling of the instrument. The tachometer (Fig. 169), with a single-dial scale three inches in diameter, reads from 100 to 1,000 revolutions per minute, and by changing the gear for the range of gas-engine indication the actual revolutions will be one-half the indicated revolutions, which divided by two will repre-

FIG. 169.—The tachometer.

FIG. 170.—The triple-indexed tachometer.

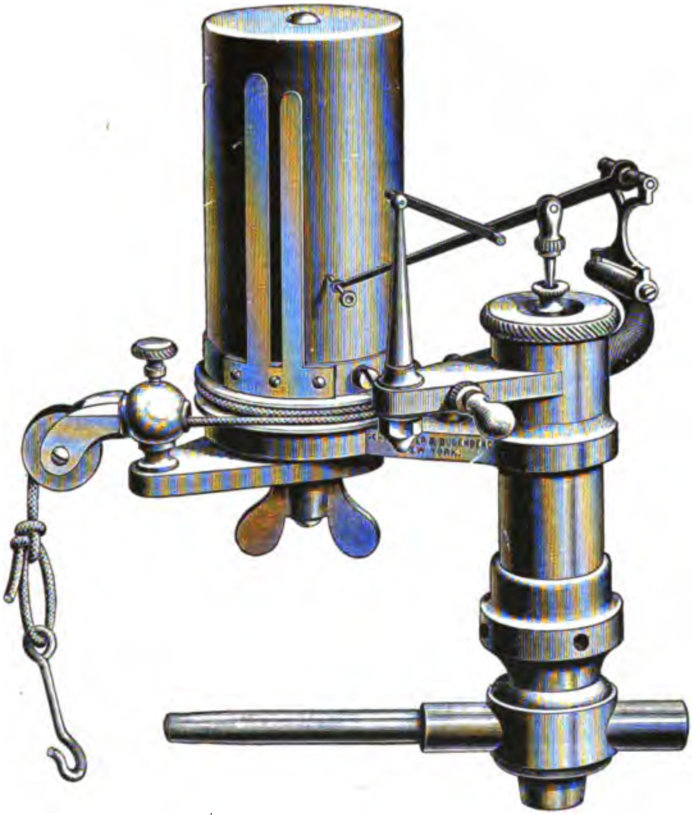


FIG. 171.—The Thompson indicator.

sent the actual speed. In this manner a very delicate reading of the variation in speed may be obtained. For testing the variation of speed in electric-lighting plants operated by gas, gasoline, or oil-engines, there is no method so satisfactory as by the use of the tachometer. The triple-indexed tachometer (Fig. 170) is a most convenient instrument for quickly testing and comparing speed of great differences, as the motor and the generator, by simply changing the driving point from one to another gear stem. These tachometers are made by Schaeffer and Budenberg, and may be obtained for any range of speed, from 50 to 500 for gas-engines and from 500 to 2,000 for generators, in the same instruments or separate as desired. Some are provided with a driving pulley so they can be placed in direct connection with the revolving elements.

THE INDICATOR AND ITS WORK

We have selected among the many good indicators in the market the one most suitable for indicating the work of the explosive engine. The Thompson indicator illustrated in Figs. 171 and 172, is a light and sensitive instrument with absolute rectilinear motion of the pencil, with its cylinder and piston made of a specially hard alloy which prevents the possibility of surface abrasion and insures a uniform frictionless motion of the piston. It is provided with an extra and smaller-sized cylinder and piston, suitable with a light spring for testing the suction and the exhaust curves of explosive motors, so useful in showing the condition and proportion

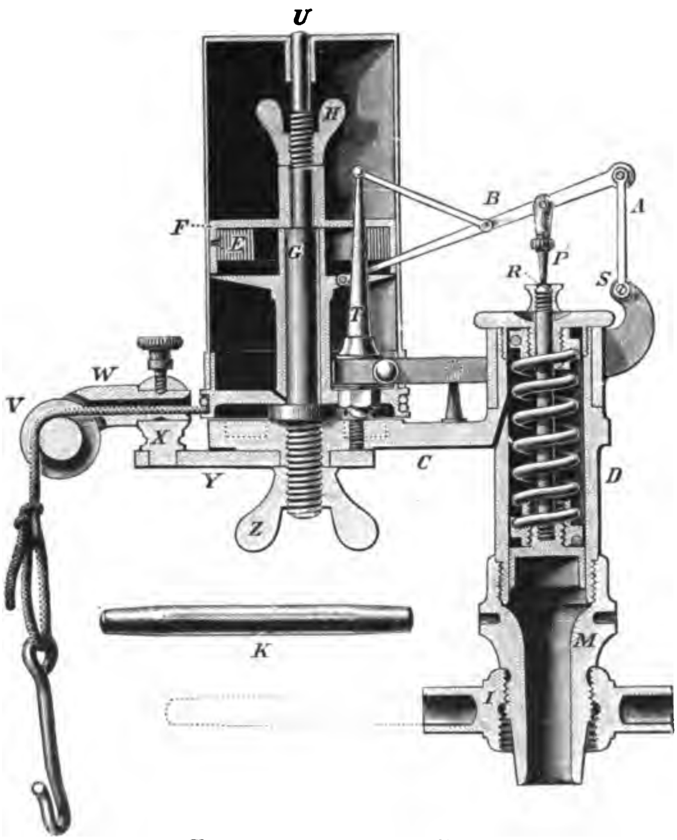


FIG. 172.—Section of indicator.

of valve ports. The large piston of the standard size is 0.798 inch in diameter and equal to ½ square-inch area. The small piston (Fig. 173) is 0.590 inch in diameter and equal to 0.274 square-inch area, so that a 50 or 60 spring may be used in indicating explosive engines with the small piston, which will give cards within the range of the paper for low-explosive pressure but full enough to show the variations in all the lines. With the 100 spring and ½-inch area of piston 250 pounds pressure is about the limit of the card, but with this size piston a 120 or 160 spring is more generally used.

The pulley V is carried by the swivel W, and works freely in the post X; it can be locked in any position by the small set screw. The swivel-plate Y can be swung in any direction in its plane and held firmly by the thumb-screw Z. Thus with the combination the cord can be directed in all possible directions. The link A is made as short as possible, with long double bearings at both ends to give a firm and steady support to the lever B, making it less liable to cause irregularities in the diagram when indicating high-speed motors.

The paper drum is made with a closed top to preserve its accurate cylindrical form, and the top, having a journal-bearing at U in the centre, compels a true concentric movement to its surface. The spring E, and the spring-case F, are secured to the rod G by screwing the case F to a shoulder on G by means of a thumb-screw H.

To adjust the tension of the drum-spring, the drum can be easily removed, and by holding on to the spring-case E, and loosening screw H, the tension can readily be varied and adapted to any speed, to follow precisely the motion of the engine-piston. The bars of the nut I are made hollow, so as to insert a small short rod, K, which is a great convenience in unscrewing the indicator when hot.

The reducing pulley (Fig. 174) is a most important adjunct of the indicator. The revolving parts should be as light as possible and are now made of aluminum for high-speed motors, with pulleys proportioned for short-stroke motors. In the use of indicators for high-compression motors it is advisable to have a stop-tube inserted in the cap-piece that holds the spring and extending down and inside the spring so as to stop the motion of the piston at the

FIG. 173.—Small piston.

FIG. 174.—The reducing pulley.

limit of the pencil motion below the top of the card. This will prevent undue stress on the spring and extreme throw of the pencil when, by misfires, an unusual charge is fired. With the smaller piston and the usual 100 or 120 spring any possible explosive pressure may be properly recorded. The proximity of the indicator to the combustion chamber is of importance in making a true record of the explosive action of the combustible gases on the card. The time of transmission of the wave of compression and expansion through a tube of one, two, or three feet in length is quite noticeable in the distortion of the diagram. It shows a delay in compression and carries the expansion line over a curve at the apex lower than the maximum pressure, and by the delay raises the expansion curve higher than the actual expansion curve of the cylinder. An indicator for true effect should have a straightway cock screwed into the cylinder.

CONSTRUCTION AND USE OF THE MANOGRAPH

The indicators in use up to the present time are fairly satisfactory when applied to low speed engines but as the speed of the engine is increased so also do the imperfections of the indicator become more manifest and in the case of very high speed engines these imperfections render the indicator quite useless. The principal defect is the number of errors due to the inertia of the relatively large mass of the moving parts of the indicator and to the friction of the joints and the indicator pencil on the cylinder. It is with a view of suppressing these defects of inertia and friction

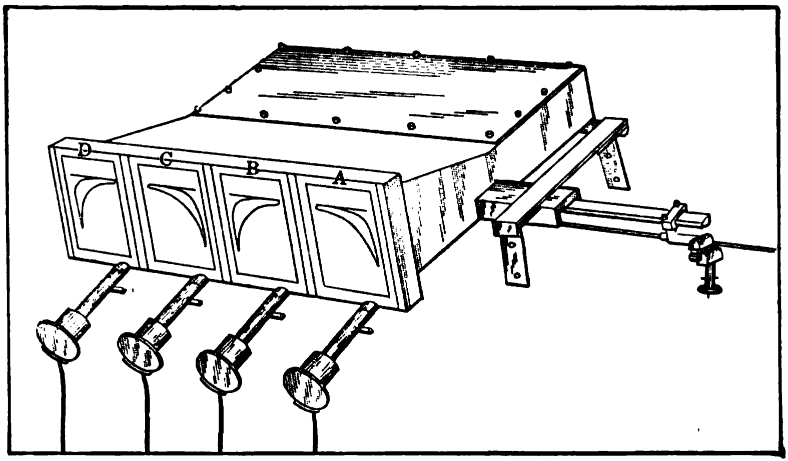


FIG. 174A.—Quadruple manograph apparatus for recording the diagrams of four-cylinder motors, thus enabling a comparison to be made. A, B, C, D indicate the four glass screens.

that the manograph has been designed. It is based on the principle of the deviation of a ray of light reflected by a mirror.

The indicating mechanism consists essentially of a mirror fixed at a single point and capable of oscillation in two planes. The oscillation in one direction corresponds to the motion of the piston in the engine. The oscillations in the other plane depend upon the variations of pressure in the cylinder. A ray of light is directed onto the mirror and is then reflected by the mirror on a ground glass screen at one end of the device on which the indicator diagram is seen. On account of the lightness and efficient balancing of the mirror, it faithfully responds to every variation of pressure in the engine cylinder even at the highest engine speeds. As is true of the regular form indicator, it is necessary to transmit the motion of the engine piston to the manograph by a system of rotary gearing and to connect it to the interior of the combustion chamber by a small water cooled tube. The diagram appears on a glass screen in the form of an unbroken continuous luminous line. The observer can follow every phase of the engine working. He can see at once any defects caused by improper admission, poor mixture, back pressure, excessive suction, defective ignition, etc.

With a manograph especially arranged for reading low pressures and the pressure scale of the diagram being increased, one can note the exact time of opening and closing the valves, the suction during admission, and the back pressure during exhaust. The student can also study the phenomena of compression and the expansion of the compressed gas. With a newly designed manograph for multiple cylinder engines, one can study simultaneously the working of all cylinders and make comparisons between the diagrams, which should be exactly alike, if the cylinders are functioning properly. This apparatus is particularly valuable in studying the relative merits of various ignition systems, the action of the carburetor, and the best size and shape for inlet and exhaust manifolds. Another valuable feature is that a record of the diagrams will be kept if the glass screen is replaced, as in a camera, by a dark slide containing photographic plates, or if preferred a sheet of sensitised paper which will give a black diagram on a white ground. For demonstration purposes, and for studying minute variations of the pressure line, a special projecting lens tube is used. It is possible by this means to project diagrams twenty-seven inches long upon a screen placed about eight feet from the tube. The manograph will work in any position and is just as suitable for steam engines as the ordinary indicator and in addition can be used on all kinds of internal combustion engines having fixed cylinders and will record faithfully up to 3000 R. P. M.

HORSE-POWER FORMULÆ

A large number of formulæ for determining indicated horsepower of internal combustion engines have been advanced, especially since the wide vogue of the internal combustion motor in motor vehicle, marine and air craft propulsion where competitive events, such as races, hill climbs, etc., make necessary some simple, easily applied rules for ready calculation. The following are the leading formulæ of European derivation and are reproduced from The Automobile Engineer Year Book for 1913. Our leading American formula, adopted by the S. A. E. is the same as that of the Royal Automobile Club. The Notation appended has been followed in all the expressions:

D = Diameter of cylinder.
N = Number of cylinders.
S = Stroke.
$R = \frac{S}{D}$ = ratio of stroke to bore.

1.—ROYAL AUTOMOBILE CLUB (R.A.C.) FORMULA:

Measurement in inches.

$.4 D^2 N$

or $\frac{D^2 N}{2.5}$

Measurement in mm.

$\frac{D^2 N}{1,613}$

This formula assumes a constant piston speed of 1,000 feet per minute and a mean effective pressure of 67.2 pounds per square inch.

2.—LANCHESTER'S FORMULA:

Measurement in inches.

$.5 D^2 N \sqrt{R}$

Measurement in mm.

$\frac{D^2 N}{1.290} \sqrt{R}$

This formula provides a correction for piston speed limited by the stroke.

3.—SOCIETY OF MOTOR MANUFACTURERS' AND TRADERS' PROPOSED FORMULÆ:

Measurement in inches.

(A) for touring engines—
$.197 D (D - 1) (R + 2) N.$

(B) for racing engines—
$.333 D (D - 1) (R + 2) N.$

Measurement in mm.

(A) $\frac{D (D - 25.4) (R + 2) N}{3.275}$

(B) $\frac{D (D - 25.4) (R + 2) N}{1,937}$

These formulæ are a simplified modification of the Lanchester formula, and embody a factor to allow for differences of cooling due to size, and assume a piston speed of 1,000 R ½ feet per minute for touring engines, and a greater speed for racing designs.

4.—DENDY MARSHALL FORMULA:

Measurement in inches.

$\frac{D^2 S N}{12}$

Measurement in mm

$\frac{D^2 S N}{200,000}$

This formula gives a close approximation of power, and assumes a revolution speed of 1,000 per minute, or with modification for effect of stroke-bore ratio on revolution speed.

$\frac{D^2 S N \text{ Revs.}}{12,000}$

$\frac{D^2 S N \text{ Revs.}}{200,000,000}$

5.—INSTITUTION OF AUTOMOBILE ENGINEERS' FORMULÆ:

Measurement in inches.

$.45 (D + S) (D - 1.18) N.$

Measurement in mm.

$\frac{(D + S) (D - 29.97) N}{1,433}$

This formula embodies a correction for mean effective pressure rising with bore, and one for effect of stroke-bore ratio on speed.

6.—BURL'S MAXIMUM RATING FORMULA:

Measurement in inches.

$$\frac{1}{2} D (D - 1.18)\sqrt{\frac{D^2 S}{M}}$$

Measurement in mm.

$$\frac{D}{1,290}\left(D - 29.97\right)\sqrt{\frac{D^2 S}{16,390 M}}$$

This formula is similar to the I.A.E. formula, but embodies a speed limiting factor based on weight of reciprocating parts, and M = mass of weight of reciprocating parts in one cylinder. For cast-iron pistons

$$M = .08 D^3 (1 + .15 R) + 1.5 \text{ lb.}$$

$$M = \frac{D^3}{204,700}\left(1 + .15 R\right) + 1.5 \text{ lb.}$$

7.—POPPE'S FORMULA:

Measurement in inches.

$$\frac{D S N}{2.5}$$

Measurement in mm.

$$\frac{D S N}{1,612}$$

This formula is based upon cubic capacity of cylinders simply.

HORSE-POWER TABLES

The accompanying table was prepared by Mr. Worby Beaumont, a well-known English Engineer, and should be useful in figuring capacities of small motors, such as used for automobiles, motor-boats and motor-cycles.

To use table: Take the bore of your motor from marginal or left-hand column, follow directly across to the column whose heading added to the bore gives stroke. The figure at the intersection of this column and your bore column gives the horse-power of one cylinder of that size at 750 revolutions, and the figure immediately below gives the horse-power at 1,000 revolutions, which is generally accepted for computing present day high-powered motors.

The table represents the calculated indicated power given by one cylinder. For multi-cylinder engines the power shown must be multiplied by the number of cylinders.

Beaumont Horse-Power Table.
Cylinder Dimensions in Inches and Fractions of an Inch.

Bore.	0	¼	½	¾	1	¼	½	¾	2
2	0.54	0.60	0.67	0.74	0.81	0.88	0.94	1.01	1.08
	0.72	0.80	0.89	0.98	1.08	1.17	1.25	1.35	1.44
2¼	0.76	0.84	0.92	1.01	1.09	1.18	1.26	1.35	1.43
	1.01	1.12	1.22	1.34	1.45	1.57	1.68	1.81	1.90
2½	1.04	1.15	1.25	1.35	1.46	1.56	1.67	1.77	1.88
	1.39	1.53	1.67	1.81	1.95	2.09	2.23	2.37	2.51
2¾	1.39	1.51	1.64	1.77	1.89	2.02	2.14	2.27	2.40
	1.85	2.02	2.19	2.36	2.53	2.70	2.87	3.04	3.20
3	1.80	1.95	2.10	2.25	2.40	2.55	2.70	2.85	3.00
	2.40	2.60	2.80	3.00	3.21	3.40	3.61	3.81	4.00
3¼	2.30	2.47	2.65	2.82	3.00	3.18	3.36	3.53	3.70
	3.06	3.28	3.52	3.76	3.99	4.22	4.45	4.70	4.92
3½	2.86	3.06	3.27	3.48	3.68	3.88	4.09	4.30	4.50
	3.81	4.09	4.36	4.63	4.90	5.18	5.45	5.72	6.00
3¾	3.53	3.76	4.00	4.35	4.47	4.71	4.95	5.18	5.41
	4.71	5.02	5.34	5.65	5.97	6.28	6.60	6.91	7.23
4	4.27	4.65	4.81	5.08	5.35	5.61	5.89	6.15	6.41
	5.71	6.06	6.42	6.78	7.13	7.50	7.85	8.20	8.55
4¼	5.13	5.44	5.74	6.04	6.34	6.64	6.95	7.25	7.55
	6.84	7.24	7.64	8.05	8.45	8.85	9.25	9.66	10.06
4½	6.19	6.53	6.87	7.22	7.56	7.90	8.25	8.59	8.93
	8.25	8.68	9.16	9.62	10.08	10.54	10.96	11.41	11.89
4¾	7.18	7.55	7.94	8.31	8.70	9.07	9.45	9.82	.....
	9.57	10.07	10.57	11.09	11.58	12.08	12.59	13.09	.....
5	8.38	8.80	9.22	9.65	10.05	10.46	10.90	.....	.....
	11.16	11.72	12.28	12.84	13.40	13.95	14.50	.....	.....
5¼	9.70	10.16	10.62	11.09	11.54	12.00	.....	.....	.....
	12.02	13.54	14.15	14.76	15.40	16.00	.....	.....	.....
5½	11.12	11.65	12.15	12.66	13.15	.....	.....	.....	.....
	14.84	15.51	16.20	16.86	17.54	.....	.....	.....	.....
5¾	12.71	13.27	13.82	14.38	.....	.....	.....	.....	.....
	16.06	17.70	18.42	19.19	.....	.....	.....	.....	.....
6	14.45	15.05	15.65	.....	.....	.....	.....	.....	.....
	19.25	20.05	20.85	.....	.....	.....	.....	.....	.....

S. A. E. Horse-Power Table.

(Four-Cycle Horse-Power Table S. A. E.)
Limit of Error, .005

$$HP = \frac{D^2 N}{2.5}$$

Calculated by J. Howard Pile.

Bore.	1-Cyl.	2-Cyl.	4-Cyl.	6-Cyl.
2½	2.50	5.00	10.00	15.00
2⅝	2.75	5.50	11.00	16.50
2¾	3.03	6.05	12.10	18.16
2⅞	3.30	6.60	13.20	19.79
3	3.60	7.20	14.40	21.60
3⅛	3.91	7.81	15.62	23.44
3¼	4.23	8.45	16.90	25.35
3⅜	4.56	9.11	18.23	27.34
3½	4.90	9.80	19.60	29.39
3⅝	5.26	10.51	21.02	31.54
3¾	5.63	11.25	22.50	33.75
3⅞	6.03	12.07	24.14	36.20
4	6.40	12.80	25.60	38.40
4⅛	6.81	13.61	27.23	40.83
4¼	7.23	14.45	28.90	43.35
4⅜	7.66	15.31	30.62	45.94
4½	8.10	16.20	32.40	48.60
4⅝	8.56	17.11	33.22	51.34
4¾	9.03	18.05	36.10	54.15
4⅞	9.51	19.01	38.02	57.04
5	10.00	20.00	40.00	60.00
5⅛	10.51	21.01	42.02	63.03
5¼	11.03	22.05	44.10	66.15
5⅜	11.56	23.12	46.24	69.36
5½	12.12	24.24	48.48	72.72
5⅝	12.66	25.32	50.64	75.96
5¾	13.22	26.44	52.88	79.32
5⅞	13.84	27.68	55.36	83.04
6	14.40	28.80	57.60	86.40

(Two-Cycle Horse-Power Table S. A. E.)
Limit of Error, .005

$$HP = \frac{D^2 N^*}{1.5151}$$

Calculated by J. Howard Pile.

Bore.	1-Cyl.	2-Cyl.	3-Cyl.	4-Cyl.
2½	4.13	8.25	12.38	16.50
2⅝	4.55	9.08	13.64	18.19
2¾	4.99	9.98	14.97	19.96
2⅞	5.46	10.91	16.37	21.82
3	5.94	11.88	17.82	23.76
3⅛	6.49	12.97	19.46	25.95
3¼	6.98	13.96	20.94	27.92
3⅜	7.52	15.04	22.55	30.07
3½	8.09	16.17	24.26	32.34
3⅝	8.67	17.35	26.02	34.69
3¾	9.28	18.56	27.84	37.12
3⅞	9.91	19.82	29.73	39.64
4	10.56	21.12	31.68	42.24
4⅛	11.23	22.46	33.69	44.92
4¼	11.92	23.84	35.76	47.68
4⅜	12.62	25.24	37.86	50.48
4½	13.36	26.72	40.08	53.44
4⅝	14.12	28.24	42.36	56.48
4¾	14.89	29.78	44.67	59.56
4⅞	15.68	31.36	47.04	62.72
5	16.50	33.00	49.50	66.00
5⅛	17.34	34.68	52.02	69.36
5¼	18.19	36.38	54.57	72.76
5⅜	19.07	38.14	57.21	76.28
5½	20.00	40.00	60.00	80.00
5⅝	20.88	41.76	62.64	83.52
5¾	21.32	42.64	63.96	85.28
5⅞	22.78	45.56	68.34	91.12
6	23.84	47.68	71.42	95.36

* This formula is evolved by multiplying the four-cycle formula by 1.65, the generally accepted ratio of two to four-cycle motors.

CHAPTER XIV

EXPLOSIVE MOTOR MANAGEMENT AND INSTALLATION

THE drift of constructive practice in the United States seems generally to be in the line of simplicity and least number of parts, in order to conform to the needs of the people that have the care of such motive power. The explosive motor now appeals, as no experience as an engineer is needed for its care and running; yet it does seem to require some common-sense as to cleanliness and the propriety of things that may assume a menacing or dangerous habit by neglect of some of the few points of attention required in persons having the charge of this rising prime mover. The ability to discover leakage of gas or oil vapors or the products of combustion in the pipe connections, through valves, or by a defective or worn piston; the thumping in journal-boxes, looseness of pins, and piston thump is easily acquired when a person assumes the care of an engine. The regulation of the explosive mixtures is fully explained in the instruction pamphlets and display sheets of the builders, and from the completeness of instructions furnished there seems nothing to fear in the first start of an explosive motor by any person of ordinary intelligence.

Cleanliness being of the first order, due attention should be given to the cleaning of the cylinder, valves, and exhaust-pipe at stated intervals; in some motors at least once a month, in other motors several months may elapse without internal cleaning being necessary, apparently without detriment. But we apprehend that the quality of the fuel has much to do with the fouling of the combustion chamber and exhaust-pipe, and therefore the quality of the fuel should be suggestive of the times indicated for internal cleaning. Excessive use of fuel or a too rich mixture is the cause of many mysterious troubles, especially in motors using the heavier oils, as with kerosene, distillate, and crude petroleum containing a large percentage of carbon, which is not burned and becomes precipitated on the interior walls of the motor and the exhaust-pipe.

The outside surfaces should be wiped off before starting or at the close of work every day, especially where the location is in a room with working people, as the odor of the lubricating oil is not agreeable when the oil is spread in excess over an engine.

In workshops or rooms where dust prevails it is most desirable to enclose motor in a small room by itself, well ventilated from without, for motor cylinders are mostly open and gather dust on their oily surfaces, and dust in the in-going air of combustion leaves grit and ashes in the cylinder. The oil for lubricating the cylinder should be the best "cylinder-oil" of the trade, and is sold by many dealers as "gas-engine cylinder-oil." It is not so expensive as to preclude its use for all the moving parts of an explosive motor, although a poorer quality is in general use.

Automatic oil-feeders are almost universally furnished with these engines, so that there should be very little waste of oil. In cleaning the internal parts from carbon and oil crust, no sharp scrapers should be used on any rubbing parts or the bearing of valves. If unable to remove the crust with a cloth and kerosene oil, a hard-wood stick and oil will generally remove the incrustation down to the metal, while the valves, if not cut, only need rubbing on their seats with finely pulverized pumice or other polishing powder. Coarse emery is not recommended, as valves often get too much grinding to their detriment by the use of this material.

In starting a motor it should always be turned over in its running direction, and when compression makes this difficult the relief-valve (most motors have one) or the exhaust or air-valve may be opened to clear the cylinder, if an overcharge of gas or a failure has been made at the first turn. In most cases turning the flywheel two or three revolutions will clear and charge the cylinder under the usual conditions for starting. With most motors a starting device is provided, which is described in the special exhibit of the explosive motors further on.

Some of the troubles to be met are severe explosions after several misfires, by which the cylinder may become overcharged with the combustible mixture. This is often caused by irregular work on the engine, and the consequent scavenging of the cylinder of the products of previous explosions, replacing with pure mixtures at the next charge. Again, by a misfire from failure in the igniter an explosive charge is intensified at the next ignition or exploded in the exhaust-pipe. Other interruptions sometimes occur, such as the sticking of the exhaust-valve open by gumming of the spindle or a weak spring. From this may also arise some of the back-firings in the muffler and exhaust-pipe. All of these explosions taking place at irregular times may be attributed, first, to irregular work; second, to irregularity in the operation of the valve gear or igniter, and although not pleasent to the ear may not be considered dangerous, because the motors and all their parts subject to explosion are made equal in working strength to the greatest pressure made by such explosions.

With the compression usual in motors, 40 to 60 pounds, the greatest force from misfire or back-fire explosives can scarcely reach 300 pounds per square inch in the cylinders and 150 pounds in the mufflers, unless, by a possible contraction of the exhaust-pipe by carbon deposit, a muffler-pot may have possibilities of rupture. In no case should an exhaust-pipe be turned into a chimney. With gas-engines the full power is sometimes not realized from insufficient gas supply. The gas bag is a good indicator of this condition, caused by a too small gas-pipe or a small meter, by which a flabby appearance of the gas bag shows that the motor is drawing more than the pipe or meter can supply with a proper working pressure. The muffler-pots have been known to accumulate water in cold weather, by condensation of the water vapor formed by the union of the hydrogen and oxygen of the gas and air, to such an extent as sometimes to cause fear in an attendant of a cracked cylinder and leakage of water in from the jacket circulation.

The water should be drawn off occasionally from the muffler-pot by a cock. Gas-motors running with electric igniters sometimes do not start at first trial from the accumulation of air in the gas-pipe. Testing by a gas-burner or a second trial will show where the difficulty lies and its remedy. And, finally, much caution should be observed in examining the interior of valve chambers and the electric exploders by taking off caps or plugs and using a light near them until assured that fuel-inlets are closed and the motor has been turned over several times to clear it of all explosive mixture. The consequences of explosions from peepholes are obvious. Even when a motor has been idle for a time it should be opened with the above caution, and electric lamps of the incandescent type always used as a source of light, if possible.

The adjustment of governors requires only care and a careful study of the directions for operating the engines, as there are too many variations in the designs and methods of adjustment for definite instructions under this head. Much care is required in renewing the ignition-tubes, especially after the spare tubes furnished with the engine have been all used. The same size gas-pipe, and of the same length as the tubes furnished with the engine, should be made and the ends welded up or capped, so that they may contain the same volume as the original tubes. This caution will ensure the uniform adjustment of the time of ignition by change of tubes; otherwise tinkering with the position of the Bunsen burner will not enable an attendant not experienced in regulating the time of ignition to regulate it with any degree of certainty. The regulation when once lost can be properly tested only by an indicator card.

With a timing valve and the amount of lead for the return fire from the tube being known, the adjustment of the timing-valve throw can be made from the position of the dead centre of the crank at the end of the forward stroke. The timing lead is the time that is required for the mixture to pass the valve and become compressed in the igniting tube and the flame to return to the combustion chamber, as measured on the circumference of the timing-valve cam. Other than iron tubes are used, such as nickel-steel, platinum, machine steel and porcelain, with satisfactory results. The porcelain tubes are made short and require a special fitting to adapt them to a chimney, or the chimney should be of special design, for a cross impact of the flame of the Bunsen burner.

There are many points in the management of explosive motors that cannot be discussed in a general treatise, arising from the varied details of design, in which special reference to the methods of operating the valve gears of igniters and governors of each individual design is required. The special instructions furnished by builders are ample for the operation of their motors, and if carefully studied lead to success in their operation by any person of ordinary intelligence or tact in handling moving machinery.

Recent experience with gas, gasoline, and oil-vapor engines has brought out more strongly the good qualities of well-made explosive motors, and placed them far ahead as a reliable, cheap, and easily managed motive power, even up to many hundred horse-power in a single installation. The application of power from explosive motors for the generation of electricity for lighting and the transmission of power is no longer a moot point of economy, but has become a fixed principle in the application of prime-moving power. The governing devices have been improved and applied in the line of uniform motion from intermittent impulse. An electric gas-governing device for controlling the flow of gas to correspond with the required amperage is a new governing application that seems to break the last objection to the use of explosive motors for generating the electric current for lighting purposes. The alternating generator is now coming into use for furnishing the igniting current with prospects of an exactitude so long as desired, and to obviate some of the exigencies of the controlling mechanism in the continuous-current system.

As it is now well known that the full firing of an explosive charge is not instantaneous from the moment of ignition in the hot tube, and that the greatest mean pressure on the piston results from perfect ignition of the whole charge at the moment of the passage of the crank over the centre, it becomes a matter of considerable importance that the hot tube and Bunsen burner should be adjusted so as to allow the compressed fresh charge to reach the part of the hot tube at which the temperature is high enough to cause ignition of the charge at a moment just before the crank reaches its centre. The variable mixture of the charge, either from misfiring of a previous charge or from the action of an over-sensitive governor, has made this adjustment heretofore somewhat difficult, especially where short-lived tubes were in use, for a change of tube usually varies the moment of ignition. Since the advent of the nickel alloy and porcelain tubes this difficulty has been greatly overcome, and the ignition tube has been restored to favor with many engine-builders who had adopted the electric system for its positive timing. The marine and automobile-engines, and all light stationary or portable forms however, will hold to electric ignition from the obvious difficulty in managing a hot tube in such service.

Many minor improvements of the past year have conduced to a general economy in running expense and to ease of management, among which may be noted a device, by the turning of which the time of sparking is retarded at starting, and the engine prevented from the possibility of starting backward by explosion before the crank reaches the centre. In this device the sparking push-blade has a double trip swiveled on the push-rod, the turning over of which changes the time of ignition.

An accumulation of air in the gas-pipe is sometimes the cause of failure in starting with an electric igniter, and often attributed to the failure of the spark. A search in both directions will find the true cause of failure. On purchasing a motor, the one who is to operate it should carefully study the mechanism and the instructions, as the detail in operating the three kinds of fuel — gas, gasoline, and kerosene or crude oil — vary enough to require special inquiry for the operation of each kind. The method of ignition is also peculiar and requires special instructions in either of the kinds of devices by which the motor is operated. Whether tube, hammer-spark, or jump-spark is selected, they are each so different in detail as to need special instruction.

One of the annoyances in explosive-motor service is the incrustation of the water-jacket by lime. Hard water, or such as contains a considerable amount of carbonate or sulphate of lime, when used as a free-running stream, has been found to choke a water-jacket in a few months so as to render the jacket almost useless as a cooling device. To obviate this difficulty a cooling tank of about twenty gallons per horse-power should be used, and filled with rain water, set above the cylinder and of such form as to give large surface to the air, with a free circulation on all sides. A round tank gives the least air-cooling surface, while a long tank of galvanized sheet-iron with vertical corrugated sides has given the most satisfactory service.

By the use of a cooling tank charged with the best water attainable, preferably rain-water, and a pound of caustic soda to each five gallons, an encrusted jacket can soon be cleaned, or the incrustation so loosened that it can be easily scraped and washed out through the core openings. Acid and water has been recommended and used; but such treatment is not convenient as the soda-circulation.

The manufacturer, if he understands his interests, usually furnishes sufficient explanatory matter to enable the operator to understand all details. Often this has been a failure, to the detriment of both maker and purchaser; but if the seller thinks he can afford to be careless about this, the buyer need not, for all shut-downs and interruptions caused by failure to operate a motor satisfactorily are more or less expensive.

For preventing the freezing of the water in the jacket or cooling tank in winter there is probably nothing better than a five per cent. addition of glycerine; or a few pounds of chloride of calcium to the water of the cooling tank will prevent solid freezing in the coldest weather. For engines exposed to outside weather, 10 to 20 per cent. glycerine may be used. Wood alcohol and glycerine combined or singly may be added to the cooling systems of automobiles to prevent freezing.

Finally, in starting a gas or gasoline-engine, it is well to remember a few facts in regard to the explosive qualities of the gas or gasoline mixture. It has been shown in other parts of this work that the proportions of gas or gasoline and air have their limits for explosive effect and that too much or too little of the fuel element is non-explosive. This is often the real trouble, when in starting a motor it refuses to go, in which case it is better to shut off the fuel and turn the flywheel over to clear the cylinder of the first charge with the relief-cock open; it should always be open in starting to save the severe work of overcoming compression resistance. The same difficulty may also occur in charging a self-starting motor of the larger size, which cannot be turned over to relieve the cylinder of the misfired charge, but by lifting the exhaust-valve and charging lightly with some pure air or fuel, as the judgment of the engineer may suggest, the start may be made. Herein lies the value of positive and full instruction that every builder of explosive motors should furnish with each motor sent out, as well as a practical lesson whenever possible to the person that is to operate the motor.

Do not think because a motor slows down by the turning on of one or two more machines than it has been giving power to, that more fuel is all that is needed, for it may have been running with more or less fuel than was due to the greatest mean pressure. It may be noted that 1 part good illuminating gas to 6 parts air or 1 part of heavy oil-gas to 9 parts air, or 1 part gasoline-vapor to 8 parts air gives the quickest explosion, the highest explosive temperature, and the greatest mean pressure. Any departures from these proportions in the mixtures are weakening in their effects, and where the highest power and efficiency of the motor is required, any variation from the above-named proportions is not the most economical in practice. As between the hit-and-miss charges and the graduation of the charge in its best mixture. there has been and is a margin for discussion in which builders of explosive motors do not agree, and may not, until long experience, trials, and new methods of regulation may lead to the best practice.

EXPLOSIVE-ENGINE TESTING

For the reason that elaborate and complicated tests have been made and exploited in other works on the gas-engine, which may be referred to for the details of expert work, the author of this work has decided to reduce the practice of testing explosive motors to a commercial basis on which purchasers can comprehend their value as a business investment for power. The disposition of builders of explosive engines to follow the economics in construction in regard to least wall surface in contact with the heat of combustion, and of maintaining the wall surface at the highest practical temperature for economical running by the rapid circulation of warm water from a tank or cooling coil, leaves but little to accomplish, save the proper size and adjustment of the valves and igniters for the engines, in order that they may properly perform their functions. The indicator card, if made through a series of varying proportions of gas or gasoline and air mixtures, will show the condition of the adjustments for economic working. The difference between the indicated power for the gas used by the card and the power delivered to the dynamometer or brake shows the mechanical efficiency of the engine. The best working card of the engine should be a satisfactory test to a purchaser that the principles of construction are correct. A brake-trial certificate or observation should satisfy as to frictional economy, and the price and quantity of gas per horse-power hour should settle the comparative cost for running. The variation in the heating power of illuminating gas in the various parts of the United States is much less than its variation in price. Producer-gas is a specialty for local consumption, and its cost drops with its heating power. Apart from the actual cost of gas in any locality and the quantity required per brake horse-power, durability of a motor is one of the principal items in the purchase of power.

In the use of gasoline, kerosene, and crude petroleum in explosive engines, their heating values are uniform for each kind, and as motors are generally adjusted for the use of one of the above hydrocarbons only, the difference of cost between these various fuels is the best indication as to the relative cost of power. No instruments have yet been contrived for giving the temperatures of combustion, either initial or exhaust, in an internal-combustion motor; for at the proper working speed the changes of temperature are so rapid that no reliable observation can be made even with the electric thermostat, as has been tried in Europe. The computed temperatures are unreliable and at best only approximate; hence the indicator card becomes the only reliable source of information as to the action of combustion and expansion in the cylinder, as well as to the adjustment of the valves and their proper action. The temperature of combustion as indicated by the fuel-constituents, and computed from their known heat values, gives at best but misleading results as indicating the real temperature of combustion in an explosive engine. There is no doubt that the computed temperatures could be obtained if the contaminating influence of the neutral elements that are mixed with the fuel of combustion, as well as the large proportion of the inert gases of previous explosions, could be excluded from the cylinder, when the radiation and absorption of heat by the cylinder would be the only retarding influences in the development of heat due to the union of the pure elements of combustion.

For obtaining the indicated horse-power of a gas, gasoline, or oil-engine, the mean effective pressure as shown by the card may be obtained by dividing the length of the card into ten or any convenient number of parts vertically, as shown in Fig. 175, for a four-cycle compression-engine. For each section measure the average between the curve of compression and the curve of expansion with a scale corresponding with the number of the indicator-spring. Add the measured distances and divide by the number of spaces for the mean pressure. With the mean pressure multiply the area of the cylinder for the gross pressure. If there have been no misfires, then one-half the number of revolutions multiplied by the stroke and by the gross pressure, and the product divided by 33,000, will give the indicated horse-power. If there is any discrepancy along the atmospheric line by obstruction in the exhaust or suction-stroke, the average must be deducted from the mean pressure.

The exhaust-valve, if too small, or with insufficient lift, or a too small or too long exhaust-pipe, will produce back-pressure on the return line, which should be deducted from the mean pressure. A small inlet-valve or too small lift, or any obstruction to a free entry of the charge, produces a back-pressure on the outward or suction-stroke and a depression along the atmospheric line, which must

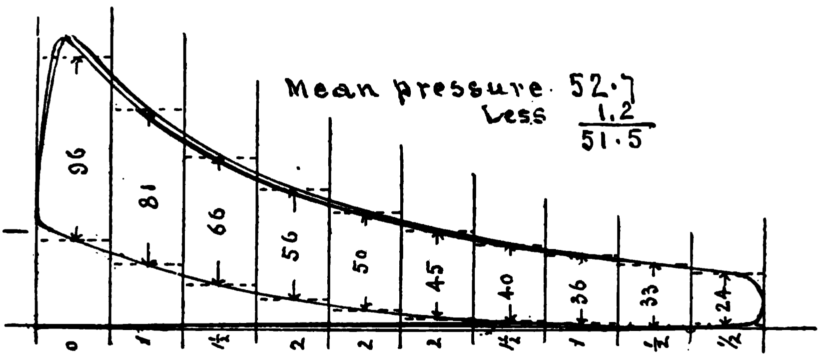


FIG. 175.—Four-cycle gas-engine card.

also be deducted from the mean pressure. It is assumed that the taking of an indicator card must be done when the engine is running steadily and at full load. During the moment that the pencil is on the card there should be no misfires recorded, in order that the card may represent the true indicated horse-power of the engine. The record of the speed of the engine should be taken at the same time as the card, but the measurement of the quantity of gas used cannot be accurately observed on the dial of an ordinary gas-meter during the few moments' interval of the card record and speed count. For the gas record, the engines should be run at least five minutes at the same speed and load and an exact count of the explosions made. The misfires or rather mischarges in an engine running with a constant load are of no importance in the computation for power because they are properly caused by overspeed, and the overspeed and underspeed should make a fair balance for the average of the run as indicated by the speed-counter.

The number of cubic feet of gas indicated by the meter for a few minutes' run, multiplied by its hour exponent and divided by the indicated power by the card or the actual horse-power by the brake, will give the required commercial rating of the engine as to its economic power. The difference as between the cost of gas for the igniter and the cost of electric ignition is too small to be worthy of consideration.

In testing with gasoline or oil the detail of operation is the same as for gas, with the only difference of an exact measure of the fluid actually consumed in an hour's run of the engine under a full load. The loading of an engine for the purpose of testing to its full power is not always an easy matter; although, when driving a large amount of shafting and steady-running machines, a brake may be conveniently applied to increase the work of the engine. In trials with a brake alone, a continual run involves some difficulties on account of the intense friction and heat produced, which makes the brake-power vary considerably and causes a like variation in the ignitions. The reader is referred to the preceding chapter for full information relative to apparatus for engine testing and practical formulæ and tables for obtaining power ratings with minimum computations.

Probably the most satisfactory method of testing the power of a motor is by its application to generate an electric current, which, if properly arranged in detail, allows the test trial to be continued for a length of time and makes the test a perfectly reliable one. For this purpose the motor may be belted to a generating dynamo of the same or a little higher rating than that of the motor. A short wiring-system with a volt and ampere-meter and a sufficient number of 16-candle-power lamps in circuit, of a standard voltage and known amperage, will indicate the power generated in kilowatts, to which should be added the loss of efficiency in the dynamo. From this data the actual horse-power of the motor may be computed, which with the fuel measurement and the speed of the motor during test trial is all that is needed for a commercial rating.

In testing motors with ordinary illuminating gas under street pressure as used for lighting purposes, the ordinary meter measurement will be found correct, but with natural or other gas supplied at high pressures, the pressure should be reduced by a pressure-regulator, or by drawing the gas from a properly weighted gas-holder. A one-inch water-pressure in an inverted glass siphon gives the proper pressure for meter measurement. The details for the finer tests of explosive motors have but little commercial value and require much expert experience in the computations in such tests; so that for ordinary purposes in testing for best effect the cylinder-cooling water should be run long enough and with the engine running at full load to establish an overflow temperature of 175° Fah., which has been found to give a good working efficiency in the cylinder temperature. This may be readily obtained by regulating the quantity of flowing water. Then the actual measurement of the gas or other fuel and its cost as compared with the brake horsepower may be said to give a fairly just measure of its fuel-economy. The test of endurance is a strictly mechanical one due to design and quality of construction, which may be obtained, first, by inspection or detailed examination of the motor, and further from guarantee of the builder.

BACK-FIRING IN EXPLOSIVE MOTORS

The so-called back-firing may be located in the exhaust-pipe or passages and is usually caused by a misfired charge being fired by the exhaust of the next impulse-charge. It may be recognized by its peculiar sound and seen at the exhaust-pipe terminal. The cause of misfiring is a frequent effect of the uncertainty of hot-tube ignition in which there is variation in the temperature of the tube at the proper point, when the greatest compression occurs. This peculiar condition has brought out the use of timing valves in large engines. The regulation of engine speed by varying the gas charge makes a variation in temperature at the ignition of the charges and so makes misfires a persistent tendency. Short-circuiting of the electric current in the break and jump-spark ignition systems is often a puzzling trouble to locate when the motor does not function properly.

There is another form of back-firing which is more perplexing still. It occurs in the inlet passage between the point of air admission or mixing-valve and the actual inlet to the cylinder. The first and most readily perceived is a leaky inlet-valve, transmitting the combustion within the cylinder to the mixture without. The other is based on the theory that the combustion of a lean mixture or a rich mixture is a prolonged one, and that a lingering flame holding over during exhaust-stroke and until the next opening of the inlet-valve fires the supply in the mixture chamber. Invariably it has been the case of the lean mixture, notwithstanding the foredrawn conclusion that it should be with the other, that the lean mixture, with its excess of oxygen, would be snapped up and quickly consumed; that the rich mixture, seeking out the last atom of oxygen, would linger in the inlet chamber, unexploded.

Irregularity of explosion, often a source of apprehension as to back-firing, is due to extreme governing action at full or partial load, which may need no further investigation than to find and correct, if the governor is not acting freely. A sticking action of the governor, often unnoticed, may lead to a suspicion of other troubles. The effect of irregular governing is shown in explosions of various strength in succession or at various intervals. This is one of the points requiring careful management in starting suction gas-motors with gasoline. The change from the feed-adjustment of a high-compression suction gas-motor for starting with gasoline should be so arranged as to allow of the least injection of gasoline that will produce an explosive charge, and thus avoid possible danger that may arise from a rich charge in a motor designed for weak charges.

VIBRATION OF BUILDINGS BY EXPLOSIVE MOTORS.

Since this class of engines has so largely superseded small steam-power, and the vast extension of their use in the upper part of buildings due to their economy for all small powers, the trouble arising from the vibration of buildings and floors by their running has largely increased. The necessity for placing motive power near its point of application has resulted in locating gas, gasoline, and oil-engines in light and fragile buildings and on floors not capable of resisting the slightest synchronal motion.

This subject has been often brought to our notice since the advent of the gas-engine in the lead for small powers. It is a difficult question to advise remedies for it, from the variety of ways in which the effect is produced. Synchronism between the time vibration of a floor and the number of revolutions of the engine is always a matter of experiment, and can only be ascertained by a trial in varying the engine speed by uniform stages until the vibration has become a minimum. Then if the engine speed of least vibration is an inconvenient one for engine economy, or for the speed layout of the machinery plant, a change may be made in the time vibration of the floor by loading or bracing. The placing of a large stone or iron slab under a motor will often modify the intensity of the vibration by so changing the synchronism of the floor and engine as to enable the proper speed to be made with the least vibration.

A vertical post under the engine is of little use unless it extends to a solid foundation on the ground; nor should a vertical post

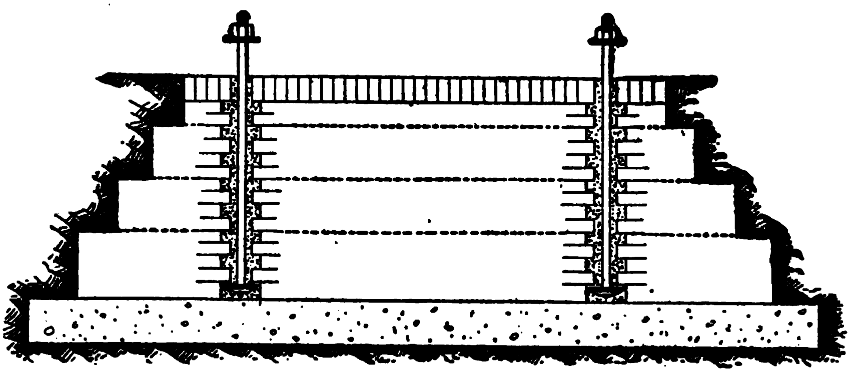


FIG. 176.—Method of securing heavy engine to concrete foundation.

be placed between the engine-floor and floor-beams above, as it only communicates the vibrations to any floor in unison with the vibrations of the engine-floor. A system of diagonal posts extending from near the centre of a vibrating floor to a point near the walls or supporting columns of the floors above or below, or a pair of iron suspenders placed diagonally from the overhead beams near their wall bearings to a point near the location of an engine and strongly bolted to the floor-beams, will greatly modify the vibration and in many cases abate the nuisance.

In the installation of reciprocating machinery on the upper floors of a building in which the reciprocating parts of the motor, as a horizontal engine, are in the same direction as the reciprocating parts of the machines (as in printing press-rooms) the trouble from the horizontal vibration has been often found a serious one. It may be somewhat modified by making the number of the strokes of the engine an odd number of the strokes of the reciprocating parts of the machine. It is well known to engine-builders that explosive motors, like high-speed steam-engines, cannot be absolutely balanced, but their heavy flywheels and bases go far toward reducing it by absorption, and the best that can be done with the balance is to make as perfect a compromise of the values of the longitudinal and lateral forces as possilbe.

The jar caused by excessive explosions after misfires and muffler-pot explosions is of the unusual kind that cannot be easily provided with a remedy where the transmitted power is not uniform, for where it is uniform there is ample regulation from the governor to make the charges regular, and if the igniter is well adjusted there should be no cause for "kicking," as our European cousins call it. A good practice in setting motors is to locate them near a beam-bearing wall or column that extends to the foundation of the building. Many motors so placed are found to be free from the nuisance of tremor. When motors are installed on the ground level, a foundation such as shown at Fig. 176, made of concrete, will entirely eliminate the disagreeable item of vibration. The duplication of cylinders and the definite counter-balancing now in use has, in a great measure, modified these troubles and two, three and four-cylinder motors are in great favor where only unstable foundations are available.

FIRE UNDERWRITERS' REGULATIONS REGARDING THE INSTALLATION AND USE OF GASOLINE-ENGINES

Rules and requirements of the National Board of Fire Underwriters for the installation and running of gasoline-engines.

As these rules are standard for practically all of the United States, they should be of interest to both the manufacturer and the user of gasoline-engines.

The rules for installation are as follows:

1. *Location of Engines* —

a. Should, wherever possible, be located on the ground-floor.

b. In workshops or rooms where dust and inflammable flyings prevail, the engine to be enclosed in a fire-proof compartment well ventilated to the outer air at floor and ceiling.

c. If located on a wooden floor the engine to be set on a metal plate turned up at the edges.

2. *Supply-tank* —

a. Shall be located outside the building, underground, where possible, at least thirty feet removed from all buildings, and below the level of the lowest pipe in the building used in connection with the apparatus.

b. If impracticable to bury the supply-tank, the same may be installed in a non-combustible building or vault properly ventilated, preferably from the bottom, always remembering that it must be below the level of the lowest pipe in the building used in connection with the apparatus.

c. Auxiliary inside tanks, if used, shall not exceed one quart in capacity, and shall not be placed on, in, or under the engine, and shall be so arranged that when the supply-valve is closed a drain-valve into the return-pipe will be automatically opened. (See also paragraph 8, Note.)

3. *Piping* —

a. None but tested pipe to be used.

b. Connections to outside tank shall not be located near nor placed in the same trench with other piping.

c. Openings for pipes through outside walls shall be securely cemented and made water and oil-tight.

d. Piping to be run as direct as possible.

e. Piping for gasoline-feed and overflow from auxiliary inside tank and feed-cup shall be installed with a good pitch so the gasoline will drain back to the supply-tank.

f. Fill and vent-pipes leading to the surface of the ground shall be boxed or jacketed to prevent freezing of earth about them and loosening or breakage of connections.

4. *Muffler or Exhaust-pot* —

a. Shall be placed on a firm foundation and be kept at least one foot from wood-work or combustible materials.

5. *Exhaust-pipe* —

a. Exhaust-pipe, whether direct from engine or from mufflers, shall extend to the outside of the building, and be kept at least six inches from any woodwork or combustible material, and if run through floors or partitions shall be provided with ventilated thimbles.

b. Shall in no case discharge into a chimney.

6. *Care and Attendance* —

Due consideration shall be given the cleaning of the cylinder, valves, and exhaust-pipe as often as the quality of the fuel may necessitate.

The rules for construction are as follows:

These rules are not to be considered as specifications for the shop construction of an engine, inasmuch as questions of design, efficiency, and operation are largely omitted. They cover only the outlines of construction of parts of special interest to the underwriters, and it should be noted that all engines conforming to the same are not of equal merit.

7. *Outside Supply-tank*—

a. Must be constructed of iron or steel plate, securely riveted together or pressed into form. Tanks should be galvanized, or painted on the outside with rust-proof paint.

b. Must be provided with a fill-pipe and a vent-pipe.

c. The fill and vent-pipes to terminate in an iron box, cover of which should be flush with the ground, and locked with a padlock.

These pipes should be provided with screen near the top and the box to be properly ventilated.

8. *Inside Auxiliary Tank* —

Note: Auxiliary inside tanks with gravity feed are not advised as their use requires extra piping and fittings, and an additional receptacle containing gasoline is introduced within the premises.

The gasoline feed-cup provided for below is sufficient for all ordinary purposes.

a. Must not exceed one quart in capacity and must be constructed in an improved manner of brass or copper of at least No. 20 B. and S. gauge or else made in a casting.

b. Must have no valves or plugs opening into the room with the exception of an air-vent.

c. Must be provided with an overflow connection draining to the outside supply-tank.

9. *Gasoline Feed-cup* —

a. Must be of cast metal rigidly secured to the engine-frame or mixing chamber, and must not exceed in capacity one-half pint.

b. Must be provided with an approved controlling-valve or regulator.

c. Must be arranged to prevent spattering, dripping, or exposure of gasoline during operation or with the engine at rest.

d. Must be provided with an overflow connection draining to the outside supply-tank.

10. *Gasoline Feed-pump* —

a. Should be of the simple single-plunger type with check-valve as close to the pump as convenient.

b. No packing should be used on plunger of pump.

11. *Igniter or Exploder* —

a. Electric ignition must be used.

12. *Muffler or Exhaust-pot* —

a. Must be made equal in strength to the cylinder or other parts subject to effects of the explosion, and should be made in cylindrical or spherical form with as few joints as possible.

b. Must be provided with a draw-off or drain-valve placed near the bottom and below the exhaust-pipe connection.

13. *Valves* —

a. Shut-off valves must close against the gasoline supply, must be made of brass and have a stuffing-cap of liberal size arranged to force the packing against the valve-stem.

b. No packing likely to be affected by gasoline to be used.

c. Regulating valves, if not designed to close against the gasoline supply, or if used as a shut-off valve, must be provided with a special stuffing-cap having a follower-gland designed to hold and compress the packing.

Note: Engine-valves of the poppet type should preferably be so placed that the gravity will act with spring to keep the valve closed.

14. *Pipings and Fittings* —

a. Tank and drain-piping must be of brass or iron, not smaller than $\frac{3}{8}$-inch size. Drain-pipe to be at least one size larger than supply-pipe.

b. Connections by right and left couplings are advised in place of unions.

If unions are used they must be of brass, with a ground conical joint, obviating the use of packing or gaskets.

c. A filter must be provided in the gasoline supply-pipe located near the engine and accessible for purpose of cleaning.

Note: A substantial flange-fitting containing fine brass gauze is recommended for use as a filter.

15. *Engine Base* —

a. Must not be used as a storage space for gasoline or any other material.

b. It is recommended that the base be constructed with a groove or channel to prevent lubricating-oil from soaking into floors.

16. *Lubricating Oil-drips and Pans* —

a. Must be provided where necessary to prevent the spilling of oil.

b. Cranks and other rapidly revolving or reciprocating parts must be shielded to prevent throwing of oil.

17. *Name-plate* —

a. Must be provided with a plate giving the name of the manufacturer, the trade-name of the engine, and its rated horse-power.

The Southeastern Tariff Association, operating in Alabama, Florida, Georgia, North and South Carolina, Virginia, and some other Southern States, uses the following gasoline-permit:

Specifications to which all gasoline-engines must conform in order to be approved for their installation:

1. Engines to be ignited by electric spark; tube-igniters not allowed.

2. Storage-tanks for gasoline shall be located underground, outside of the engine-room, and top of tank shall be below the level of the base of engine and not less than ten feet away from any building. Gasoline must be drawn from the general supply-tank, either to the engine, or the auxiliary or secondary reservoir or receptacle into which the pump discharges, and out of which the gasoline is fed into the engine. The overflow of said auxiliary or secondary reservoir or receptacle must lead back to the main storage-tank and be of four times the capacity of the pump.

3. Tanks to be cylindrical in shape and constructed as follows: viz., less than 200-gallon capacity to be of not less than $\frac{1}{8}$-inch steel throughout. Tanks of 200 to 300-gallon capacity to be of not less than $\frac{3}{16}$-inch steel throughout; heads to be stayed with iron; seams of all tanks to be securely riveted and caulked. Tanks to be coated with tar before being placed in the ground. No tank of larger than 300 gallons allowed.

4. Pipes leading from storage-tank to engine must be put together at every joint, metal to metal with pipe-screw connections. Supply and overflow-pipes to incline toward tank in order that surplus gasoline may drain back to tank from building when engine is not in operation; hand-valves to be placed in each supply and overflow-pipe outside of building, said valves to be closed when filling tank and when engine is shut down for the night. A vent provided with screw-cap must be attached to tank, said pipe to be open during filling. Storage-tank must be always filled by daylight, and all attachments between supply-wagon, tank-car, or barrels shall be tight-fitting screw-connections.

5. Any form of carburetor or vaporizer (that is, engines with a carburetor or vaporizer so constructed that by the passing of air over or through the gasoline the explosive mixture is formed within the carburetor or outside of the engine cylinder) is prohibited. This rule will apply except where vaporizer or carburetor has been specifically approved by this Association.

KEROSENE OIL-ENGINES.

In New York City stationary gasoline-engines are prohibited. The following are the requirements of the New York Board of Fire Underwriters for the installation and use of kerosene oil-engines:

Location of Engine —

Engine shall not be located where the normal temperature is above 95° F., or within ten feet of any fire.

If enclosed in room, same must be well ventilated, and if room has a wood floor, the entire floor must be covered with metal and kept free from the drippings of oil.

If engine is not enclosed, and if set on a wood floor, then the floor under and three feet outside of it must be covered with metal.

Feed-tank —

If located inside the building, shall not exceed five gallons in capacity, and must be made of galvanized iron or copper, not less than No. 22 B. and S. gauge, and must be double seamed and soldered, and must be set in a drip-pan on the floor at the base of the engine.

Tanks of more than five-gallon capacity must be made of heavy iron or steel, be riveted, and be located, preferably, underground outside of the building. If there is no space available outside the building for a tank, it may, by written permission from this Board, be located in an approved vault attached to the building, or in a non-combustible and well-ventilated compartment inside the building, but no such tank shall exceed five barrels capacity.

Tanks, irrespective of the method of feed, must not be located above the floor on which the engine is set.

The base of an engine must not be used in lieu of a tank as a receptacle for feed-oil. A tank if satisfactorily insulated from the heat of the engine, and approved by the Board, may be placed inside of the base

In starting an engine, gas only, properly arranged, must be used to heat the combustion-chamber.

A high-grade kerosene oil must be used, the flash test of which shall be not lower than 100° F.

Oily waste and rags must be kept in an approved self-closing metal can, with legs to raise it six inches above the floor.

The supply of oil, unless in an approved tank outside the building, or in a non-combustible compartment, as above provided for, shall not exceed one barrel, which may be stored on the premises, provided same is kept in an unexposed location ten feet distant from any fire, artificial light and inflammable material, and oil is drawn by daylight only.

A drip-pan must be placed under the barrel.

Empty kerosene barrels must not be kept on the premises.

CHAPTER XV

TYPES AND DETAILS OF STATIONARY EXPLOSIVE MOTORS

THE leading features of two-cycle engines are essentially an embodiment of the Day model, shown at Fig. 177, as first made in England, and noted for the absence of valves for inlet and exhaust, and for a compression of the initial charge in a closed crank chamber, made by the impulse-stroke of the piston and a final

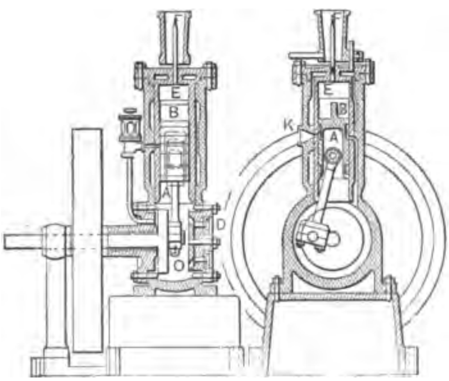


FIG. 177.—The Day model.

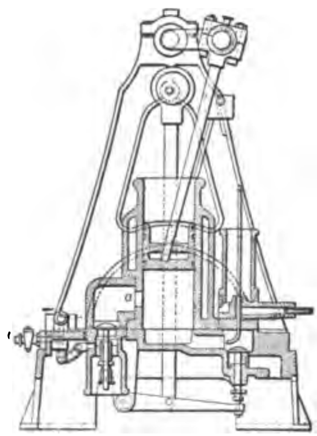


FIG. 178.—Root engine.

compression and explosion of the charge at every revolution of the crank-shaft. The air and gas or vapor are drawn into the crank chamber by the action of the piston and the mixture completed by the motion of the crank. From the absence of cylinder-valves and valve gear this type of explosive engine has the peculiar advantage that it can be run in either direction by merely starting it in the direction required. This type of motor receives its charge and exhaust through cylinder-ports at the end of the impulse-stroke of the piston. In some modifications of the Day model a supplementary exhaust is provided for by the use of a valve in the cylinder-head or near it, which facilitates the passage of the fresh charge to meet the ignition-tube or electrodes, and thus contributes to the regularity of ignition. This has become a leading type with many variations of detail, which are illustrated and described in other portions of this work.

Among the many designs for increasing the power of a gas-engine the Root model for a duplex explosion shown at Fig. 178 seemed to be a step in the right direction. It is a four-cycle compression type with a secondary explosion chamber and cylinder-port, which is closed by the piston at about half compression stroke and shutting off part of the explosive mixture, which is exploded at about one-third of the impulse-stroke by the heat of the primary explosion in the clearance space at the beginning of the stroke. The gas and air mixture was injected through the supplementary chamber, thus leaving a strong charge for the secondary explosion, and so largely increasing the pressure during

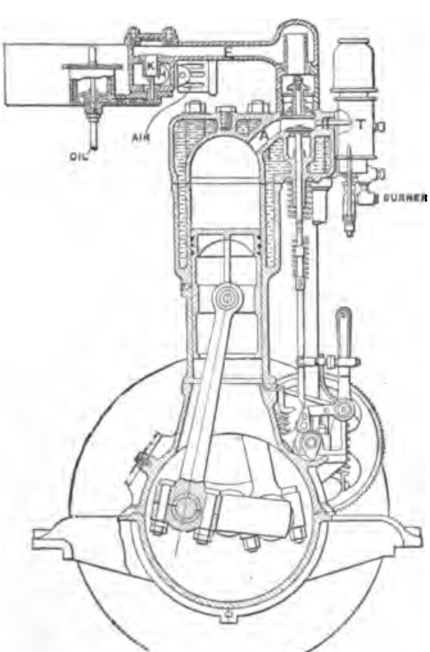


FIG. 179.—Early vertical marine or automobile model.

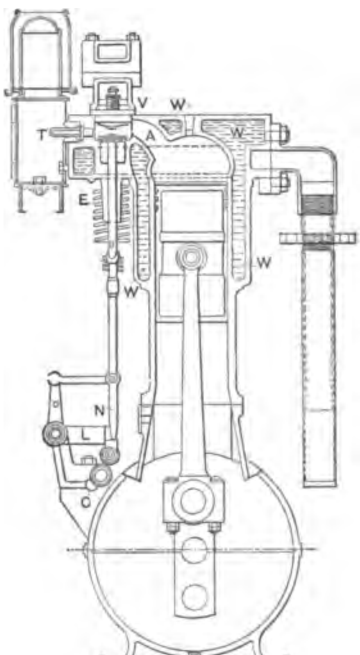


FIG. 180.—Early vertical stationary model.

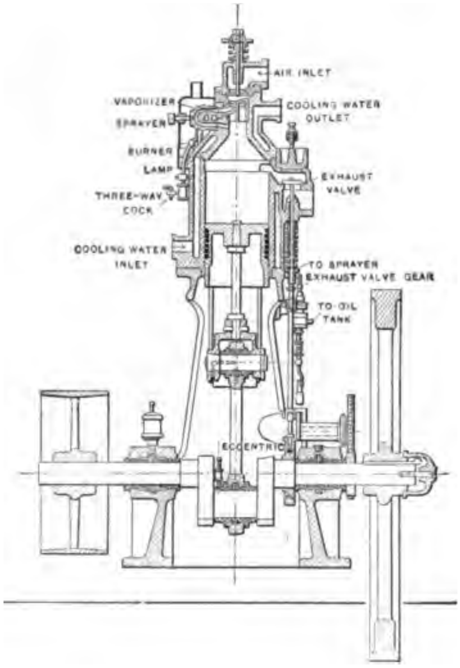


FIG. 181.—Sectional elevation.

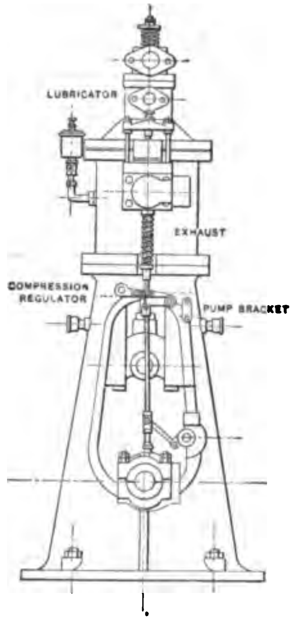


FIG. 182.—Side view.

expansion of the exploded charge. This type has not proved of practical value and the author knows of none in use in the United States. It was an English invention.

In Fig. 179 is illustrated in section a two-cylinder marine or automobile-motor of early European design, with platinum hot-tube igniter. The gasoline is fed through a regulator to a jet-nozzle at the bottom of the atomizing chamber K and mixed with the incoming air through the cage and air chamber H, and finally vaporized in the passage E. In Fig. 180 is illustrated a vertical stationary model, also of early European design, and also with a platinum hot-tube igniter and similar feed as described above. The cylinder-heads of both motors are water-jacketed, integral with the cylinder. The exhaust-valves of both motors are operated by a pick-blade action from cams on the secondary shafts; but by what means the speed is governed is not made clear.

In Fig. 181 is illustrated a vertical motor of European design with cross-head and guides, in section, and in Fig. 182 a side-view of the same motor. This type relieves the piston of side-thrust, but involves a longer gait or shorter connecting-rod—a disadvantage not approved of by our best engineers. It is derived from the lean of designers toward steam-engine practice. It is a departure from the most approved explosive-motor practice and is not recommended as the basis of simplicity in motor design.

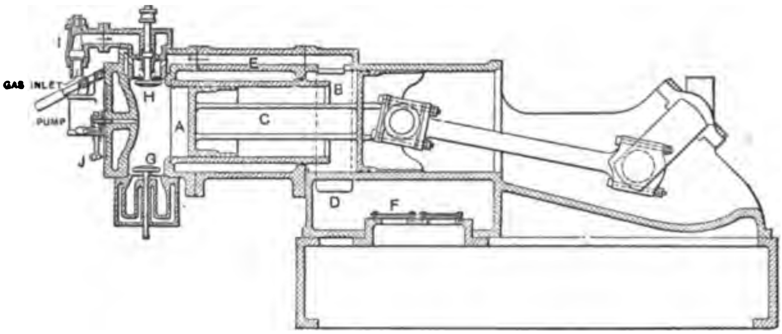


FIG. 183.—Differential piston-motor.

In Fig. 183 is illustrated a gas-engine of the scavenging class, of European design, in which a piston of larger size than the engine-piston acts as a cross-head for the connecting-rod and as a pump for compressing the air charges. Each outward stroke of the differential pistons draws air through the valves F, and by the return strokes compresses it in the chamber D, which communicates with the passage E, for furnishing the charge under pressure. The inlet-valve H opens during the last moment of the exhaust-stroke, forcing a scavenging blast from the accumulated pressure in the passage E. The double piston largely adds to engine friction and complication, which lessens the mechanical efficiency to a greater extent than the value of the scavenging effect.

In Fig. 184 is illustrated a vertical section of a gas and gasoline-engine of American construction. Its design has been toward the fewest parts that will give efficiency, ready adjustment, and renewal of vital wearing parts, together with a gas and gasoline attachment that allows of interchange of fuel elements without stopping the engine, if necessary. It has a supplementary exhaust through a port in the cylinder, opened by the piston at the end of its stroke, which has been shown to be a great relief to the work and wear of the exhaust-valve, as by this exhaust arrangement the exhaust-valve opening follows the piston-port opening. The governor controls the gas and air charge by holding or throttling the inlet duplex-valve, the lower section around the spindle being

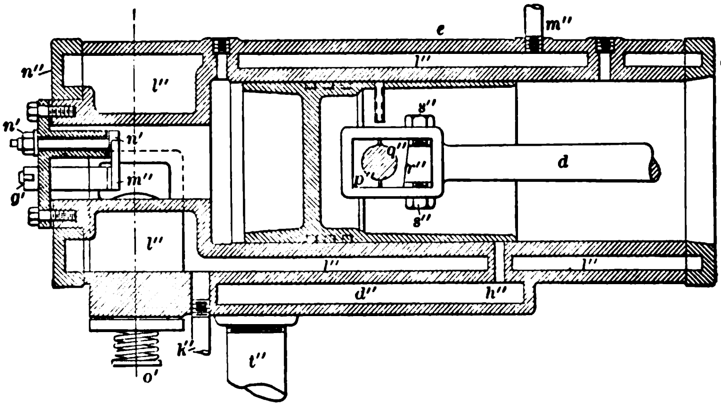


FIG. 184.—Vertical section of cylinder.

a gas chamber fed by the pipe y (Fig. 186), while the annular chamber receives the air through a side inlet, the mixture taking place between the two valves. The spindle of the gas-valve is hollow, through which the spindle of the inlet-valve passes beyond the spring-block x, at o', so that the cam-operated lever opens the

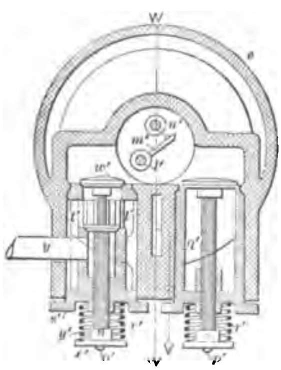


FIG. 185.—Valve-cases.

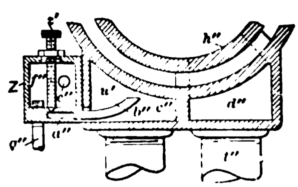


FIG. 186.—Gasoline attachment.

inlet-valve first and wider than the gas-valve. Both valves are fitted and seated in removable cases, the cylinder and head being cast in a single piece. The hole through the cylinder-head serves the work of boring the cylinder, and to receive the igniter device, which is a contact-break with a wiping motion, which prevents fouling of the electrodes, as shown at *m″*, *n′*. In Fig. 186 is a section of the gasoline attachment, consisting of a constant-level chamber *f″*, an inlet-pipe *g″*, overflow exit *e″*, a small needle-valve *z′*, and tubes *b″*, discharging into the air-mixing chamber *u′*. The cylinder and its water-jacket is cast in one piece with an open water space at the crank end, which is covered with ring flanges *o″* and *n″*. The ignition and valve chamber are water cooled as shown at *t″*.

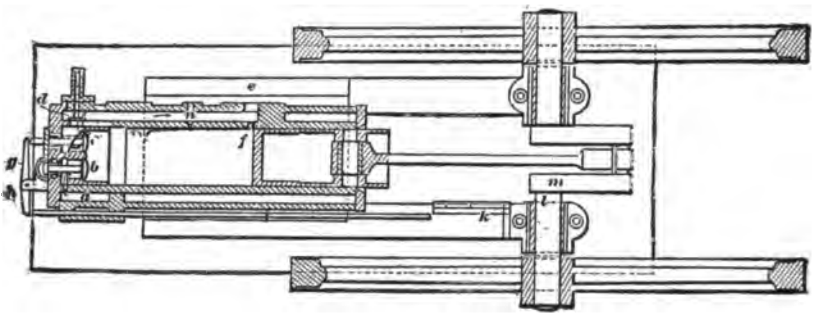


FIG. 187.—Sectional plan of the White and Middleton engine.

In Fig. 187 is shown a sectional plan of the White and Middleton motor of the four-cycle compression type, with the principal exhaust-port opened by the piston at the end of its impulse-stroke. The supplementary exhaust-valve is operated by a lever across the cylinder-head and a push-rod direct from a differential-slide mechanism, which does away with the reducing gear used on other engines. An arm on the push-rod operates the gas-valve stem, which is provided with a regulating adjustment. A small roller-disk on the push-rod mechanism is under the control of a centrifugal governor and a spring, being thrown out of gear with the shaft-cam whenever the speed of the engine exceeds the normal rate, and thus failing to open the gas supply and the supplementary exhaust-valve until the speed of the engine has returned to its

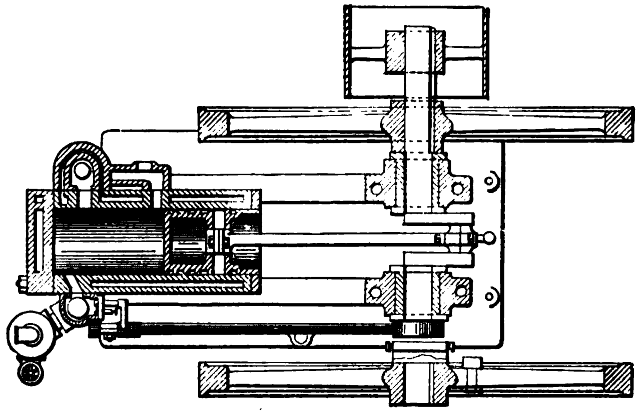


Fig. 188.—Lewis motor.

normal rate. There is a relief-valve opening into the supplementary exhaust-passage for relieving the pressure in the cylinder when starting the engine. The whole design of the engine is exceedingly simple and its action noiseless. When gasoline is used the gas-supply valve is replaced by a small pump, which is operated by the push-rod, and its hit-or-miss stroke is governed by the action of the push-rod and its governor.

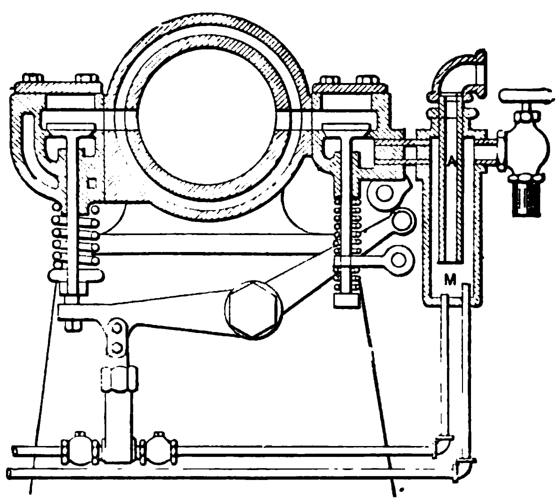


Fig. 189.—Vertical section of motor and vaporizer.

We illustrate the special construction of the Lewis gas and gasoline-motor in Figs. 188 and 189. The principal feature of this motor is the addition of the cylinder-port exhaust as an auxiliary to the regular exhaust-valve, which is now a conceded measure of economy in reduced exhaust back-pressure and in the saving of wear on the exhaust-valve. The vaporizer is shown in section in Fig. 189, which consists of a chamber M, with an air-pipe A, by which the mixture of gasoline and air is regulated by drawing the air-pipe to or from the surface of the gasoline constant-level, which is regulated by the overflow-pipe at M. A further regulation of the charge mixture is made by the valve at the right of the vaporizing chamber. The gasoline-pump is operated from the arm of the exhaust-valve lever. The igniter is of the hammer-break type and is attached by a flange to the side of the inlet chamber and operated directly from a snap-cam on the reducing

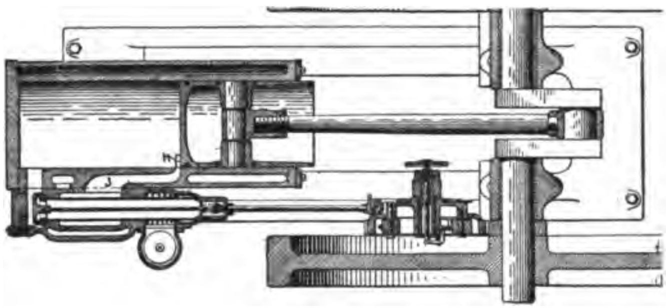


FIG. 190.—Plan of the Olin gasoline-engine.

shaft. The governor limits the lift of the inlet-valve through the arm on its spindle.

In Fig. 190 is shown a sectional plan of the Olin gasoline-engine. It is of the four-cycle type with an exhaust-port opened by the piston at the end of its impulse-stroke by which the exhaust with its terminal-stroke heat is impinged upon a tube through which the charge is fed and vaporizes the gasoline. The exhaust surrounds the vaporizing tube by the passage and chamber J. The exhaust is continued after the closure of the piston-port by an annular valve around the inlet-valve.

In Fig. 191 is shown the sectional detail of a small motor first brought out in France. The engraving has been made on

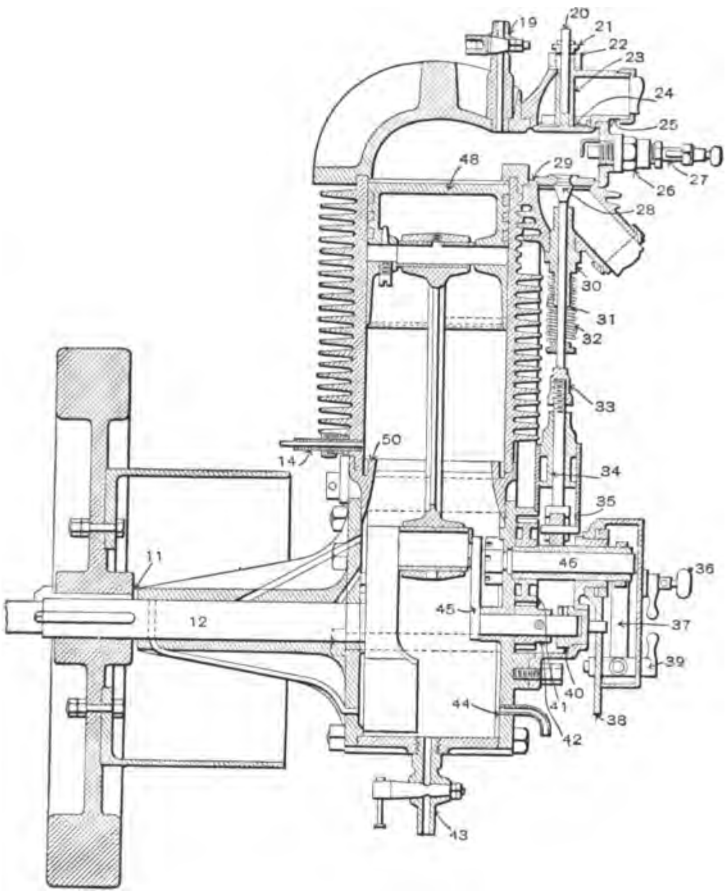


FIG. 191.—Section of air-cooled motor.

Figured parts of the motor: 12. Crank-shaft. 13. Oil-cooling tube. 14. Oil-duct. 19. Pet-cock. 20. Key. 21. Washer. 22. Spring. 23. Valve-guide. 24. Admission-valve. 25. Valve-seat. 26. Igniter. 27. Porcelain. 28. Exhaust-valve. 29. Exhaust-valve seat. 30. Exhaust-valve stem guide. 31. Exhaust-valve stem. 32. Spring. 33. Collar. 34. Exhaust-valve operating rod. 35. Cam-roller controlling exhaust. 36. Thumb-screw. 37. Contact. 38. Platinum contact. 39. Screw-controlling platinum contact. 40. Distributing-crank bearing. 41. Distributing-gear wheel. 42. Distributing pinion. 43. Drain-cock. 44. Waste-pipe. 45. Distributing-crank. 46. Cam-shaft for exhaust. 48. Piston. 49. Pin of piston-rod. 50. Oil-groove in frame.

a scale of about $\frac{3}{16}$ of an inch to one inch, the diameter of the cylinder being $3\frac{1}{4}$ of an inch, with 4-inch stroke. It is rated at 4 horse-power at full speed, and is a very simple type for the amateur mechanic to study and build. It can be used to run many different kinds of machines used in the shop or on the farm.

The latest design of the Nash gas-motor is illustrated in section in Fig. 192. It is of the four-cycle type, with one, two, or three vertical cylinders. The speed is controlled through the governor by missed charges. The air chest surrounds the passage by which gas enters and is drawn with the air into the mixing chamber A. The admission valve B is open during each suction-stroke and the mixture passes through that valve to the cylinder to be compressed upon the succeeding stroke and then exploded. The toe which lifts the gas-valve is carried upon the stem of the admission-valve and is kept from engaging with the latch upon the gas-valve stem when explosion is not required. The admission is operated by a positive cam upon the side-shaft in an obvious manner, and the fact that it is opened every fourth stroke insures an indraft of fresh air, even

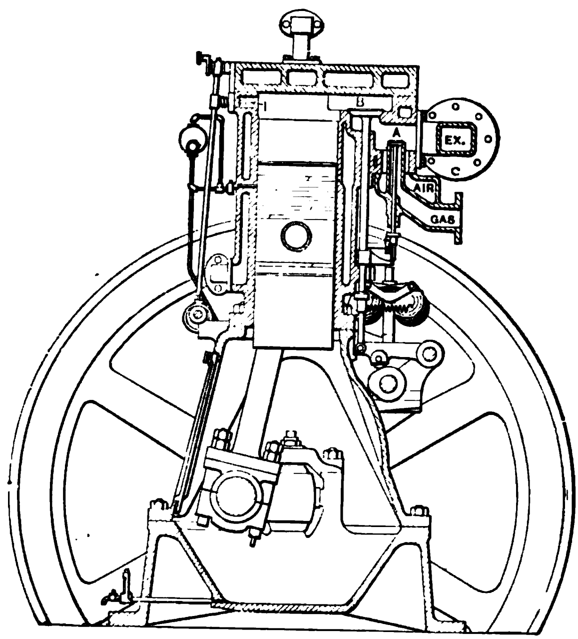


FIG. 192.—The Nash gas-engine.

when no gas is admitted, scavenging the cylinder of any products of combustion remaining. The exhaust-valve is similar to the admission-valve, but its roller can be thrown to a cam, relieving the compression when starting up. The igniter is at I and is operated by an eccentric upon a side-shaft on the opposite side of the engine, this side-shaft being operated by a cross-shaft geared to the other side-shaft, which in turn is geared to the main shaft with two-to-one spur gears. The governor is driven from the first side-shaft and simply regulates the position of the latch upon the gas-valve stem.

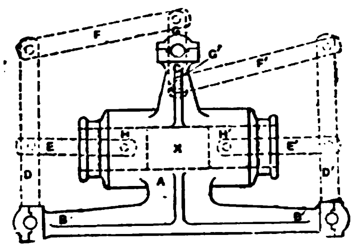


FIG. 193.—Balanced motor.

One of the ingenious but complicated designs for balancing the explosive shock is the balanced explosive motor of the Secor type in Fig. 193. The charge is fired in the chamber X between the two pistons H H′ whose motion is transmitted to the cranks G G′, having equal throw and set at 180° apart on the crankshaft. The pistons are connected by the short connecting rods H H′ to the vertical levers D D′, which transmit motion to the cranks through the connecting rods F F′.

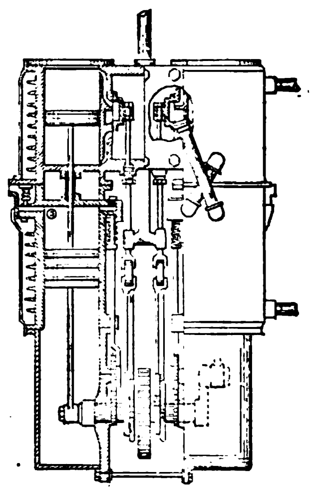


FIG. 194.—Combination motor.

A more curious than practical design of a motor is a combination of a steam and an explosive motor in one machine, as shown in Fig. 194, and is thus described: In this design the piston of the explosive motor is made the cross-head for the connecting-rod. A duplex steam-engine with a duplex explosive motor as an auxiliary power in which the exhaust of the steam-engine may also be turned into the explosive-motor cylinder as an additional power and lubricant when the explosive motor is not in use.

Another unconventional type of internal combustion engine to receive consideration is shown at Fig. 195 and is intended to utilize the exhaust gas from regular pattern high pressure cylinders in an intermediate member having proportionately greater piston top area to compensate for the lower pressure of gas discharged

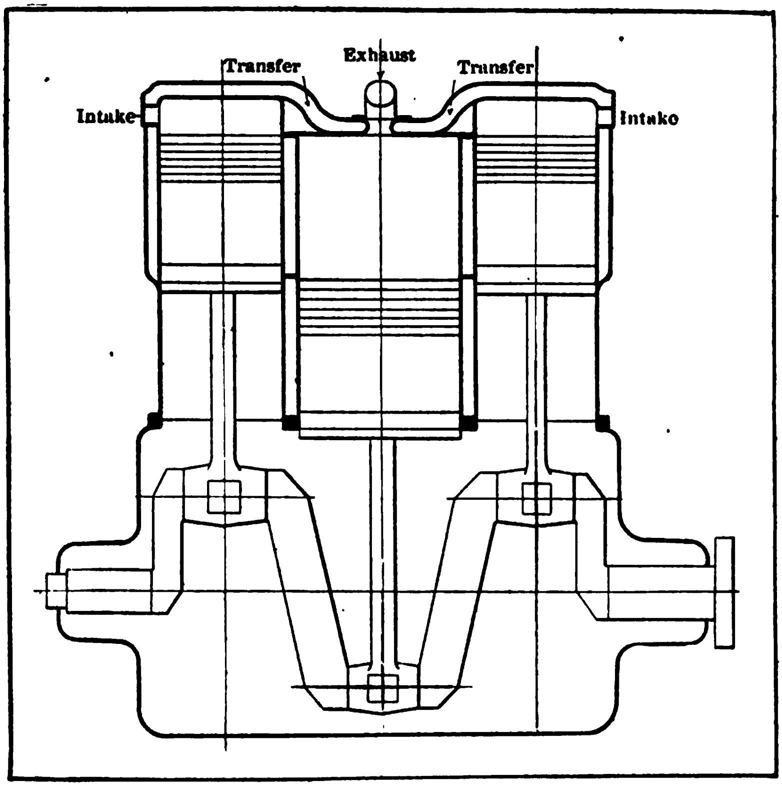


FIG. 195.—Diagram showing principle of construction involved in compound engine.

into it. The following excerpts from a paper by the late Eugene P. Batzell, read before the S. A. E., will throw some light on this form of motor.

"The well-known German authority on gas-engines, Hugo Güldner, states that compound gas-engines have no future in the industry. Here the definition 'compound engine' is given to those in which the whole charge of explosive mixture is ignited and partly expanded in one cylinder, then further expanded in other cylinders successively. Regardless of whether the above statement is true or not, one continues to hear occasionally about efforts being made toward the production of a practical and advantageous compound gas-engine. For this reason it might be well to bring out again some discussion about them and about their efficiency which could be reached in practice. The aim of a compound gas-engine lies in further utilizing the gases exploded and then expanded at a rate approximately equaling the rate of compression in the cylinder, whereby at the end of this expansion stroke the pressure still amounts to several atmospheres. This remaining gas energy is completely wasted in the exhaust of ordinary four-cycle engines, and the question arises — what proportion of this energy can be gained by adding a low-pressure cylinder?" . . . .

"The actions of the high and low-pressure cylinders of a gas-engine can be combined in different manners. We will consider the construction having opposite reciprocating pistons with the corresponding cranks at 180° to each other. One low-pressure cylinder is working in connection with two high-pressure ones, which are 360° apart in their action from each other, because the cycle of the low-pressure cylinder is completed in one revolution. A sketch of such a combination is shown diagrammatically in Fig. 195. The suction, compression and expansion strokes of the high-pressure cylinders occur like in a common four-cycle engine, but their exhaust stroke is different, because then the low-pressure cylinder becomes connected to them by a valve transfer passage. When the piston in the high-pressure cylinders expels the burnt gases within, these enter into the low-pressure cylinder and, exerting a certain pressure upon its piston in the direction of its forward stroke, cause it to deliver an amount of work. The low-pressure cylinder must have a dead space initial clearance between the back of the piston and the cylinder head, but it should be of as little volume as possible, so that at the moment of connecting the high-pressure exhaust with the low-pressure cylinder the full exhaust pressure acts upon the piston in the latter." . . . .

"A pressure loss, formerly not accounted for, is taking place when connecting both cylinders, because the dead space and the connecting passage have to be filled up as above mentioned, and this causes the initial pressure of the low-pressure cylinder to be a little lower than the final expansion pressure in the high-pressure one. However, the volumes to be filled generally can be made very small as compared with the whole high-pressure cylinder

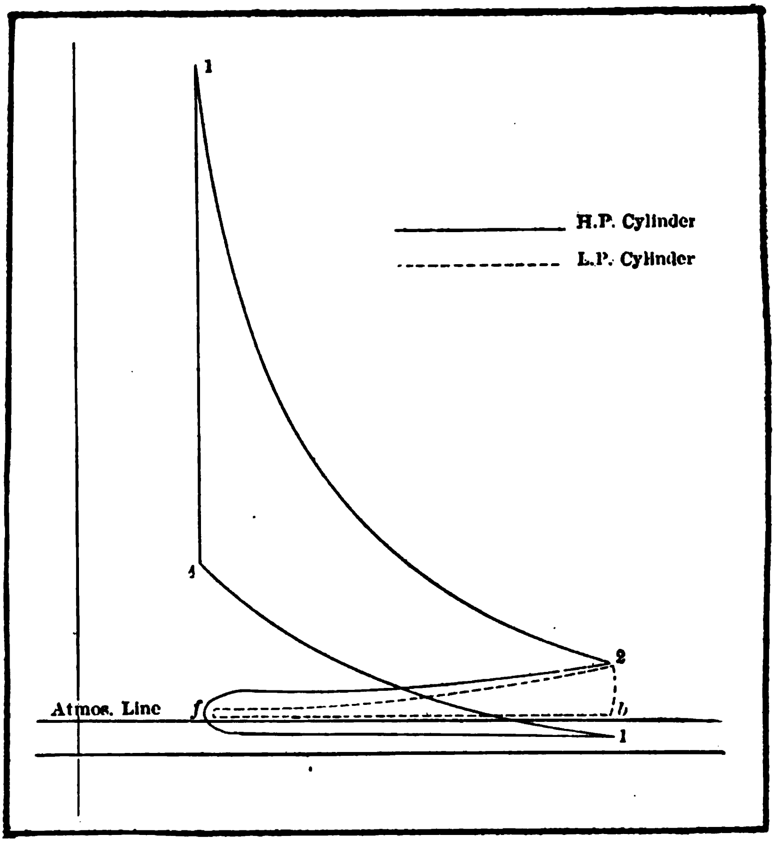


FIG. 196.—Indicator diagrams of high and low-pressure compound engine cylinders.

volume and consequently the loss of pressure will be small, hardly over 5 per cent. No further losses might be considered when correcting the ideal diagram, because the heat, etc., losses have been taken care of by setting the expansion line exponent at $n = 1.5$ for the low-pressure cylinder. It will also follow from the foregoing that the actual decrease to be made by all these corrections in the work of this cylinder can reach 20 to 25 per cent., thus bringing the gain in indicated motor power by compounding down to 11 per cent. over that of the common engine type." . . . .

"An ordinary well-built four-cycle two-cylinder motor will show a much higher efficiency than has been indicated in the A. L. A. M. horse-power formula. The mechanism of the low-pressure cylinder, when added to this existing pair of high-pressure cylinders, will absorb by its friction and other resistances less than one-half of the power absorbed in the two-cylinder motor, and it will be approximately 6 per cent. of the power developed by the latter. Thus, the low-pressure cylinder, acting alternatively with each of the high-pressure ones, in the example taken will show a gain in actual developed power of only 5 per cent., this being reached only at full motor load. In a partially loaded engine, working with closed throttles, the total of power gain will be rapidly decreased following the decrease of the load, and the power gain can not only vanish entirely, but it can be transferred into loss, depending on circumstances. An idly running engine doubtless will operate its low-pressure cylinder at the expense of the others, because the mean effective pressure in this cylinder will be practically equal to nothing due to a very low final expansion pressure in the high-pressure cylinders."

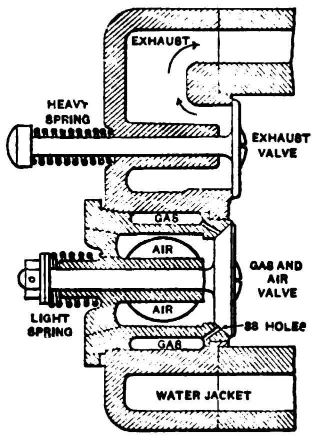


Fig. 197.—The valves.

In Fig. 197 is shown a horizontal section of cylinder-head having novel features. It is seen that the fitting of the inlet-valve casing is recessed on its outside so as to make an annular gas chamber immediately behind the valve seat and through which 38 small holes are drilled around the face of the seat, thus making a simple and thorough mixture of the charge at the moment of entrance to the cylinder, the air entering through a side passage, as shown by the circle in the valve chamber. The motor is of the four-cycle

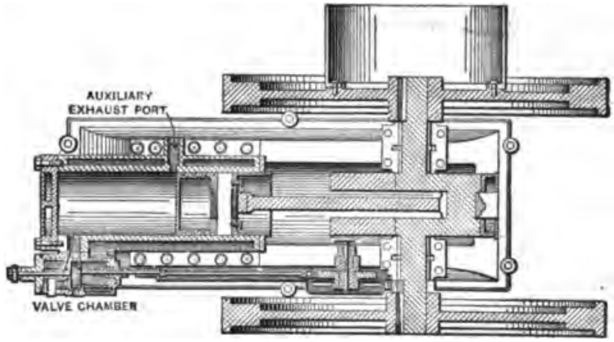


FIG. 198.—The Lazier motor. Sectional plan.

type and the exhaust-valve governs by the hit-or-miss action from the fly-wheel centrifugal governor. The regulation is by holding open the exhaust-valve by a stop-lever that catches the push-rod when the valve is open and holding it until released by the governor. It may be noticed that the valves in this design are as large as can be made practical in a cylinder-head and that the inlet-valve is larger than the exhaust-valve, which allows for a low lift for better mixing of the fuel and air.

The Lazier motors have a peculiar valve arrangement, which we illustrate in Figs. 198, 199, 200. The design is of the four-cycle type, with the hit-or-miss governing gear, but is peculiar in the fact that its exhaust-valve is the only one mechanically operated, and is so constructed that when the engine needs to miss an explosion it is held open, telescoping over the seat of the air-suction valve, cutting off all fuel supply, and allowing the piston to travel in the cylinder without compensation, during which time the valves remain in a state of rest. Fig. 199 shows a plan in section of the

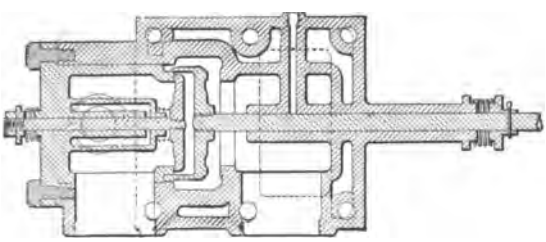


FIG. 199.—Vertical section of valves.

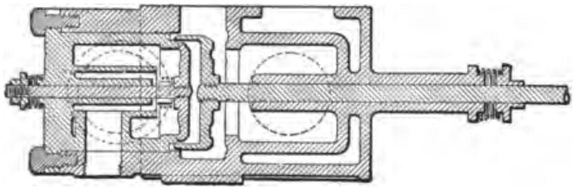


FIG. 200.—Horizontal section of valves.

cylinder, while Figs. 198 and 200 are horizontal and vertical sections, showing the valve-mechanism upon a larger scale. Fig. 201 shows the position of the valves during a suction-stroke, the admission-valves *a* A being drawn open by suction, the explosive charge entering as shown by the arrows, and the exhaust-valve E being seated. On the next stroke the charge is compressed; the next is the explosion or working stroke. At the end of the power stroke the piston uncovers the automatic port in the side of the cylinder, which allows the high-terminal pressure to be reduced, thus permitting the main exhaust-valve to open at atmospheric pressure, at which time the piston sweeps back, clearing the residue gas from the cylinder, and is then ready to take in a new mixture if governor permits, and on the next the exhaust-valve is held open, allowing the products of combustion to escape. All this time the pressure on the cylinder has been greater than the outside of the admission-valve, and there has been no tendency for the

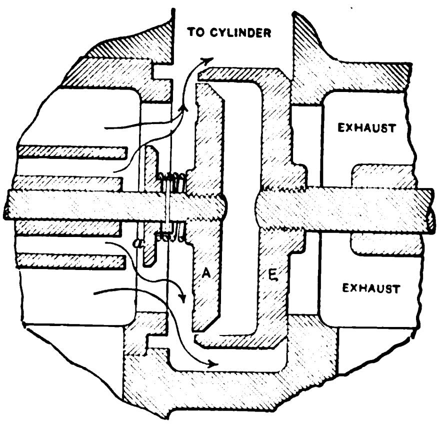


FIG. 201.—Inlet valve open.

latter to open. In fact, during the exhaust-stroke the valve is in the position shown in Fig. 201, completely covering the admission-valve. When the speed exceeds the normal, the exhaust-valve remains in this position, so that on the suction-stroke there is no vacuum created, the exhaust-passage being open, and even if there were the admission-valve is effectively closed by the telescoping of the exhaust-valve. Neither is there any useless compression, the exhaust remaining open and the valve remaining motionless until another admission is required. The air-suction and fuel-valves are mounted in a cage with ground seats with ports registering with openings in the valve chamber proper, thus allowing the valve cage to be taken out without disturbing the piping.

In Fig. 202 we illustrate in a vertical sectional view the Oil City motor. An auxiliary exhaust by a cylinder-port is one of the features of this four-cycle motor. The gas-inlet and atomizing

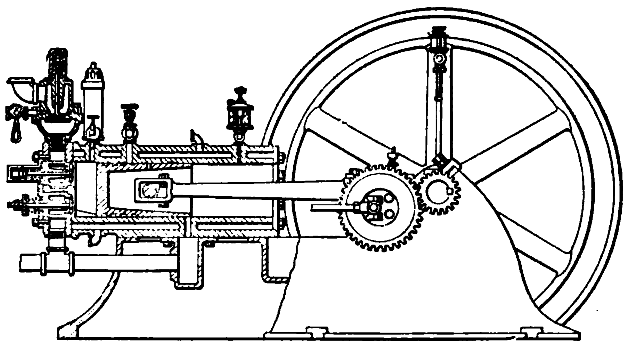


Fig. 202.—Section, Oil City motor.

valve for gasoline, seen at the top of the cylinder-head, is an annular chamber around a perforated valve seat, with space between it and the final inlet-valve for thorough vaporization of the gasoline and mixing with the incoming air. In their smaller motors regulation is made by holding the exhaust-valve open by the governor. In the large motors the throttling system is used.

The Bessemer engine is of the two-cycle type and its operation is as follows: During the backward stroke of the piston, Fig. 203, the mixture of air and gas is drawn into the front end of the cylinder through the port A, while at the same time the previous charge is being compressed in the back end or combustion chamber

B. As soon as the piston completes the stroke, the charge is ignited and the piston driven forward by the burning gases. When the piston reaches the end of the stroke in the direction of the shaft, the exhaust-port C is opened, and at about the same time the gas-port D, at the top of the cylinder, is opened, admitting the fresh charge, which was compressed by the piston during the working stroke. The incoming charge enters the cylinder under moderate pressure and drives the burnt gases before it, thus filling the cylinder very quickly with the fresh mixture.

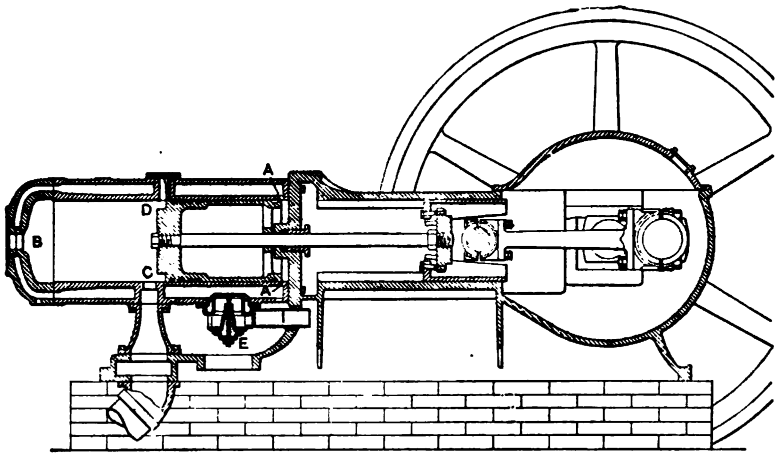


FIG. 203.—Longitudinal section of engine.

The air and gas are drawn into the front end of the cylinder through the gas-valve E, located beneath the cylinder, Fig. 204 being an enlarged view of this valve. The air enters through the large annular opening F, while the gas is admitted through a series of small holes or ports G. The valve H when seated closes the opening F and the small ports G, both being opened simultaneously by the valve, which is raised by the suction of the piston. Air enters the valve-body through the air-pipe I (Fig. 204), which is connected with the interior of the bed to avoid drawing in dust and dirt.

The governor is located in the gas-pipe at the side and on a level with the top of the cylinder, the speed being regulated by throttling the gas and thus modifying the force of the explosion to meet the requirements of the load. The cylinder and back cylinder-head are water-jacketed, the front head having no jacket, since it is subjected to the low temperatures due to the slight compression of the fresh charge or mixture. This engine is provided with a piston-rod, cross-head, and guides the same as a steam-engine; in fact, the construction throughout is in accord with the practice in steam and gas-engine construction. The stuffing-box in the front head is subjected to only moderate pressures and temperatures, consequently no trouble is experienced in maintaining a tight and durable joint. The working parts are enclosed by a neat hood and crank-case which not only prevent dust and dirt from reaching the vital parts, but render the engine self oiling and adapted to making long continuous runs with the minimum of attention. The connecting-rod is of the marine type and extra heavy. The pins are also large and provided with means for obtaining ample lubrication. The main shaft-bearings are provided with chain-oilers which ensure copious lubrication at all speeds, and at the same time prevent any waste of oil.

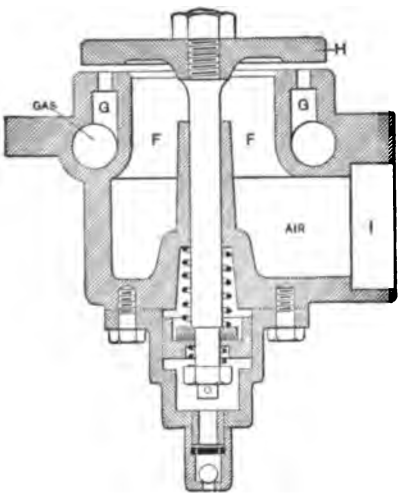


FIG. 204.—Section of air and gas valve.

The piston is oiled by means of a special automatic sight-feed oiler. The piston is very long, thus providing liberal wearing surfaces and is provided with four wide packing rings. The engine is not only very simple, but is unusually massive, being designed for all kinds of service for which gas-engines can be employed.

The Dudbridge gas-engine has some peculiarities worthy of record, and which we illustrate in Figs. 205, 206, and 207. The cylinder is overhung and bolted to the bed-piece and made in two pieces. The jacket and cylinder-head are cast in a single piece and the liner made of a specially hard mixture of iron for wearing quality and easy replacement when worn out. The valve-casings are all contained in the cylinder-head, which is spherical and

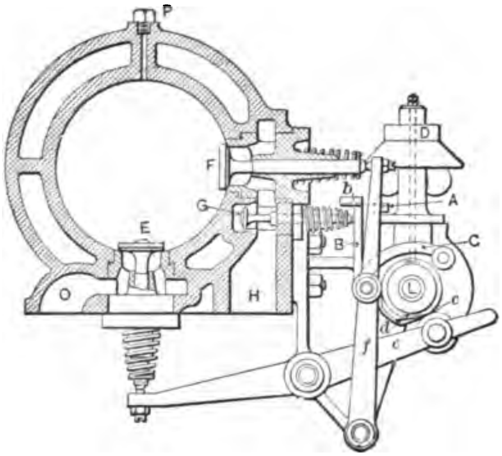


FIG. 205.—Valve gear.

water-jacketed. All valves are contained in casings with flanges and shoulder joints, easily removed for cleaning or repairs. Ignition is of the hot-tube type, as shown at J I, and the gas-inlet is regulated by an index-cock at V (Fig. 207). The governor, as will be seen by reference to the illustrations, is of the fly-ball type, controlling the engine on the hit-or-miss principle.

The construction of the valve gear may be more readily under-

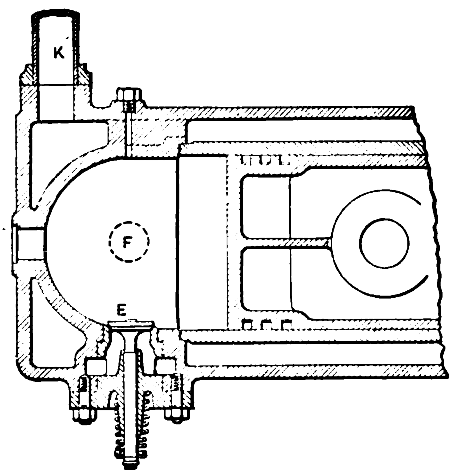


FIG. 206.—Cylinder and inlet valve.

stood by reference to the figures. All valves are worked from the reducing shaft L, which is driven from the crankshaft by means of helical gears. F and G are air and gas-valves respectively, valve G opening directly into the air-inlet H. The exhaust-valve E opens directly into the exhaust outlet O. The air-valve F is driven through the lever *f* by means of the cam *c*. The exhaust-valve is controlled by the lever *e*, operated by the cam *d*. The gas-valve is opened by means of a small arm B, and the striker-blade A attached to the air-lever arm. Small arm B also carries a striker

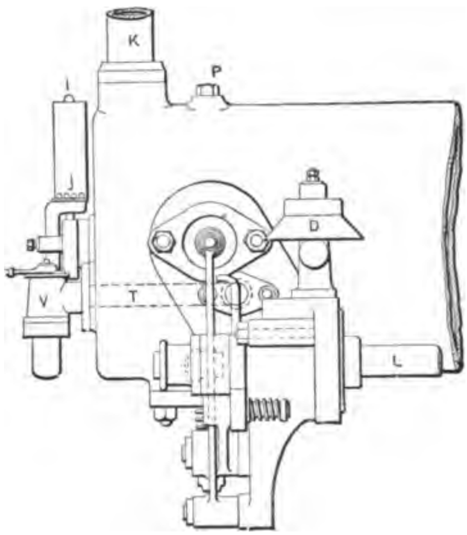


Fig. 207.—Governor and tube igniter.

which is met by the striker-arm A as it moves toward the cylinder to open the air-valve. Arm B is under control of the governor through the arm C, and so connected that, as the governor rises, lever B is lifted and the striker *b* is lifted out of the path of A. In this manner, when the speed rises above the limit, the gas-valve G is not opened, and the cylinder takes in a charge of pure air, thus missing impulses and developing less power. The speed of the engine may be increased by putting on extra weights as shown at D, or the speed may be decreased by removing weights on the governor at D.

In Fig. 208 are shown some of the details of the Wayne Motor.

A double cam on the reducing gear-shaft operates the exhaust-valve E through a push-rod and lever across the cylinder-head and also a supplementary gas-valve, independent from the free opening inlet-valve. The igniter of the make-and-break type is operated by a pick-blade on the firing-rod which engages with the arm of the igniter-spindle. The throw of the firing-rod is controlled by the governor. Fig. 209 shows a cross section of the cylinder, valve chamber, valves, and exhaust-valve bell-crank lever of the Elyria engine, a simple and compact device. Ignition is by means of an electric spark, the plug for which is placed in the

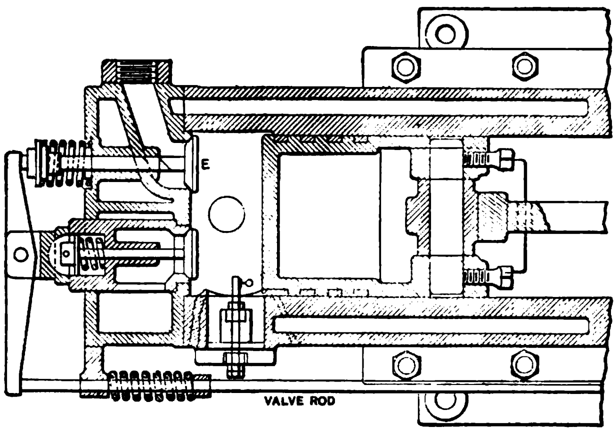


FIG. 208.—Section, Wayne motor.

inlet and exhaust-valve, where it has the benefit of the cooling effect of the incoming air, thereby prolonging the life of the sparking points. It will be seen that the inlet-valve is enclosed in a flanged bushing large enough to allow the exhaust-valve to be drawn out through the inlet-valve opening.

Fig. 210 is a cross section of the Wayne gas and gasoline-engine, showing the position and operating gear of the gas-valve, inlet-valve, and exhaust-valve. The operating cam, which is mounted on a short secondary shaft geared to the main shaft, throws the rocker-arm to the left; this movement, being imparted to the valve-rod, opens the exhaust-valve A. The spring D returns the rod when it is released by the cam and opens the gas-valve C, as the spring D is much stronger than the seating spring on the gas-valve stem. The gas-valve delivers fuel to the valve B, which is opened directly into the combustion chamber by atmospheric pressure. Thus during a normal charging-stroke the valve-rod is entirely released by the cam, and by means of the spring D holds open the gas-valve, which it releases at the end of this stroke, and the rod takes an intermediate position during the compression and working strokes. At the end of the working stroke the cam comes into position and pushes the valve-rod clear out, thus opening the exhaust. This cycle is repeated so long as the speed is at or near the normal value, but when the speed is excessive the governor raises the end of a latch, which engages a lug on the rocker-arm actuating the valve-rod, thus holding it back and allowing the gas-valve to remain closed so that air only enters the cylinder through the admission-valve.

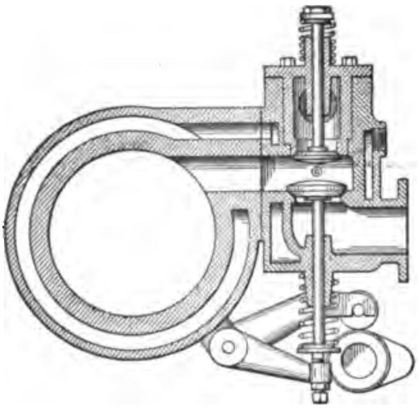


FIG. 209—Cross-section through the cylinder and valve chamber.

In Fig. 211 is represented a detailed section of a type of the Olds gasoline-engine. A notable feature, apart from the position of the valve chambers in the head of the cylinder, is the making of the cylinder and jacket in two pieces bolted together by contact with the head, which is bolted to lugs on the cylinder. The inlet-valve and seat are encased in a double-seated flanged cage, which is easily removed to allow the exhaust-valve to be drawn out through the opening.

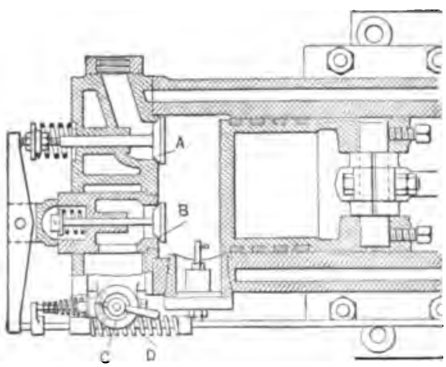


FIG. 210.—Cross-section of cylinder and valves of the Wayne engine.

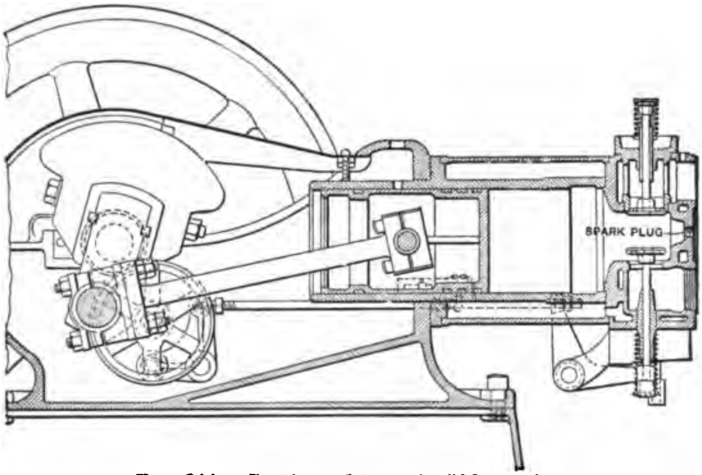


FIG. 211.—Section of type A, Olds engine.

The exhaust-valve is operated by a cam on the reducing shaft, two bell-cranks, and a push-rod.

In Fig. 212 we illustrate an unconventional two-cycle design of English origin (the Lister), in which two pistons are connected to a single crank-pin, by which a direct impulse is given to the crank when it is on the centre. The three positions of the pistons and crank-pin are shown in the three sections of the cut. It will be seen that two cylinders, A and B, are arranged parallel to each other above the crankshaft, A being the exhaust and B the inlet cylinder, connected by a common compression chamber at their inner ends. The pistons are joined by the connecting rods, R^1 and R^2, to two corners of the triangular frame, as shown, the other corner being attached to the crank-pin C. The movement of the frame is constrained by the radius-rod L, the other end of which is jointed to the casing of the engine. Ignition of the compressed charge takes place when the pistons are in the position shown by 2. The crank rotates in the direction of the arrow, so that piston B travels faster than piston A, and has approached the end of its out-stroke by the time the latter piston has arrived at the exhaust-port. Their positions are then as in 3. When the exhaust-port is uncovered the pressure drops to atmospheric, and piston B, then passing an inlet-port communicating with the enclosed crank chamber, allows a volume of air to pass through

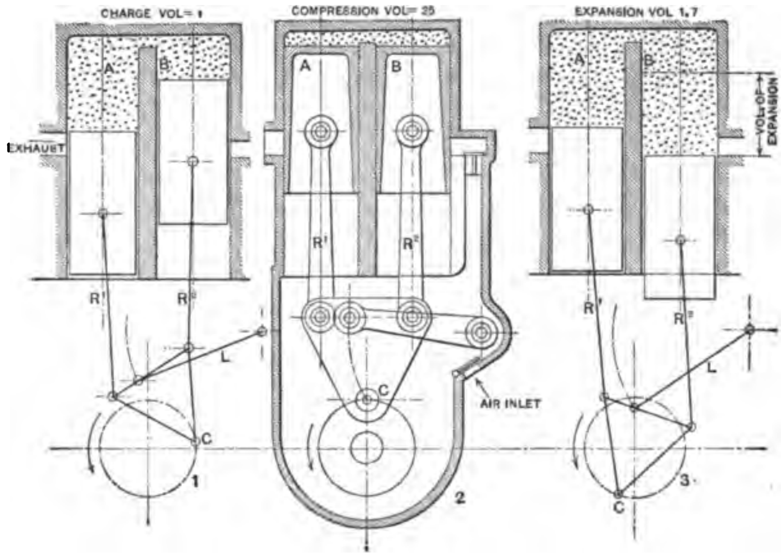


FIG. 212.—The Lister two-cylinder motor.

the check-valve into the cylinder B, in order to scavenge the cylinders from the products of the previous explosion. The piston B then commences its stroke again in advance of piston A, forcing out a quantity of air, and nearing the end of its in-stroke at the time the exhaust-port is closed by piston A. The position of the pistons before compression is shown in 1.

Shortly before the closing of the exhaust-port a charge of gas or gasoline is pumped into the cylinder B, forming an explosive mixture with the air previously drawn in. In engines, the close governing of which is not essential, the charge may be drawn into the crank chamber with the air, and thence delivered to the cylinder, thus doing away with the necessity for pump-charging, though the advantages of scavenging are lost by this arrangement. This mixture is compressed as the pistons approach the upper end of the cylinders, ignition is effected by any of the usual methods, and the cycle is repeated as before, one explosion taking place to every revolution of the crankshaft. It will be noticed that the initial volume of the charge is increased by from 50 to 70 per cent. before the exhaust, allowing more work to be obtained from the fuel together with a lower exhaust pressure. The ratio of expansion volume to compression volume is as 6 to 8. The design permits of the connecting-rods being kept very short, and they are so proportioned that at no point of the stroke do they make a greater angle with the centre line of the cylinders than 5°; thus the pressure on the cylinder walls and the consequent wear are very small. The combined effective-power strokes of the two pistons are approximately equal to 1.8 times the crank-stroke, the compression portion of the return stroke amounting to 1.2 times the crank-stroke.

In Figs. 213, 214, 215 we illustrate some of the details of a novel type in a four-cycle gas-engine of the scavenging type, made by

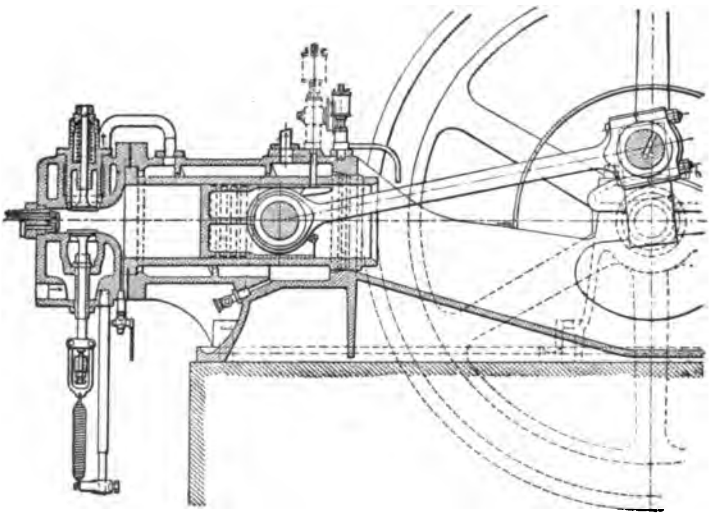


FIG. 213.—Sectional elevation of Bollinckx gas-engine.

the Société Anonyme des Moteurs à Gaz A. Bollinckx, at Buysinghen, Belgium, in which compression is carried up to 165 pounds per square inch, and a special scavenging arrangement expels the burnt gases after the explosion, thereby increasing the efficiency and preventing premature explosion. The governor is of the hit-or-miss type and ignition is effected by electric spark, produced by a magneto machine. The frame is very heavy and strong, being cast in one piece in the smaller sizes, and is designated to serve as an oil catcher. The bearing brasses are in four parts, of cast iron lined with white metal. The cylinder, which is shown in section in Fig. 213, is separate from the frame, and the latter is provided with spiral fins in the water-jacket, so that the cooling water is compelled to follow a spiral path round the cylinder, producing the maximum effect. On withdrawing the cylinder it is easy to clean the water spaces of sediment and incrustation. The crankshaft is of steel and is provided with rings to receive oil from a fixed lubricator, the oil being driven into the crank-pin by centrifugal force. Complete automatic lubrication has been avoided, as the makers believe that the attendants trust too implicitly in such devices, with the consequence that accidents result. The crankshaft is very massive, and is fully counterbalanced by counter-weights attached to the crank-webs. The crank end of the connecting-rod is fitted with phosphor-bronze bushings, and the small end with a cylindrical cast-iron bushing, working on a pin of hardened steel.

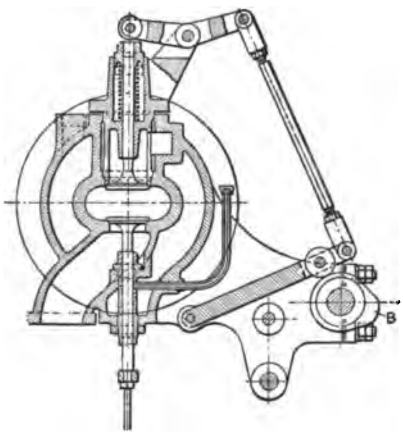


FIG. 214.—Section through admission and exhaust-valves.

The piston, as usual, is provided with a large surface bearing on the part of the cylinder which is not directly heated by the hot gases, the diameter of the piston being reduced at the back end. Only one ring is exposed to the highest temperature the remainder working in the cooler portion of the cylinder. The admission and exhaust-valves work vertically, as shown in Fig. 214, the former above the latter, and are especially easy to inspect, while their arrangement tends to prevent wear. A drain-cock, shown in Fig. 213, permits the removal

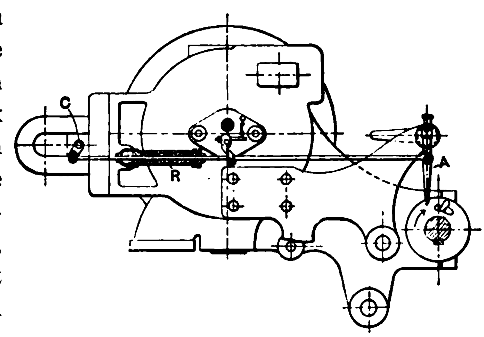


FIG. 215.—Ignition mechanism.

of oil, which might collect in the bottom of the cylinder and cause premature explosions. The valves are driven by means of a cam-shaft and cam B (Fig. 214), actuated from the main shaft by skew gear; at the end of the explosion-stroke the exhaust-valve is opened and allows the burnt gases to escape, and at the end of the return-stroke the admission-valve is opened to admit the scavenging current of air, which is sucked in by virtue of the high velocity and inertia of the exhaust gases, producing a partial vacuum in the cylinder. The vertical arrangement of the valves is more costly than other systems, but has been preferred on account of its superiority. The cylinder is lubricated by a special sight-feed lubricator, with a catch-feeder for the piston-pin.

Ignition is produced by means of a small magneto-dynamo carried on the engine. Inside the cylinder there is a fixed insulated contact and a finger, which normally rests against the contact, under the control of a spring. The armature of the magneto C (Fig. 215) is pushed round through an angle of about 90° by lever A, operated by the cam-shaft, and on its release is quickly pulled back by spring R, thus causing a momentary but powerful current to flow through the finger to the contact in the cylinder; at the same moment the finger is suddenly drawn from the contact, breaking the circuit and producing a very intense spark. Moreover, the spark is just as intense when the engine is being started, and the compression is weak, as when the engine is running at full speed. The action of the igniting device is a most novel one and well worthy of study in regard to the part revolution of the armature of the magneto generator, taking place at a uniform speed at the starting of the engine, however slow, and the trip of the circuit-breaker at a positive and adjustable time. The governor is of the centrifugal type, with provision for adjusting the speed while running, and actuates a small fork which determines whether the admission-valve shall be opened or remain closed for one or more cycles.

THE SCAVENGING ENGINE

A slight increase in the power of an explosive motor is claimed from the discharge of the products of combustion in the clearance space at the moment of the close of the exhaust-stroke, by holding open the exhaust-valve until the crank is slightly past the centre and mechanically giving a free opening to the air-inlet valve or a supplementary valve arranged to give a free air-inlet at the right moment. The addition of a lengthy exhaust-pipe, with bends instead of elbows, gives the rapid-flowing exhaust a momentum that produces a slight vacuum or draught in the combustion chamber and through the air-inlet valve, which sweeps out the products of combustion and fills the clearance with fresh air, while the piston is nearly stationary at the end of the exhaust-stroke. An exhaust-pipe of about 100 times the length of the stroke, with the muffle-pot at the end of the pipe, has been found to give the best effect. A saving of about 20 per cent. per brake horse-power has been shown by scavenging over non-scavenging engines as constructed by the Crossleys in England. For this type the valves must be located on opposite sides of the cylinder and so arranged that the gases of combustion will pass out with as little friction as possible. The Crossley four-cycle scavenging engine was designed with the curved cylinder-head and piston-head to conform to least friction, but any motor with valves in line on opposite sides of the cylinder can be given the scavenging effect, more or less efficient according to the valve and exhaust-pipe arrangement. Nor is it necessary to adjust the inlet-valve for air alone to enter at the moment of scavenging, as there can be but little loss in scavenging with the explosive charge. A considerable increase in the explosive pressure may be obtained, with a consequent increase in the power of the motor, from a full charge of explosive elements.

THE FAN-COOLED MOTOR

In Fig. 216 is shown a motor with a fly-wheel fan cooling system consisting of light wings attached to the front face of a fly-wheel, and the wheel and fan encased to direct the air-blast directly on to the motor-head and cylinder air-cooling flanges. This system has been the subject of English experiments with the following results: When enclosed in a suitable case, arranged to concentrate the whole blast on the engine, it took only $\frac{1}{20}$ of a horse-power at full speed, and gave a blast of 25 to 28 miles per hour. It kept the engine rather cooler than when running full speed on the road without a fan. The flywheel fan absorbed so little power that it was very difficult to detect or measure the power absorbed. By employing a small electric motor to run the flywheel alone in its

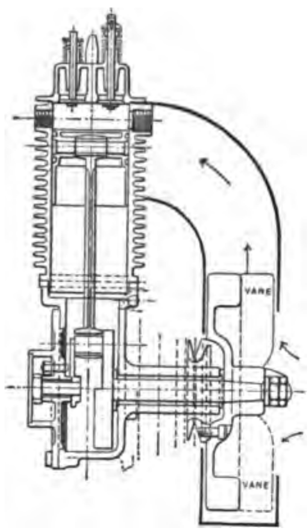


FIG. 216.—Motor-driven air-blast.

bearings (the piston and the rest of the engine gear being removed) with and without the fan attached, it appeared that the power absorbed at 2,000 revolutions did not exceed $\frac{1}{30}$ of a horse-power, which is quite negligible in comparison with other losses. In slightly modified form this system of cooling has been used successfully in connection with automobile and stationary power plants as previously described.

THE GEMMER VERTICAL INVERTED CYLINDER MOTOR

In this unconventional design the cylinder is placed downward, with the crankshaft up, as it can be lubricated much better, and some parts, especially the piston and connecting-rod, are much easier to get at than if the cylinder were the other way up. The governor is attached to the reducing-gear. AA are the weights. When the speed is above normal, these weights fly out, overcoming the tension in the spring B and sliding hardened steel collar C toward the gear-wheel D, when the steel piece E on the lever F engages with the steel catch G and holds open the exhaust-valve. The exhaust-valve is opened by a cam W on gear D, pressing down on roller J mounted on a hardened steel pin in lever F, which presses down the stem H, opening the valve. When the speed returns to normal and the cam again presses down stem H the tension of the spring B brings the weights AA together, moving the collar C so that the E returning misses the catch G, permitting the valve to close, when the engine takes up its regular cycle. This governor is very sensitive and holds the speed constant, making the engine suitable for operating a cream-separator or any machine requiring a steady speed. (Fig. 217).

The pump draws the gasoline from the supply tank, which may be placed outside of the building, thus complying with the insurance regulations. The engine is shipped with this tank in

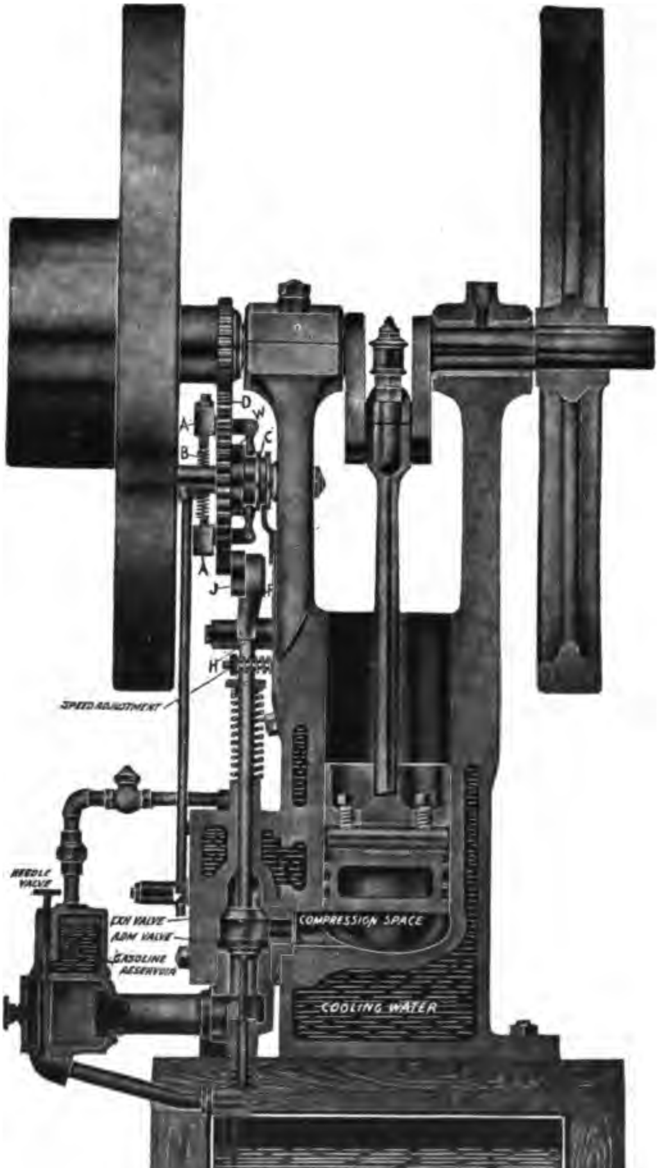


Fig. 217.—Gemmer vertical engine.

the wooden sub-base, as shown in the section. The pump may be worked by hand at will, which is a great convenience, as the gasoline-vaporizer reservoir must be filled before starting. In the vaporizer the gasoline is fed through a sight-feed needle-valve and drops onto a brass wire screen, where it is caught by the incoming air and sprayed through other screens of graduated meshes, atomizing it perfectly. This vapor passes through the fuel-valve, opened at the proper time by the governor, and is mixed with the necessary amount of air for perfect combustion and enters the cylinder through the inlet-valve. Any possible mixture of vapor and air desired is obtained by simply turning the brass knob, which controls the passages to both the gas and air chambers of the vaporizer.

ROOT AND VANDERVOORT MOTORS

The Root and Vandervoort motors are of the horizontal and vertical four-cycle type, and are well designed for all kinds of power service and for pumping and hoisting. In Fig. 218 we illustrate their horizontal gasoline-engine, showing the valve-gear and gasoline-pump operated by a side-shaft driven by spiral gears from the main shaft, at half-speed for the four-cycle effect. A fly-ball governor driven from the side-shaft controls the flow of gasoline to the atomizer and vaporizer, so that the engine speed is governed by the varying volume of fuel. The ignition is electric, of the hammer-spark type, operated by push-rod from a crank-pin at the end of the side-shaft, as shown in the cut. In Fig. 219 we illustrate the horizontal engine, direct connected to a four-pole generator on a substantial base bolted to the engine-base. The running of this engine of eight, fourteen, and eighteen horse-power direct connected to a four and one-half, eight and one-half, and twelve-kilowatt generator of 110 to 120 volts is so steady that the voltage does not

FIG. 219.—Direct-connected engine and generator.

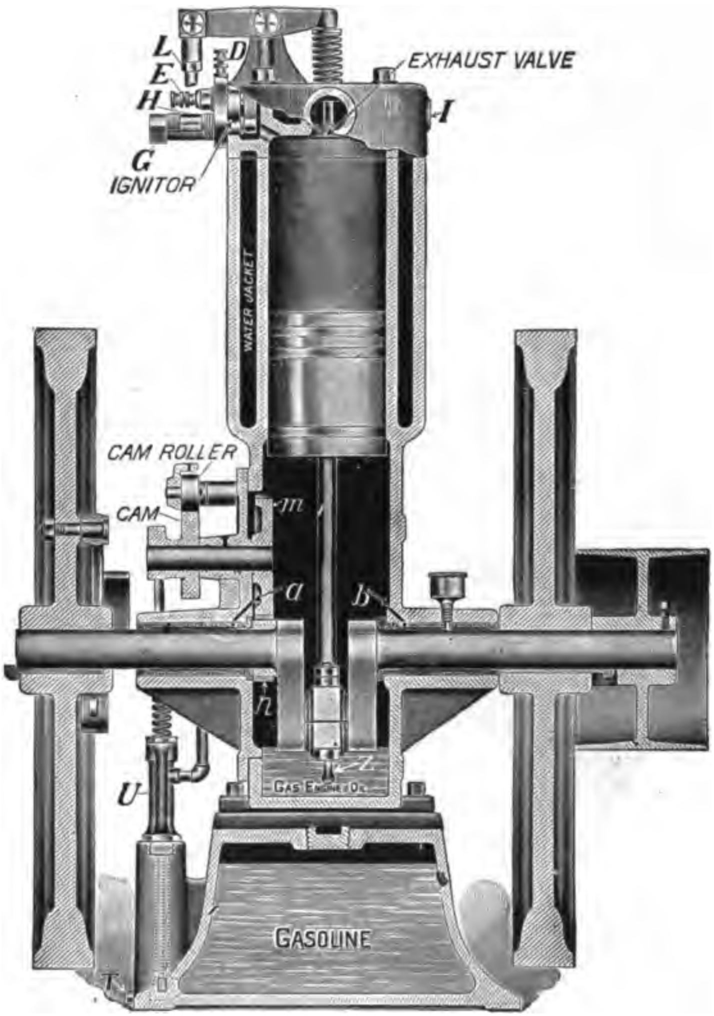


FIG. 220.—Section of R. & V. vertical engine.

fluctuate to exceed one volt. The vertical engines of this company are of the same cycle type as before described, but the arrangement of the valve and pump-gear are made to meet the vertical position of the cylinder. A pair of spur-gears on the inside of the crank-chamber drives a short shaft on which are fixed the exhaust and pump cams. The exhaust push-rod also carries a short igniter-rod, which by a double motion of the exhaust-rod operates the hammer-stroke of the igniter.

In Fig. 220 is illustrated a section of the vertical engine, showing details of the parts. The pump, operated from a cam on the small shaft, pumps an excess of gasoline to the small constant-level reservoir at the top of the cylinder, and overflows to the main reservoir in the base of the engine, which holds a day's supply. By this means a constant level of gasoline is maintained

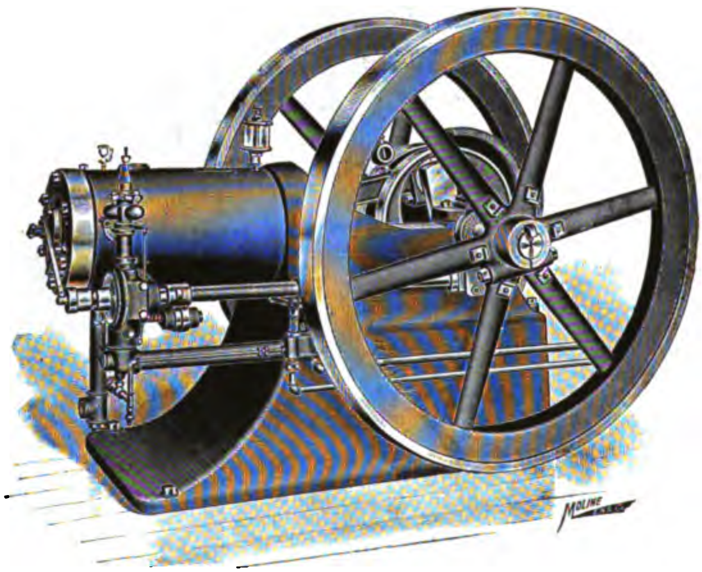


FIG. 218.—The R. & V. horizontal gasoline-engine.

at the mixer, assuring a uniform charge. The governor of the vertical engine is of the centrifugal type, with a single weight and arm, adjusted by a spring, making a hit-or-miss charge by holding the exhaust open.

BLAISDELL DOUBLE ACTING ENGINE

In Figs. 221 and 222 are shown the details of the valve gear, valves, and ignition gear of the Blaisdell double-acting four-cycle engines, having two cylinders placed tandem.

The valves are of the poppet type, working vertically, and are held to the seats by means of springs. The inlet-valve, it will be noticed, is located immediately above the exhaust-valve, thus causing the incoming charge to impinge upon it and to pass over the exhaust-valve, thus keeping the temperature comparatively low and rendering it unnecessary to circulate water through the valves. The inlet-valve is placed in a cage, which is readily removable, thus exposing the exhaust-valve, the latter being readily removed through the opening normally filled by the cage. The exhaust-valve chamber is water-jacketed, thus preventing the overheating of its stem and guide. The igniter and valves are operated by means of a cam on the side-shaft, one cam being used to operate both the inlet and exhaust-valves. The igniter mechanism is illustrated in Fig. 222, and represents a special form of make-and-break contact operated by the eccentric. The eccentric-rod rests in a small forked timing lever forming one arm of the rock-shaft, which, however, is not a part of the igniter proper, thus permitting the latter to be removed without disconnecting or otherwise disturbing any other parts. The engine is started with compressed air, one cylinder being operated by air pressure until the other cylinder receives an impulse, after which the engine continues to run on its own fuel.

LARGE EXPLOSIVE MOTORS

The Nurnberg engine has been designed especially for the use of blast-furnace gas, and consequently all the details of construction have been developed with a view to adapting the engine to the perfect utilization of this fuel, as well as coke-oven gas, producer gas, and Mond gas. Thus far the Nurnberg engine has been built in large sizes only, viz., in units ranging from 250 to 3,200 actual horse-power. The engine is of the four-cycle, double-acting type. The operations taking place at each end of each cylinder are on the Otto cycle, hence the results accomplished in each end of the cylinder are the same as in the single-acting Otto engine, and, therefore, each end of the cylinder is provided with three distinct valves. First, the inlet-valve, admitting either air or combustible mixture into the cylinder; second, the gas-valve, regulating the amount and period of gas admission to the cylinder for each impulse; and third, the exhaust-valve.

Fig. 223 is a longitudinal section of the engine, showing the

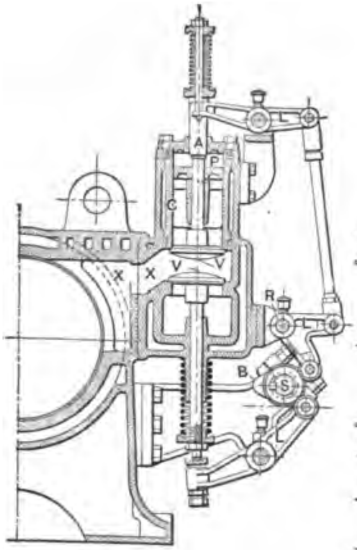


FIG. 221.—Section of valves.

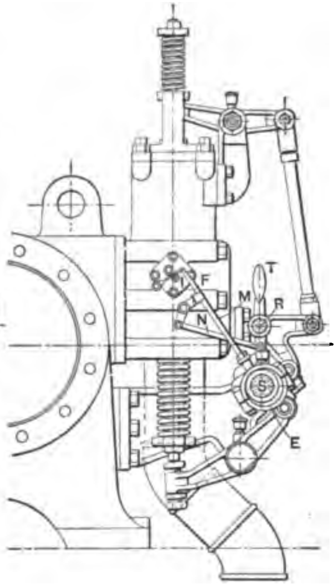


FIG. 222.—Valve gear.

general arrangement of the interior and the location of the valves, while Figs. 224 and 225 are cross sections between the cylinders and through the valve chambers, respectively. The inlet and exhaust-valves are of the usual poppet type, positively operated by a simple form of valve gear, a general view of which is shown in Fig. 226. The inlet-valves open approximately when the crank reaches one dead centre and close approximately when the crank reaches the opposite dead centre. The gas-valve is operated by a governor-controlled mechanism illustrated in Fig. 225. This type of gear is what is known as the "Marx" patent gear, which has proved to be especially well adapted to operating the valves of large-sized gas-engines.

Referring to Fig. 226, the forked rod A is actuated by the eccentric on the lay-shaft, the upper end of A being carried by the swinging link B. To the pin C is pivoted the hook D, which engages the outer end of the rolling lever E, the inner end of which is connected to the gas-valve stem. Lever F is provided with a curved upper edge, upon which lever E rests. One end of the lever F is fulcrumed upon a pin fixed in the valve-bonnet, while the

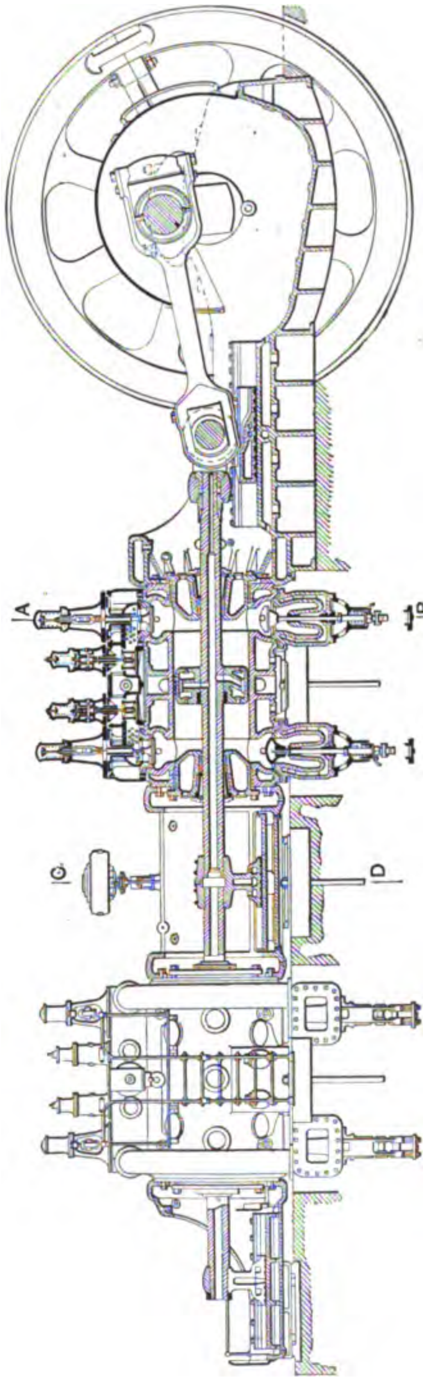


FIG. 223.—Section and view of the two-cylinder, double-acting, four-cycle engine for blast-furnace fuel. Nurnberg type.

outer end is raised and lowered by the arm G, which is actuated by the governor through the arm I. When the outer end of the lever E is drawn downward by the hook D, the rocking motion imparted to E lifts the inner end, and with it the gas-valve, the hook releasing the lever E at the end of the piston stroke. The easy seating of the gas-valve is assured by means of the dash-pot J. It will be seen that as the outer end of the lever F is lowered by the governor, the motion of lever E is modified so that the gas-valve is lifted later in the stroke of the piston. Thus, by varying

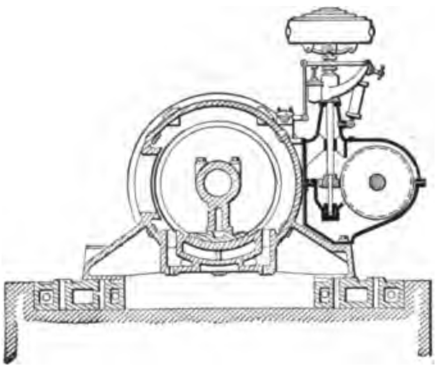


FIG. 224.—Section of piston-rod guide.

the position of the lever F, the opening of the gas-valve can be effected at any point in the stroke according to the power demand and the consequent speed and position of the governor. The gas-valve opens quickly and closes instantaneously, but is prevented from pounding the seat by the dash-pot. The exhaust-valve is opened by a simple rolling lever operated by an eccentric on the lay-shaft as shown. The results obtained by this simple valve gear are the opening and closing of the air and mixing valves, as well as the exhaust-valves, while the crank is close to the dead centres, and the opening of the gas-valve earlier or later in the stroke according to the variations in the load. The retardation of the opening of the gas-valve is accompanied by a proportionate throttling of the gas.

The Westinghouse vertical motor is a model of compactness and is shown in sectional detail in Fig. 227, and as built for natural gas has a usual compression of 120 pounds, with an explosive

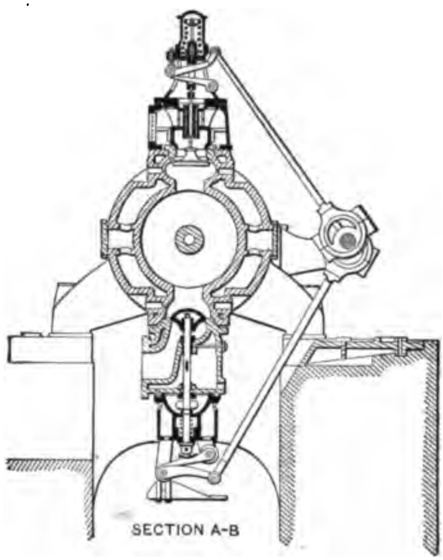


FIG. 225.—Section through valves.

pressure of 350 pounds per square inch, exhausting at 30 pounds at full load, which decreases as the load falls. All valve movements are operated from a single-cam shaft A. One of the features in this design is the location of the admission and exhaust-valves

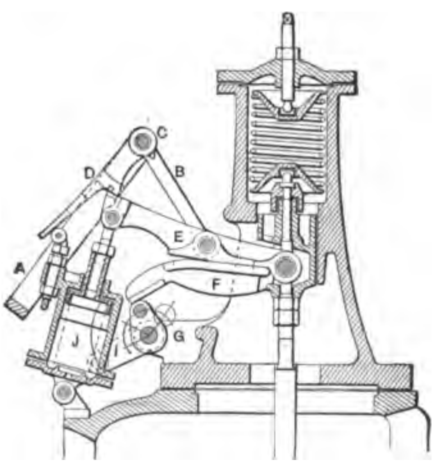


FIG. 226.—Gas-valve mechanism.

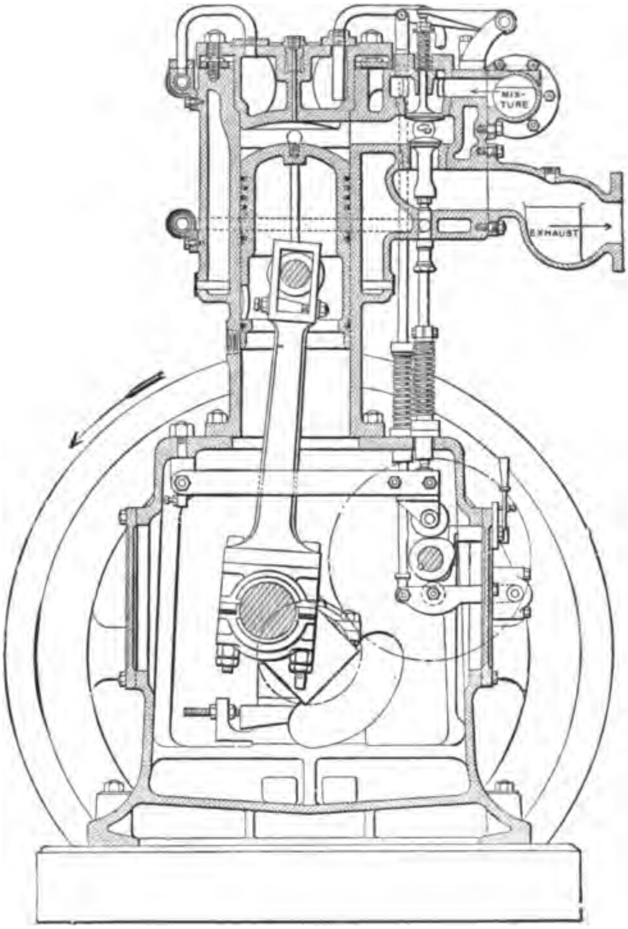


FIG. 227.—Westinghouse Standard vertical gas-engine.

in line, and both operated by push-rods and levers from cams on the shaft A, both valves being held to their seats by springs. The admission-valve B is mounted in a bonnet C, and can be removed without removing other parts. This also allows room for taking out the exhaust-valve and its seat F when required. Duplex hammer-spark ignition is employed and, when convenient, with a direct reduced current from a lighting circuit. A conspicuous feature in this design is the housing of the cranks, trunk-pistons, camshaft, cams, and push-rod rollers; all of which can be quickly got at through movable doors in the box-frame.

In the sectional view (Fig. 228) are shown some of the details of construction of the double and opposite-cylinder engine of the American type of the Crossley engine. Some notable features of this design are the casting of the cylinder, water-jacket, cylinder-head, and exhaust-valve chamber in separate pieces and bolting them together. This allows of the novelty of water-cooling ribs on the cylinder. The water cooling of the piston for large engines

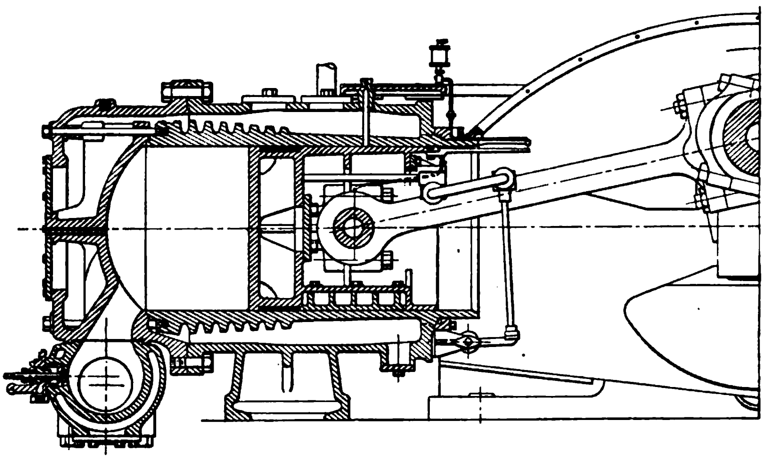


FIG. 228.—Sectional view of one-half of engine.

is accomplished by circulating sections in the piston and a flexible pipe-connection to traverse with the piston. The crankshaft has a centre-crank and the connecting-rods work on one crank-pin, one rod having a single box and the other a forked end with a box in each fork. In Fig. 229 are shown the details of a double engine working upon a single crank and pin, one rod having a single box and the other a forked end with two boxes.

In Fig. 230 is shown a section of a water-cooled balanced exhaust-valve used on the American Crossley engine. It is of the poppet type and the relation between the valve and its seat is the same as in the ordinary mushroom form of valve. An oscillating arm, receiving motion from the cam on the secondary shaft, operates the valve. The valve and stem are hollow and

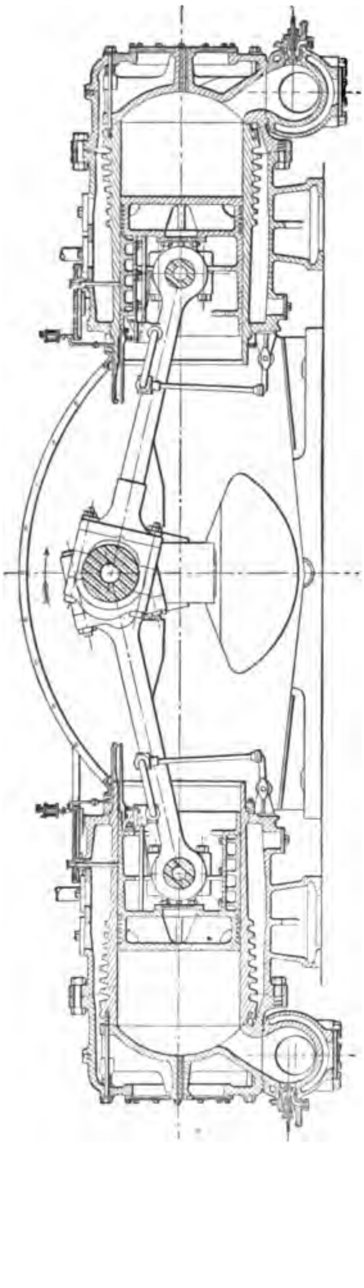


Fig. 229.—Section of the opposed two-cylinder American Crossley gas-engine, as built by the Power and Mining Machinery Company, New York City. In sizes from 150 to 650 horse-power.

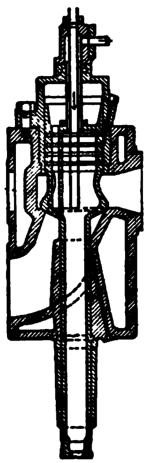


FIG. 230.—Water-cooled balanced exhaust valve.

water for the purpose of internal cooling is conveyed through the pipe shown at the top in the cut. The water escapes around this pipe through a second pipe, the direction being indicated by arrows. This valve has the unusual advantage of travelling in double guides, one on each side of the exhaust, which prevents the pressure from within from throwing it out of alignment with the seat. Oil-ducts, for the purpose of lubricating the guides of the valve-stem and valve-shell, are shown in the cut. The exhaust-valve chamber is a separate piece, bolted to the under side of the cylinder, and can be taken off without interfering with any other working parts of the engine.

In Fig. 231 are shown the vaporizer and water-cooled valve chambers of the new Crossley oil-engine. It is essentially a kerosene and distillate-oil engine, but a claim is made that crude oil may be equally useful as explosive fuel. It will be noticed in this design that an air-shifting valve makes a water-spray into the vaporizer near to the oil-spray inlet, making an explosive compound of oil, air, and water-atoms to be ignited by compression and an igniter-tube projecting within the vaporizer, as shown in the small cross section. The outside ribs on the vaporizer facilitate the heating of the chamber when starting and are also for regulating the temperature while the motor is running. The water element in this combination of explosive fuel allows of excessive compression without preignition, otherwise possible.

In Fig. 232 is shown a vertical section of the Walrath three-cylinder engine of the four-cycle type, and in Figs. 223 and 224 a plan of the cylinder-head and valve-levers and a vertical section of the water-cooled exhaust-valve as applied to the larger engines of 50 horse-power. The general style of construction is shown in Fig. 232, which gives a cross-sectional view of the engines with cylinders 12 $\times$ 12 inches or smaller. The base, cast in one piece, is bored to receive the cylinders, crank, and camshaft bearings The main bearings, being a separate casting made to fit a corresponding circular bore in the base, can readily be removed without disturbing the crankshaft. The cylinders are bolted on the top of the base, fitting into the bore made to receive them, as shown.

The valves, of the poppet type, are two in number, one serving as the inlet for the explosive mixture and the other acting as the exhaust-valve. In all engines of over 10 horse-power the valves are placed in cages which fit into the cylinder-head. By having the joint between the cages and the head ground, it is the work of but a few minutes to remove either valve when desired. In the larger engines a special water-cooled valve, illustrated in Fig. 234, is employed. The valves are operated by a camshaft revolving at just one-half the speed of the crankshaft. This is

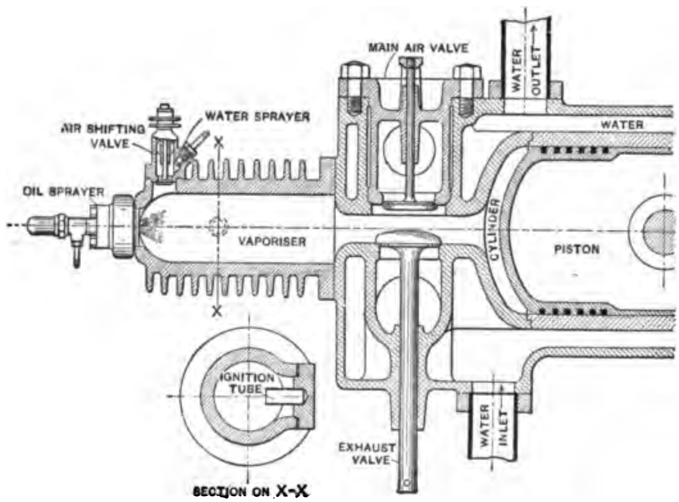


FIG. 231.—Section of vaporizer and valve chamber. New Crossley.

accomplished by a train of three spur gears, which, with those used to drive the governor, are the only gears used on the engine. This camshaft operates both the valves and the igniter for all of the cylinders.

The pistons are extremely long to give enough surface to reduce the wear on the cylinder and pistons to a minimum. This is a vital point in cases where the piston must perform the additional services of a cross-head, for when short, undue wear will result, giving necessity for extensive repairs and large repair bills. To reduce the friction and wear on the pistons from the angularity of short connecting-rods they are all made three strokes in length.

The boxes at both ends are of bronze, while the rod itself is of forged steel.

The igniter is of the break-type and consists of a casing holding two electrodes, one of which is stationary and insulated from the main body of the casting. The other electrode is movable and operated by a cam, which causes it to make and break contact with the insulated electrode. The contact points are composed of a special metal, which is adapted to withstand great heat.

The governor is of the fly-ball type, driven by means of bevel gears. It operates a piston-valve which regulates the amount of explosive mixture required for each impulse to maintain a steady speed under all conditions and variations of load. This method of governing gives an impulse every second revolution for the one-cylinder type, every revolution in the two-cylinder, and every

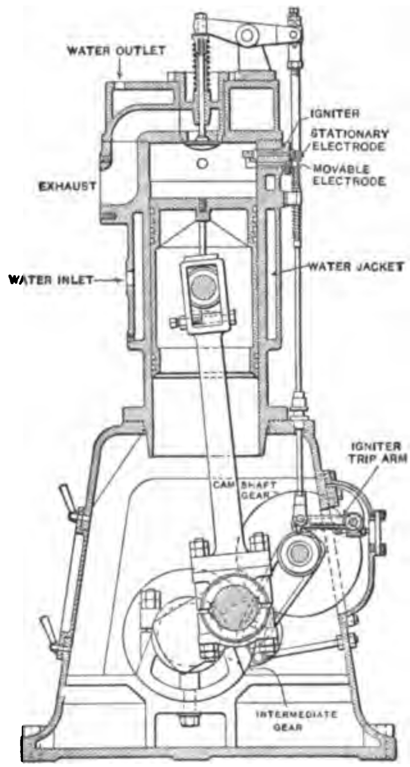


FIG. 232.—Cross section of vertical engine.

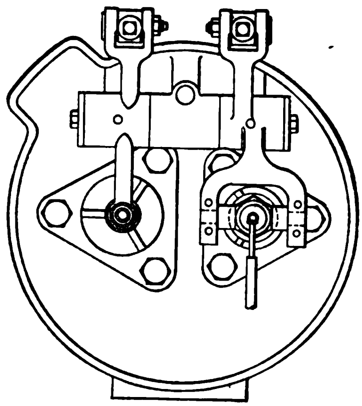


FIG. 233.—Cylinder-head and valve levers.

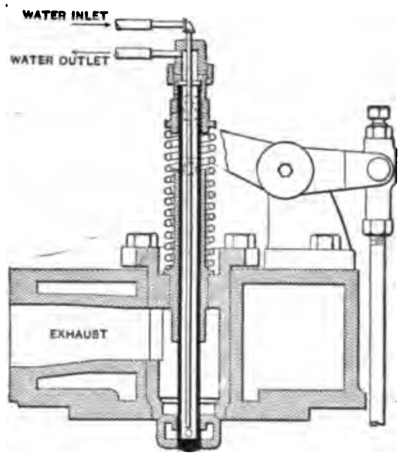


FIG. 234.—Water-cooled exhaust-valve.

two-thirds of a revolution in the three-cylinder type, no matter what the load may be.

A starting device is provided upon all engines above 20 horse-power, and can be supplied on the smaller sizes. An air-pump, generally driven by a small pulley on the engine crankshaft, charges a storage tank with air at a pressure of from 100 to 200 pounds. A starter-lever of the piston type, operated by a cam, admits the air above the piston, which moves downward. The valve then opens communication between the engine-cylinder and the atmosphere, which causes the air to be exhausted. The engine goes through a series of such operations until an explosion of the gases takes place.

In Fig. 235 we illustrate the large double-acting gas-engine of the Westinghouse Company. In construction the engine embodies many established features of modern steam-engine practice. From crank to cylinders the construction is that of a horizontal steam-engine suitably strengthened in proportion to the increased maximum pressure due to the explosion of the charge. The design of cylinders, pistons, and valves, of course, departs materially from steam-engine practice. The cylinders are double-walled, with the outer walls split peripherally to permit independent expansion and contraction without placing the cylinder-casting under stress. The many difficulties arising in providing a suitable packing-

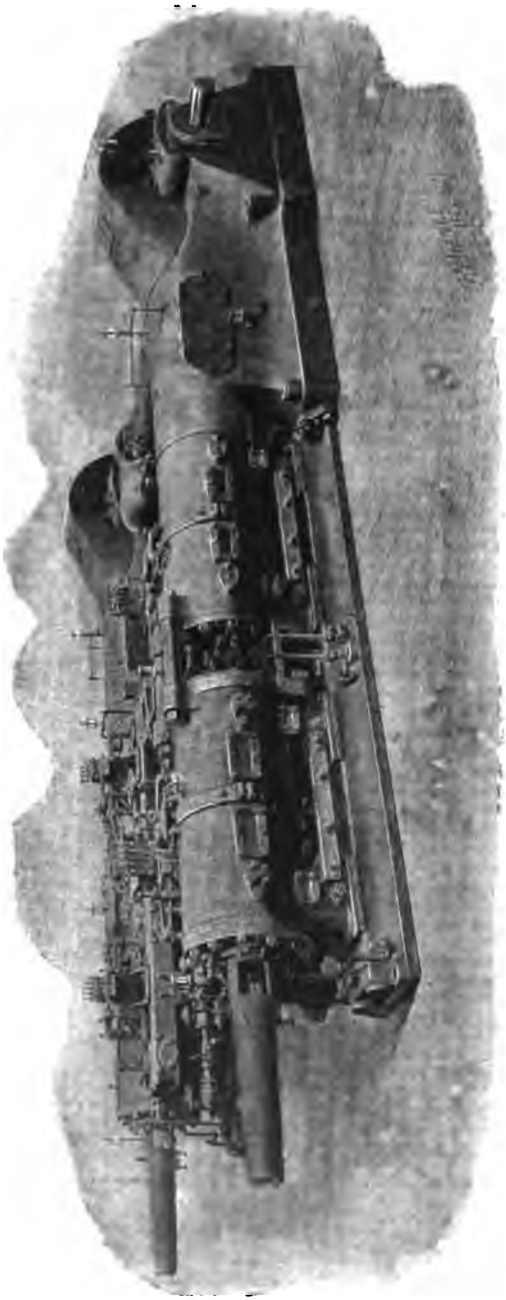


FIG. 235.—The Westinghouse four-cylinder, double-acting gas-engine.

Four-cycle type, cranks 90° apart, giving four impulses per revolution, in units from 400 to 2,000 horse-power. The pistons, piston-rods, and exhaust-valves are all water cooled. One-lay shaft, parallel with and running between the cylinders, operates through independent cams the inlet and exhaust valves and the igniters, so that the action of the valves and igniters may be timed in order to secure the best results.

gland for the cylinder-heads have been overcome by means of a simple metallic packing similar in some respects to that used on high-pressure steam-engines.

Both valves are of the single-beat poppet type and seat vertically along the same axis, the admission-valve opening downward and the exhaust upward. The admission-valve is mounted in a separate bonnet which, together with the valve, may be readily removed without dismantling any parts of the engine other than the tappet-lever through which the cam motion is imparted to the valve. Both admission and exhaust-valves are of steel and are held to their seats by spiral springs. The exhaust-valve is water cooled.

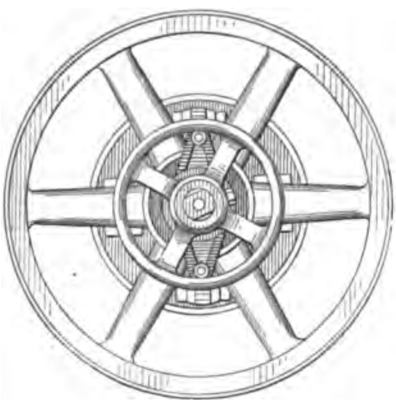


FIG. 236.—Front view of clutch.

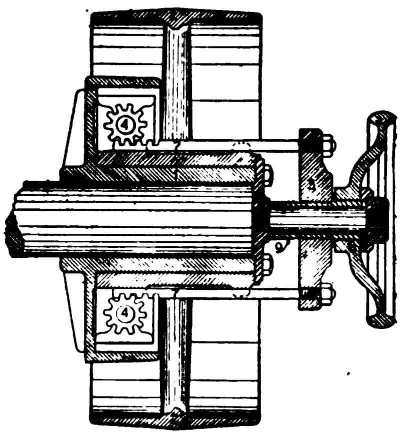
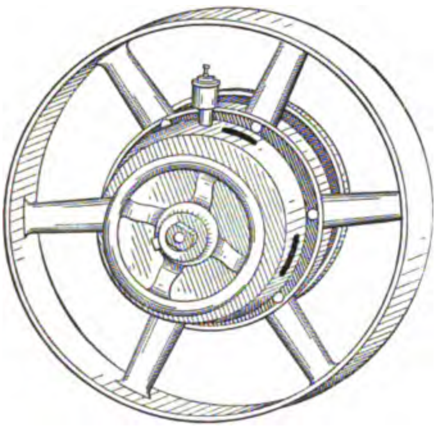


FIG. 237.—Section of clutch.

It is bored hollow throughout its length, and this canal conveys cooling water to the head of the valve; the water returns in the opposite direction through an inner concentric tube, finally emerging at the lower end. By spraying a small part of the jacket-water into the exhaust-pipes, the temperature of the pipe may be kept at a comfortable point through the absorption of the latent heat of evaporation of the water used.

Both pistons and the piston-rod are water cooled, as well as other parts subjected to internal heat. Means for introducing the cooling water is secured by a telescopic-pipe connection bolted to the inside of the cross-head guide. The inner tube of this telescopic joint is attached to the cross-head at such a point as to convey the cooling water to the end of the piston-rod bore, whence it proceeds in succession through the two pistons, emerging through a bronze tail-rod extending through the rear cylinder-head. Each piston is a one-piece casting, cored hollow to accommodate the circulating water, and packed by cast-iron packing-rings set out with flat, steel springs. In order to convey the water in and out of the piston, deflecting plugs are inserted at the proper points in the rod-bore. A cast-iron jacket surrounds the tail-rod and receives the water emerging from it, whence it is drained away.

The one-lay shaft paralleling the cylinders operates, through cams, all of the valve movements of the engine. Independent



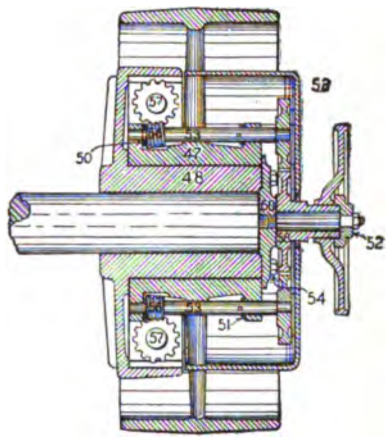


FIG. 238.—View, worm-gear clutch. FIG. 239.—Section, worm-gear clutch.

cams are provided for inlet-valves, exhaust-valves, and igniters, so that the action of each valve may be timed in order to secure the best results. The main cams are all of cast iron with working surfaces chilled and ground. The engine is started by compressed air, and for this purpose a special disengaging gear is provided which isolates the rear cylinder, and on admitting the compressed air allows the cylinder to operate as an air-motor until the regular combustion cycle is taken up in the forward cylinder; the rear cylinder may then be thrown into normal action.

THE EXPLOSIVE-MOTOR CLUTCH

The clutch for facilitating the starting of explosive motors has become a most essential adjunct of every motor plant. The later designs are automatic in their action, and when once closed with the driven machinery increase their frictional resistance by automatic closure. The creeping of clutches, with its consequent loss of power and wear due to the impulse operation of the explosive motor, has been overcome, and creeping is automatically arrested by increase of frictional pressure.

In Fig. 236 we illustrate a front view and in Fig. 237 a section of a pulley or gear-clutch of the Carruthers-Fithian type, as used on motors of from 5 to 35 horse-power. The hand-wheel 8 locks the screw-sleeve 9 by pushing and turning the wheel in the direction that the motor is running, which pushes the cross-head 3 and the rack-bars in, revolving the gears 4 on right and left screws, which throw out the friction-shoes to contact with the friction rim. Then drawing the hand-wheel back locks the wheel in the dentals of the nut and screw-sleeve, when the motion of the motor tightens up the friction automatically. In Figs. 238 and 239 we illustrate their worm-gear clutch for the larger motors of from 40 to 150 horse-power. The operation of throwing the clutch in is much the same as with the smaller clutch, only that the transmission is through three spur-gears and worm-gears on the right and left screws, which operate the friction-shoes with great power.

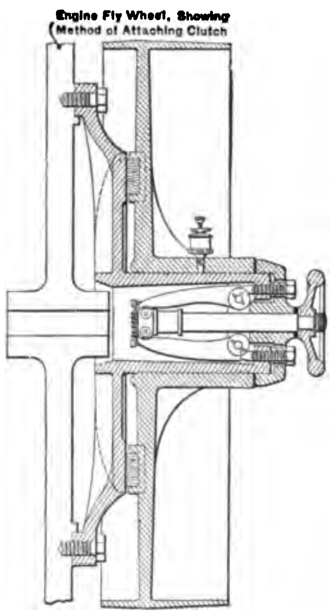


FIG. 240.—Gas-engine clutch.

In Fig. 240 is a section of the B and C gas-engine clutch, which consists of three main parts: the pulley, the carrier, which is bolted to the arms of the engine fly-wheel and acts as a journal of the pulley, and the gripping mechanism; which consists of a gripping plate, spindle, and cam-levers. The clutch has a side-grip which eliminates the effect of centrifugal force and insures a positive release. Two rollers are mounted on the end of the spindle, which works in and out through a hole in the gripping plate, and journaled on the end is the operating hand-wheel, which can be held in the hand regardless of the speed of the engine. Bearing on the rollers are cam-levers, which in turn are pivoted on the gripping plate, and lugs on the levers abut against the adjusting screws. These adjusting screws go through a flange on the carrier, and are locked in place by lock-nuts, which also hold the gripping plate in position. In the operation of the clutch, when the spindle is pulled out against the stop, the pulley is free to turn on the carrier-journal and when pushed in is gripped in a circular vise and turns with the engine flywheel. The load can be taken up as gradually as desired by pushing in the hand-wheel slowly, and released at will by pulling it out.

CHAPTER XVI

MARINE MOTORS

THE explosive motor has of late acquired a success in its application for marine power, in which its use has developed a marvellous speed in small craft that has outstripped anything heretofore accomplished by steam-power. Racing launches and yachts are passing the 40-mile mark, and their speed limit may be far beyond our earlier dreams; all due to the new element of power. For the accomplishment of this ideal purpose a marine motor must be as compact and light in weight (compatible with strength) as possible, and should be so designed that any part can be adjusted, taken out, or renewed without disturbing anything else, for the quarters in which engines of this type are placed are oftentimes cramped and dark; and accessibility, after reliability, is a prime necessity. When these points are given proper consideration in the design and construction of marine motors, far greater success and pleasure will attend their use than has been experienced in the past. Yet the era of advancement during the past decade has had its salient points of interest and pleasure in sailing speed, and the present designs of marine motors are fast approaching the perfection of action and convenience of management so desirable in the motor service for pleasure craft.

MARINE ENGINES AND THEIR WORK

The oft-repeated inquiry as to the proper size of motor and wheel for certain-sized boats has induced the author to gather, in the following table, the leading points for moderate-speed boats, as derived from a leading yacht and launch motor-boat concern. The conditions are much too high for auxiliary power for sailing craft, and too low for racing craft, which in all cases requires special design of boat lines and allotment of power as well as of size and pitch of screw. The approximate speed of launches and larger boats as scheduled may be obtained by deducting from 20 to 25 per cent. of the product of the revolutions per minute and the pitch of the wheel in feet and decimals which gives the speed in feet per minute. Multiply this product by 60 and divide by 5280 for the miles per hour, or divide the first product by 88, which is $\frac{5280}{60}$, a shorter way.

APPROXIMATE SIZES OF ENGINES, PROPELLERS AND BOATS.

Size.	Cylinder.		Revolutions.	Propeller-Wheel.		Launch or Boat.	
	Diam.	Stroke.		Diam.	Pitch.	Length.	Beam.
3 H. P. Single-cylinder....	5 in.	7 in.	480	16 in.	24 in.	18 ft.	5 ft.
4 " " "	5½ in.	7 in.	450	18 in.	26 in.	25 ft.	6 ft.
5 " " "	5½ in.	9 in.	425	20 in.	28 in.	28 ft.	6½ ft.
6 " " "	6½ in.	9 in.	400	21 in.	28 in.	30 ft.	7 ft.
6 " Two-cylinder......	5 in.	7 in.	475	18 in.	26 in.	30 ft.	7 ft.
8 " " "	5½ in.	7 in.	400	23 in.	32 in.	32 ft.	7½ ft.
10 " " "	6½ in.	9 in.	410	26 in.	34 in.	35 ft.	8 ft.
16 " " "	7½ in.	11 in.	325	30 in.	38 in.	40 ft.	8½ ft.
25 " " "	9 in.	13 in.	300	34 in.	48 in.	45 ft.	9 ft.
16 " Three-cylinder....	6½ in.	8 in.	380	28 in.	38 in.	40 ft.	8½ ft.
16 " Four-cylinder.....	5½ in.	7 in.	375	28 in.	35 in.	40 ft.	8½ ft.
20 " " "	6½ in.	9 in.	360	30 in.	40 in.	42 ft.	8½ ft.
32 " " "	7½ in.	11 in.	330	36 in.	48 in.	48 ft.	9½ ft.
50 " " "	9 in.	13 in.	300	40 in.	54 in.	50 ft.	10 ft.

TYPICAL MARINE MOTORS

The motors of the Bridgeport Motor Company, Bridgeport, Conn., are of the marine and stationary two-cycle type and are of compact and simple design. The ignition by hammer break-spark and the circulating-pump are both operated by the pump-rod from a cam on the motor-shaft, the igniter being a separate rod lifted by a trip-block on the pump-rod and let go by contact with an adjusting timing-screw. The gasoline is fed to the crank-chamber by an atomizing carburetor with an adjusting needle-valve opening. The feed to the cylinder is regulated by a revolving perforated damper as shown in the drawings (Figs. 241 and 242), which are to a scale for small-sized motors. By this adjustment the charge-mixture is regulated in its proportions in the crank-case, and the quantity of each charge is also regulated for the speed of the motor. Views of the two- and three-cylinder types are shown at Fig. 243. These are the latest three-port type, whereas the simple form previously illustrated is a two-port pattern.

The Bridegport motor runs equally well in either direction, dispensing with the necessity for a reverse clutch or reversing

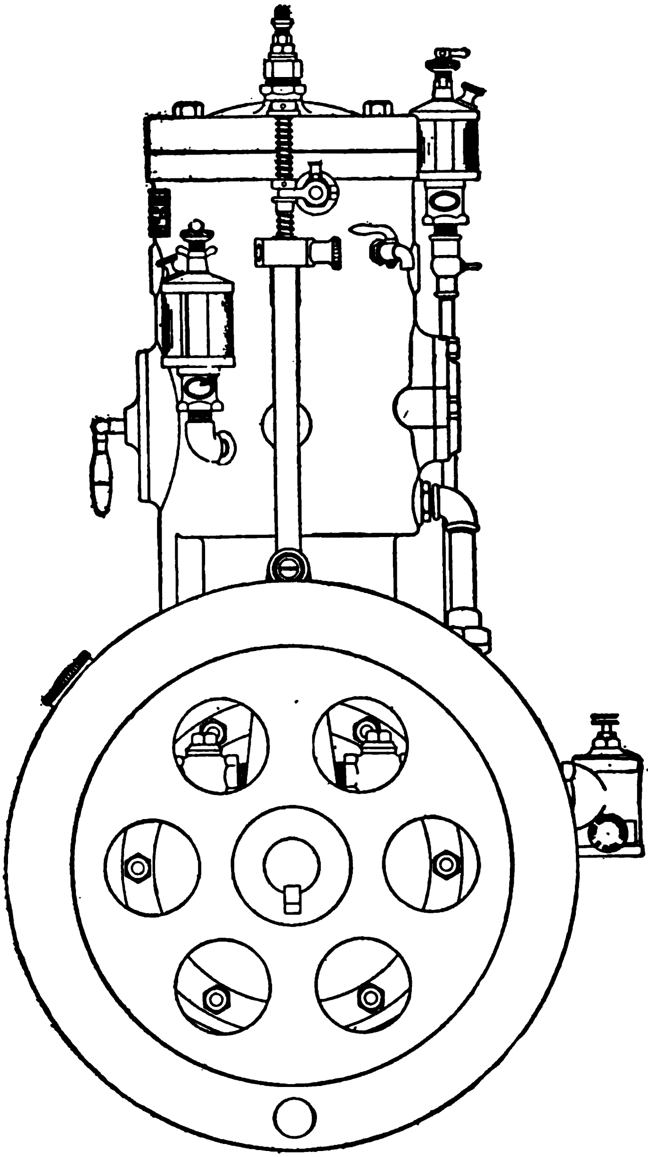


FIG. 241.—Front view, Bridgeport motor.

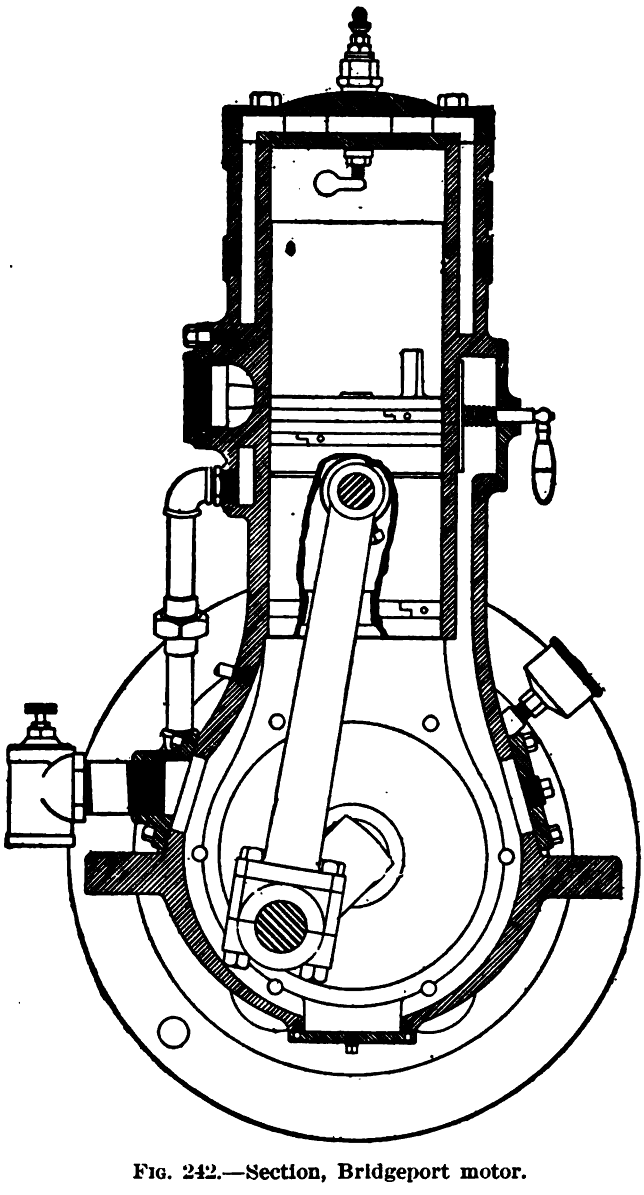


FIG. 242.—Section, Bridgeport motor.

propeller, except in the larger sizes. With a solid propeller-wheel, in any size up to six and one-half horse-power, if it is desired to reverse, the switch is thrown off as in stopping engine, and when the engine flywheel is near to last revolution and nearly on centre, switched on again, and engine is thus reversed without stopping.

SPECIFICATIONS OF BRIDGEPORT MARINE GASOLINE-ENGINES.

Engines	1½ H. P.	2½ H. P.	3½ H. P.	5½ H. P.	6½ H. P.	8 H. P.	12 H. P.	20 H. P.
Cylinders, number	1	1	1	1	1	1	2	3
Bore, inches	3¼	3¾	4½	5¼	5½	6½	5½	5½
Stroke, inches	3½	4	5	5½	6½	6½	6½	6½
Revolutions per minute	600	500	475	450	425	400	400	400
Diameter balance-wheel, inches	12	13	15	17	18½	18½	22	22
Diameter engine shaft, inches	1	1⅛	1¼	1¾	1¾	1¾	2	2
Size of base, inches	7½x10½	8x12	9x13½	12x16½	13x18	13x18	19½x26	19½x38
Height of engine shaft line, inches	12½	14	16½	21½	23½	23¾	24	24
Weight, pounds	125	170	210	415	485	575	1,000	1,300
Diameter propeller-shaft, inches	⅝	¾	1	1¼	1¼	1¼	1½	1½
Diameter propeller-wheel, inches	12	14-16	15-18	16-20	18-22	20-24	22-26	24-26

The above dimensions are given for the study of all desiring to fit up a launch. The following are the boat dimensions suitable for the horse-powers in the above table of motor dimensions.

Dimensions of Stock Sizes.	Standard Models.					Comfort Models.	
Length, over all	18 ft.	22 ft.	25 ft.	28 ft.	30 ft.	17 ft.	22 ft.
Beam, extreme	5 ft.	6 ft.	7 ft.	7 ft.	7 ft.	7 ft.	7 ft.
Depth, least	25 in.	27 in.	30 in.	30 in.	36 in.	24 in.	27 in.
Draught	20 in.	22 in.	27 in.	27 in.	30 in.	20 in.	22 in.
Engine, horse-power	2½	3½	6½	6½	6½	3½	6½

In Fig. 244 is shown a section of the Lozier two-cycle marine motor. The principal features are the throttle-valve to regulate the charge from the crank chamber and the operation of the hammer spark-break from a cam on the shaft. A rotary circulating pump is driven by chain from the main shaft and the discharge of the water from the cylinder is around the exhaust-pipe. The thrust is taken by ball-bearings in the cam-hub. A throttle-valve in the passage from the crank chamber to the cylinder, with an index handle, regulates the charge. The starting handle is located within the rim of the fly-wheel and held by a light spring. To start the motor the handle is pulled out and flies back the moment

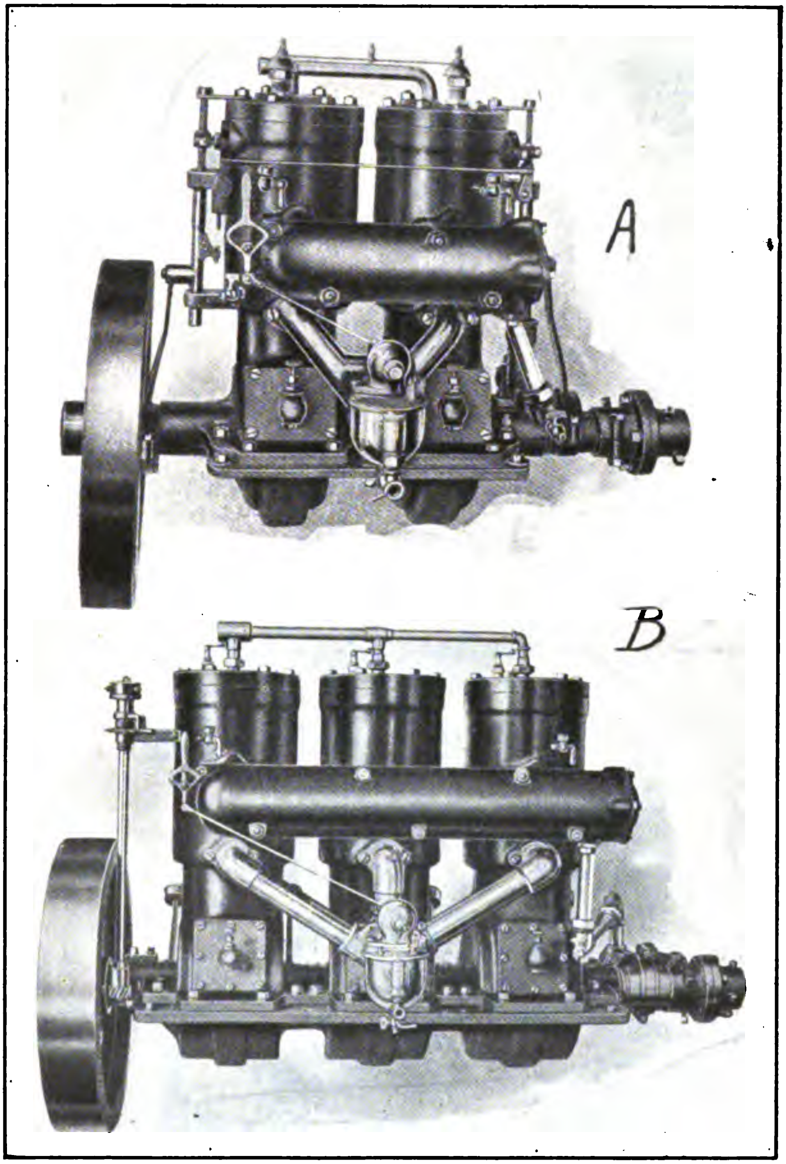


Fig. 243.—Two and three-cylinder Bridgeport motors when viewed from the carburetor side.

the motor starts by its own impulse, thus saving much annoyance from starting crank-wrenches.

We illustrate in Fig. 245 a section of the Hubbard two-cycle vertical motor. Its characteristics of construction are similar to the general type of this class of motors. Its movement is simple, complete, with the ignition device driven by a single push-rod connected to a cam-rod and which also carries the plunger of the circulating-pump. In upper right-hand corner of the cut is shown the quick-acting spark-break device. The action of the spark is very simple and easily understood. The slide S, which carries both the plunger of the pump P and the spark-trigger T, is moved by an eccentric on the flywheel, so that it is at the top of its stroke simultaneously with the piston. When it nears the top, T strikes plunger H and lifts it against spring U, allowing the inside spark-lever R and outside spark-lever K, which are firmly pinned together, to be pressed upward by spring U till

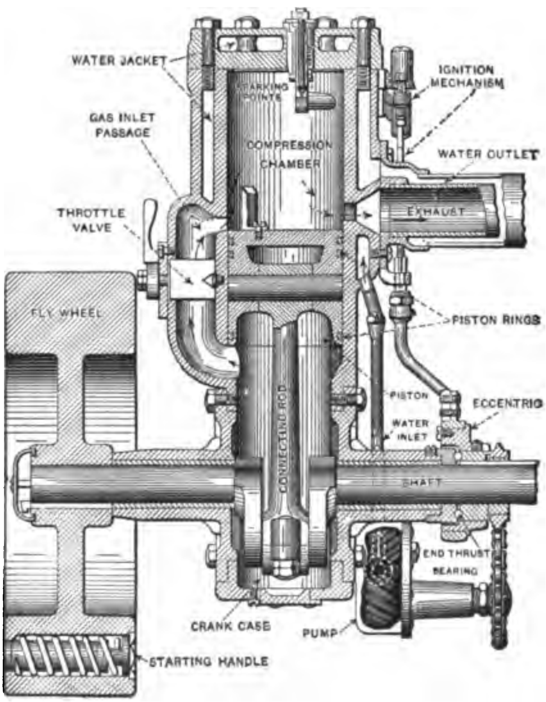


FIG. 244.—Lozier gasoline-motor.

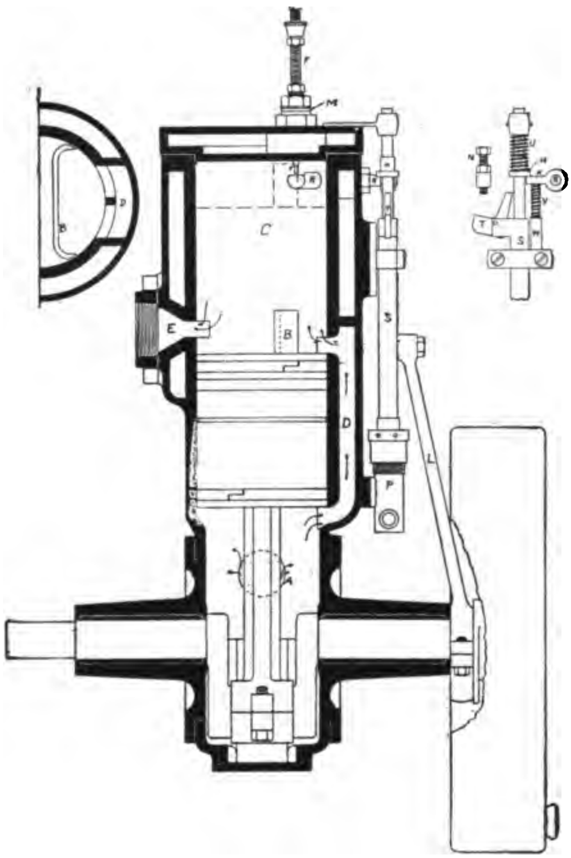


FIG. 245.—Section of the Hubbard motor.

R touches F. Then T strikes screw N, causing H to be released and strike K sharply, thus snapping R quickly away from F and making a bright spark. In order to advance the spark, N is screwed down, and to retard it, screwed up.

During the up-stroke of the piston a mixture of air and gasoline is drawn from the mixing-valve through the opening A into the tightly enclosed crank-chamber. At the beginning of the downstroke the mixing-valve is automatically closed, and when the piston passes the inlet-port D the mixture in the crank-chamber is sufficiently compressed so that it rushes through port D into the cylinder, where it is deflected upward by the baffle-plate B, and forces out any remaining burnt gases through the exhaust-port E. When the piston goes up again the charge is compressed into the space above the dotted outline of the top of the piston and fired by a spark between firing-pin F and inside spark-lever R. This makes a pressure of about 300 pounds per square inch, which drives the piston down on its lower stroke, at the end of which the charge is exhausted through E when that port is uncovered by the piston.

THE STANDARD ENGINES

We illustrate in Fig. 247 the six-cylinder Standard marine

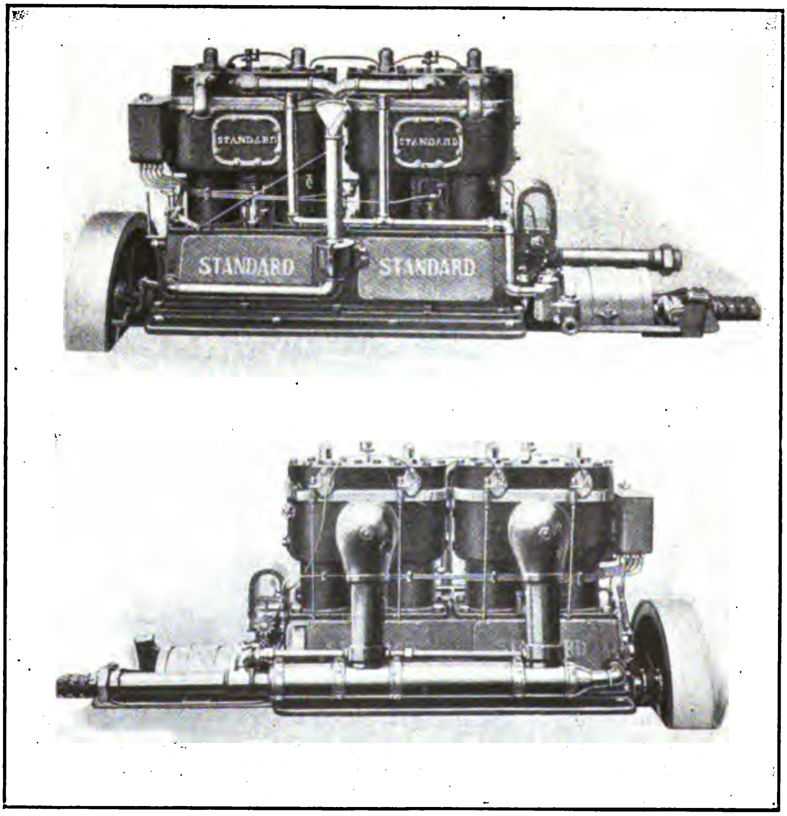


Fig. 246.—Inlet and exhaust side of the Standard four-cylinder marine motor.

motor. The cylinders are 8 inches in diameter, 10-inch stroke, and the motor runs 600 revolutions per minute, driving a propeller 36 inches in diameter. The "Standard" is of the four-cycle type and reversed by shifting the valve motion; receives the explosive fuel through a single atomizing vaporizer, with a controlling-valve and index. The motor is started by compressed air, and, having no dead centres, instantly starts on opening the compressed-air valve. A small air-pump keeps an air-tank at sufficient pressure for starting several times without continuous running. Another type of Standard engine which has received wide application mounted on a substantial base with a reverse gearing integral is

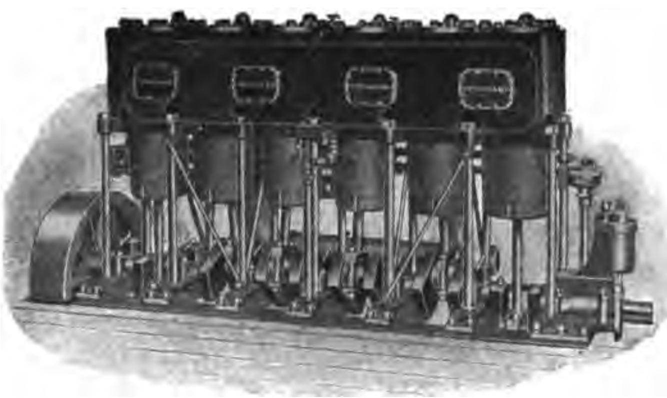


FIG. 247.—The "Standard" 100 horse-power motor.

shown at Fig. 246. While in the engine previously shown the crankshaft is exposed as are the connecting rods and other interior parts, that outlined at Fig. 246 has the entire interior mechanism incased in the engine base. The four-cylinder engine shown is made in three sizes, 20-24 and 32-37 and 65-75 H. P. The complete data relating to the important dimensions of this type of engine is clearly outlined at Fig. 249 while that of the compressed air starting and reversing types is presented at Fig. 248.

The auto marine type of Standard engine is especially designed as a light, powerful, high speed motor that is intended for use in racing boats and various forms of pleasure craft where speed is a desired factor. The reversing form of engine offers many

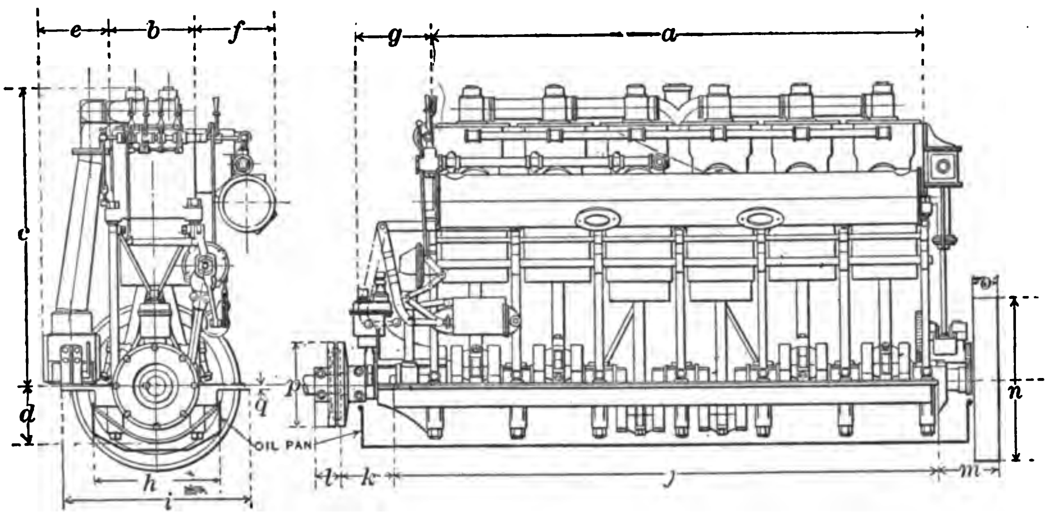


TABLE OF DIMENSIONS FOR STOCK ENGINES
Compressed Air Starting and Reversing Type

Horse Power	No of Cyls.	Weight lbs	Revs. per Min	Diam. Prop'l'r Shaft	Diam Prop'l'r	A ins.	B ins.	C ins.	D ins	E ins.	F ins.	G ins.	H ins.	I ins.	J ins.	K ins.	L ins.	M ins.	N ins.	O ins	P ins.	Q ins.
100 Sing. Act.	6	5,800	350	3	36	77 1/2	12 3/4	46 3/4	9	13 3/4	11	23 1/2	21	28	81 3/4	13 3/4	4	11 3/4	24	5	12	3/4
300 Sing. Act.	6	9,500	300	3 1/2	44	96	16	58	11	....	15 1/2	15	30	41	105 3/4	10 1/2	5	12 3/4	32	5	16 1/4	6 3/4
300 Do. Act.	6	8,000	300	3 3/4	......	88 3/4	27 1/2	57 1/2	9	9 1/4	8 3/4	7 1/4	22 3/4	32	93 3/4	13	5	7	No Fly Wl.		16 3/4	1/2
500 Do. Act.	6	1,600	300	4 1/2	60	143	36 1/2	76	14 1/2	12 1/2	9 3/4	10 1/2	33	46	118 1/2	18	7 1/2	7 1/4	No Fly Wl.		24	Flush

FIG. 248.—Table of dimensions of Standard reversing type marine engines.

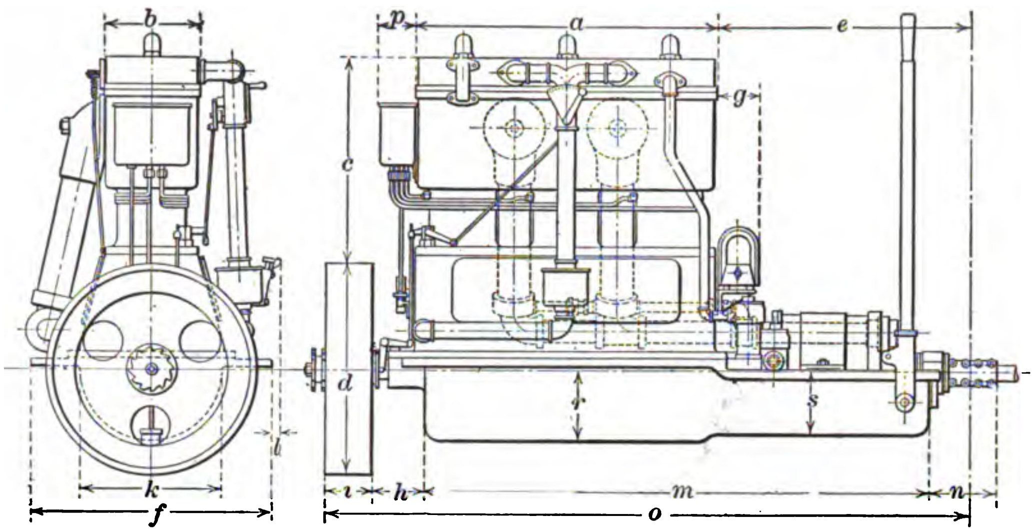


TABLE OF DIMENSIONS FOR STOCK ENGINES
Reverse Gear Type

Horse Power	No. of Cyl's	Weight lbs	Revs per minute	Diam. Propeller Shaft	Propeller Diam.	A ins.	B ins.	C ins.	D ins.	E ins.	F ins.	G ins.	H ins.	I ins.	K ins.	L ins.	M ins.	N ins.	O ins.	P ins.	R ins.	S ins.
10–12	2	850	400	1¼	24	18¼	8¾	17⅛	19½	24½	21	5⅜	5⅛	4½	12	..	37½	6¼	59¼	4⅛	6	5⅜
16–18	2	1,200	350	1⅜	28	20⅝	9⅞	21¾	22	25¾	25	5⅜	5	5	15	¾	41¼	6¾	55½	4⅛	7½	6¼
20–24	4	1,600	400	1⅜	26	37½	8¾	16⅜	21	24¾	21	5⅜	5	4½	12	..	56½	6¾	70⅜	4⅛	6	5⅜
24–27	3	1,800	350	1⅜	30	31⅝	9⅞	21¾	22	27	25	5⅜	5	5	15¼	¾	53⅛	7¼	66⅝	4⅛	7⅝	7⅛
25 Auto-Mar	4	500	900	1⅛	18	23½	9	10⅛	18	19¾	13⅛	5¾	1½	2½	10⅛	...	40¾	4⅞	48	4⅛	4¾	3¾
32–37	4	2,800	350	1⅞	32	42⅛	9⅞	20 3/16	25	30 1/16	25⅜	5⅜	5	5	15½	¾	65⅜	9⅛	81½	4⅛	7¾	7¾
50–54	6	3,200	350	1⅞	34	63⅝	9⅞	20¼	25	30 3/16	25⅜	5⅜	5	5	15½	¾	86⅞	9⅛	103	4⅛	7¾	7¾
55–75	4	5,300	350	2⅛	36	56¾	12¾	25¼	30	41¼	33	5⅜	6⅛	6	20¾	...	89¼	12½	109¾	4⅛	10⅜	10⅜
60 Auto-Mar.	6	1,500	700	1⅜	24	53⅛	13½	15⅛	18	27⅛	20¼	4⅛	1⅜	3¼	15	5¾	77¼	5⅞	85½	4⅛	6⅜	5⅞
90–100	6	8,000	350	2⅛	44	86½	12¾	25¼	30	41¼	33	5⅜	6¾	6	20¾	..	118¾	12½	139½	4⅛	10⅜	10⅜

FIG. 249.—Table of dimensions of Standard reverse gear type marine engines.

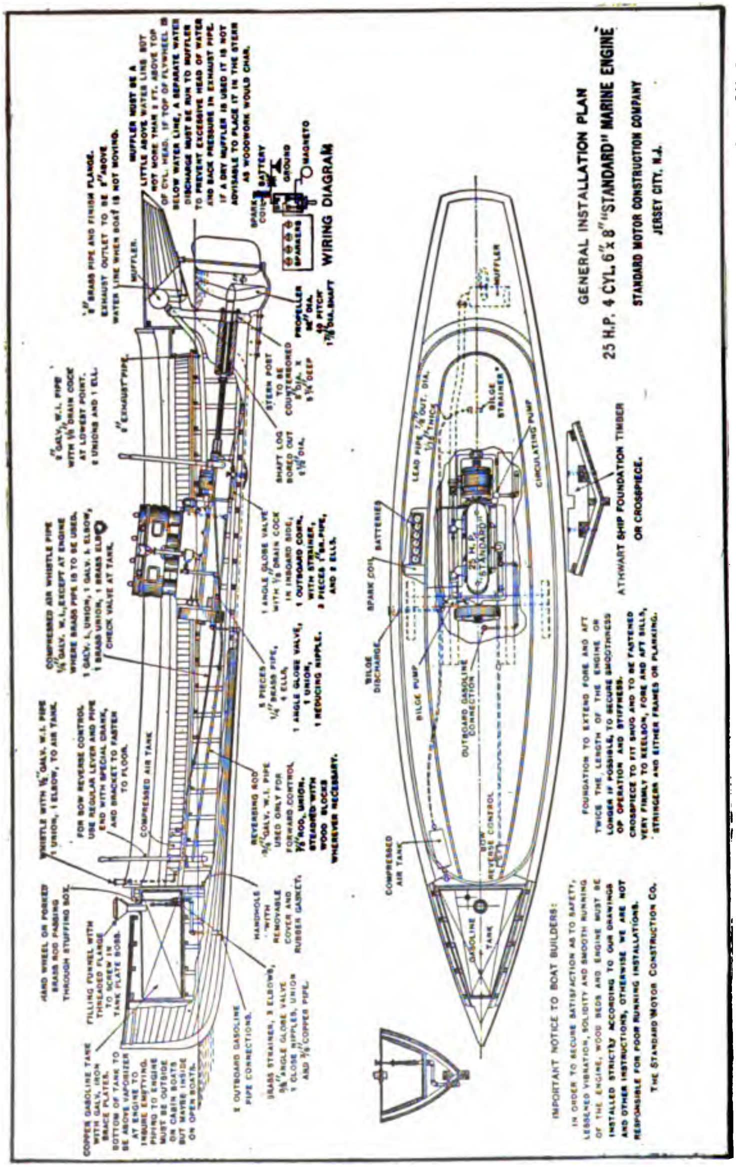


FIG. 250.—General installation plan of Standard marine engines, showing the application of a 25 horse-power marine engine to an open launch.

features that are really valuable in power plants for large craft. Heretofore the marine gas-engine was not considered as practical in the larger powers as it was in the smaller ones but the development of the six-cylinder reversible balanced marine engine, which can be furnished in sizes up to 2000 H. P. has changed this condition. With the Standard engine, starting and reversing operations are as positive as with steam, as the engine is readily controlled by altering the valve motion from ahead to astern. A positive initial start in either direction is given to the engine by the opening of a compressed-air throttle, as there are no dead centres, the working cylinders instantly take hold and the engine functions in its normal manner. Sufficient compressed air is stored up in the steel tank by a small air pump on the engine to start thirty or forty times without repumping. The engines employ mechanical lubrication, water-cooled exhaust pipes, large water-feed pipes, balanced, water-cooled valves and extremely large bearings. They are not only practically vibrationless but are quiet running and are an ideal marine power plant. The method of installation of the 25 H. P. four-cylinder 6″ x 8″ marine engine in an open launch is clearly outlined at Fig. 250. The diagrams at Figs. 248 and 249 are very valuable inasmuch as all important dimensions necessary for installation are given.

The illustration Fig. 251 represents an oyster sloop 32 ft. long and 11 ft. beam. The boat was not originally intended for power, and is of the type usually found in Long Island Sound, and in Great South Bay, on the south side of Long Island. It is equipped with one 6-horse-power two-cycle gasoline-engine. The engine is connected to one double-drum hoister, built especially for the 6-horse-power engine. It is so designed that both drums are placed on one shaft. This does away with one set of gears and considerable extra machinery, which is necessary when connected in the usual way. This greatly simplifies the hoister, and reduces the cost considerably, but does not decrease efficiency of hoist in any respect. This is a one-man boat, *i.e.*, one man can run the engine, operate the hoist, and steer the boat. The hoister consists of two drums 12 inches in diameter, 10 inches long, and will hold 100 feet of ¼-inch chain; hoisting shaft, 1¾ inches in diameter; tube to cover the shaft; set of side-rollers, friction-clutch, all necessary axle-boxes, levers, fittings, etc.

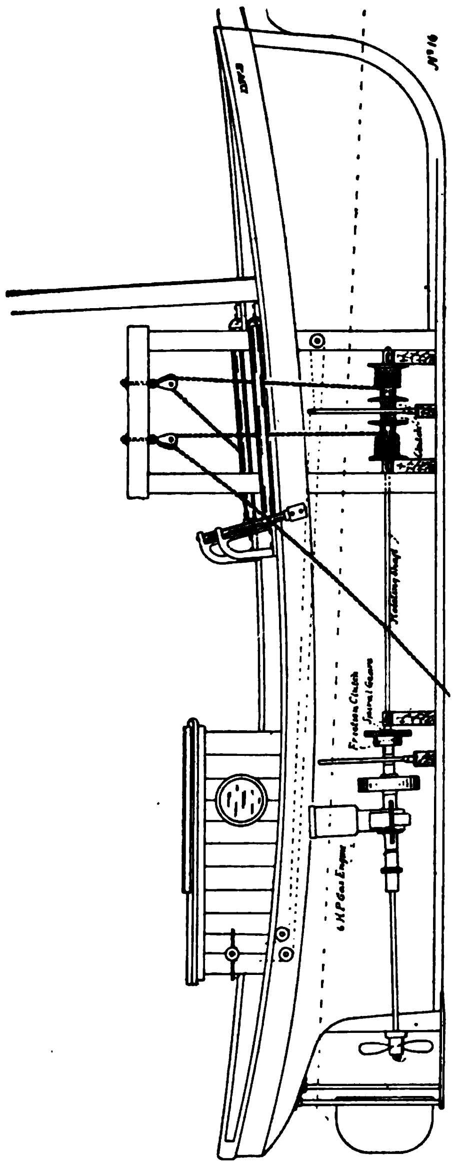


FIG. 251.—Oyster boat, motor, and hoist.

LOZIER FOUR-CYCLE MOTORS

The larger Lozier marine motors are of the four-cycle type, with four cylinders, and are a model of compactness and lightness. The twenty-five horse-power motor, with the bed-plate, flywheel, and reversing-gear weighs 850 pounds, or only thirty-four pounds per horse-power. In the four-cycle type of the Lozier motors the admission-valves, as well as the exhaust-valves, are mechanically

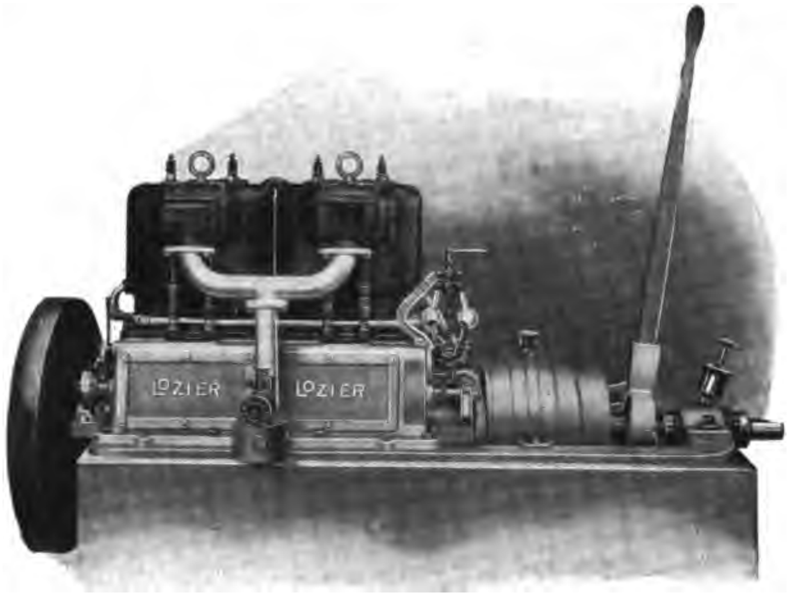


FIG. 252.—Four-cycle auto-marine motor. four-cylinder, 25 horse-power.

actuated, and the principal governor, of the ball type, operating on the admission-valves throttles the gas as it enters the firing-chamber. This governor automatically responds to any change in the load, and is a feature which cannot be applied to a motor, the admission-valves of which are operated by suction. A valuable point to be noticed in connection with this governor is the fact that the speed may be reduced, with a corresponding reduction in the amount of gasoline consumed.

The time of ignition may be changed by means of the timing-lever, which enables the speed of the motor to be controlled at the will of the operator, making a great range of speed possible. The admission and exhaust-valves are on opposite sides of the motor, giving it a well-balanced appearance. The valves, being mechanically lifted, are positive in action, and there can be no sticking or fouling, as is liable to be the case where valves are operated by suction. By unscrewing the covers, which are set

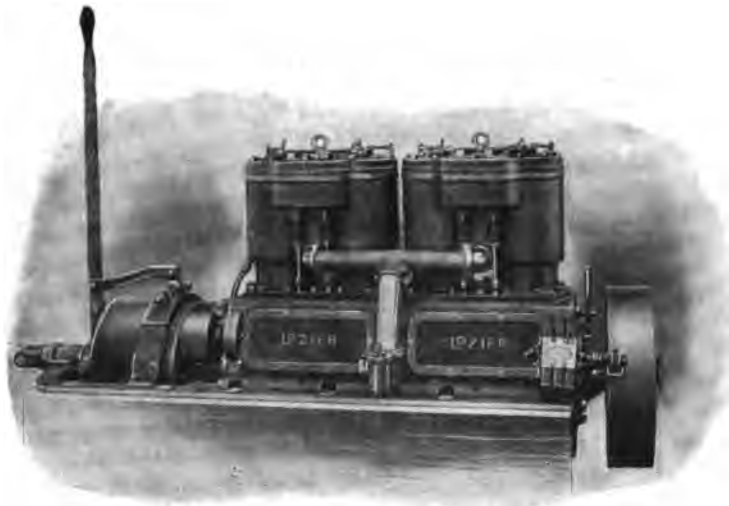


FIG. 253.—Four-cycle auto-marine motor, four-cylinder, 40 horse-power.

in the cylinder-heads directly over the valves, they may be easily removed and examined. The valves are of nickel-steel and not easily affected by the intense heat, thus removing one of the prevalent sources of trouble with four-cycle motors. The exhaust-valves may be lifted by means of a single hand-lever, which relieves the compression and allows the flywheel to be turned in starting with very little exertion. A safety locking-device makes it impossible for the operator to start the motor without setting the timing-lever at "safety." The igniter mechanism is of the make-and-break type. The firing-plug for each cylinder contains both the firing-pin and rocker-arm, and occupies a central position in the cylinder over the firing-chamber.

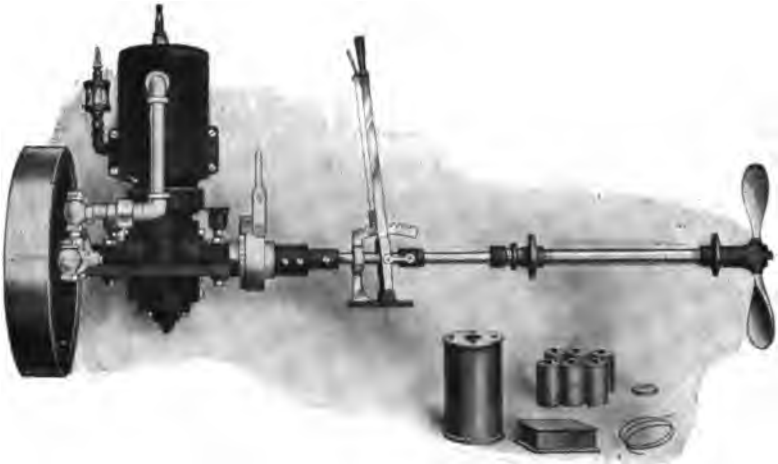


FIG. 254.—Cushman marine motor and equipment.

CUSHMAN MARINE MOTORS

In Fig. 254 we illustrate the Cushman high-speed one-cylinder marine motor. In the design of these motors, simplicity in the arrangement of all their parts has been followed, with the result that a light-weight, high-speed motor, suitable for any service of the pleasure or racing boat, has been attained. Their product is

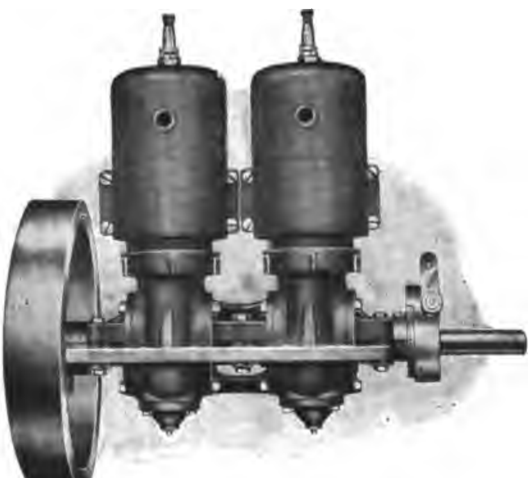


FIG. 255.—Two-cylinder high-speed marine motor.

in one and two-cylinder motors of two, four, seven, eight, and fourteen horse-power, and stationary motors of three and six horse-power. Fig. 255 represents their two-cylinder motors of eight and fourteen horse-power. In Fig. 256 the entire power plant and auxiliaries is shown. The atomizing carburetor discharges its gasoline and air mixture into an annular chamber at the lower end of the cylinder, where it is perfectly vaporized, and enters the cylinder on the opposite side through pressure from the crank-chamber and ports in both cylinder and piston, opened at the charging end of the stroke.

THE SMALLEY MOTORS

In Figs. 257 and 258 we illustrate the details of a novel marine gasoline-engine. The method of admitting the charge at the top of the cylinder through a by-pass from a port in the piston is a distinct feature and a valuable one in defining the boundary of the new charge and the exhaust of the last explosion. When the piston A moves upward, a charge of vaporized gasoline is compressed above the piston top and exploded in the usual manner, and a fresh charge is drawn in through the opening B to which the vaporizer is connected into the crank-chamber C. As the piston

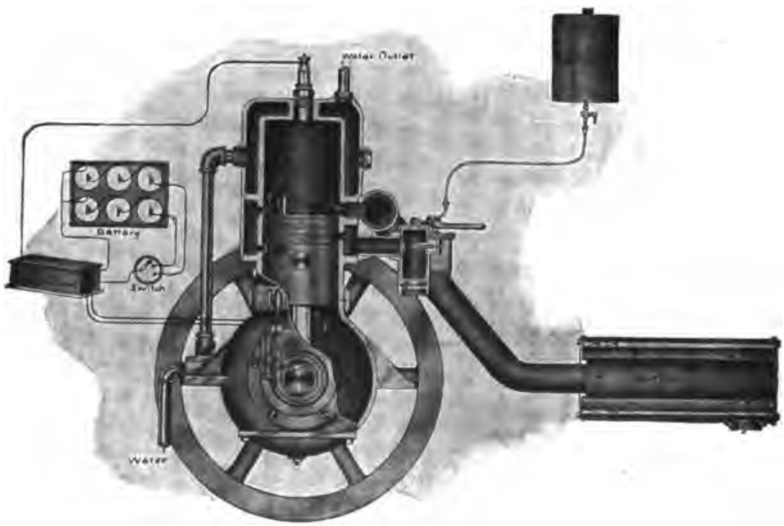


FIG. 256.—Section of motor, wiring, and muffler.

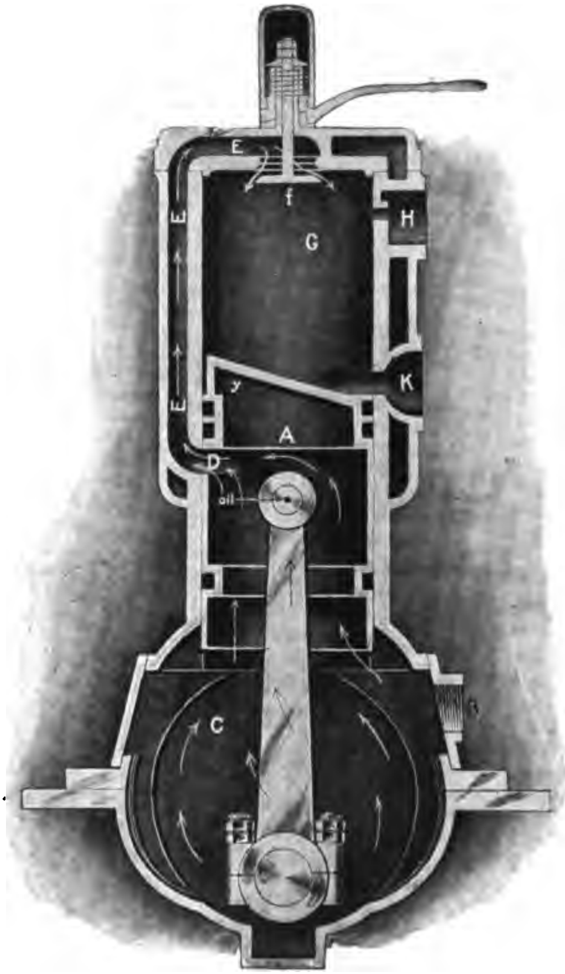


FIG. 257.—Section showing charging by-pass.

reaches the lower end of its stroke it brings the admission-port D (Fig. 257) in the hollow piston opposite the by-pass opening E E E, thus allowing the vapor-charge in the crank-chamber to pass into the upper end of cylinder or combustion-chamber G, through the admission-valve f, which is forced open. At the beginning of the upward stroke of the piston, the valve f is closed by the tension of the spring S, and the gas thus held in the chamber G is compressed by the piston moving up against it. The charge is then ignited by an electric spark in the ignition-chamber H (Fig. 258). The expansion caused by the explosion of this gas forces the piston downward. As the piston passes down-

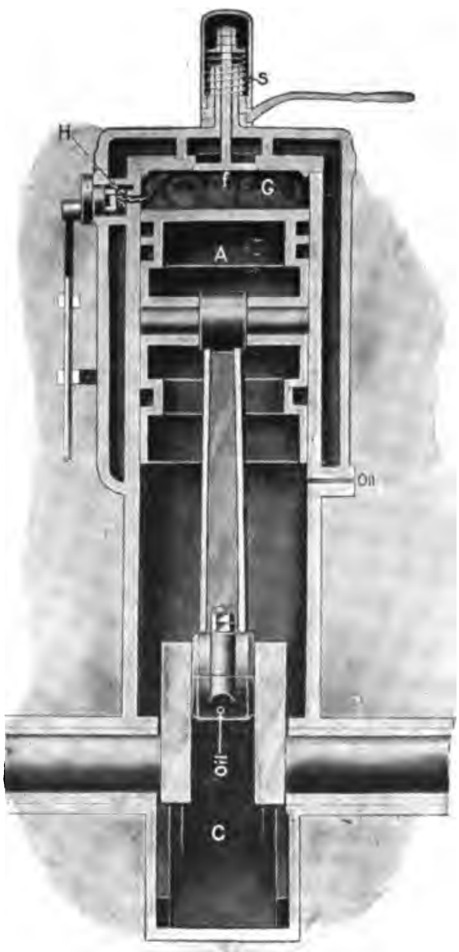


Fig. 258.—Section of ignition-chamber and break-spark device.

ward, the exhaust-port K is opened and the burned products of combustion are entirely exhausted from the cylinder, upward pressure on the valve f is thereby relieved, and the new vapor, which has been compressed in the crank-chamber by the downward stroke of the piston, is again allowed to pass through the port D and the chamber E E E, and thus, by its pressure, forces open the valve f, which allows a new charge to enter the cylinder-chamber G. A special feature of both types of design in the Smalley motors shown is the charging-port through the wall of the piston, which by its position effects a cooling influence on the piston not attainable otherwise than by water circulation, which

Fig. 259.—Three-cylinder Ferro engine, starboard side.

is complicated and troublesome. The method of oiling the piston and crank-pin is also notable in these motors. The piston-pin and connecting-rod are hollow and receive oil through the piston-pin from the cylinder oil-cup and cylinder oil-hole at the moment of exhaust.

FERRO MARINE ENGINES

The Ferro marine engines are a popular design that include a number of excellent features in their construction. They are of the three-port two-cycle type and are built in various standard sizes. All have jump-spark ignition. Each cylinder is cast separately. Fig. 259 shows the general appearance of the starboard side. For a speed boat, launch, or family boat, where the minimum of vibration is desired, the three-cylinder engine is most suitable; being low and compact, it takes up very little room.

Fig. 260 illustrates the forced feed oiling system used on this engine. The oil-tank, on the side of the base or crank-case, holds from one to two gallons of cylinder oil. The down-stroke of the piston, which slightly compresses the mixture in the crank-case, has a passage at the top of the oil tank connecting with the crank-case, which is controlled by a check-valve. At each revolu-

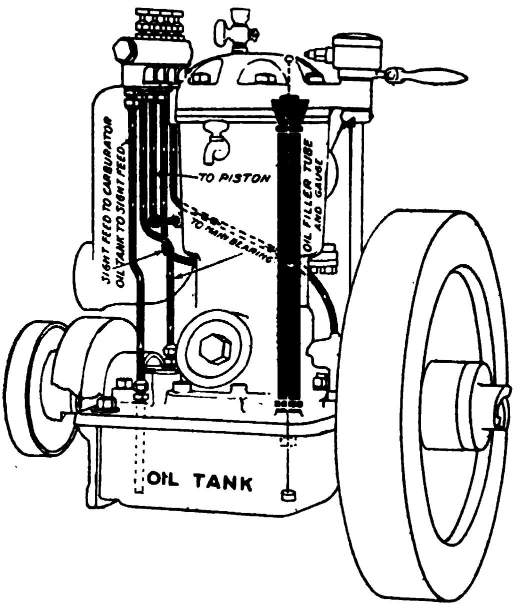


FIG. 260.—Showing oiling system.

tion pressure is stored in the reservoir, and thus serves to force oil up to the sight-feed distributor through a feed tube. From the bottom of each sight-feed valve an oil-tube leads directly to each bearing and every moving part of the engine. As will be seen, the feed tube from the oil-tank, close to the bottom, conveys the oil—as shown by arrows in the cut—up to the sight-feed distributor, the tubes leading to the main bearings and the crankshaft being bored diagonally. The oil from the main bearing passes through the crankshaft to the connecting-rod, the centrifugal force carrying the oil to connecting-rod bearing, thus insuring perfect lubrication. An addition to the regular "splash-feed" system, universally used by marine-engine builders, is supplied as an auxiliary safeguard against carelessness or ignorance. This consists of two wicks in the end of the connecting-rod cap, operating on the crank-shaft, which constantly feeds oil to its bearing.

The oil which settles into the bottom of the crank-case forms a pool, which is thus splashed all over the interior by the rapid revolution of the crank. On the other hand, should one fail to provide oil for the splash-feed system, the pressure feed supplies oil to the crank and connecting-rod, preventing the possibility of

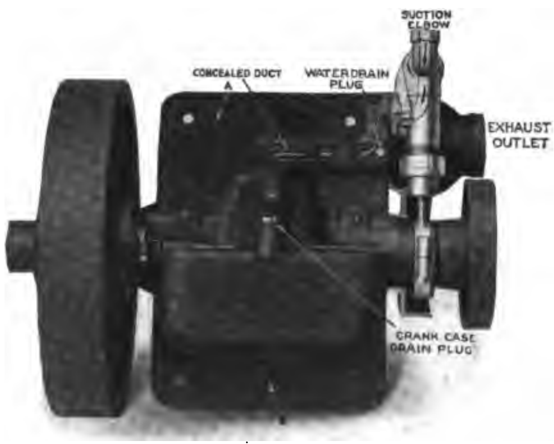


Fig. 261.—Showing water inlet, exhaust, and drainage.

burning out the connecting-rod bearings or cutting the crank. The tube leading to the cylinders supplies the oil direct to its inside walls at a point in line with the hollow piston-pin and oil grooves of pistons. The oil passes through the piston-pin to opposite walls of cylinders, is conveyed and distributed by grooves to all parts of the cylinder walls, is picked up by the piston-rings, and is distributed by the movement of the piston, thus thoroughly lubricating every part of the cylinder. The oil-filling tube contains a float showing the amount of oil in the tank, and a screw cap for filling which may be filled while the engine is running. At the top of the filler cap is a release-valve, by turning which the pressure from the tank is relieved when the engine is not running. The oil-tank is entirely separate from the crank-case, as shown in Fig.

261, which is a view of bottom of engine. This illustration shows how the tank is drained; also how the water is taken from the circulating pump through the case, and how this is drained. It will be seen that no piping is necessary.

In order to secure the best results, it is very important that the cylinders should be properly cooled. With marine engines water is used as shown in Fig. 262. The water passes from the pump through passage in base, through the walls of the cylinder to the extreme top of cylinder-head and out through the water-cooled

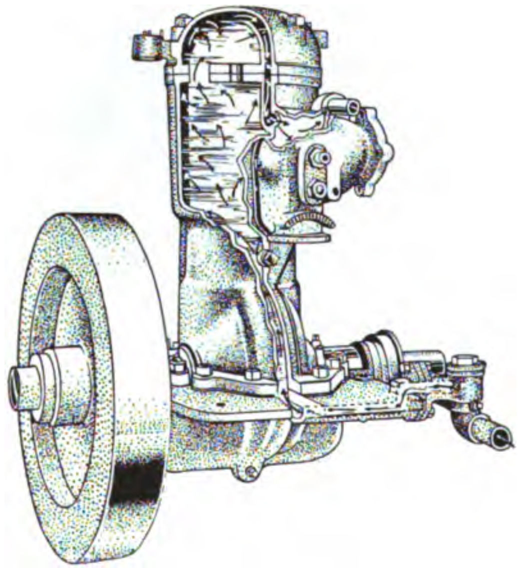


FIG. 262.—Showing water circulation.

exhaust-jacket, which cools the exhaust-flange and helps to condense the exhaust. A portion of the circulating water is expelled direct into the exhaust condenser or silencer, passing over a deflector, further condensing the exhaust. The rest of the circulating water passes out at the top to the right, as shown by an arrow in cut, and may be piped in any direction or conveyed into the exhaust-pipe beyond the condenser. When the cylinder-head is bolted to the cylinder, both are ground, and a copper-asbestos gasket is used, making leakage impossible, if only common-sense be employed. The check-valve of pump should be examined, if a proper water supply is not obtained. If grit should cut the valve-seat, it is a simple operation to grind it in by applying a little emery and oil on the valve-seat turning the valve. The water intake in bottom of boat should always be provided with a screen and a scoop — the opening facing the bow, which forces the water upward to the pump.

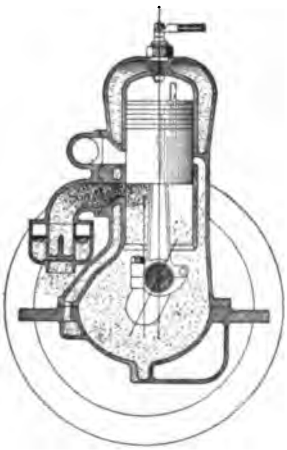


FIG. 263.—Offset cylinder.

As will be seen by Fig. 263, the cylinder is offset from the line of the centre of the crankshaft. Combustion takes place when the piston is at the top, as shown: the crank has passed over the top centre, the force is applied on a working part and not on a dead centre. This relieves the bearings of the constant thrust and jar, lessens the side thrust, avoids dependence upon the momentum of the flywheel to take it over the centre, prevents kicking back, and increases the power of the engine. To further illustrate the advantage of the offset cylinder, one may take the operation of a treadle, in starting which the first thing is to turn the crank over the centre before applying the power — as with a bicycle foot-lathe or grindstone. No energy is wasted and no undue shock given the bearings in producing the maximum power. The offset cylinder is adopted on some of the most successful automobile motors in use to-day.

The dynamo and storage battery for ignition and lighting, used in connection with any standard electric ignition, can be employed to supply electricity for a group of low voltage incandescent lamps for lighting the boat. (See Fig. 264.) The storage battery may be used singly or in a series, depending upon the capacity or duration of current required to operate a system without recharging. The dynamo furnishes the electricity to the batteries; and from then it is fed to the ignition and lighting systems. The dynamo is usually belted to the fly-wheel of the motor, but it can be used with a friction wheel or spur gear. An automatic speed governor is generally furnished with the dynamo, and serves to maintain a steady volume of current to the battery. An automatic switch breaks the dynamo circuit, when the batteries have been charged to their full capacity. This system furnishes a constant and steady current, and obviates the necessity of replacement or renewals, which exists in the case of dry and wet batteries. It is inadvisable to depend upon the dynamo alone, without any batteries, to start a motor, unless, by cranking the motor, the speed of the dynamo can be made high enough to furnish sufficient strength of electricity for ignition. Some other source of current should be used.

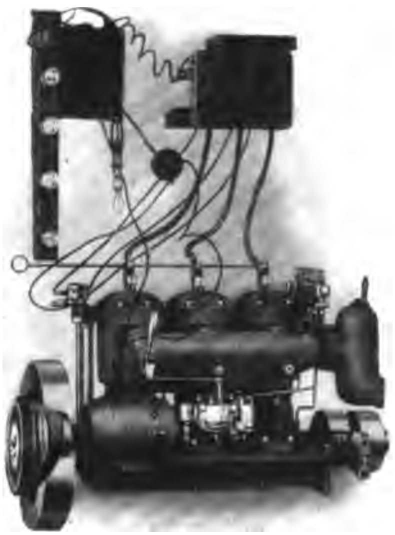


FIG. 264.—Showing lighting storage and ignition.

MOTORS FOR FISHING BOATS

Those who follow fishing for a livelihood usually need something different from other motor-boats — both as to style of boat and engine. On the Delaware and Chesapeake bays and tributaries for drift nets, what are known as skiffs and bateaux are used for fishing for shad, sturgeon, herring, etc. While the ordinary old-style shad skiffs are good sailers and fine sea boats, they are not capable of extra speed with motor. These will, however, make very good speed if not overpowered. These boats usually require 4 to 8 horse-power, according to size. Still, some builders have now changed the model, giving more of a speed boat, bottom and stern; and here 10 to 12 horse-power may be used. As these fishing-boats use a drift net 100 to 600 fathoms long, it is necessary in taking on the nets to be able to regulate the speed to a nicety both in backing and going ahead. A reversible propeller and not a reverse gear, is the proper thing to use. A fisherman does not usually have time to slow down an engine, nor can he take any chances of the engine stopping in an attempt to slow down. As his boat is out in all kinds of rough weather and storms, often taking in considerable water from the

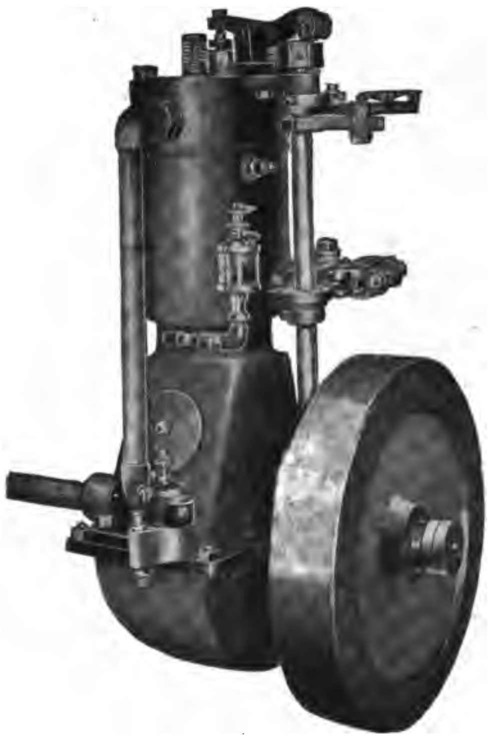


FIG. 265.—The fisherman four-cycle engine.

net as well as from the sea, the fisherman favors a make-and-break engine in order to do away with the high-tension wire and jump-spark coil. Though more complicated than the jump-spark, the make-and-break engine is thought to be preferable for open working boats for fishermen, oystermen, and others of similar occupation. Several jump-spark systems have been evolved, such as the Perfex, that reduce the liability of short circuit by combining the induction coil and spark plug.

Fig. 265 represents a 6-horse-power four-cycle engine designed solely for fishermen, crabbers, tongers, oystermen, and ferrymen. Bore, 5″; stroke, 6″; outside diameter of cylinder, $8\frac{1}{4}$″; crankshaft, $1\frac{5}{8}$″; propeller-shaft, 1″; diameter of flywheel, 20″; swings a two-blade 18″ propeller; 32″ pitch weighs about 375 pounds. Normal speed, 450 to 600 revolutions per minute; will throttle to as low

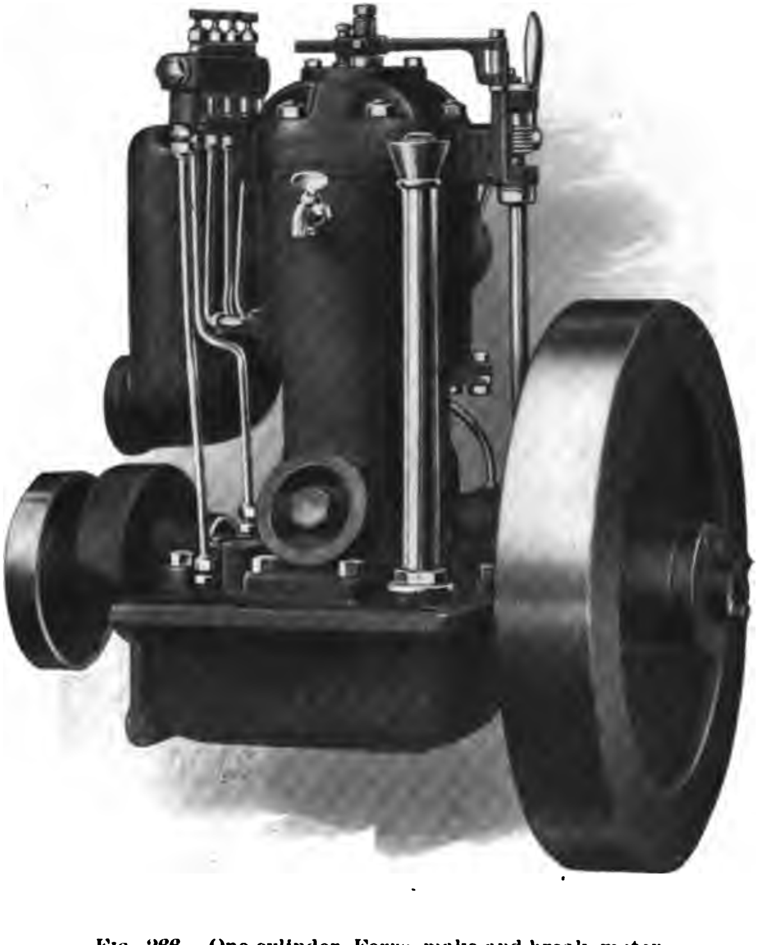


FIG. 266.—One-cylinder Ferro make-and-break motor.

a speed as 75 revolutions. Fitted with either jump-spark or make-and-break. The cylinder is offset. It is a heavy-duty engine. The Ferro one cylinder make and break engine is shown at Fig. 266. The general features are the same as in the engines of this make previously discussed.

THE POWELL OPEN-BASE TWO-CYCLE ENGINE

This marine engine represents a distinct type of its own, as the illustration, Fig. 267, and the sectional view, Fig. 268, show.

The piston is hollow, the charge being compressed beneath it and not in the crank-case. Compression at the bottom of the piston-chamber results in an extra-strong primary compression and quick-firing mixture at the time of explosion. In other two-cycle engines the mixture is introduced into the base or crank-case. It is claimed that the strong primary compression of the

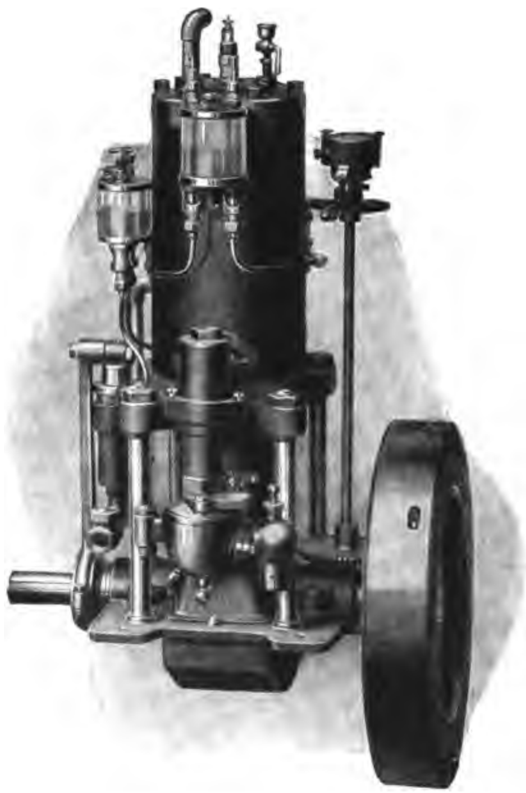


FIG. 267.—Powell open-base two-cycle engine.

open-base engine is more perfect in the scavenging of the cylinder, and that leakage at the crankshaft is eliminated. The carburetor is below the level of the cylinder, preventing flooding. The advantage claimed by the open-base construction is that all parts are accessible without taking the engine apart or disturbing one cylinder to get at any part of another. The engine is fitted with

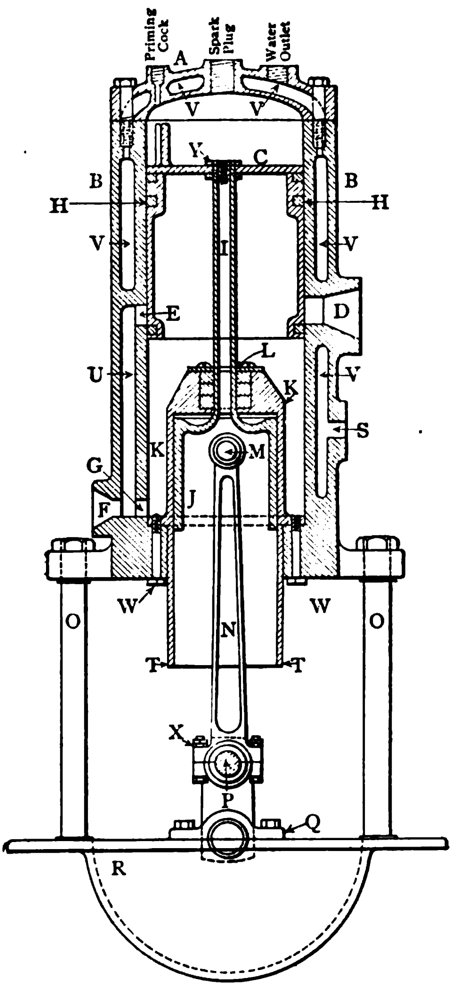


FIG. 268.—Powell open-base engine, sectional view.

oil-guards which surround the crank-arms and are easily removed, and prevent oil from being thrown about the boat. Bearings are reversible, interchangeable, and renewable. The engines are manufactured in one, two, three, and four cylinders, of 5 horse-power each, and are designed to run at 525 and 500 revolutions. The sectional view, Fig. 268, gives a list of parts showing the construction.

GRASSER COMBINATION TWO-PORT, THREE-PORT ENGINE

In this engine the most important and distinct feature is the combination. It is claimed that it has all the advantages of both, without any of the disadvantages. The difference between the combination two-three port and the ordinary two-cycle will be readily seen, if reference be made to the sectional illustrations, Fig. 269 and Fig. 270. As the piston starts on its up-stroke, the charge is drawn through the generator valve A into the crank-case B, until the third port C is opened, allowing the remaining vacuum to be displaced by air through the third port C, which would not otherwise be displaced. When the piston starts on its down-stroke, valve A closes; the result is a full charge in crank-case, which means a larger charge in the cylinder, more power,

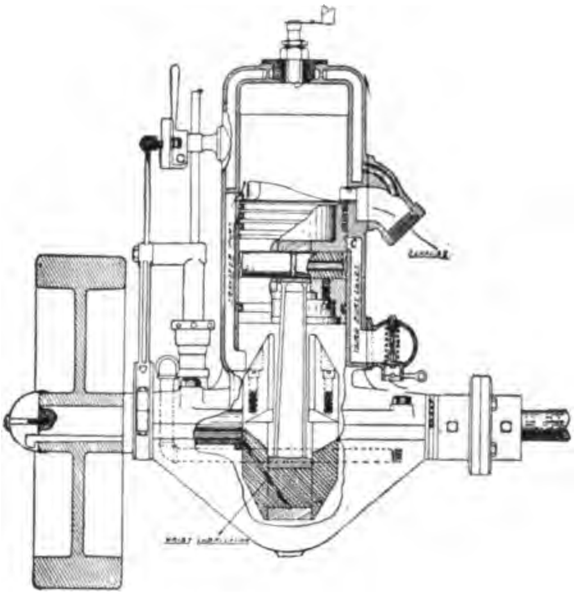


Fig. 269.—Grasser marine engine—sectional view.

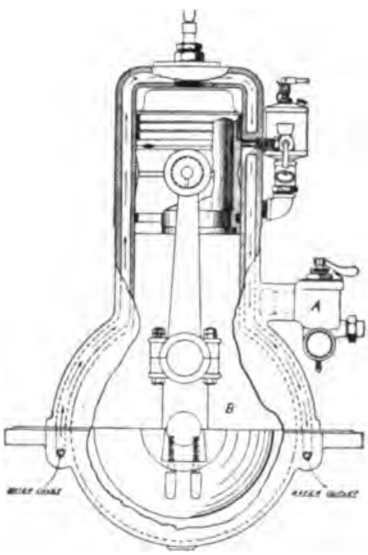


Fig. 270.—Grasser marine engine—sectional view.

and an absence of crank-case firing. The ports are placed in the forward and aft sides of the cylinder. The reason for this is that the thrust of the piston is always sideways; and, by maintaining a solid wall on the sides without ports, the wear will be longer than when there is no thrust fore and aft on the piston. This style of engine is made in 3, 6, 10, and 15 horse-power. On the starboard side of the engine a generator is connected direct to the crank-case, using a check-valve as in a two-part engine, forming the two-port system; another generator on the port side

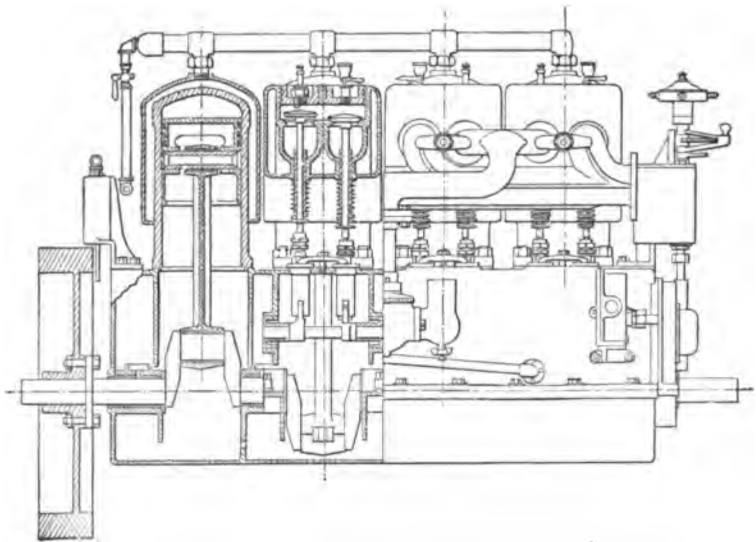


FIG. 271—Scripps motor, longitudinal section.

of the cylinder supplies gas through the third port. An improved generator-valve with throttle control is used. Each cylinder has its own generator. The water circulation as well as the general construction is shown in the two illustrations.

SCRIPPS FOUR-CYCLE ENGINES

Scripps motors are of the four-cycle type, having both inlet- and exhaust-valves on the same side of cylinder and operated by the same cam-shaft (Figs. 271 and 272). They are provided with roomy hand-holes on side of crank-case, giving free access to connecting-rods and main bearings. The main bearings are

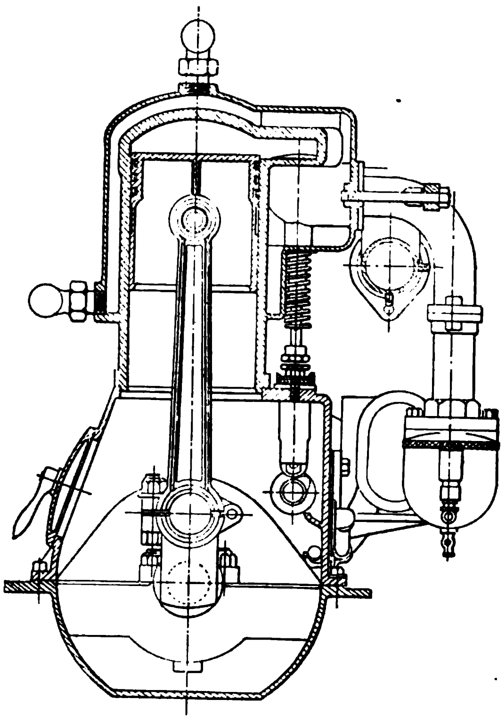


FIG. 272.—Scripps motor, lateral section view.

supported by webs or partitions which divide the lower half of crank-case into separate compartments for each cylinder. Thus the two-cylinder size has three bearings, the four-cylinder has five, and the six-cylinder, seven. They are lined with the highest grade of babbitt obtainable and are scraped to a perfect bearing. The same applies to connecting-rod bearings. All gears are encased, and by the use of a bronze intermediate gear are rendered practically noiseless. Control levers are located at a convenient point, so that the operator may easily reach them while sitting in a position to reach the reverse lever.

WATERMAN OUTBOARD ENGINE, FOR TENDERS AND ROW-BOATS

This engine or complete power plant can be shipped like a rudder, and actually takes the place of a rudder, besides driving the boat. Fig. 273A shows it ready for use. The ignition

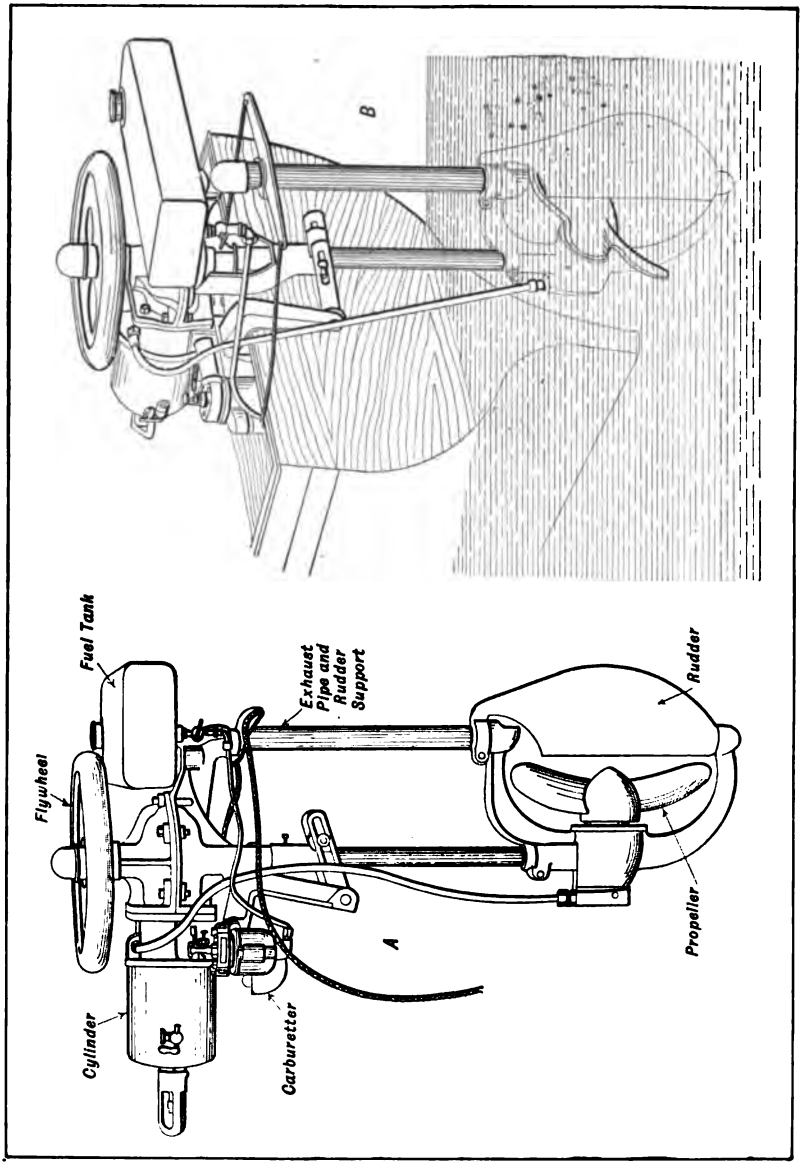


Fig. 273.—View of the Waterman Outboard Power Plant, complete and ready for installation at A, and method of application to rowboat stern at B.

equipment may be placed in any part of the boat — usually it is near the stern. Fig. 273B illustrates how to attach it, and the construction. This engine may be quickly attached to or detached from any row-boat, and fills a distinct want. It will enable one to convert a row-boat into a motor-boat in a very short space of time.

The sectional view at Fig. 274 shows clearly the construction of the small power plant with its auxiliaries conveniently placed, the method of propeller drive through bevel gears, and the very

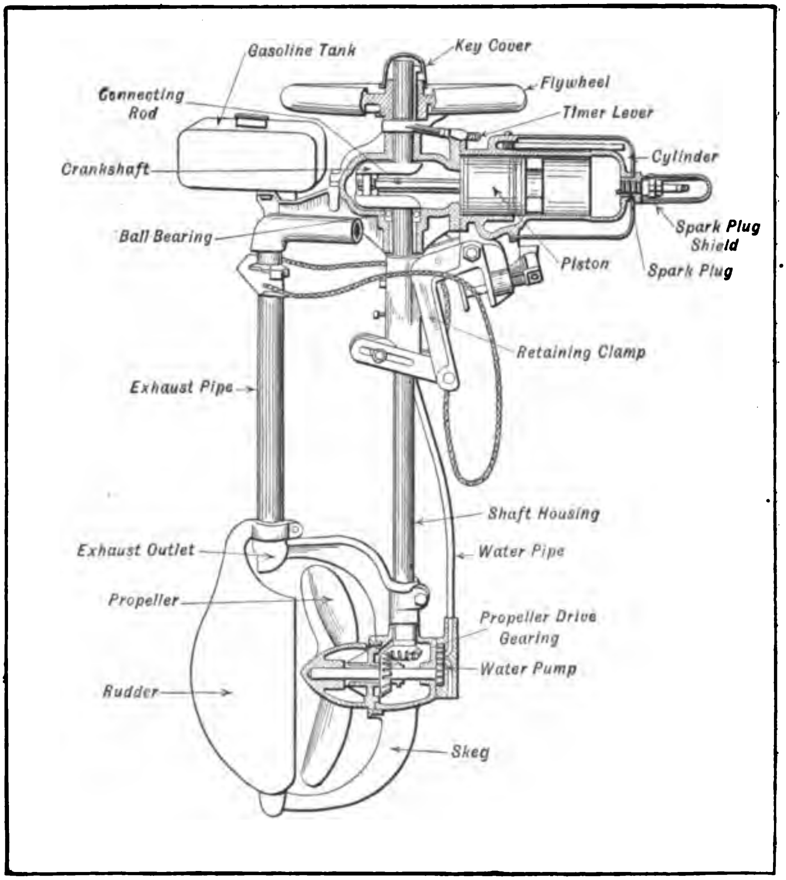


FIG. 274.—Sectional view detailing internal construction of the Waterman outboard motor.

effective and ingenious attaching bracket. In general features of construction it is the same as the inboard motors.

WATERMAN INBOARD MOTOR

A very popular light weight motor intended for application to light hulls is also marketed by the Waterman interests. This is shown at Fig. 275 with important parts clearly outlined. A feature of merit is the method of applying the sheet metal water jacket to the cylinder casting. It is claimed that the use of this construction makes it possible to obtain much smoother and cleaner cylinder castings inasmuch as considerable intricate core work is eliminated. Ample opportunity is given to smooth off the exterior of the entire cylinder casting and to remove all roughness and sand from the casting before the water jacket is applied. The method of retaining the water jacket in place may be clearly understood by referring to the illustration. The cylinder has a circular flange cast integral which is turned off to form a seating for the steel water jacket. This member is pushed down in place and is held in position by a clamped ring which is shrunk in place around the lower portion of the water jacket. A tight joint is made at the upper end by a clamped bushing which also serves as a carrier for the spark plug. But one boss projects from the cylinder and this is to make allowance for the application of a compression relief and priming petcock. This presses the steel water jacket firmly in place against the boss when screwed in place and there can be no leakage of water at that point. The engine base is of aluminum in order to obtain extreme lightness and it is said that the port proportions are such that speeds of rotation varying from 200 to 1200 revolutions per minute may be obtained by spark and throttle control. The form shown is known as the K-1 and is the lightest of the inboard type of motors.

THE PERFECTION ENGINE

The Perfection marine motors, one of which is shown at Fig. 276, are of the two-port, two-cycle type and are made in a variety of sizes ranging from 2 to 8 horse-power single cylinder and also in a number of multiple cylinder models. The carburetor is attached to a fitting carrying the intake check valve which is bolted to the side of the transfer passage through which the gas

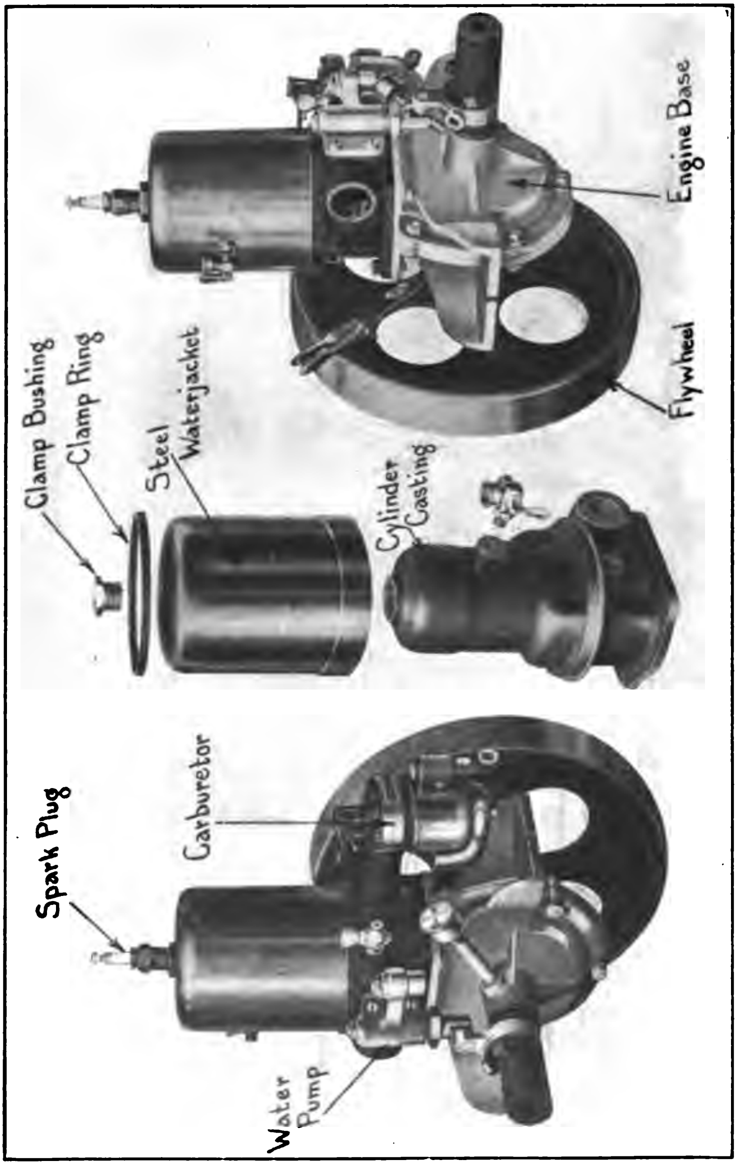


FIG. 275.—Views outlining construction of the Waterman inboard marine motor with special reference to the method of installing the applied water jacket.

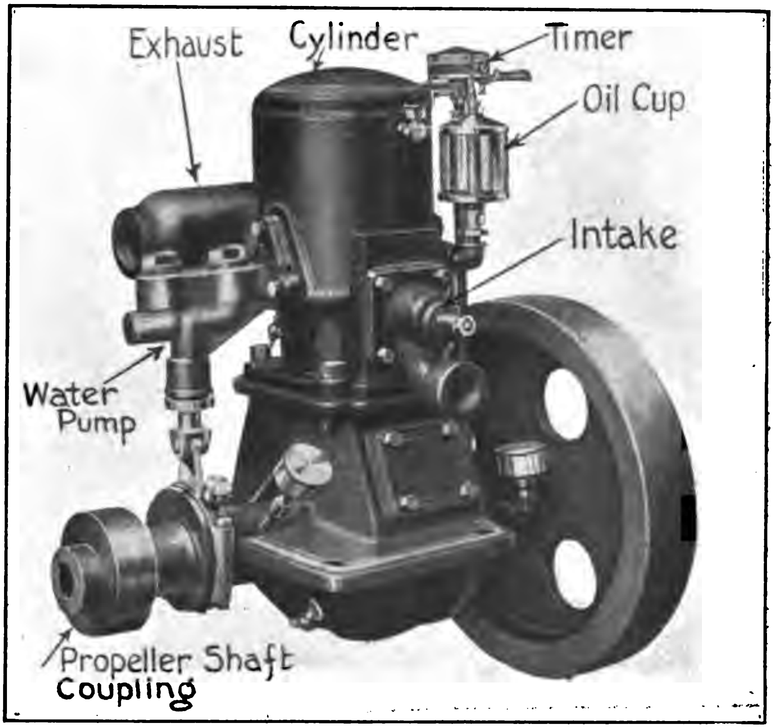
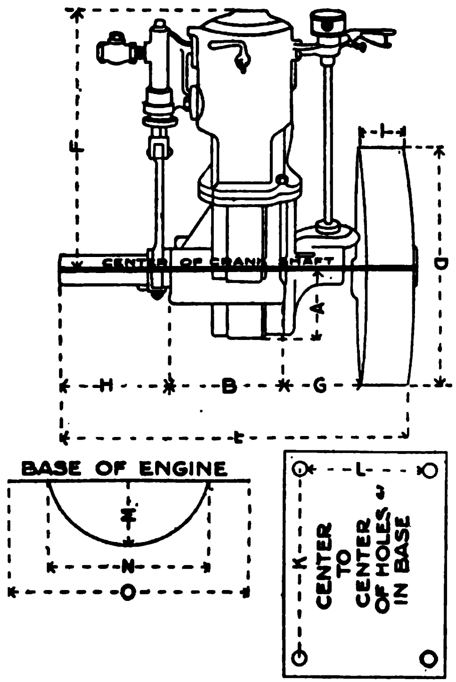


FIG. 276.—The Perfection 8 horse-power, two-port, two-cycle, medium duty marine motor.

flows from the engine base where it has received a preliminary compression of about 5 pounds to the cylinder interior. The water pump is bolted on to the cylinder and is of the plunger type driven by an eccentric on the engine shaft. The check valves of the pump are easily reached in event of trouble, and in fact all of the parts that require attention are readily accessible. The lubrication system is extremely simple: a sight feed oil cup attached to the cylinder lubricates that member, while the main bearings are lubricated from compression grease cups. These grease cups not only provide positive lubricity, but it is said that the heavy grease will stay in place better than the lighter oils generally used, and assist in retaining the crank case compression even after the bearings are worn to some extent. The crank case is cast in one piece and has solid bearings. A large inspection



DIMENSIONS OF "PERFECTION" ENGINES

Letter on Drawing	HORSE POWER									
	2	2½	3½ to 4	6	7 to 8	8	12	14	18 to 20	30
A	3½	3½	3½	4	3⅜	5	3⅜	4½	5	4½
B	6¼	6¾	6¾	7½	10½	9½	16½	13¼	16½	26½
D	12	12	14	15	14	18	14	15	18	15
E	18½	19	19	21	27¼	27¼	32½	31½	36	46¼
F	13½	13¾	13¾	17¾	13	25¾	13	17½	21	17½
G	4¼	4¼	4¼	4½	4¼	6¼	4¼	4⅛	5¾	4⅛
H	5½	5½	5½	6½	10	11¼	10	11	10½	11
I	2⅜	2⅜	2	2¾	2	2¾	2	2¾	3	2¾
K	9¾	9¾	9¾	11¾	11½	14½	14⅜	11	14¾	25¾
L	5	5	5	5½	8½	7½	11⅝	13½	14¼	13½
M	3¼	3¼	3½	4	3¼	4¾	3¾	4¼	4½	4¼
N	7½	7½	7½	9½	7½	11¾	7½	9½	11	9½
O	11⅜	11⅜	11⅜	14	13	16½	13	15½	16	15½

FIG. 277.—Diagram giving principal dimensions of Perfection marine engines.

plate is provided so that the interior of the engine may be readily examined without taking it apart.

A complete table showing the specifications of the various types of Perfection motors is given at Fig. 277. As these are typical of a number of small power plants, they will serve as a basis for designing engine foundations in varying types of craft. The engine shown is 6 horse-power, is suitable for launches up to 26 feet in length; it has a bore of 4¾ inches and a stroke of 4½ inches and weighs 200 pounds. It is capable of turning a 16-inch two-blade, 20-inch pitch or 16-inch, three-blade 18-inch

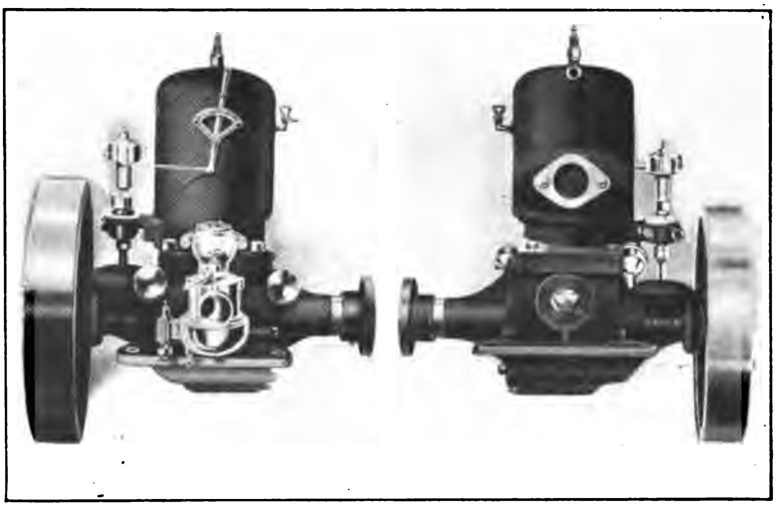


FIG. 278.—Views showing the construction of Gray Model R marine engine.

pitch from 100 to 800 revolutions per minute. All Perfection engines except the 18 horse-power heavy duty are supplied with the jump spark system. The make-and-break or low tension ignition is used mostly in open boats and is not apt to be affected by moisture as is the jump spark system, though the latter will give good results if care is taken in installing the batteries so they will be in a water-tight box, carefully protecting the secondary wire and using a water-proof form of spark plug.

THE GRAY MARINE ENGINES

One of the most popular of the marine power plants and one that has received general application in all classes of boats is the Gray engine. These are made in a number of different types, ranging from 3 to 12 horse-power in the single-cylinder form, from 6 to 24 horse-power in the two-cylinder, and from 21 to 36 horse-power in the three-cylinder types. The model R engine, which is shown at Fig. 278, is one of the most popular, as it is not only very simple in construction, but is very compact and

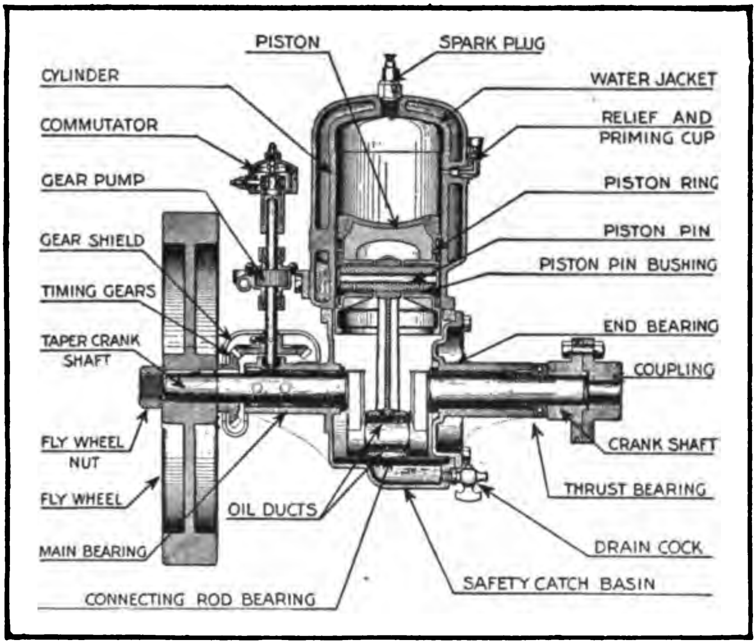


FIG. 279.—Part sectional view showing important details of the Gray Model R marine motor.

easily installed. The 4½ horse-power model has a bore of 4 inches and a stroke of the same dimensions. It weighs complete 190 pounds and is suitable for boats from 16 to 22 feet in length. The 6 horse-power has the same stroke as the 4½ horse-power model, but has a bore of 4¾ inches. This power plant is suitable for hulls from 18 to 35 feet in length. The sectional view of the model R Gray engine at Fig. 279 shows clearly the interior construction of this simple power plant. Attention is directed to

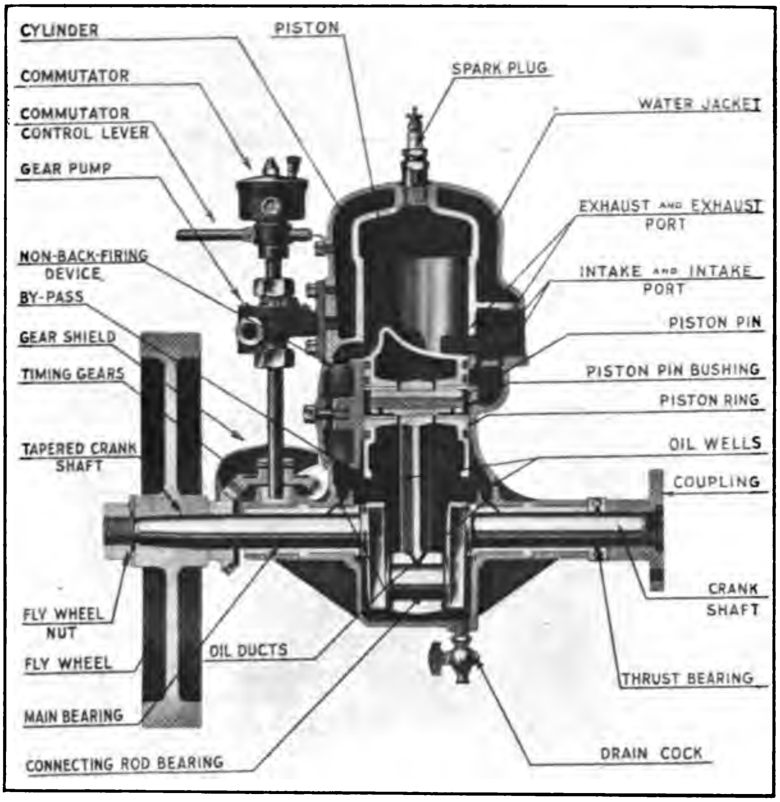
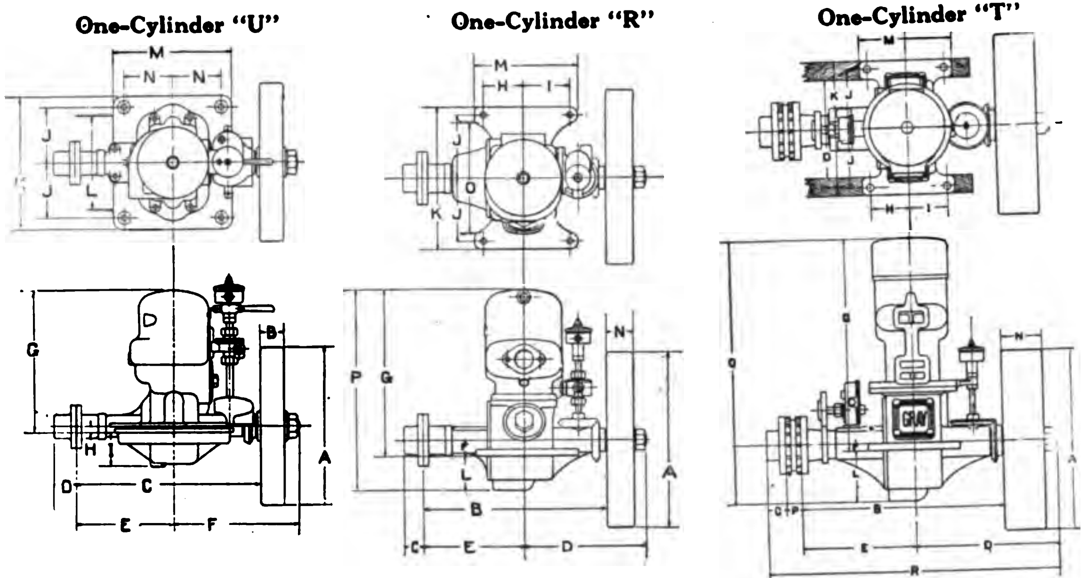


Fig. 280.—Part sectional view showing arrangement of ports in the Gray Model U marine motor.

the strength of the parts, the length of the main bearing and the general substantial design throughout.

Another form of single cylinder Gray motor is shown in section at Fig. 280. This shows clearly the arrangement of the by-pass passage with its non-backfiring device, the light piston with its efficiently formed deflector and the large size of the exhaust ports. This engine is of a three-port type, whereas the model R previously shown is a two-port design. When the piston is in the position shown both inlet and exhaust ports are uncovered and the fresh gas is flowing from the crank case interior into the cylinder while the exhaust gas is discharged through the fully opened exhaust ports. When the piston reaches the upper



One-Cylinder "U"

HP	A	B	C	D	E	F	G	H	I	J	K	L	M	N
3	15 1/4	2	15 5/8	1 1/8	8 3/4	10 1/2	12 1/16	1/2	2 3/8	4 3/8	11 1/8	7 3/8	10	4 1/16

One-Cylinder "R"

HP	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
4 1/2	16	17 1/4	1 3/4	11 9/16	9 1/4	1 15/32	14 15/16	3 11/16	4 1/4	5 3/4	13	3 1/16	9 7/16	2 1/2	9	18
6	16	17 1/4	1 3/4	11 9/16	9 1/4	1 15/32	15 1/16	3 11/16	4 1/4	5 3/4	13	3 1/16	9 7/16	2 1/2	9	18 5/16

One-Cylinder "T"

HP	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R
7	16	17 1/4	1 3/4	11 9/16	9 1/4	3/8	17 3/4	3 11/16	3 11/16	5 3/4	13	4 5/16	8 7/8	2 1/2	9 1/2	1 1/4	22 1/16	25 5/8
8	17 3/8	19 3/16	1 7/8	13 3/8	11 1/16	3/8	20 1/4	3 1/4	3 3/4	7	16	5 1/16	9	4	12	1 1/2	25 3/8	28 1/4
12	17 3/8	19 3/16	1 7/8	13 3/8	11 1/16	3/8	[illegible]	3 1/4	3 3/4	7	16	5 1/16	9	4	12	1 1/2	26	28 1/4

FIG. 281.—Tables giving principal dimensions of the Gray one-cylinder marine motors in three main types.

end of its stroke, it uncovers the intake port and admits a charge of gas into the crank case interior. As a guide for the installation of these motors all important dimensions of the various single cylinder types are presented at Fig. 281.

The engine at Fig. 282 is a model T, two-cylinder combined two- and three-port type. A sectional view through the cylinder of the model T engine at Fig. 283 shows clearly the arrangement

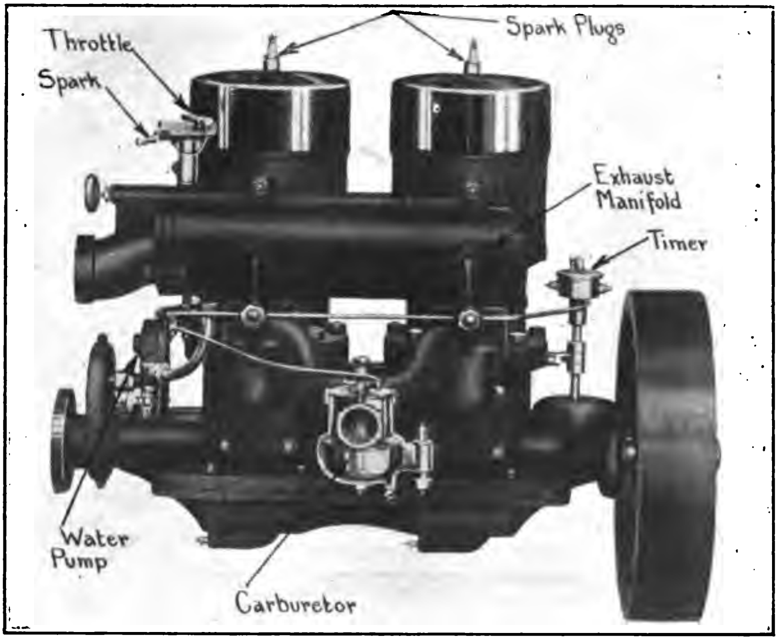


FIG. 282.—Side view of the Gray two-cylinder Model T marine power plant.

of the ports. It is said that it can be changed from a two- to a three-port engine in a few minutes or it will operate as a combination type. Attention is directed to the safety screen forming part of the by-pass cover which prevents back firing. It will be apparent that a hole is bored in the lower portion of the piston wall that registers with a similar opening in the cylinder casting when the piston reaches the end of its down stroke. When the piston starts to go up, as soon as the transfer passage is covered the second port which is controlled by the piston walls is opened

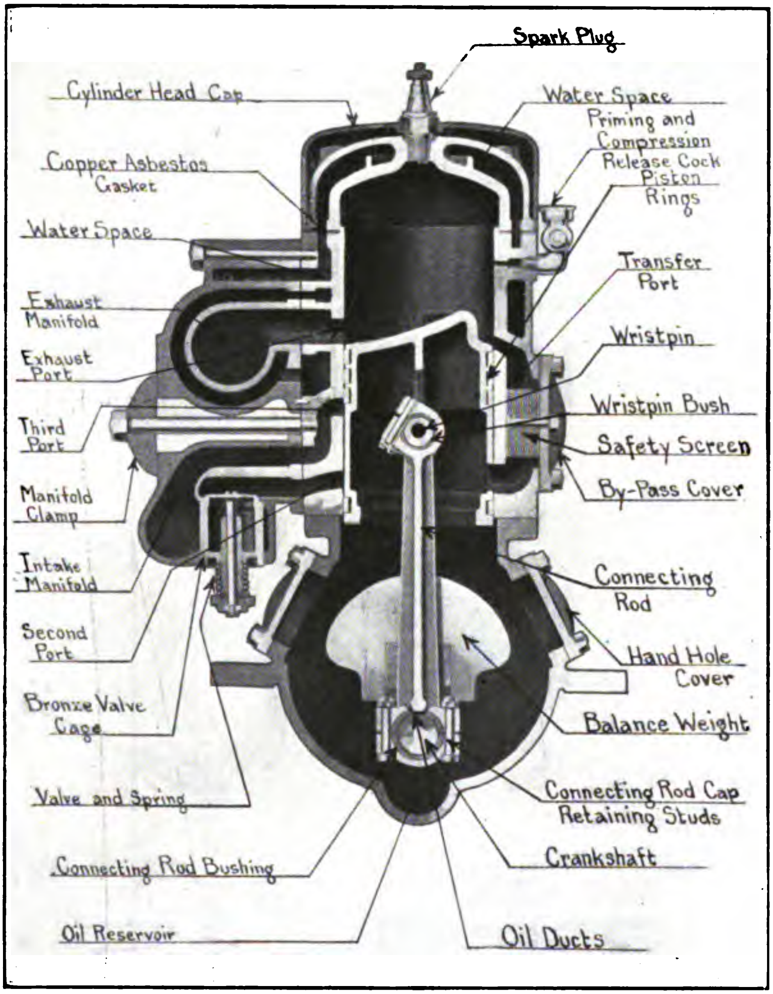
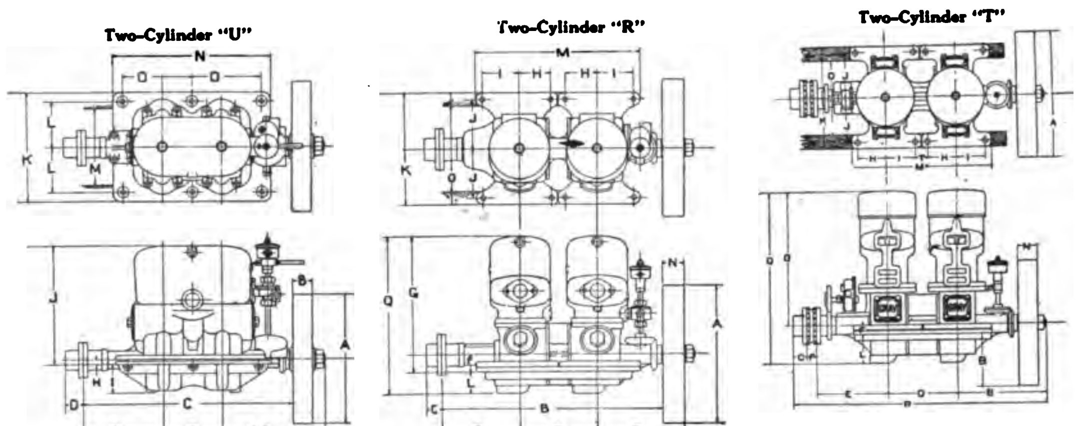


Fig. 283.—Sectional view showing arrangement of parts in the Gray combined two and three-port Model T engine.

and suction begins from the carburetor, through the inlet check valve which opens at this time. When the piston reaches the top of its stroke it uncovers the third port which is also in direct communication with the intake manifold and provides two ports through which the crank case can be charged. The exhaust manifold is provided with a water jacket which is in communication



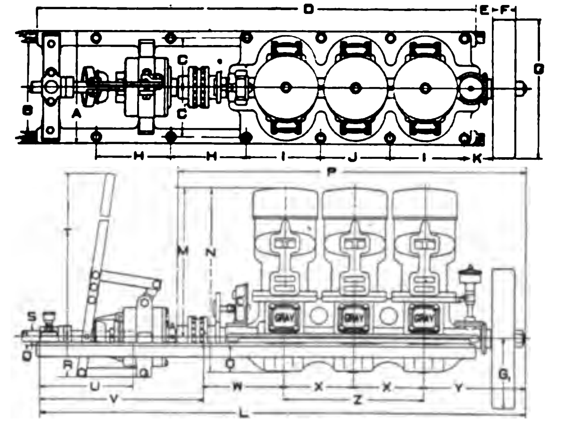
HP	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O
6	13¾	2	21⅝	1⅞	6¼	6	10¼	½	2⅝	12¼	11⅜	4⅜	7⅝	16	7¼

HP	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q
9	16	20¼	1¾	11⅝	9¼	2 1/16	15½	5 5/16	4¼	5½	13	2½	19	2½	9½	9	18⅝
12	16	20¼	1¾	11⅝	9¼	2 1/16	15 9/16	5 5/16	4¼	5½	13	7½	19	2½	9½	9	18⅝

HP	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T
14	16	[illegible]	1¼	9	9½	1¼	17½	3⅛	3	5½	13	3½	17½	2½	10¼	1¼	12½	12	11	1½
16	17½	7½	1⅜	10½	11¼	1¼	19	3¼	1½	7	16	4½	19½	4	12½	1½	25½	30	13¼	3
24	17½	7½	1⅜	10½	11¼	1¾	19	3¼	3½	7	16	4½	19½	4	12½	1½	26	30	13¼	3

FIG. 284.—Diagrams giving principal dimensions of the Gray two-cylinder marine plants.

Three-Cylinder "T"—also Two and Three-Cylinder "T"'s with Gear Attached



	2 CYL		3 CYL	
HP	14	24	21	36
A	13	16	13	16
B	9[illegible]	11[illegible]	10[illegible]	[illegible]
C	5[illegible]	7	5[illegible]	7
D	36[illegible]	47[illegible]	50[illegible]	60
E	2[illegible]	3[illegible]	[illegible]	1[illegible]
F	2[illegible]	4	2[illegible]	4
G	16	17[illegible]	16	17[illegible]
H	7[illegible]	11[illegible]	8[illegible]	10[illegible]
I	7[illegible]	7[illegible]	8[illegible]	10[illegible]
J	1[illegible]	3	8	10[illegible]
K	4[illegible]	4[illegible]	2[illegible]	3[illegible]
L	51	55[illegible]	55[illegible]	66[illegible]
M	17[illegible]	20[illegible]	17[illegible]	20[illegible]
N	22[illegible]	26	21[illegible]	24[illegible]
O	3[illegible]	4[illegible]	2[illegible]	3[illegible]
P	32[illegible]	38[illegible]	39[illegible]	49[illegible]
Q	1[illegible]	1[illegible]	1[illegible]	1[illegible]
R	3[illegible]	5[illegible]	4[illegible]	5[illegible]
S	1[illegible]	1[illegible]	1[illegible]	1[illegible]
T	16[illegible]	25[illegible]	21[illegible]	24[illegible]
U	1[illegible]	11[illegible]	11[illegible]	11[illegible]
V	21[illegible]	20[illegible]	18[illegible]	20[illegible]
W	9[illegible]	11[illegible]	9[illegible]	11[illegible]
X	—	—	8	10[illegible]
Y	11[illegible]	13[illegible]	11[illegible]	13[illegible]
Z	9	10[illegible]	—	—
G1	8	8[illegible]	8	8[illegible]

Fig. 285.—Diagram giving principal measurements of Gray two and three-cylinder Model T engines with reverse gear attached.

with the water space of the cylinder. The various parts are so clearly shown that further discussion seems unnecessary. Complete specifications of the various types of two-cylinder Gray engines with all dimensions necessary for installation are shown at Fig. 284, while the three-cylinder forms are given at Fig. 285.

THE VAN BLERK MARINE MOTORS

The Van Blerk motors, which are illustrated at Figs. 286 and 287, are of the four-cycle type, and are made in two main forms, the medium duty or model B and the type C or high speed motors. The former are intended for use in cruising and work boats, and rotate normally at from 500 to 700 revolutions per minute. The high speed motors are made to deliver their maximum power and operate at the greatest efficiency at speeds ranging from 100 to 1400 revolutions per minute. These motors are intended for use in speed runabouts and racing craft. The cylinders are of the T head type, and on all the motors are cast individually. On the medium duty motors the cylinder water jacket is of cast iron formed integral. On the high speed motors the cylinders are cast with open water jacket, and this opening is covered with a brass plate securely fastened by means of a screw. The crank

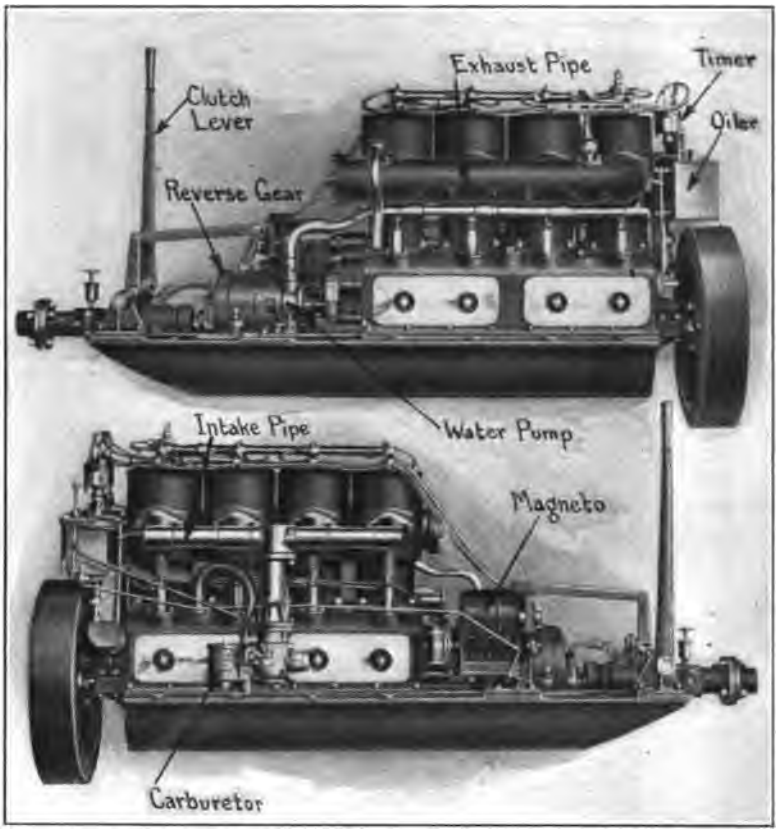


Fig. 286.—Exhaust and inlet side of the Van Blerk four-cylinder marine motor with reverse gear attached.

cases are re-enforced with stay rods which run through and connect the two halves and thus afford rigid support to the top of the case, holding it rigid and steady against vibration because of the firm support afforded by the engine base.

The crank case is provided with openings that are sufficiently large to enable the operator to take out connecting-rod bearings and inspect all internal parts after the hand hole cover plates are removed. The lower half of the crank case is provided with webs or partitions between each of the crankshaft throws. These add strength to the case, support the bearings, and divide the case into compartments which retain oil. The crankshafts are forged from selected special carbon crank steel on the medium duty model and are made of chrome nickel steel on the racing model or high speed motor. The connecting-rods are dropped forgings of steel and are of the four bolt marine type.

Lubrication is by a special circulating system on the racing model and by a mechanical four feed oiler on the medium duty type. The circulating system includes a reservoir of about one

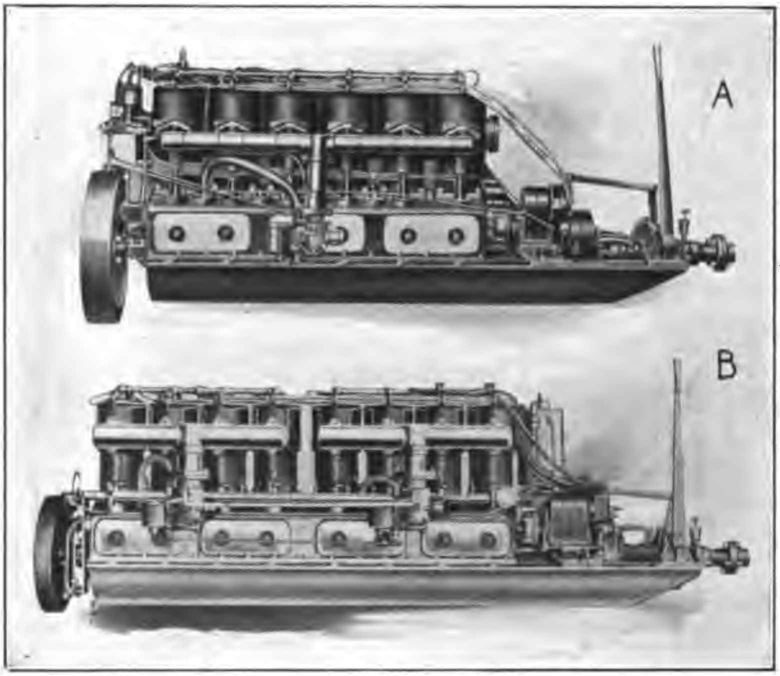
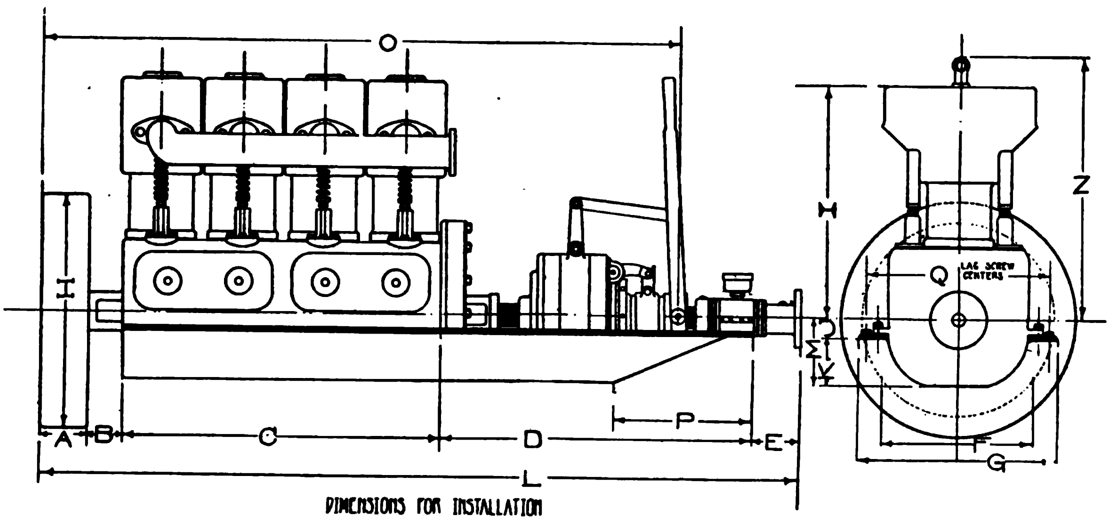


FIG. 287.—Showing construction of six and eight-cylinder Van Blerk marine motors.

gallon capacity and two gear pumps. One pump takes the oil from the reservoir and forces it through the main bearing and into the hollow crankshaft, from which it is discharged through the connecting-rod bearings. The oil is under pressure, which is shown by a gauge, and any desired pressure can be secured by altering the adjustment of the safety valve located on the oil reservoir. The other gear pump takes the oil from the crank case and returns it to the reservoir through a stream, which is



Type	Number Cylinders	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q
B-2	Two	3¼"	3½"	13"	27½"	3"	12"	17"	19"	20⅛"	¾"	5¼"	51¼"	6½"	22¾"	43"	12"	16¼"
B-4	Four	3¼"	3½"	27"	27½"	3"	12"	17"	19"	20⅛"	¾"	5¼"	65½"	6½"	22¾"	57"	12"	16¼"
B-6	Six	3¼"	3½"	41"	27½"	3"	12"	17"	19"	20⅛"	¾"	5¼"	79½"	6½"	22¾"	71"	12"	16¼"
C-4	Four	3¼"	3½"	27"	27½"	3"	12"	17"	17"	20⅛"	¾"	5¼"	65½"	6½"	22¾"	57"	12"	16¼"
C-6	Six	3¼"	3½"	41"	27½"	3"	12"	17"	16"	20⅛"	¾"	5¼"	79½"	6½"	22¾"	71"	12"	16¼"
C-8	Eight	3¼"	3½"	56¼"	27½"	3"	12"	17"	16"	20⅛"	¾"	5¼"	94"	6½"	22¾"	86¼"	12"	16¼"

FIG. 288.—Dimension sketches giving all necessary information for installation of the Van Blerk marine motor.

large enough to permit free passage of oil but which keeps out foreign substance. The bearings always receive cool oil, which is very important on high speed racing motors. Ignition is by Bosch high tension magneto, the two spark dual system being employed on the high speed motors.

The complete power plant shown at Fig. 286 includes a reverse gear and clutch which is secured to the engine base and is the four-cylinder B-4 motor, having a bore of 5 inches and a stroke of 6 inches. It will develop 30 horse-power at the normal speed of 600 revolutions per minute and weighs about 1000 pounds with cast-iron crank case. The motor shown at the top of Fig. 287 is the B-6 having six cylinders, 5-inch bore and 6-inch stroke. This delivers 40 horse-power at the normal speed of 600 revolutions per minute and weighs about 1400 pounds with cast-iron crank case. The form shown at Fig. 287 B is the C-8 or special 8-cylinder racing motor. In this, the cylinders are 5½-inch bore and 6-inch stroke. The range of speed is from 300 to 1500 revolutions per minute. It will deliver 130 horse-power at the normal speed of 1200 revolutions per minute. With an aluminum crank case this motor weighs but 1300 pounds, or considerably less than the medium duty motor of but 40 horse-power shown at *A*. Diagrams showing the principal dimensions of the Van Blerk motors are given at Fig. 288.

ROBERTS TWO-CYCLE MOTORS

While the Roberts motors are made in many forms, they are not radically different from the other two-cylinder motors previously described as relates to the small models. The Roberts motor shown at Fig. 289 is a six-cylinder form rated at 60 horse-power. The engine is of the three-port type and utilizes two carburetors, one for each set of three cylinders. A feature of the Roberts motors is a safety cellular by-pass which consists of alternating flat and corrugated plates of thin sheet metal placed in usual open by-pass to form a preventative of back firing. This phenomena, which is commonly termed a "base explosion," is caused by the flame of the explosion in the cylinder reaching the new charge in the crank case, through the usual open by-pass. When the cellular by-pass is employed, the heat of the flame is prevented from reaching the charge in the base and prevents the

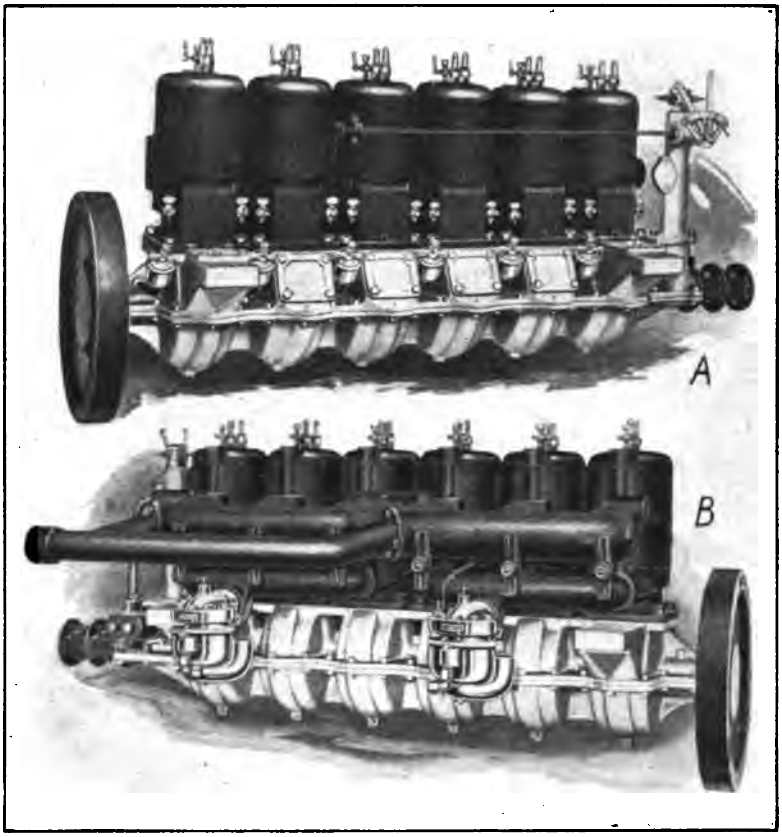


FIG. 289.—The Roberts six-cylinder, three-port, two-cycle marine engine for speed boats.

base explosion. It is said that in the cellular by-pass the combined area of the tubes is equal to the required area and that there is no choking effect due to their use. It is said that these corrugated plates also materially assist in the vaporizing of the gas if any liquid particles of fuel are contained therein. The plates are raised to a temperature of about 150 degrees F. because of the heat of the explosion, and this heat assists materially in breaking up the liquid particles into vapor or gas. The type shown is intended for use in speed boats, though a lighter form operating on the same principle is manufactured for speed boats and areoplanes where extreme lightness is desired.

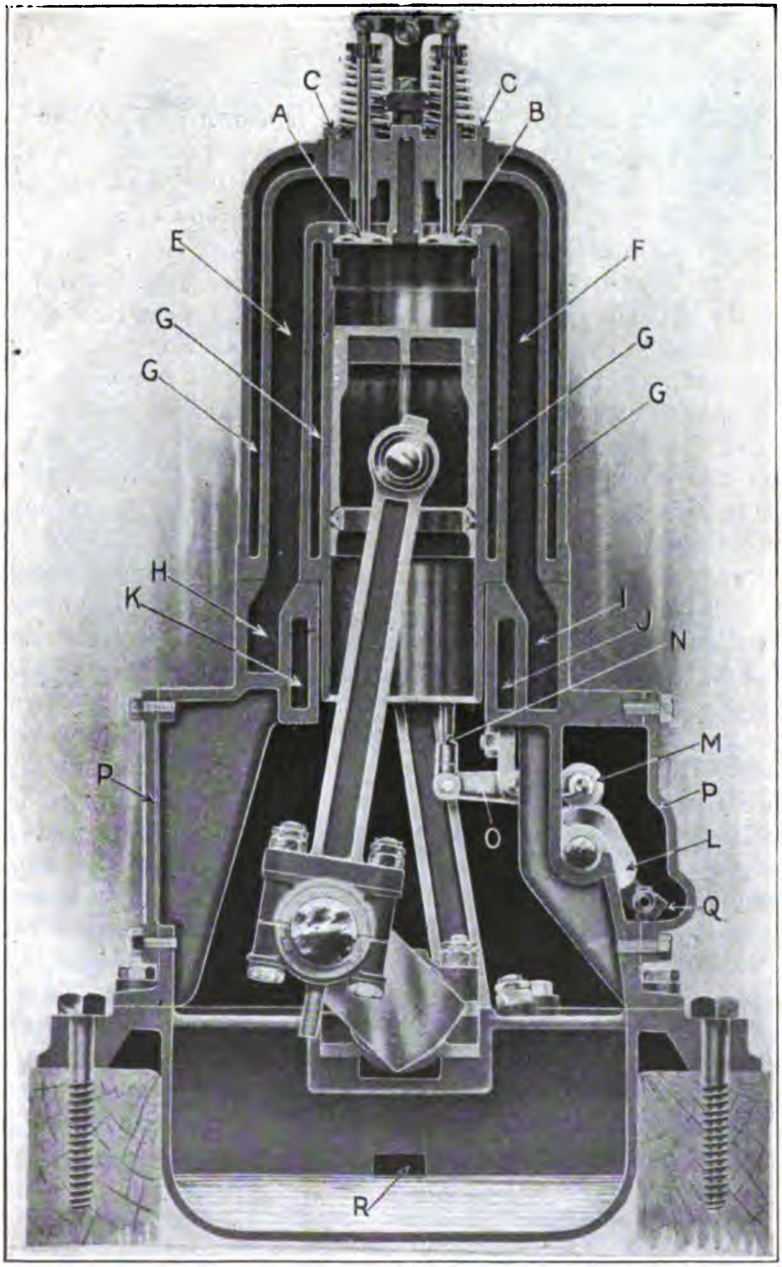


FIG. 290.—Sectional view through cylinder of Ralaco four-cycle marine motor having completely water-jacketed inlet and exhaust gas passages integral with the cylinder casting.

THE RALACO FOUR-CYCLE MOTOR

The Ralaco four-cycle engine, which is shown at Fig. 290, is especially adapted for marine use, inasmuch as the valve operating mechanism is entirely enclosed and both inlet and exhaust passages are properly water-jacketed. This produces an exceptionally smooth cylinder casting. In the sectional view, Fig. 290, the valves A and B are located in cages which are placed directly into the cylinder head in such a way that they open right into the combustion chamber. Both valves are operated by a centrally fulcrumed rocker arm, which is rocked by a tappet rod N worked by a lever O carrying the roll M which bears on the combination cam L. This cam is so formed that it will permit the roll M to drop into a depression when it is required to open one valve and to ride on the cam profile when it is necessary to operate the other valve. Even if the valve stem should break the valve is prevented from dropping into the cylinder by a ridge of metal below it which also forms a stop to prevent the piston being forced into the cylinder so far that the top piston ring will not be permitted to spring out in the counter-bored portion of the cylinder. The valve cages open or communicate with the cored passages E and F which are surrounded by the water-jacket spaces GG. One of these is employed for the exhaust gases, the other for the inlet or fresh mixture. Attention is directed to the large hand hole cover P which may be removed to permit ready access to the connecting-rod adjusting bolt, or the main bearings. Another hand hole cover on the camshaft side of the motor, also marked P, gives access to the valve operating mechanism. With a water-jacketed inlet manifold it is possible to thoroughly vaporize the fuel gas on account of the heat of the jacket water which promotes efficient action even with low grade fuel. The water-cooled exhaust pipe is also advantageous because the exhaust gases are cooled and their volume and pressure reduced by the reduction in temperature so that proper silencing is obtained with minimum back pressure.

PROPELLERS—DESIGN AND SELECTION

The manufacturers of engines usually supply a wheel of the best size and style to suit a particular engine, in order to give the best results, as it requires a propeller of a certain size and pitch to give the desired number of revolutions to develop the horse-power of the engine; and the user will do well to take their regular equipment, and then, if not satisfied with results, experiment for his own gratification. It should be remembered that when the size and pitch of propeller are increased the speed of the engine is decreased and the horse-power is reduced. The propeller should always be entirely submerged; the deeper it is in the water, the greater the hold it will take on the water. For high-speed or very high-speed engines the universal opinion is that a two-blade propeller gives the best results for engines below 15 horse-power. The novice who has his own ideas about propelling and propellers should remember, if a propeller screws

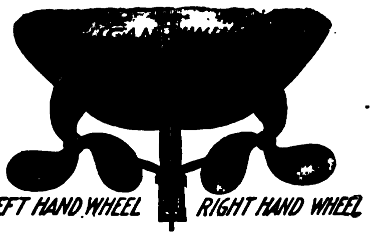


FIG. 291.—Right-hand and left-hand propeller wheels.

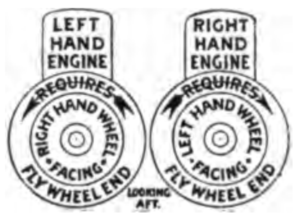


FIG. 292.—Right and left engines.

through the water eight miles per hour and there is no slip (but there is), it would be necessary to increase the speed of the propeller or use a greater pitch or a larger wheel to get more speed out of the boat. This cannot be done with the same power, of course admitting that, according to design and build of the boat, some styles, sizes and pitches will give better results; this is purely a question of experimenting, and a novice will be more apt to get best first results by the manufacturer's selection than by his own. In ordering an engine outfit or a propeller, always state the probable size of the boat, giving style of boat, length, beam, and draught, and whether heavy or light build.

HOW TO TELL A RIGHT-HAND FROM A LEFT-HAND PROPELLER

Right- or Left-hand Engine, Fig. 292.—In facing the flywheel looking aft, if top of flywheel turns from the right to the left, a right-hand propeller is required; if top of flywheel turns from the left to the right, a left-hand propeller.

Right or Left Propeller, Fig. 291.—In standing aft of the stern of boat, facing the bow, a right-hand propeller enters the water, turning to the right, just as a right-hand screw. A left-hand propeller enters the water, turning to the left, similarly to a left-hand screw, both taking the water on the back or flat side of the blades. The crowning side of the propeller should be next to the boat, the flat side or working surface aft.

THE TWENTIETH CENTURY SPEED PROPELLER

This propeller, manufactured by Michigan Wheel Co., Grand Rapids, Mich., is designed for speed. It takes the water at the shaft, and gives a continuous push to the extreme end of each blade. It may be used on all kinds of boats for racing and pleasure, particularly on light boats; and its design is such that it will pass over logs, rocks, etc., with less liability to break or bend than the ordinary propeller, and through weeds without liability to foul, as it is almost a weedless propeller. The width of the blades is one-fourth the diameter of the propeller. The following table gives the diameter, pitch, and horse-power per 100 and 400 revolutions.

Two-Blade.				Three-Blade.			
Diameter, Inches.	Pitch.	Horse-power. 100 Rev.	Horse-power. 400 Rev.	Diameter, Inches.	Pitch.	Horse-power. 100 Rev.	Horse-power. 400 Rev.
10	14	.3	1.25	12	16.8	.8	3
11	15	.4	1.50	14	19.6	1.	4
12	16	.5	2.	16	22.4	1.75	7
13	18	.6	2.40	18	25.2	2.5	10
14	19	.75	3.	20	28.0	4.	16
16	22	1.25	5.	22	30.8	5.	20
18	25	2.	8.	24	33.6	7.	28
20	28	3.	12.	26	36.4	9.	36
22	30	4.	16.	28	39.2	10.	40
24	33	6.	24.	30	42.0	12.	48
				33	46.2	14.	56
				36	50.4	16.	64
				40	56.0	18.	75
				44	61.6	22.	90
				48	67.2	24.	100

In calculating the size of propeller required, if not shown

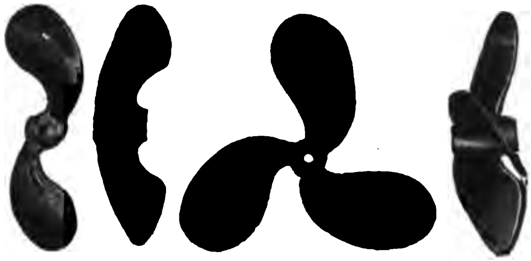


FIG. 293.—Twentieth century speed wheel semi-weedless.

in the tables, it will be seen that a 1 horse-power engine at 100 revolutions requires a 16-inch diameter (see Towing Propellers) propeller, with 17.6-inch pitch. A 4 horse-power at 400 revolutions requires the same propeller. An 8 horse-power engine at 800 revolutions would require a propeller of about the same size as a 1 horse-power at 100 revolutions. The pitch of a propeller is the distance it would advance at one revolution if turning in solid material, as a screw in metal.

SELECTING PROPELLERS

If an engine is designed for and does develop a certain horse-power at 350 revolutions, and the propeller is not properly selected, so that the engine will turn up only 250 revolutions, then the engine is not developing the rated horse-power; in order to develop its full power such a diameter and pitch as will turn at 350 revolutions, should be used. If designed to run at 700 revolutions and it turns up only 500, the propeller is not suited for the engine, as above. Either the slow- or high-speed engine may be run at a slower speed if desired with less power. On the other hand, if your engine is designed to run 700 revolutions and it turns up 1,000 revolutions, better results would be obtained, with most boats, if a propeller with greater pitch to bring it down to 700 or 800 revolutions were used. Should the boat be a heavy, wide-beam one, a larger

FIG. 294.—The Michigan weedless speed propeller wheel.

propeller with less pitch is preferable to a smaller one with greater pitch, and usually a two-blade propeller will give better results with most boats, particularly when there is a wide dead-wood, or for auxiliary use. For speed boats under 15 horse-power, the two-blade is undoubtedly the best. More solid water is obtained with two blades than with three.

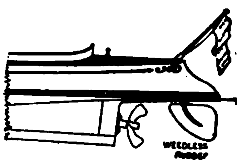


FIG. 295.—Showing weedless rudder.

WEEDLESS PROPELLER WHEELS

This is a perfectly weedless speed propeller. In operation, it rotates the forward edge of the blades through the water in a curve at an acute angle to that edge, thus cutting the water with a sliding motion. Any weeds, grass, or other obstruction engaged thereby will slide along the forward edge toward the periphery, and be discharged from the outer edge of the blades. The propeller is thus made self-cleaning and will not become fouled. This propeller is greatly appreciated by hunters among the marshes and weeds, making motor-boating a pleasure in lakes, rivers, and bays, where weeds would otherwise be troublesome.

It is easily attached to any propeller shaft, and should be placed close to the stern bearing, a weedless rudder, shown in Fig. 295, being fixed as far back of the propeller as possible. Set-screws, if used on the propeller, should be flush with the outside of the hub.

MICHIGAN WEEDLESS TWO-BLADE PROPELLER.

Diameter, Inches.	Pitch, Inches.	Horse-power. 100 Revolutions.	Horse-power. 400 Revolutions.
10	14	.20	.75
11	15	.25	1.
12	16	.40	1.50
13	17	.45	1.75
14	19	.60	2.
16	22	1.00	4.
18	25	1.50	6.
20	28	2.25	9.
22	30	3.	12.
24	33	4.	16.
26	36	6.	24.

PROPELLERS FOR HEAVY LOADS

It often happens that, with a heavy boat, or extreme beam, or deep draught, when the propeller is deeply submerged, the propeller usually sent out by the manufacturers with the ordinary outfits does not give the best results, and the engine fails to run near the number of revolutions intended in the average light boat. In this case the pitch of the propeller should be reduced or a propeller with about 3 inches less pitch should be substituted.

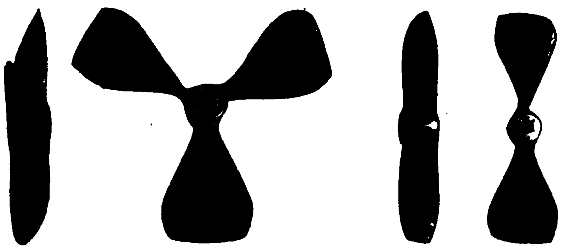


FIG. 296.—Michigan towing propeller wheels.

For instance, if employing a 15 horse-power engine with propeller 20 inches in diameter and 27-inch pitch, with above results, use in its place a 20-inch propeller with 24-inch pitch. Remember this is a suggestion to work by, and not a set rule. Some boats may require still less pitch. It depends upon how fast the engine runs with the 27-inch pitch, etc. Taking into consideration the speed the engine is intended to run to develop its rated horsepower, by ascertaining the revolutions the propeller is making, it can be readily determined whether one of more or less pitch, or of greater or less diameter is needed.

TOWING PROPELLERS

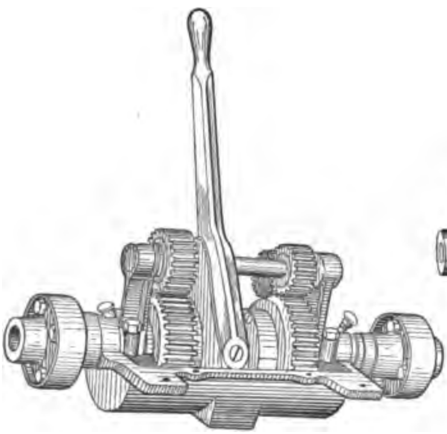
The "Michigan" is specially designed as a towing propeller and is designed on the true screw pitch basis (see Fig. 296), allowance being made for slippage at the hub. It is adapted for heavy boats with powerful engines requiring a large blade area. The width of the blade is one-third its diameter. A portion of the back of the blade is flat, to give power in backing. This propeller may also be employed to advantage with high-speed engines, with a heavy, deep-draught, extreme beam, requiring a larger diameter than that of the speed propeller. The following table gives diameter, pitch, and horse-power for propellers of two and three blades:

Two-Blade Towing Propeller.				Three-Blade Towing Propeller.			
Diameter, Inches.	Pitch, Inches.	Horse-power. 100 Rev.	Horse-power. 400 Rev.	Diameter, Inches.	Pitch, Inches.	Horse-power. 100 Rev.	Horse-power. 400 Rev.
16	17.6	1.	4	16	17.6	1.50	5
18	19.8	1.5	6	18	19.8	2.25	9
20	22.0	2.25	9	20	22.0	3.	12
22	24.2	3.	12	22	24.2	4.	16
24	26.4	4.	16	24	26.4	6.	24
26	28.6	6.	24	26	28.6	7.5	30
28	30.8	7.5	30	28	30.8	9.	36
30	32.3	9.	36	30	32.3	10.	40
				33	36.3	12.	48
				36	39.3	15.	60

Should a propeller be found too large to enable the engine to turn up the required number of revolutions, trim off the outer edges a little at a time. Care should be used to take the same amount off each blade.

REVERSING GEAR FOR MARINE ENGINES

A five-spur gear-reversing clutch (Fig. 297) is much in use on marine engines in which the gears are constantly oiled by the



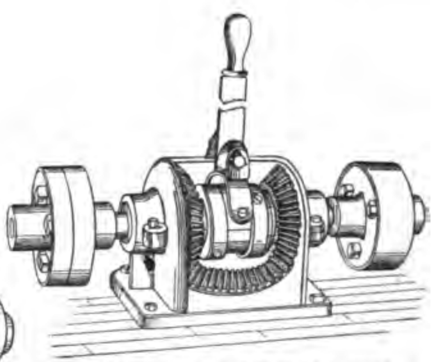


FIG. 298.—Brodie reversing gear.

FIG. 297.—Reversing gear.

dipping of the shaft gears in the oil-trough below. The gear on the wheel-shaft is fixed to the shaft and driven for forward motion by a friction-clutch sleeve feathered on the end of the motor-shaft. For reversing, the yoke-lever is thrown over and engages the feathered sleeve in the clutch of the idle gear on the motor-shaft, when the back-motion is transmitted through this reverse-gear train to the propeller-shaft. The clutches are of the expanding ring type.

A simple and effective reversing gear for marine motors is illustrated in Fig. 298. It consists of three bevel gears and a clutch-sleeve; the sleeve is on the motor-shaft with a traverse

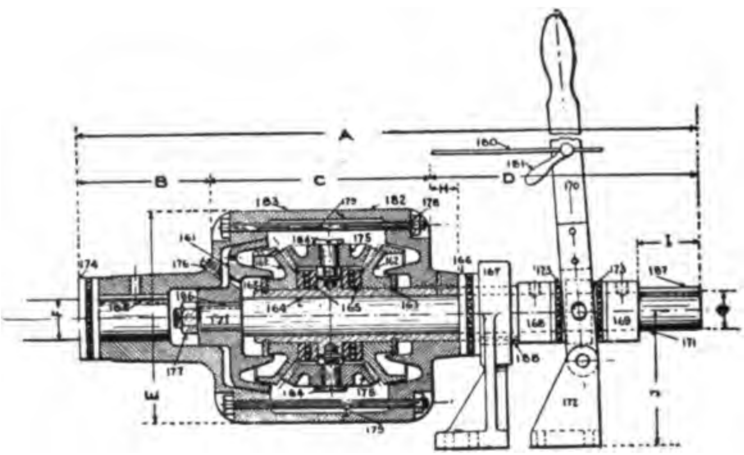


Fig. 299.—Mietz & Weiss reversing clutch.

spline and friction-drive on the shaft bevel wheel. The bevel wheel on the propeller-shaft is fixed to the shaft, while the bevel wheel on the motor-shaft runs loose.

The third bevel wheel runs on a pin fixed to the box-frame. When the lever is thrown forward the sleeve is thrust against the friction surface of the propeller gear and the other bevel gears run loose. When the lever is thrown to the centre all the gears are in repose. For the back motion of the propeller, the lever is thrown back and the sleeve engages the friction of the loose motor gear and reverses the propeller through the action of the idler gear.

In Fig. 299 is shown a sectional detail of the Mietz and Weiss reversing friction-clutch as used on their marine oil-engines. It consists of an oil-tight cast-iron drum made in two sections and is keyed to the shaft. Inside of this drum is a steel stub-shaft on the inner end of which is keyed a friction-driving cone. There are two friction-disks with beveled gears and interposed pinions on a bronze sleeve, which is prevented from rotating by a key in its bearing, screwed to the base of the engine. This bronze sleeve is free to slide longitudinally. The two friction-disk bevel gears rotate around this sleeve and are held in place by means of split-washers. On the forward motion the stub-shaft, to which the propeller-shaft is coupled, is brought forward by means of the lever, and the friction-driving cone on its inner end engages the inner surface of the drum, imparting a forward motion direct from the engine-shaft to the propeller, the thrust of the propeller thus acting directly on the friction-cone and on the thrust-ball collar at the engine-bearing.

On the reverse, the lever is thrown back, thus releasing the forward thrust of the driving cone, and bringing its inner friction surface directly in contact with the friction surface of the first bevel gear, while the second gear, engaging the inner friction of the drum, imparts through the interposed pinions a reversed motion to the sub-shaft and thence to the propeller. A central position of the lever disengages the friction surface from the drum entirely, so that the engine may continue to run idle while the propeller is at rest. The thrust of the propeller on the forward motion exerts its entire force directly against the friction surfaces, without the assistance of toggle-levers or cams, the whole connection from the engine to the propeller acting as one shaft. On the reverse, the tension of the propeller upon the shaft exerts its force against the reverse-gearing and inner-driving surface of the drum in the desired direction. The lever must be locked in its forward, central, or reversing position.

CHAPTER XVII

KEROSENE, DISTILLATE, AND CRUDE-OIL MOTORS

The incentive to explosive motor design in the line of economy of power has been the means of producing remarkable results in the adaptation of the use of the cruder and cheaper fuel-oils for motive power. The rise in the cost of gasoline gave an impetus to experiments for utilizing the heavier petroleum products, and kerosene and distillate came into successful use, and finally crude petroleum in its cheapest form is at the head of fluid fuel as an all-around and portable element of power and obtainable the world over. For stationary motive power there is a further economy in the producer and blast-furnace gases that is greatly expanding the field of operation for the explosive motor and will continue during the coming years, when its power, like that of steam, will become stationary in its economical progress. The heavy-oil-engine differs from the gasoline-engine in essential details of its mechanism, due to the different natures of the two fuels. Low-grade fuel being less volatile, no carburetor is used to convert the fluid into gas. The oil is introduced into the cylinder as a spray, mixed with air, and is changed into a gaseous condition within the cylinder before ignition occurs by means of heat which must be within well-defined limits. If the vaporous fuel comes in contact with too great a heat the petroleum is disintegrated, its hydrogen escapes, and its carbon deposits in stone-like scales upon the cylinder-head, while too little heat will not produce the gaseous condition necessary to perfect combustion.

A simple engine designed to burn low-grade fuels is shown in Fig. 300, and Fig. 301 in section, showing the position of the piston when ignition is taking place, and when the exhaust-gases are escaping and the fresh charge entering. When the piston has risen nearly to the end of the upward stroke the air-inlet A is uncovered, and as there is a partial vacuum in the air-tight crank-case C, the air rushes in. As the piston descends, impelled by the ignited gases, the air in the crank-case is compressed, the pressure extending to the passage D. The descending piston finally uncovers the exhaust-passage, through which the inert gases of combustion escape, impelled first by the pressure remaining in the cylinder, and then by the rush of air through the inlet E, which is opened after the exhaust, as will be noticed in Fig. 300. The moment the inlet E is opened the compressed air in the crank-case and air-passage rushes into the cylinder, driving out the remaining gases, except a small amount left in the cavity

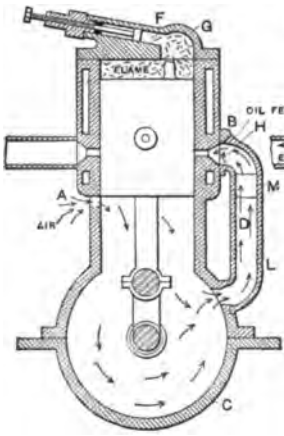


FIG. 300.—Ignition.

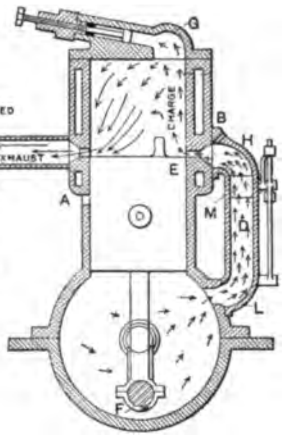


FIG. 301.—Exhaust.

holding the firing-plug F. This small residue plays an important part in the successful operation of the engine, for it keeps the new charge away from the red-hot firing plug until the proper time for igniting. As compression takes place above the piston during its upward stroke, the new charge forces the burned gases farther into the cavity, until the new charge comes in contact with the plug and is ignited. The plug has a screw-stem, by which its position in the cavity may be adjusted by a nut on the outside of the cylinder-head to correspond to the power required. Withdrawing the plug into the cavity delays ignition, thus furnishing greater power. Advancing it nearer to the opening of the cavity advances the ignition, and less power is developed.

The charge of fuel is introduced through the inlet B, either from a pressure-tank or by the suction created by the partial vacuum of the crank-case. A check-valve prevents the oil from returning through B after it has been admitted. The oil as it enters the engine is received on a gauze screen H, and by capillary attraction forms a thin film upon it until the entrance E is uncovered and the air rushes into the cylinder, passing through

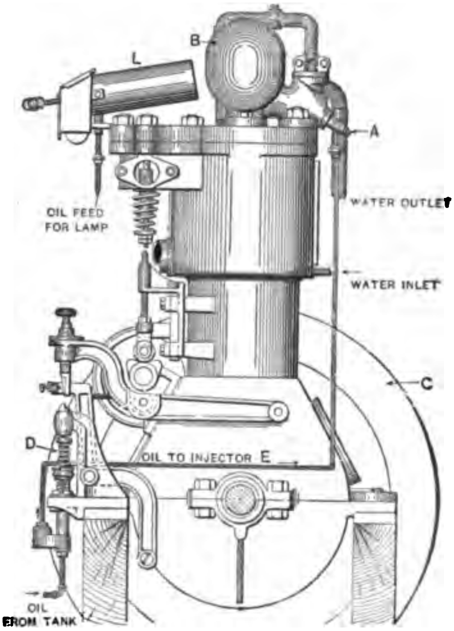


FIG. 302.—New York kerosene marine motor.

the gauze, taking the oil with it in a very fine spray, so fine that if permitted to escape into the air it would float. The intermingled air and oil-spray passing up into the cylinder strikes the hot cylinder-head at G. The heat of the metal is not enough to ignite the oil, nor to cause it to "crack," *i.e.*, give off hydrogen and deposit carbon. Instead, this heat converts the oil and air into a gaseous mixture, which is maintained by the heat of compression until the moment of igniting. The ignition-plug replaces the torch, which need be applied only in giving to the plug its initial red heat for the first discharges, after which the heat of ignition is all that is necessary. For changing the speed of the engine a butterfly-valve, shown at M, Fig. 300, in the air-transfer box, is employed for throttling and is automatically controlled by the governor, which is of the usual type.

The engine shown at Fig. 302 was one of the earliest American designs to use fuel injection in connection with hot head ignition. It is provided with a combustion-chamber B, preparatory to starting. The air inlet-valve and the exhaust-valve are actuated by cams in the ordinary manner on a secondary shaft, the engine being of the four-cycle type. The injection of oil is accomplished by the pump D, actuated by one arm of a rock-lever, which is oscillated by a cam on the secondary shaft. The charge of kerosene is regulated by the stroke of the pump, which is controlled by a lever in the marine motors and by a governor in stationary motors.

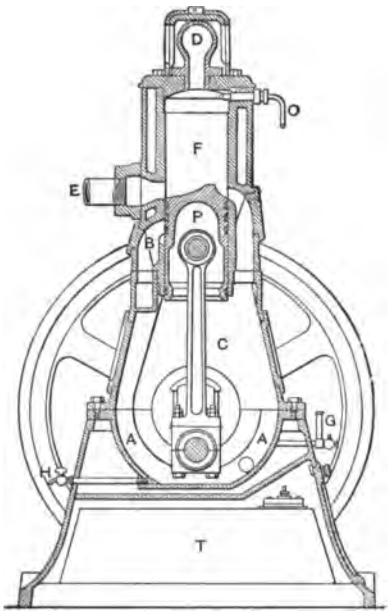


FIG. 303.—Section of oil-motor.

The injection of the oil is in a very fine stream under considerable force, by which it is atomized in the hot-chamber B. The blow-pipe lamp L is made permanent in the stationary engines with an air-pressure combination for gas or gasoline. In the marine motors a kerosene-torch is used which heats the combustion-chamber ready for starting the motor in about five minutes. The clearance is so adjusted that the compression is carried to eighty-five pounds, at which point, or just before the piston reaches the dead centre, the charge of oil is suddenly injected and vaporized by the heat of compression and the walls of the vaporizing-chamber. By the late injection of the oil pre-ignition is impossible and the atomizing of the oil being instantaneous is followed by its perfect vaporization in its mixture with the hot air. The firing of the charge of partially mixed oil-vapor and air is exact and instantaneous as to time, and owing to the small volume of the clearance space carries the pressure up to about 190 pounds, and by continuous combustion during the impulse-stroke gives a higher expansion-curve than is due to the adiabatic line, and showing by the indicator card a mean effective pressure of seventy-four pounds. This exceeds the usual mean pressure in gas and gasoline explosive motors.

Another form of engine operating on the two-stroke principle is shown at Fig. 303. It will be noticed that the ignition is accomplished by the usual ignition hot-dome D, at the upper end of the cylinder, the dome being protected by a damper-cap to prevent heat radiation after the engine is started. A concentric cap fits over the inner cap. When both apertures coincide, the heating-lamp for starting is placed inside; after starting the outer cap is rotated till the apertures are covered. The operation of the engine is as follows: the ignition-dome D is heated for five minutes or more by a Primus kerosene blue-flame torch, then the handle of a small oil-pump is operated a few times, to force the oil up from the tank T through the nozzle O into the cylinder F. One or two quick turns of the fly-wheel are given, then the engine starts.

On the up stroke of the piston P, air is drawn in through two holes A in the base, and follows the piston through the port B into the crank-case C as soon as the piston uncovers the port. On its descent the piston slightly compresses this air in the crank-case until its upper end uncovers the exhaust E and also the air-inlet, then the exhaust-gases pass out of E, and by the curved top of the piston the air from the crank-case is projected upward at the same time into the cylinder and locked there upon the upward stroke of the piston P, which closes the air-inlet and exhaust-port E. The air in the cylinder is then further compressed and heated by the continuation of the up stroke of the piston, and just as the latter is about to descend a minute quantity of kerosene is injected by the oil-feed pump and is immediately vaporized and mixed with the air, forming an explosive mixture that is in turn ignited by the hot dome D, the explosion driving the piston downward. The combustion is so perfect that the cylinder always remains clean and the piston is never clogged by soot. There is thus a positive entrance of the air and oil to the

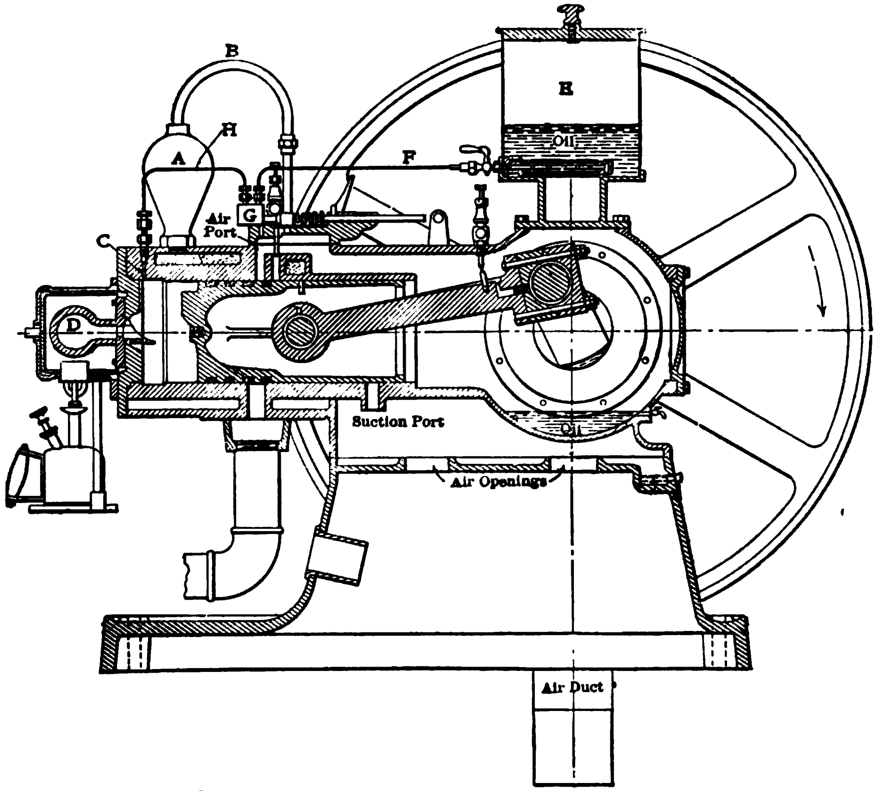


FIG. 304.—Section of steam, air, and oil-engine.

cylinder in regular sequence. G is an oil-well for one of the main bearings, and H is a faucet for drawing off the oil collecting in the bottom of the crank-case. An eccentric on the main shaft with a variable throw, regulated by a simple governor, changes the stroke of the oil-feed pump to suit the load. The engine responds very quickly to the varying quantities of fuel it receives, and the governing action is consequently positive and very close. This results in high efficiency, and makes it possible to obtain a brake horse-power with 0.7 to 0.8 pound of oil, or a little less than a pint.

MIETZ AND WEISS OIL-ENGINE.

In Fig. 304 we illustrate in section the details of the new style Mietz and Weiss kerosene-oil engine. The new feature is the use in the cylinder of steam generated in the water-space of the cylinder-jacket, which is different from all other explosive motors in principle and effect. It utilizes a large part of the heat ordinarily lost through the cylinder-walls and cooling-water, and considerably reduces the trouble from deposition of carbon in the cylinder (probably by the dissociation of the water-vapor furnishing oxygen to the hot particles of carbon). The new parts are a small steam-dome A, a short steam-pipe B, connecting the steam-dome with the air-port, where it is admitted with the charge of air into the cylinder, when the piston is at the forward end of the stroke. When the piston reaches the correct position, a small quantity of oil is drawn by the oil-pump G through the pipe F from the oil-tank E, and delivered through the pipe H to the opening C, where it falls upon the lip of the red-hot igniter-ball D, and is exploded along with the air and by its heat dissociates the steam, which adds further elements of combustion to the unconsumed carbon; thus increasing the mean pressure of the expansion-curve.

An efficiency is claimed for the steam, air, and oil mixture of from fifteen to twenty per cent. higher than for the oil and air mixture alone, the total thermal efficiency in a test being forty-four per cent. with a compression pressure of 100 pounds gauge, and 170 pounds explosion pressure by gauge, using one pint of oil per brake horse-power per hour. The tests were made on a fifteen horse-power engine of the two-cycle type in which the air is drawn into the crank-case, during the compression-stroke through the suction-port from the engine-base; is compressed during the impulse-stroke and passes through the side-port, taking a portion of steam in its passage. Since there is no circulation through the water-jacket, the level of the water in the jacket is maintained at a constant level by a float-trap in a side compartment, and only water is fed to equalize the evaporation, with a water temperature just below the boiling-point and which has been found to be the best working temperature for an explosive motor. In Fig. 305 we illustrate a sectional view of the Mietz and Weiss vertical marine oil-engine with reference figures showing the detail parts. Kerosene oil, the most economical and conveniently obtained fuel for explosive motor service, has been the incentive for bringing the oil-engine to its utmost perfection in design and working

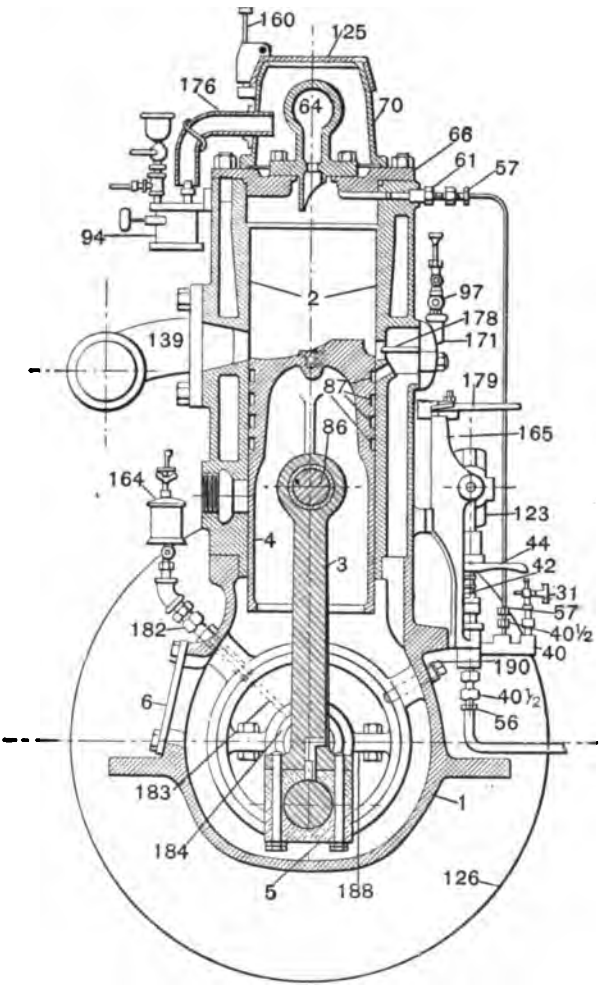


Fig. 305.—Mietz & Weiss marine oil-engine.
References: 34. Ignition-ball. 70. Mantle. 125. Damper. 160. Damper-regulator. 176. Blow-pipe. 94. Lamp. 61. Oil-injections. 179. Regulator-handle. 42. Oil-pump plunger. 44. Oil-pump handle. 31. Air-relief cock. 40½. Suction and pressure oil-valves. 123. Centrifugal governor operating the throw of the pump-plunger. The other parts shown are self-explanatory as to the general arrangement of the two-cycle engine.

power, and for marine motors, safety as well as economy has made it of primary importance for launch, yacht, and auxiliary service.

THE HORNSBY-AKROYD OIL-ENGINE

This engine is of English origin, the invention of Mr. H. Akroyd Stuart, who has lately made many improvements in its design by perfecting the charging-mixture. These engines are of the four-cycle compression type, using kerosene and any of the heavy mineral oils as fuel. In Fig. 306 is shown a sectional elevation, details of design of the cylinder, piston, combustion-chamber, and its case. It may be noticed that the combustion-chamber is made in two parts, flanged together, so that by a special water-jacket the front half is kept cool and to limit the firing-plane in the combustion-chamber to a definite position. The oil-reservoir, located in the base of the engine, is partitioned to allow of traversing the intake-air over and around the oil to take any vapors or odors from the oil and constantly sweep them into the cylinder.

An extension of a chamber from the cylinder-head, somewhat resembling a bottle with its neck next to the cylinder-head, performs the function of both evaporator and exploder. Otherwise these engines are built much on the same lines of design as gas and gasoline-engines, having a screw reducing-gear and secondary shaft that drives the governor by bevel-gear. The bottle-shaped extension is covered in by a hood to facilitate its heating by a lamp or air blow-pipe, and so arranged as to be entirely closed after the engine is started, when the red heat of the bottle or retort is kept up by the heat of combustion within. The narrow neck between the bottle and cylinder, by its exact adjustment of size and length, perfectly controls the time of ignition, so that of many indicator cards inspected by the writer there is no perceptible variation in the time of ignition, giving as they do a sharp corner at the compression terminal, a quick and nearly vertical line of combustion, and an expansion curve above the adiabatic, equivalent to an extra-high mean engine-pressure for explosive engines.

The oil is injected into the retort in liquid form by the action of the pump at the proper time to meet the impulse-stroke, and in quantity regulated by the governor. During the outer stroke

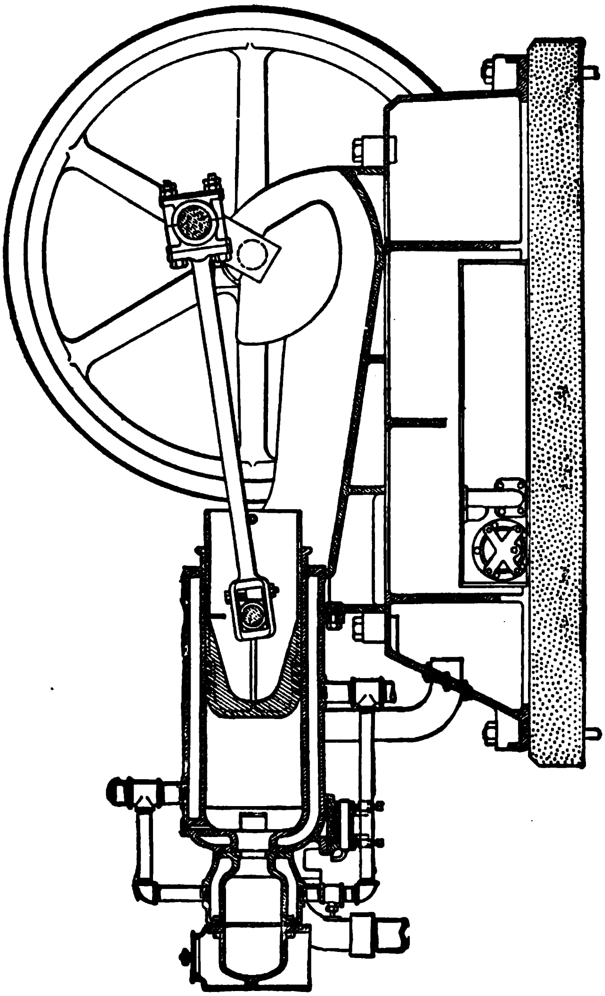


FIG. 306.—Sectional elevation, Hornsby-Akroyd oil-engine.

of the piston, air is drawn into the cylinder and the oil is vaporized in the hot retort. At the end of the charging-stroke there is oil-vapor in the retort and pure air in the cylinder, but non-explosive. On the compression-stroke of the piston the air is forced from the cylinder through the communicating neck into the retort, giving the conditions represented in Fig. 307 and Fig. 308, in which the small stars denote the fresh air entering, and the small circles the vaporized oil. In Fig. 308 mixture commences, and in Fig. 309 combustion has taken place, and during expansion the supposed condition is represented by the small squares. At the return stroke the whole volume of the cylinder is swept out at the exhaust, and the pressure in the retort neutralized and ready for another charge.

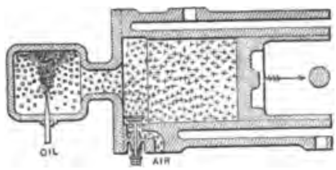


FIG. 307.—Injection, air, and oil.

It is noticed by this operation that ignition takes place within the retort, the piston being protected by a layer of pure air. It is not claimed that these diagrams are exact representations of what actually takes place within the cylinder; nevertheless, their substantial correctness seems to be indicated by the fact that the piston-rings do not become clogged with tarry substances, as might be expected. This has been accounted for by an analysis of the products of combustion, which shows an excess of oxygen as unburned air; which indicates that the oil-vapor is completely burned in the cylinder, with excess of oxygen.

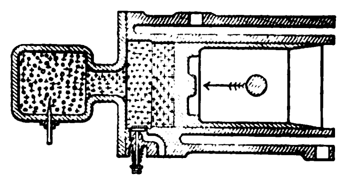


FIG. 308.—Compression.

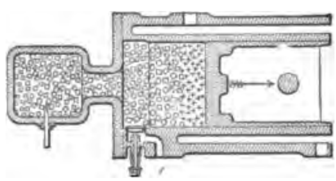


FIG. 309.—Combustion and expansion.

WEISS OIL-ENGINE

In Fig. 310 is illustrated the working detail of the Weiss kerosene-oil engine in a sectional elevation showing the conical vaporizer E D enclosed in a shell for confining the lamp flame when starting and to keep the outer walls hot when the engine

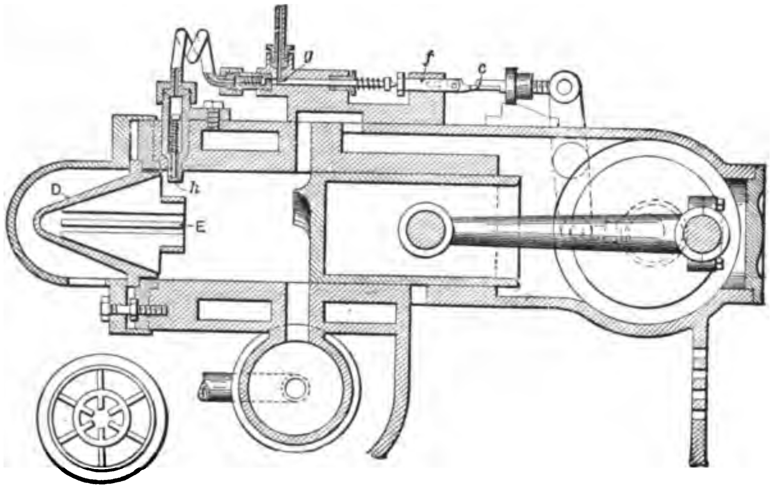


FIG. 310.—Section Weiss kerosene-oil motor.

is running. A front view of the vaporizer at the lower left-hand corner of the cut shows the extended web surface. The small spring-held oil-valve at *h* holds the oil between it and the pump intact during the impulse stroke. The small oil-pump at *g* is operated by the pick-blade *c*, with a hit-or-miss charge, governed by the momentum of a small weight sliding on an inclined plane, the amount of charge and the interruption being readily adjust-

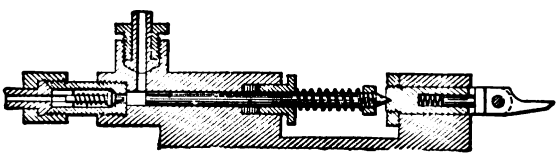


FIG. 311.—Oil-pump and pick-blade.

able. In Fig. 311 is shown an enlarged section of the oil-pump and pick-blade. The injection by the movement of the motor-piston is of pure air drawn into the crank-case by the forward motion of the piston and compressed; when at the opening of the cylinder-port at the end of the impulse-stroke, the compressed air is injected into and guided to the head of the cylinder to meet the vaporized oil in the vaporizing cone. Compression and the heat of the vaporizer fire the charge at the proper moment.

NOVEL KEROSENE VAPORIZING METHOD

A novel method of vaporizing kerosene and other low-grade fuels is outlined at Fig. 312. Kerosene is kept in a tight tank or reservoir. Pressure is put on the fuel by connecting the upper part of the reservoir with the engine crank-case and interposing a check-valve V in the pipe between them. The kerosene is drawn from the bottom of the reservoir and passes through a coil C in the combustion chamber, where it is turned into gas or vapor.

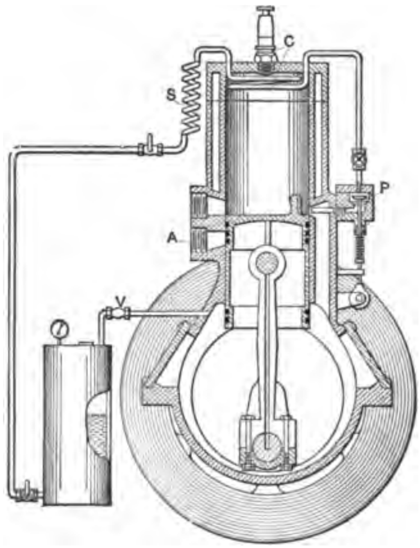


FIG. 312.—Section of motor.

While the engine is running the oil is heated to form the kerosene-vapor in the coil C and is then let into the cylinder through the poppet-valve P. This valve P is moved by a cam in such a way as to time the inlet of the gas a little later than the completing of the exhaust and a little later than the beginning of the inlet of fresh air from the crank-case. Incidentally the engine, like the old Day engine — the original two-cycle engine uses no inlet-valve to the crank-case, but uses an air-port which is uncovered by the piston at the highest point of its stroke.

In starting up, a secondary vaporizing coil S, in the supply-pipe outside the cylinder, is heated by a blow-torch. This coil S

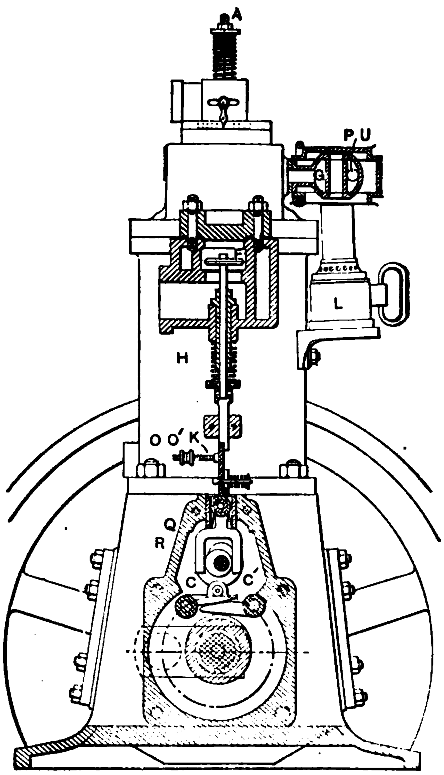


FIG. 313.—Millot engine, showing vaporizer and governor.

is kept heated only until such time as the heat from the explosions gets the coil C in condition. The advantages of using kerosene in vapor form are very pronounced. In this condition it makes a perfect mixture, free from fine drops of liquid. Such mixtures permit of much higher compression and much higher economy than is possible with oil spurted directly into the cylinder. A mixture of air and kerosene "gas" burns without depositing soot.

An air starter supplies air through a poppet-valve moved by an eccentric, and since the air must pass through a check-valve before reaching the piston, it follows that the engine changes automatically from the air-starter to fuel-burning. No vapor can reach the crank chamber from the vaporizing coil C, as the mechanically operated inlet-vapor valve P is closed during the up-stroke of the piston, and the check-valve V prevents vapor from passing to the crank-chamber from the kerosene-tank. The air-inlet port at A furnishes sufficient air at or during the terminal of the up-stroke of the piston. The air for starting is compressed in a small cylinder by a pump operated by hand or in multi-cylinder motors by the motor for storage.

THE MILLOT OIL-ENGINE

This is of French origin and has some novel features of construction. The oil, which is kept in a separate reservoir,

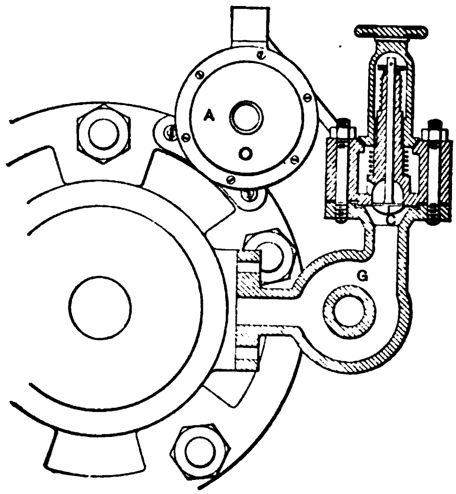


FIG. 314.—Petroleum-engine on the Millot system; top of cylinder.

comes into a chamber where it is kept at a constant level. The oil which is drawn into the engine passes through a spraying device to a very small opening which compels the oil to spurt out forcibly. This spraying is made still more active by the air coming from the valve C (Fig. 314), this valve being opened by the suction from the descent of the piston. The vaporized oil arrives by the opening P U (Fig. 313), in the gasifier G, which is a kind of cast-iron bowl kept at a dark-red heat by means of an oil-lamp with Bunsen flame. The oil in vaporized state passes through the orifice G (Fig. 313) into the compression chamber and then into the cylinder. At the end of the induction stroke, the cylinder and the combustion chamber are filled with a mixture of gas and air. The piston, rising, gives a high compression to this mixture as it can occupy a volume only equal to that of the compression chamber. The pressure of the mixture striking upon the walls of the gasifier G (Fig. 314), which are at red heat, determines the time of explosion. After a few minutes of running, the heat produced by explosion is sufficient to keep the walls at red heat and the lamp L can then be removed.

The governor is a novel feature in a vertical engine, it being of the inertia type. This consists of a stem K, fastened to the

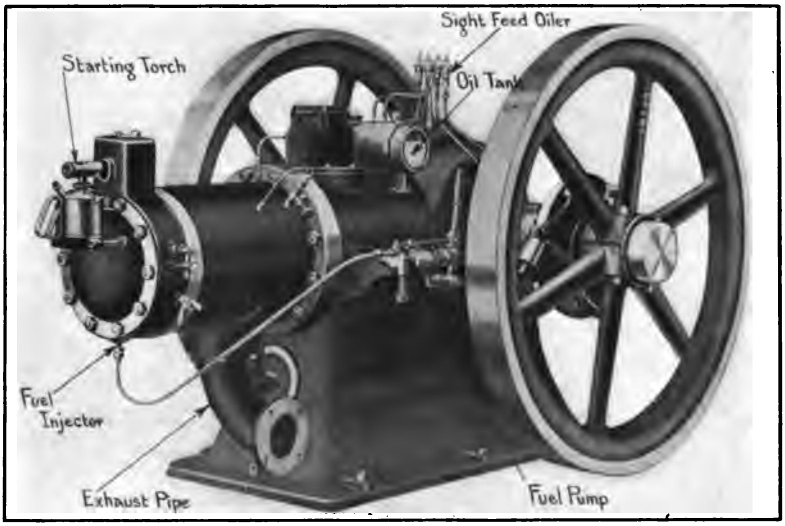


FIG. 315.—The Anderson oil-engine operating on the two-stroke cycle principle.

side of the escapement, which is pivoted at the lower end. During normal running, the pawl is held by a spring in a vertical position. The catch C has an oscillating movement given it by the lever Q, which is driven by the cam R. The tension of the spring which holds the catch is such that the inertia of the weights O Ó is not sufficient to prevent the catch from following this movement when the engine turns at its normal speed; but when this passes the proper limit, the inertia of these weights makes the catch oscillate and leave its contact with the stem of the escapement. Consequently the valve F is not raised by the escapement and there is no exhaust, the cylinder retaining the products of combustion from the preceding explosion. No explosive mixture is drawn in, therefore, and no ignition can be produced so that the motor slows down. When the engine reaches its normal speed, the governor ceases to act and exhaust commences again.

REPRESENTATIVE HOT BULB ENGINES

The illustration at Fig. 315 conveys a fair idea of the general design of the Anderson oil-engine, adapted to the use of crude oil as it comes from the ground, fuel oil, kerosene or distillate. The engine is of the two-cycle type, thereby giving an impulse to each revolution of the shaft — this insures plenty of power and uniform speed. It is simplified to the lowest possible minimum, there being no inlet or exhaust valves, cams, shafts, gears, springs or electric sparking devices. The carbon found in all low-grade oil is consumed in three different ways: 1. By excessive heat. 2. By a blast of air or introduction of oxygen into the combustion-chamber. 3. By a combination of these two. The Anderson oil engine has claims covering these distinctive principles, both as to method and process. The highest possible degree of heat is maintained in the combustion-chamber through heat of combustion and compression, and at the same time the oil is atomized to promote the most rapid and perfect combustion. By the introduction of air into the combustion-chamber, the carbon is prevented from depositing on the walls of the cylinder to any extent, the same being disintegrated and blown out through the exhaust, rendering the engine free from carbon and sticking. It is not of the high compression type of engine, the compression in the cylinders averaging about 75 pounds per square inch. The ignition is effected by compression and is entirely automatic. Provision is made for a kerosene burner furnished with each engine, to be used for a few moments before starting.

The lubricator is of the automatic compression oiler type, of large capacity. The oil for this purpose is forced by pressure to all parts needing lubrication, while the engine is in motion. The governor is of the centrifugal type, absolutely automatic in its action — operates from the main shaft direct without any belting or extra shaft, graduating the charge from 0 to full load. The speed of the engine can be changed at the will of the operator without stopping same. This governor instantaneously varies the fuel-oil pump stroke to suit the fuel requirements of all changing loads.

The Robey engine shown at Figs. 316 and 317 is an English design working very much on the same principle as the Anderson

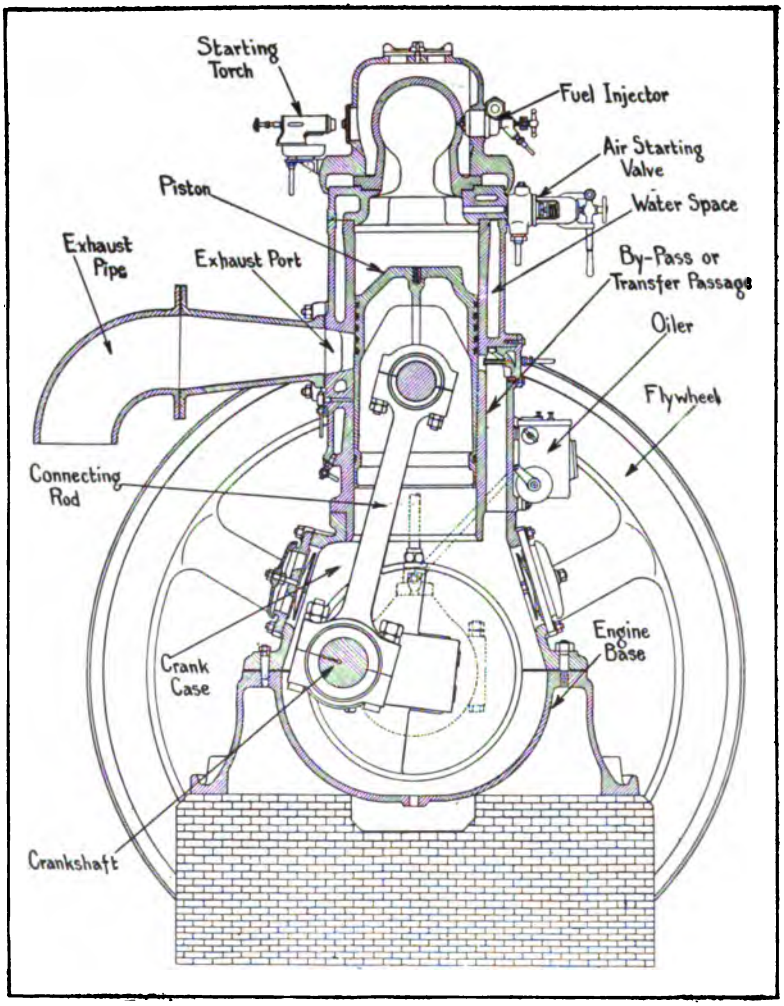


FIG. 316.—Sectional view through cylinder of the Robey crude oil-engine, employing hot bulb ignition and injections of the fuel charge.

as it is a two-cycle type. The sectional view Fig. 316 shows the arrangement of the parts very clearly and also makes the size and location of the cylinder ports clear. The other view presents the method of mounting the engine on a substantial foundation for stationary work and outlines the very simple and practical design clearly. The fuel is injected directly into the hot bulb.

Beneath the injector in the end sectional elevation is to be seen the air-starting valve. Compressed air, however, is needed on the larger models only, and even then is not made to be operated automatically by the engine; this may be in order to avoid the use of a cam or eccentric, which, on a four-cycle engine, would be a matter possessing little disadvantage, but on a two-cycle engine would lead to a great increase in the number of parts. Introduction of compressed air is effected by the perpendicular handle after the engine has been turned to starting position and must be operated in unison with the movements of the flywheel, a performance that might be awkward at first, but would soon become easy. On the larger engines water injection is employed to limit the temperature of the combustion-chamber when the engine is working under full load. Unlike most hot bulb engines, the water is not sprayed directly into the bulb, but is fed in at the top of the trunk that carries the scavenging air to the cylinder. The amount is regulated by a drip feed and needle valve in the pipe that carries the water from the jacket to the nozzle by gravity and it is introduced by the rush of the scavenging air into the cylinder. In other engines the amount of water is fed by pump and the water pumps are mounted on the same plate as the fuel pumps and the supply is obtained from the same source, cocks allowing it to be directed either to the pumps or down the gravity pipes.

Jacket cooling water is circulated by a pump driven by an eccentric near the fuel pump eccentric, but not shown in the illustrations. From the top of the jacket water is taken to the cooling apparatus, being circulated around the injector on its way, thus preventing any trouble from overheating of this part. Care has been taken that the lubrication of the piston shall be adequate, there being four leads that conduct oil to various parts of the circumference. One is situated in the path of travel of the gudgeon and so keeps this important bearing supplied. The

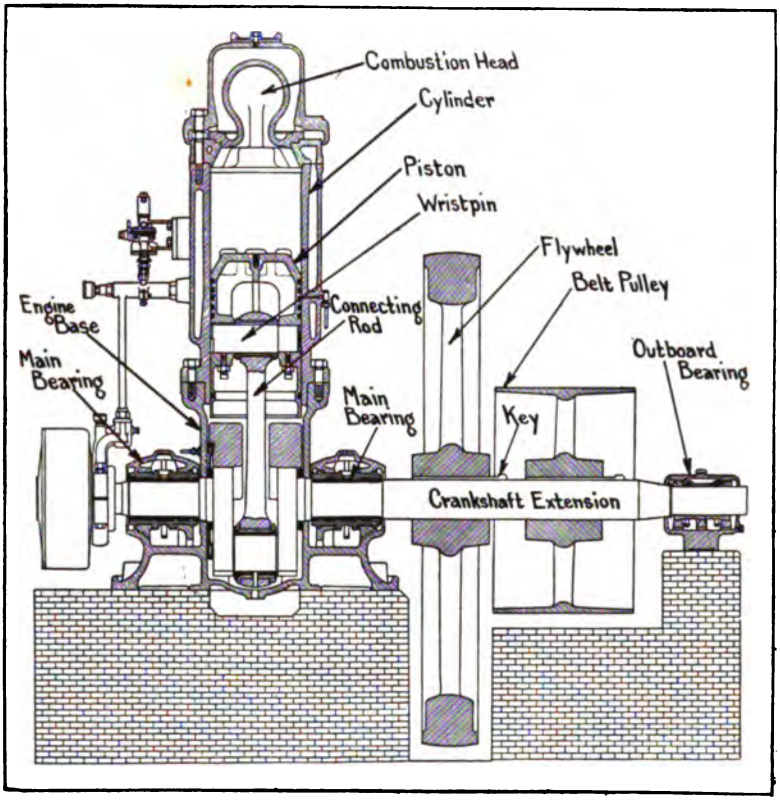


FIG. 317.—Sectional view of Robey two-cycle stationary power plant, showing method of installation on substantial foundation.

big end is lubricated by oil fed to a centrifugal ring on the shaft. Piston lubrication is maintained under pressure produced by a light arm attached to the pump eccentric operating the oiler. The simplicity of the engine is a particularly commendable feature, and the way fuel, water injection and lubrication all vary with the load automatically is worthy of notice.

ECONOMY OF HOT BULB ENGINES

The marked economy of the hot bulb type of engine is clearly shown by comparing the figures given in following table. The figure given for gasoline is low, as are the items of cost of bituminous coal. In view of the marked saving shown by the oil engine, even under conditions where the costs of gasoline and coal are considerably less than current market prices, it is not difficult to understand why this power producer is receiving wider application as its economy becomes better known.

TABLE OF POWER COSTS

Type of Power.	Kind of Fuel.	Price of Fuel.	Fuel Consumed per Horse-Power per Hour.	Cost per Horse-Power per Hour.	Cost per Horse-Power per Year of 300 Days, 10 Hours per Day.	Cost of Generating Electricity, per Kilowatt Hour (Generator Efficiency 89½ per cent.).	Saving of Oil Engine per Horse-Power per Year (Fuel 2¼ cents per gallon).
Steam Simple Engine	Bituminous Coal	$3.00 per ton	8 pounds	$0.01200	$36.00	$0.01800	$27.60
Steam Compound Non-Condensing	Bituminous Coal	$3.00 per ton	5 pounds	$0.00750	$22.50	$0.01125	$14.10
Steam Compound Condensing	Bituminous Coal	$3.00 per ton	3 pounds	$0.00450	$13.50	$0.00675	$ 5.10
Gas Engine	Illuminating Gas	$0.75 per 1,000 cubic feet	18 cu. ft.	$0.01350	$40.50	$0.02025	$32.10
Gas Engine	Natural Gas	$0.30 per 1,000 cubic feet	12 cu. ft.	$0.00360	$10.80	$0.00540	$ 2.40
Gasoline Engine	Gasoline	$0.12 per gallon	1 pint	$0.01500	$45.00	$0.02250	$36.60
Electricity (Motor Efficiency 85 per cent.).	....	$0.02 per Kilowatt hour	878/1,000 kw.	$0.01760	$52.50	....	$44.10
Oil Engine	Fuel Oil	$0.02¼ per gallon	1 pint	$0.00280	$ 8.40	$0.00420	......

DEVICE FOR CONVERTING CRUDE OIL TO GAS

In Fig. 318, and following, we illustrate one of the later devices for generating the cheapest of power fuels yet obtained from fluids or their vapors. Crude petroleum has become directly subservient to the requirement for power-fuel in explosive motors by an evaporative process that utilizes all its available properties and at the same time allows the waste tar products to be discharged, and also of the thorough cleaning of the evaporating surface when required. The generator consists of a chamber of two compartments separated diagonally by a partition on which projects a series of ribs that causes the oil to flow in a ziz-zag course down the surface heated by the exhaust through the chamber beneath. The crude oil is fed at the top, as shown in the cuts; the vapor is drawn to the motor through the pipe and small chamber around the exhaust-pipe as shown. A three-way cock regulates the quantity of the exhaust required for evaporative effect in the generator. A small injection of gasoline into the air-pipe at the side of the cylinder is used for starting. When the generator is warmed up the crude oil is turned on. The governor regulates the mixture-charge.

THE DIESEL MOTOR

This motor is an innovation upon all former ideals in explosive power and indicates the "Ultima Thule" of explosive-motor compression, and possibly the limit of fuel economy in this type of prime movers. The late Mr. Diesel had attempted to realize, within the limitations of practice, an approach to the conditions of the "Carnot cycle" by the production of a motor of very high thermal efficiency. In order to accomplish this result it was evident that a much higher degree of compression was necessary than that used in existing motors, since it was demanded that the charge be compressed adiabatically to the maximum initial pressure at which the motor was to be operated, this pressure not to be exceeded by the gases generated during the combustion. Such a compression would naturally produce an increase in temperature sufficient to ignite the combustible, and hence it became apparent that the fuel must not be introduced with the air, but that the air must first be compressed adiabatically and that the fuel must then be introduced and burned during the out-stroke of the piston isothermally, if the desired cycle was to be practically realized.

In the Diesel motor the high temperature attained by the compression of the air is sufficient to provide for the ignition of the combustible, and it is only necessary for the fuel to be injected into the heated air for its ignition and combustion to take place. In his theoretical discussion of the subject, Mr. Diesel laid down four conditions as essential to the realization of the highest economy: First, that the combustion temperature must be attained not by the combustion, and during the same, but before, and independent of it, by the compression of pure air. Second, that this is best accomplished by deviating from the pure Carnot cycle to the extent of combining two of the stages of the cycle, and directly compressing the air adiabatically, instead of first isothermally from two to four atmospheres, and then adiabatically to

FIG. 318.—The crude-oil generator.

thirty or forty fold. Third, that the fuel be introduced gradually into the compressed air, and burned with little or no increase in temperature during the period of combustion. Fourth, that a considerable surplus of air be present.

It will be seen from these conditions that a motor to meet them, although operating upon the so-called "four-cycle" principle, must differ essentially from engines of the Otto type, and it was to realize these conditions that the Diesel motor was designed. In general construction it resembles the design of a vertical gas-engine, except that all parts are built to stand the high pressure employed (Fig. 320). In the Diesel engine, compression is entirely independent of the quality of the fuel, for the simple reason that no fuel is introduced until it is wanted

FIG. 319.—Outside view of generator.

to ignite. Pure air alone is compressed, and therefore the intensity of compression is limited only by two factors — the ability of the mechanical construction to withstand the stresses, and the thermal possibilities involved. The high compression produces a temperature sufficient to cause ignition of the fuel, and this ignition takes place as soon as the fuel is introduced to the heated atmosphere in which it burns.

The working cycle is as follows: On one down-stroke the main cylinder is completely filled with pure air, the next up-stroke compresses this to about thirty-five atmospheres, creating a temperature more than sufficient to ignite the fuel. At the beginning of the next down-stroke the fuel-valve opens, and the petroleum, atomized by passing through a spool of fine wire netting, is injected during a predetermined part of the stroke into this red-hot air, resulting in combustion controlled as to pressure and temperature. This injection is made possible by the air in the starting-tank, which is kept by the small air-pump at a pressure some five or ten atmospheres greater than that in the main cylinder. A small quantity of this air enters with the fuel-charge, which it atomizes as described. When the motor is running at full load, a very small quantity of injected air suffices, and the pressure in the air-tank steadily rises. At half load, with less fuel injected, more air passes in. For this reason, the starting-tank is made large enough to equalize these differences, and a small safety-valve is provided on the air-pump.

The petroleum is pumped into the fuel-valve casing by a small oil-pump bolted to the base-plate. This pump is arranged to pump a fixed maximum quantity of petroleum. A by-pass is provided so that this whole quantity, or any portion of it, can be returned to the supply-tank. The governor controls the action of this by-pass valve, closing it just long enough to compel the exact quantity of the fuel required to pass into the fuel-valve casing. The full charge of air being always supplied for complete combustion, it matters not whether the governor permits one or fifty drops of petroleum to enter the working cylinder at each motor-stroke, and the combustion is always complete. To stop the motor it is only necessary to close the valve which admits the petroleum into the fuel-valve casing. The valve-gear consists of a series of cams placed on a shaft journaled on brackets cast on the cylinder. The highest efficiency indicated has been found to be thirty-seven per cent. at full load and forty-one per cent. at half load, with a brake efficiency at full load of twenty-seven per cent. and at half load nineteen per cent. These high efficiencies are probably due to perfect combustion under high pressure, which is an essential feature of this motor.

As a machine, the Diesel engine may be fully as frictionless as a steam-engine, and recent tests of a Diesel engine have shown that this is the case. It is also found that an indicated horse-power hour can be got for about 0.32 pound of crude oil with a calorific capacity of about 19,000 B. T. U., and this points to a very efficient utilization of the heat-value of the fuel. This high efficiency is a result due largely to the high compression which is possible only with the Diesel system of fuel admission. It is also partly due to diminished friction and diminished jacket losses.

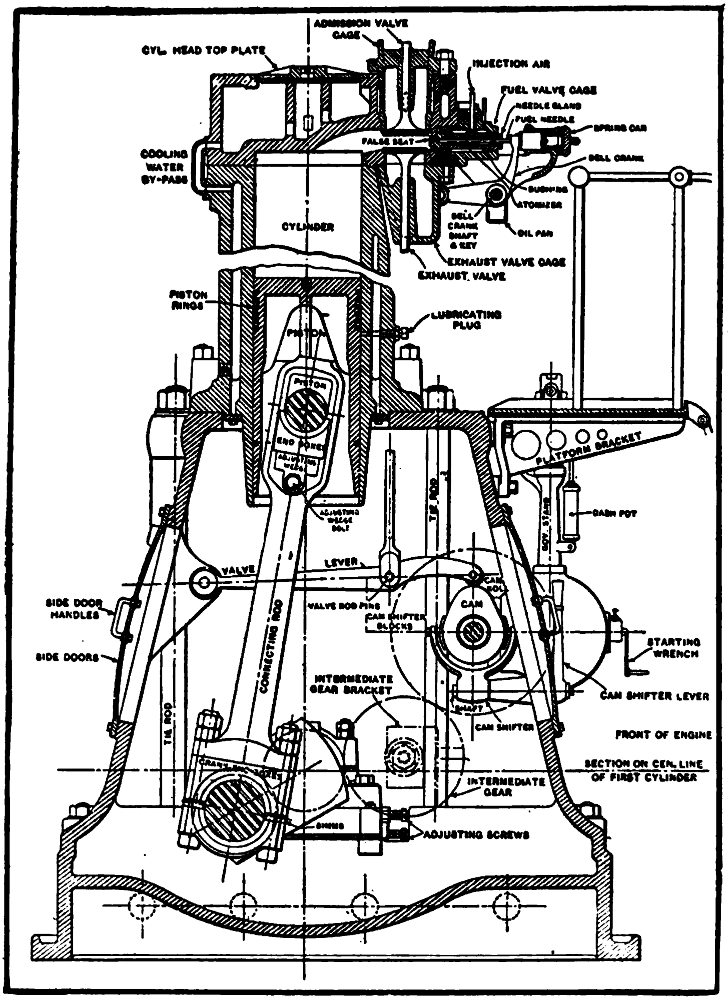


FIG. 320.—Internal construction of Diesel stationary power plant.

The future improvement of internal-combustion engines lies so much along the lines followed by Diesel that this motor may be studied to good advantage, for its system of compression removes the most serious limitations of the ordinary motor, and in weight of combustible per unit of energy output its record is far ahead of any other motor.

The following excerpts from a very interesting discussion on the Diesel engine in the *Sibley Journal of Engineering* give the results of actual tests and other pertinent information of value:

"It is amazing to note how many manufacturers of gas and steam-engines have taken up the manufacture of Diesel engines,

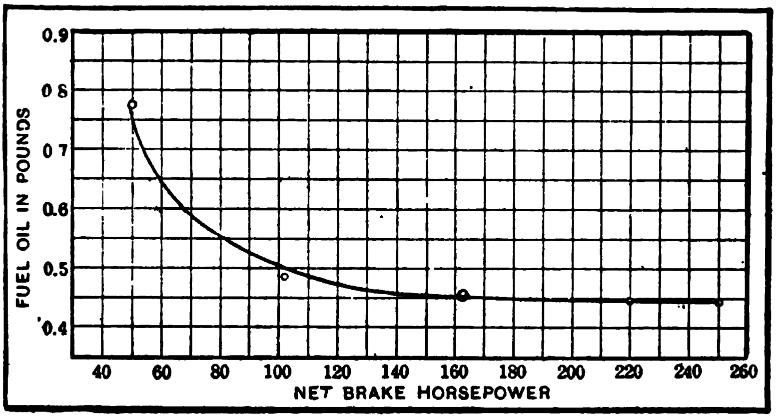


Fig. 321.—Curve showing fuel consumption of Diesel engine.

because they found that the sale of suction-gas producer plants and smaller steam-engines has fallen off alarmingly within the last few years. The reason is plain. The single-acting four-cycle, single or multi-cylinder Diesel engine, but particularly the former, is comparatively simple in construction and operation. It does not require upkeep and attendance of boilers or gas producers, and its cost, compared with that of a steam or gas plant, is reasonable. It can be installed in the basements of buildings below occupied dwellings. One of the greatest advantages, however, is the fact that the actual fuel consumption of Diesel engines taken over long periods of operation does not materially exceed the guaranteed figures; whereas, in gas producer and steam plants this excess is quite considerable. In a Diesel plant the human element—the skill of the operator—has much less influence upon the fuel economy than in a steam or producer gas plant, where everything depends upon the efficiency and intelligence of fireman and producer attendant."

"On account of its low cost and great simplicity and variety of fuels which can be utilized the engines are being used and installed for various purposes, as for power transmissions, for operating flour mills, for lighting factories, office buildings, department stores, and for electrical works, etc. Lately they have

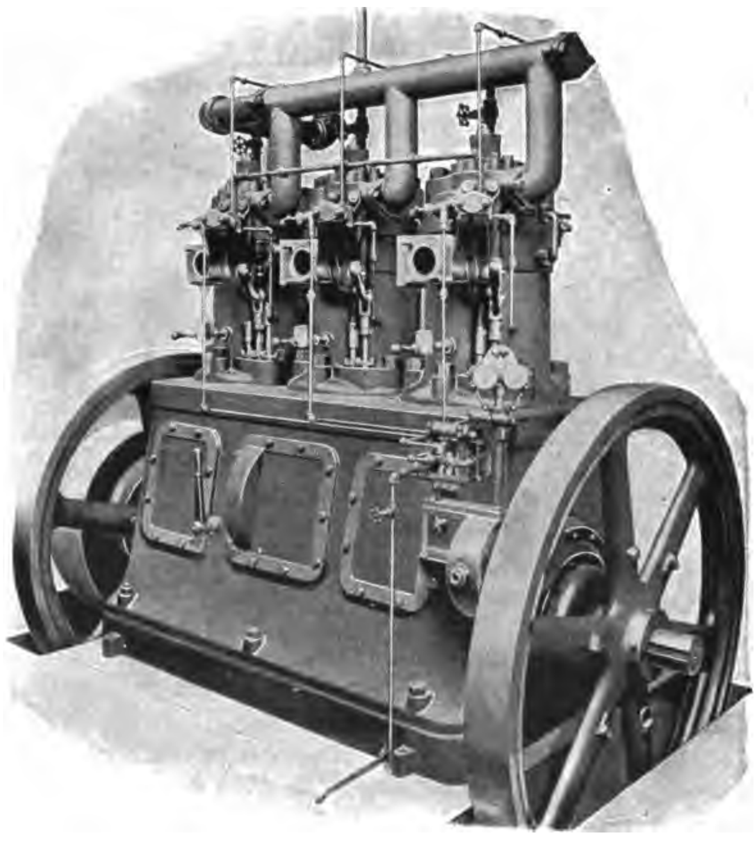


FIG. 322.—Three-cylinder Diesel motor, 225 horse-power.

been applied to the production of power where new and additional equipment is found necessary. It has been found that this engine can be very economically maintained as a reserve for relieving the original units of the peaks of the load and for like purposes."

EFFICIENCY OF THE DIESEL ENGINE

The efficiency of this type of engine and its characteristics may be readily obtained from the results of a test on a typical Diesel engine doing actual work. This test was conducted by A. C. Scott at the Scott Engineering Co., Dallas, Texas, on a 225 horse-power Diesel engine; the following data have been obtained which will give an excellent idea of the applicability of this engine to the generation of power. The engine tested operates on the four-stroke cycle Diesel principle. It is a vertical three-cylinder unit, rated at 225 net brake horse-power, with allowance for power necessary to drive auxiliaries. It is directly connected to a Fort Wayne 200 K. V. A., three-phase, sixty-cycle, 2300-volt generator of 164 revolutions per minute. The auxiliaries consist of one belt-driven three-stage air compressor; one motor for driving compressor, 25 horse-power; and one exciter, 10 K. W., speed 850 revolutions per minute, belt-driven from the engine.

In many cases the air compressors for the starting and injection of air for Diesel engines are belted from the crankshaft of the engine; or when the compressor is motor driven, the power is supplied from the generator. During the series of tests, the power for the compressor was furnished from an outside source. Therefore, in order to obtain the net available power developed by the engine, readings were taken of the power consumption of the motor driving the compressor and a deduction made from the kilowatt output of the generator. No allowance was made for the efficiency of the motor; so that with the compressor belted to the engine shaft, the fuel consumption per brake horse-power hour would have been less than shown in the results given.

FUEL OIL USED

To obtain the amount of fuel oil consumed by the engine, it was only necessary to weigh the oil fed to the fuel pump, where the amount of oil actually fed to the engine cylinders is automatically controlled by the governor, and varied according to load requirements. For weighing the oil a 15-gallon tank and platform scale were used, and readings were taken at ten-minute intervals. A funnel of sufficient capacity was installed in order that a small amount of oil might be stored ahead of the governor, and the weighing tank properly refilled with oil. The oil used for these tests was taken from regular stock in ordinary use in this engine. Samples of oil were taken periodically, and the analysis made upon the total combined samples. The analysis of the fuel oil is given below:

British Thermal Units....................	18,986 per pound of oil
Specific Gravity, 25.5 degrees C. to 27 degrees C. (78 degrees F. to 81 degrees F.)................................	0.8531
Viscosity, 33.3 degrees C. (92 degrees F.)	1.63
Flash Point	143.6 degrees F.
Burning (Fire) Point....................	181.4 degrees F.
Sulphur................................	0.2 per cent.
Water..................................	Trace
Free Acid	None

The cost of this oil was $1.22 per barrel of 42 gallons, delivered; or practically 2.9 cents per gallon. The instruments used in making the tests were all calibrated before using and due corrections made in the figures obtained. A water rheostat was found entirely satisfactory for the adjustment of all loads. Six tests, each of three hours' duration, were made with load changed for each hour period as follows:

Test Nos.	1	2	3	4	5	6
Net B. H. P.	2.25	49.7	111.39	162.97	219.63	245.62

FUEL CONSUMPTION

The fuel consumption for these various loads was in each instance obtained from an average of the readings taken at ten-minute intervals on the generator and the motor, and the records of oil used also taken every ten minutes. Fig. 321 is a curve showing in pounds the fuel oil consumed per net B. H. P. hour at the various loads. Test No. 1 is not included in this curve, as it was practically a no-load test. This curve was plotted by taking the average readings of output of the generator at the switchboard, corrected for the previously ascertained generator efficiency at the given load. From this was subtracted the actual kilowatt in-put to the motor and the resulting figure reduced to brake horse-power. It will be noted from the above curve, that the consumption of oil per brake horse-power hour increases but very little when the output is decreased from full load to about half load. At full load the fuel consumed was 0.441 pound per brake horse-power hour. When running at practically half load, the fuel consumed was 0.482 pound per brake horse-power hour, or about 6.8 gallons per hundred net brake horse-power hours. At quarter load the engine consumed 10.8 gallons of oil per hundred net brake horse-power hours."

LARGE DIESEL ENGINES

"Owing to the steady application of its power, the lack of vibration, comparative noiselessness, reliability, and to the small space which it occupies, the Diesel engine has become an almost ideal engine for marine work where oil may be had at nominal prices. An idea of the ease with which a Diesel engine may be controlled may be had from experiments run with the British motor ship *Eavestone.* This vessel is a ship of 4310 tons displacement recently built by Sir Raylton Dixon Co., Ltd. The 800 horse-power Diesel engine was largely built and installed by Richardsons, Westgarth & Co., Ltd., of Harthpool. She was capable of doing somewhat over nine knots. The action of the engines was very remarkable; they were reversed from full ahead to actually running astern in from nine to ten seconds, and this without any haste; in fact, with intentional deliberation. It was stated that the reversal had actually been completed in the remarkable time of six seconds. Observations showed that it only took about three seconds from the ringing of the telegraph till the engine was actually running ahead, the gear being already in the 'ahead' position. Units of this character have been built for marine service up to about 2000 horse-power, single-acting type."

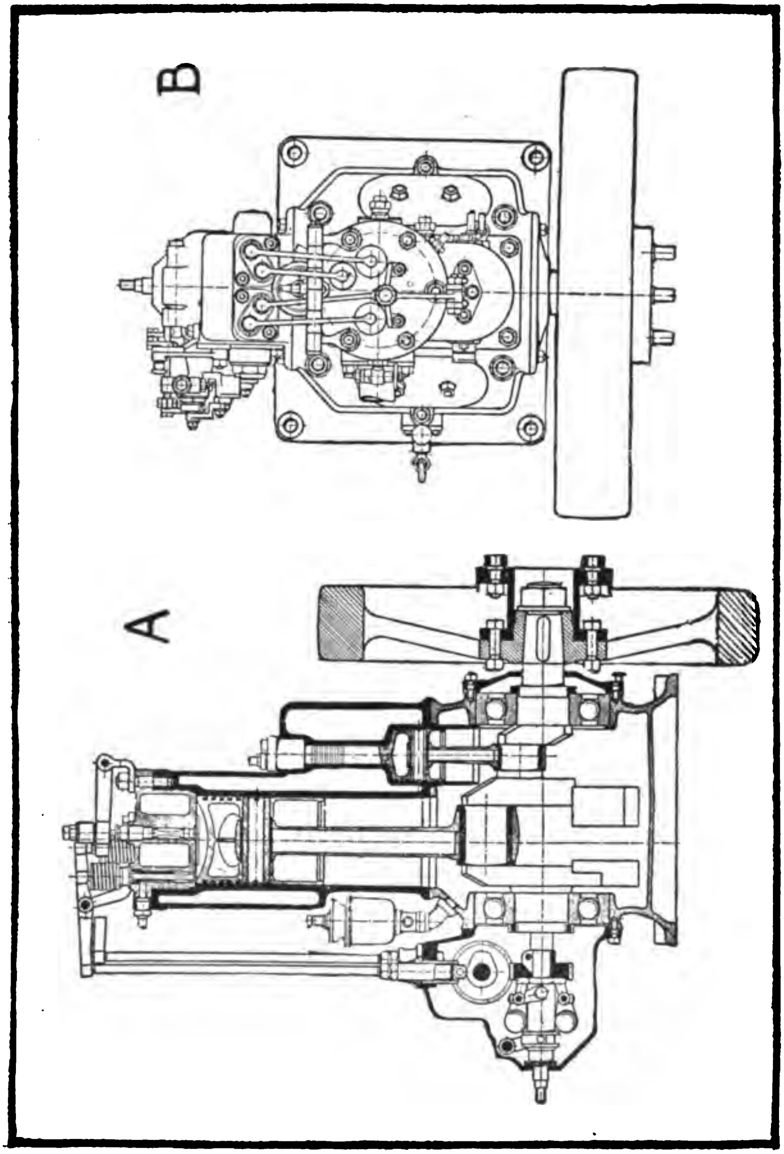


FIG. 323.—Section of 5 horse-power Diesel motor with ball-bearing crankshaft at A. Plan view of 5 horse-power Diesel engine, showing arrangement of valves in cylinder head and disposition of actuating mechanism.

SMALL DIESEL ENGINES

That the Diesel engine, or similar type of heavy-oil motor, will be generally adopted for large power yachts there can now be no doubt, as the extra accommodation and additional cruising range, as compared with steam, are too important to be overlooked by yachtsmen, while economy and safety make this power far more desirable than gasoline machinery. But whether the Diesel will ever be within the reach of the small-boat owner is a very difficult question to answer. It is not generally known, however, that there are already six Diesel yachts afloat, of which four are under 50 horse-power, and that eight more are under construction. The latter, however, range in powers up to 2000 horse-power. The present great difficulty in the way of more general adoption is, of course, the heavy cost of such machinery; but this is partly due to the large amount of experimental work which the makers had in connection with the first engines. If there is a wide market for them it should be possible to systemize the construction and to turn out large numbers at a cost that should not be much above the prices of first-class gasoline motors. If the small Diesel engines are built non-reversible and supplied with reverse gears, then much of the constructional expense will be eliminated. Making the engine directly reversible is one of the expensive features of the construction.

Semi-Diesel, or hot-bulb oil-engines, are being considerably adopted for power yachts, but the objection of the yacht owner to this class of motor is that it is necessary to use a blow-lamp for starting purposes, also when running slowly, while only paraffin and the lighter residual oils, such as gas-oil, can be used; whereas the Diesel type can use practically any heavy oil fuel when necessary, so an owner can be sure of obtaining fuel of some kind at every port.

There are already two small successful Diesels in use, one of 6 horse-power being installed in the Elbe motor Light-ship *Burgermeister O'Swald,* driving an air compressor at 600 revolutions per minute. It was built by Sulzer Bros. of Winterthur and is of the four-stroke type. The other of 5 horse-power is illustrated at Fig. 323, and was built at Dr. Diesel's own Munich works. Its bore is 116 mm., by 150 mm. stroke, and its rated power is developed at 600 revolutions per minute. At 828 revolutions per minute as much as 7.65 horse-power is developed, the fuel consumption being 0.555 pints per B. H. P. The normal load consumption is 0.498 pints. The oil used was Galician resid-

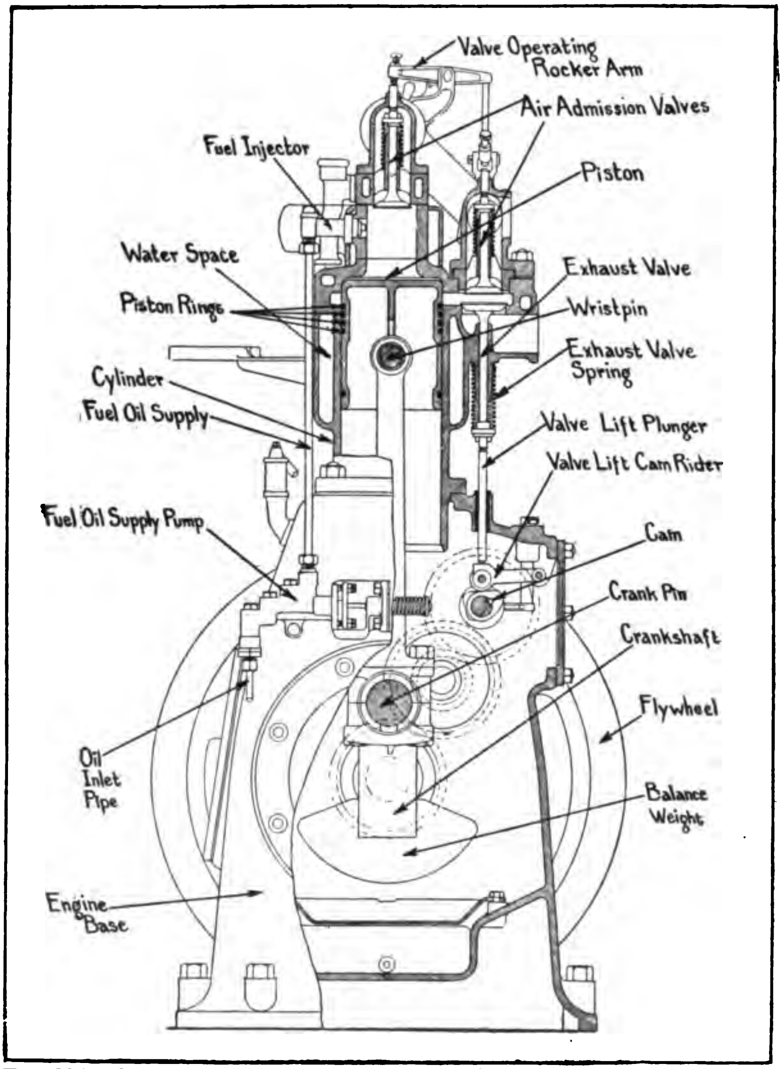


FIG. 324.—Sectional view of the Cross semi-Diesel engine for stationary purposes.

ual of .865 specific gravity with a heat value of 19,850. An engine like this only costs 2 cents per hour to run at full load, so it will be seen that even at this low power the Diesel is economical.

CROSS SEMI-DIESEL ENGINES

The views at Figs. 324 and 325 show the construction of the Cross semi-Diesel or hot-bulb engine very clearly. The end sectional view of one of the larger models shows the disposition of the parts and outlines the valve arrangement as well as other related parts. The longitudinal sectional drawing at Fig. 325 depicts the form adapted for marine use, fitted with a hand-starting crank carried on a suitable support attached to the front end of the engine. The motion of the hand crank is imparted to the crankshaft by chain connection. Attention is directed to the grouping of the four oil supply pumps at the front end and the method of operating them by cams at the front end of the camshaft. The engine operates on the four-cycle principle and follows conventional practice. The cylinders are individual castings, the crankshaft is a five-bearing four-throw type. A flywheel is provided to insure smooth running and easy starting by the hand crank. Starting torches are used to heat the cylinder head to a point where the highly compressed charge will explode.

LARGE DIESEL MARINE ENGINE

The following excerpts are taken from an article describing the power plant of a typical oil-engine ship, the *Fordonian,* that appeared in "Gas Review":

The *Fordonian* was built by the Clyde Shipbuilding and Engineering Company, Ltd., of Glasgow, for the Canadian Inter-lake Line, Ltd., of Toronto, for carrying grain on the Great Lakes. The boat is propelled by a two-stroke cycle Diesel oil-engine. The boat is two hundred and fifty feet long, forty-two feet six inches beam, sixteen feet ten inches moulded depth to the main deck and twenty-six feet six inches to the awning deck. The *Fordonian* has a two-foot frame pitch, and a dead weight carrying capacity of three thousand three hundred tons on sixteen feet six inches draft. The draft on service is restricted to fourteen feet and the dead weight capacity is two thousand two hundred tons. She is built to Lloyd's highest class.

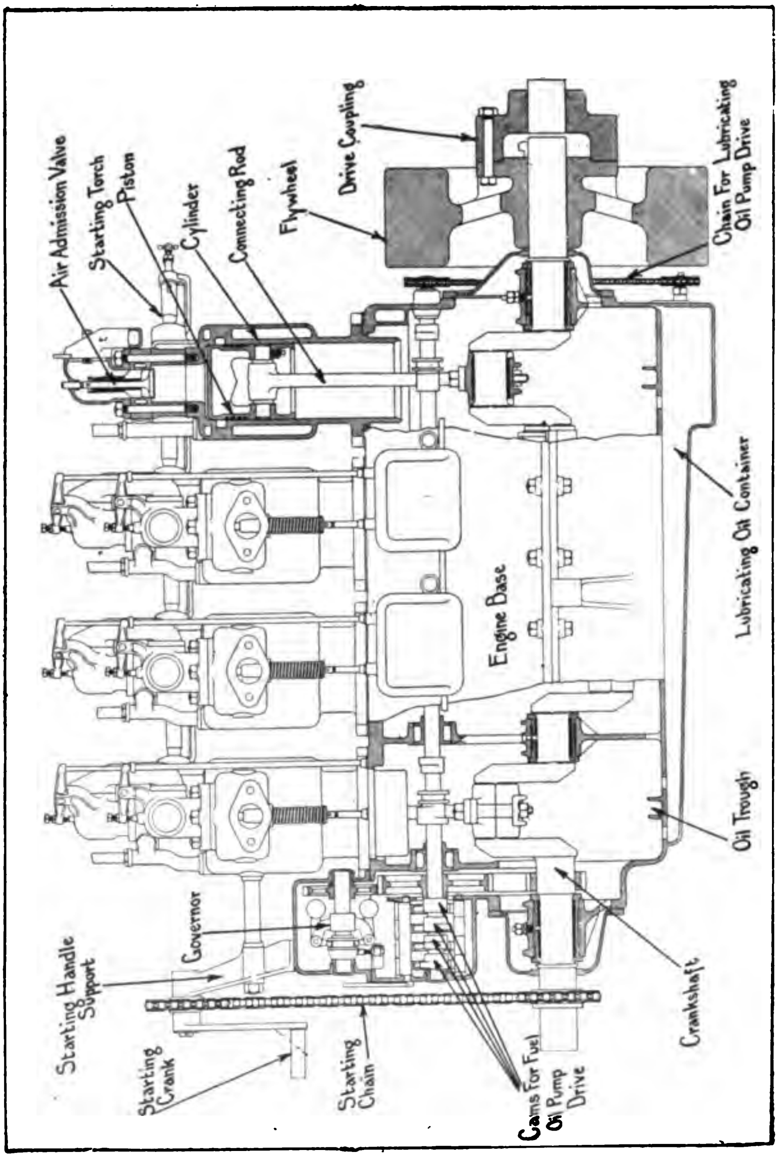


FIG. 325.—Longitudinal part sectional view of the Cross semi-Diesel, marine engine of the four-cylinder type.

The main propelling engine shown at Fig. 326 is a four-cylinder two-stroke cycle single-acting Carels type of Diesel oil-engine. The cylinder dimensions are eighteen and one-tenth inches in diameter by thirty-two and one-quarter inches stroke and the engine runs normally at about one hundred revolutions per minute. The propeller is eleven feet in diameter by nine feet pitch. The bed-plate is of cast iron and is of the usual marine design, having a flat bottom and being supported in the center as well as at the sides. This design contrasts with that evolved by many continental makers, who prefer the bed-plate supported at the sides only, with the cross-members of deep box section carrying the main bearings, which have forced lubrication. This last method has advantages, but, as will be described later, the *Fordonian* has not a forced system of lubrication. The columns of the engine are of the usual box section, bolted rigidly together at the top, and are very thick, to withstand the tension stresses consequent upon the high pressures of the Diesel cycle. These tend to give great rigidity; the engine runs entirely free from vibration. With this design of support the bed-plate must be strong to take the bending stresses between the column feet and the main bearings.

The cylinders have separate liners pressed in, and the liners have exhaust ports round the whole of their periphery, and communicate with an exhaust belt of large cross-sectional area running round the cylinder. The water spaces are large, as the elevation of the cylinder shows, and ribs are cast on the inside of the cylinder to aid water circulation and to give to the cylinder wall strength to resist the direct pull passing through it. On each cylinder there are six bolted doors, of about nine inches diameter, permitting of ready inspection and cleaning of the water-cooling spaces. At the bottom of the cylinder there is a lantern-ring, which serves to keep the joint between the cylinder and liner water-tight, and, further, to prevent the escape of cylinder gases into the engine room, which would be a serious drawback with an open engine. Any leakage of gases goes into an annular chamber, which is connected to the scavenging pump suction, and is thus kept at a pressure below that of the atmosphere. This means is efficient to insure an engine room free from gases, although necessitating a long piston. The cylinder head is of cast steel, to gain the necessary strength, and has large cooling-water spaces,

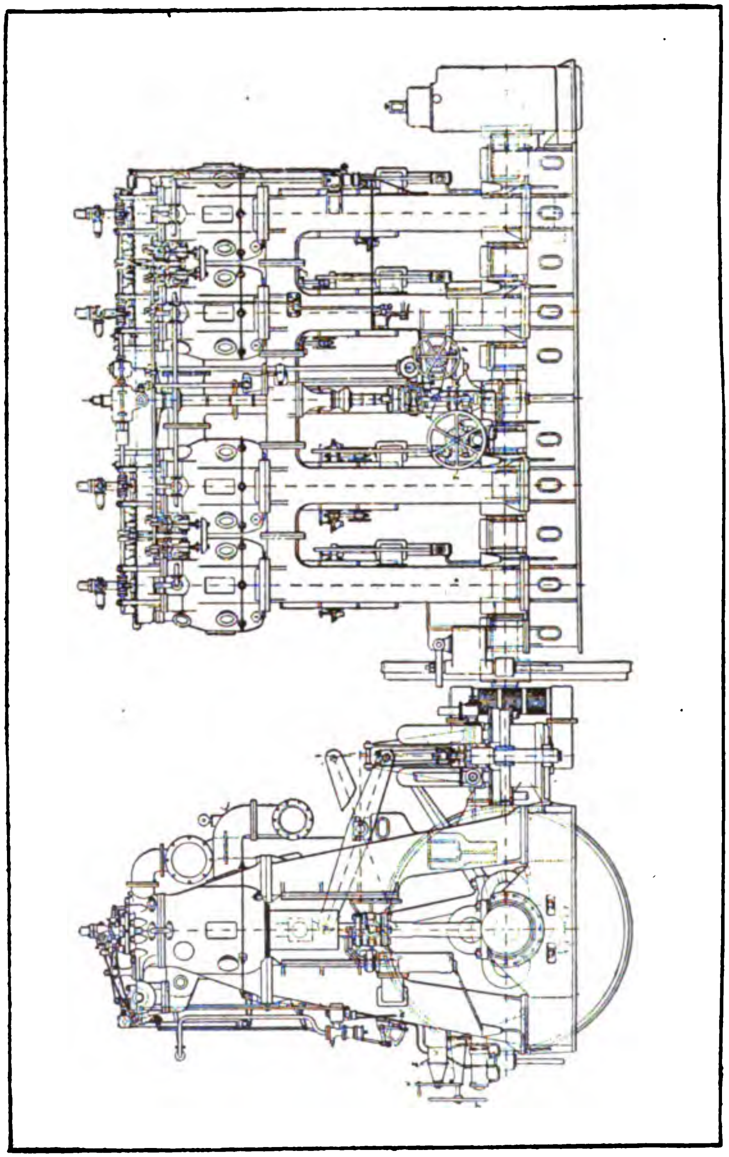


Fig. 328.—Typical Diesel engine adapted for ship propulsion.

with seven valve openings; one fuel, four scavenging, one starting air, and one safety. It is probable that at an early date such covers will be made of cast iron.

The piston of the Carels engine is in two pieces. The top piece is carried by a shoulder on the piston rod, and the bottom piece, or shroud, is carried at its bottom by another shoulder on the piston rod. The forming of the piston in two pieces makes for simple castings, and when the high temperatures are considered, this is a desirable end. Water cooling is adopted for the piston, and the water is circulated by the action of the plungers, as shown, as opposed to the system of walking pipes. The piston in this engine ran cool on trial, and the stresses due to temperature would thereby be minimized. Moreover this advantage makes it possible to reduce clearances, consequent upon the large expansions due to the high temperature, and there is thus less chance of losing compression.

The arrangement of the engine into two units of two cylinders each permits of a two-piece crankshaft in interchangeable halves, of the vertical spiral drive for the valve gear being taken from the center of the engine, and also of the scavenging pumps being driven from the two center crossheads by links, as with the air pump of steam-engines. The dimensions of the double-acting scavenging air pumps are twenty-seven and one-quarter inches in diameter with a twenty-three and one-half inch stroke, and give thus a ratio of free combustion air taken into the main cylinders of one and sixty-five one-hundreths, which is higher than the usual practice. The pressure of the scavenging air is three pounds per square inch. There are four valves in the cylinder head for the inlet of the scavenging air to cope with the large volume of low-pressure air used in the engine. This combination of large volume of scavenging air, a large reservoir, and four valves in the cylinder head, makes for good combustion and a clean exhaust. The scavenging pumps are controlled by two piston valves worked by slipping eccentrics driven from the aft part of the two piece interchangeable built-up crankshaft, and the change of angular position permitted by the slipping of the eccentric on the crankshaft automatically reserves the scavenging pump piston valves. The scavenging air is led by cast iron pipes from the valves to a built-up three-sixteenths inch lap-riveted steel plate reservoir running along the cylinder top and supplying the four scavenging valves on each cylinder.

The exhaust is led down by bent cast iron pipes from the cylinder belt to the main exhaust pipe running along the engine to the cast iron silencer. These bends have internal water injection, and the silencer is also internally water cooled and is of the cascade design. This very effectively silences the exhaust, but it is difficult to judge the combustion of the engine at the overboard exhaust, since the water must cleanse unburnt products. The exhaust is led overboard under the counter. The funnel is for the exhaust gases from the donkey boiler. Separate leads are provided from the water cooling pumps worked off the links driving the scavenging pumps, and cocks are provided on all these leads to regulate independently the amount of cooling water supplied to each part. The temperature of this cooling water may be felt, as there are open discharges into the funnels leading to the bilges. These open discharges can clearly be seen, beside the engine columns, in the illustration.

The system of lubrication is interesting. For the main bearing, solidified oil is used; for the crank-pin bearings the ordinary drip feed suffices. The bearing pressures for the main and crank-pin bearings are respectively about three hundred pounds and six hundred and fifty pounds per square inch. For lubricating oil a forcing pump is attached to each crosshead, and worked by the swing of each connecting-rod, as shown. This system of lubrication permits of an open crank case, and the bottom end bearings can always be easily felt by the engineer on watch. There are two guides for each, such being Messrs. Carels' practice for oil-engines. The piston is lubricated by four Mollerup lubricators, which force the oil between the piston and the cylinder; there are four inlets to the cylinder, and they are arranged to enter on the fore and aft and athwartship center lines.

As already enumerated, there are seven valves and an indicator cock in the cylinder covers. These valves are operated in the usual way by cams and cam levers. The fuel valve has a rotary, and not a reciprocating, gland; this makes for tightness. The levers are of cast steel. The cams are of hardened cast iron, and the cam-shaft is driven from the crankshaft by a helical gearing through a vertical shaft, with cast iron helical wheels and pinions.

Reference to Figs. 1 and 2, diagrammatical sketches of the valve gear at Fig. 327, will make the valve gear and the reversing and maneuvering mechanism quite clear. There are two shafts running fore and aft at the cylinder tops, and supported by brackets to the cylinder body in the usual way. The outer shaft *A* is the maneuvering shaft, and to it are keyed, firstly, cams *B* for operating the fuel pump's suction valves by means of the bell crank lever *C* and rod *D*, as seen in figure 1; and, secondly, the cams *E*, Fig. 2, which serve to regulate the amount of lift of the fuel and starting air valves *F* and *G* respectively, and also to lift and replace

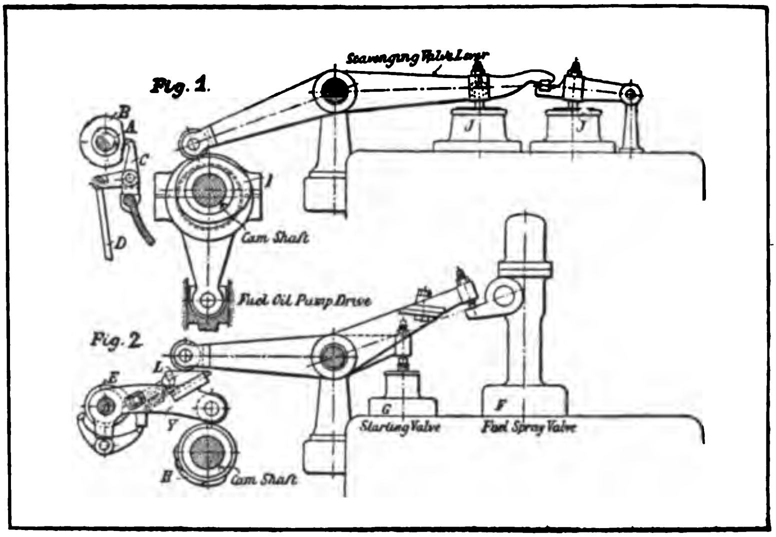


FIG. 327.—Valve mechanism of typical Diesel engine used for ship propulsion.

the inter-starting and inter-fuel levers from off the main cams *H*. Loose upon this shaft there are two inter-starting and inter-fuel levers *Y* for each cylinder.

Turning now to a consideration of the reversing mechanism and its operation, there are two scavenging cams *I* operating the four scavenging valves *J* and these are reversed by turning the camshaft through approximately thirty degrees by extending the driving vertical shaft by means of a compressed air servo-motor. It must clearly be understood that the scavenging cams do not move fore and aft. There are two fuel cams and two starting air cams to each cylinder, and the cams E, by means of the roller-raising lever, lift the inter-starting and inter-fuel lever Y from off the cam upon which it has been working. Then the maneuvering shaft is moved longitudinally, and the inter-starting and inter-fuel levers are brought into line with the fuel or starting air cam for the required direction of rotation. Further rotation of the maneuvering shaft rotating cams E — the inter-starting and the inter-fuel valve levers are loose upon this shaft—causes the inter-fuel and inter-starting air levers, of bronze, to descend upon the requisite cam. Still further rotation of the maneuvering shaft and its cams E actuates the wedge-piece L through a roller and spindle, and so first causes the opening of the starting air valve. Starting air is thus admitted to all four cylinders, and then for two cylinders a rotation of the maneuvering shaft cams, actuating now the wedge-piece L, causes the starting air to be gradually cut out and the fuel to be gradually cut in.

The fuel pumps, of which there are four, one for each cylinder, are operated by eccentrics from the camshaft, and at the same time as the fuel oil is being cut into the first two cylinders the cams shown at B operate through bell crank levers, and so control the suction valves of the fuel pumps, and fuel oil is thus delivered to the cylinder. After the two cylinders are firing, the further rotation of the maneuvering shaft causes exactly the same cycle for the other two cylinders, and all cylinders will then be running on fuel. The working of the valve gear, the exact relationship between the camshaft, maneuvering shaft, and cam levers can well be understood from reference to the drawings herewith. The action of the wedge-piece is obvious, and the spring which controls it is shown. It will be noticed that the fuel and starting air cam levers are in two pieces, to facilitate the inspection of the valves. The scavenging air valve gear is also clearly shown, and the manner in which the four scavenging valves are operated will be readily understood.

The feature about this interesting valve gear is the wedging action whereby the starting air is gradually cut out and the fuel oil gradually cut in. This gives an even turning moment all the time. At the commencement the air pressure, eight hundred pounds to one thousand pounds per square inch, insures that there is a large starting torque; further, the design of the starting valve

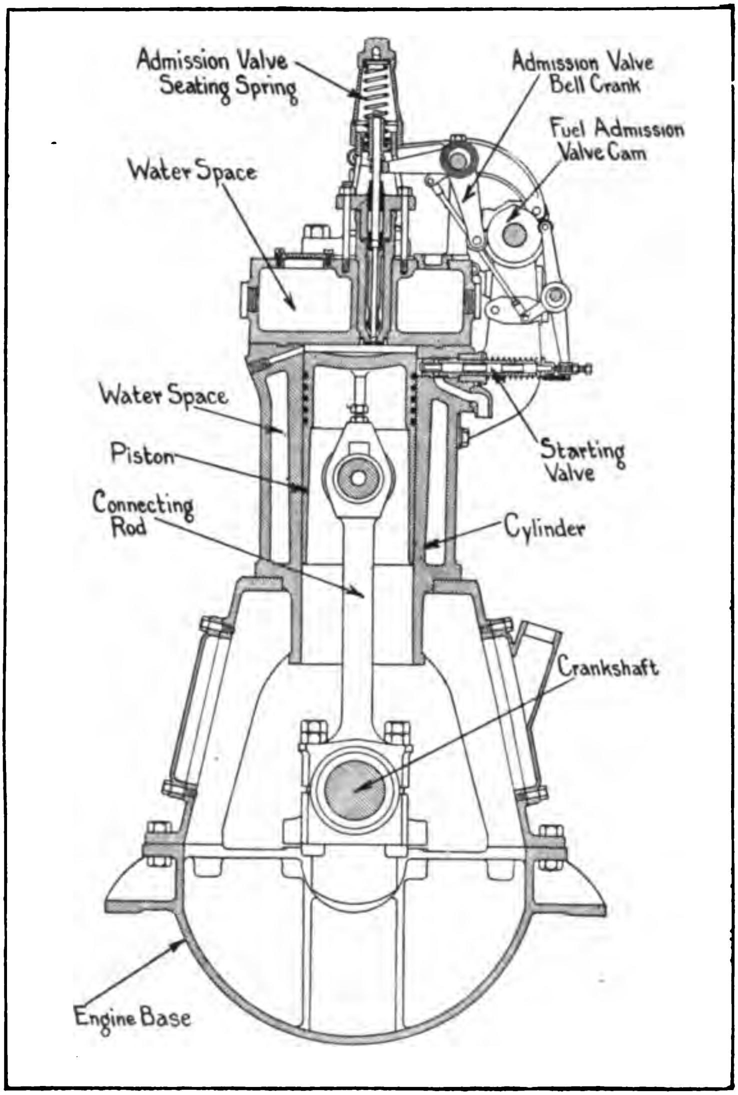


FIG. 328.—End sectional view of the Gardner-Diesel engine, showing arrangement of parts of four-cycle type power plant.

mechanism necessarily gives that large starting torque at all positions of the cranks, and the wedge action makes for an even turning moment throughout the period of engine acceleration. There is no shock due to the air being suddenly cut off and the fuel suddenly cut in. It is a gradual process, the one merging into the other.

The control of the engine is by means of one wheel and two levers on the starting platform; one lever, *X*, (see Fig. 326) controls the compressed air engine, which gives the camshaft its angular displacement by raising or lowering the vertical driving shaft, and also gives the maneuvering shaft its fore and aft movement. The other lever, *W*, controls the fuel. The wheel *V*, operated by hand, gives the maneuvering shaft its rotary motion. As seen in Figs. 1 and 2, the cams upon the maneuvering shaft act upon the suction valves of the fuel oil pump. Hand control is also provided by the handle on the column, which actuates a shaft running fore and aft on the engine, and so sets all the fuel pump suction valves. The small dial seen above the hand wheel indicates the position of the valve gear. Although compressed air is used, as stated, for actuating the vertical shaft, causing the angular rotation of the camshaft and the rotation and displacement of the maneuvering shaft, hand gear, in emergency may be used — viz., wheel *T*.

THE GARDNER 100 H. P. DIESEL ENGINE

The following description of a typical high speed engine of the Diesel pattern is reproduced in part from a descriptive article in "Internal Combustion Engineering" and the drawings were also made or adapted from the illustrations accompanying the discussion. The vertical end section or front view is given at Fig. 328, and the longitudinal part section Fig. 329 makes clear the details not outlined in other view.

This being a high-speed engine, the enclosed type of crankpit has been employed. The cylinders are cast independently, each with its own jacket and walls, and not fitted with liners, and the crowns are detachable. From the four cylinders of 10 in. diameter, with a piston stroke of 8 ins., and at a speed of 500 revolutions per minute, the power measures about 100 H. P. on the shaft. Relatively the stroke is very short, which ordinarily is not accounted an advantage in Diesel design, because it entails for a given

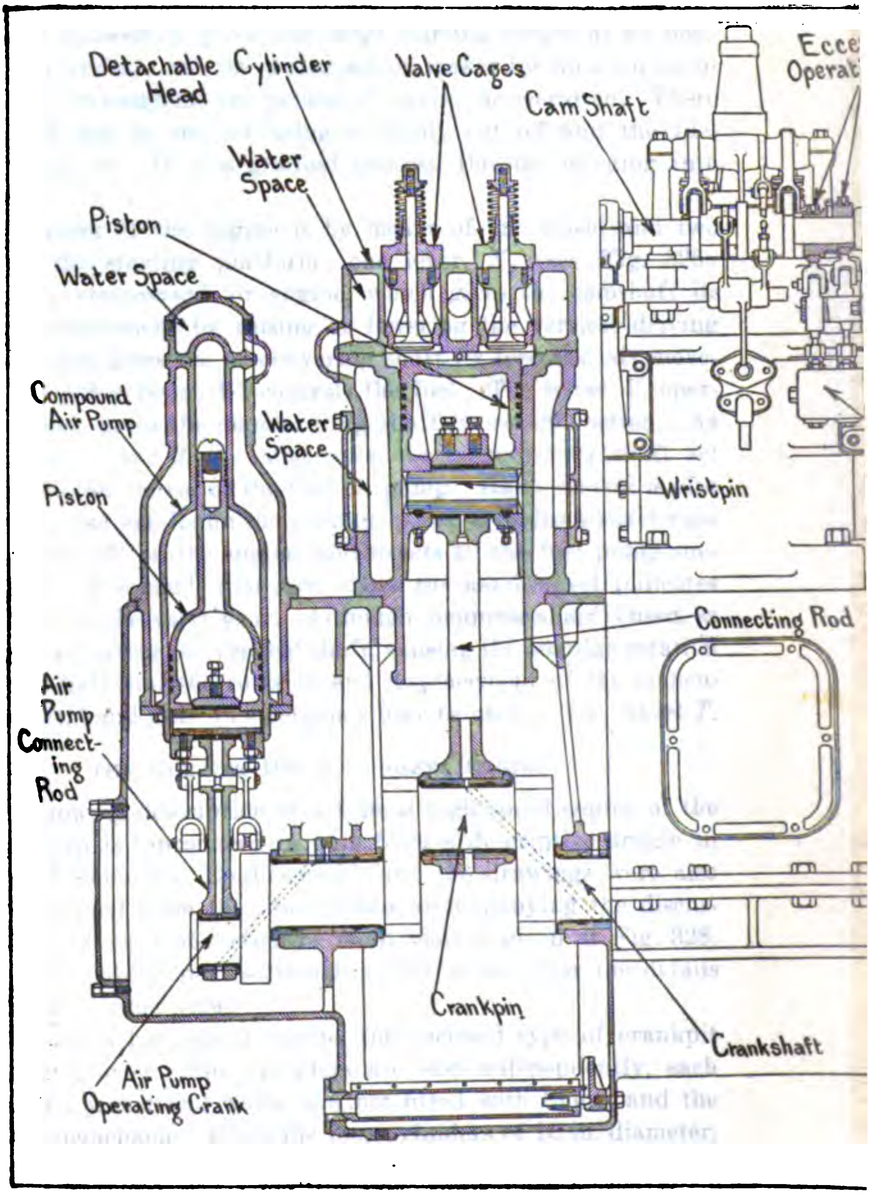
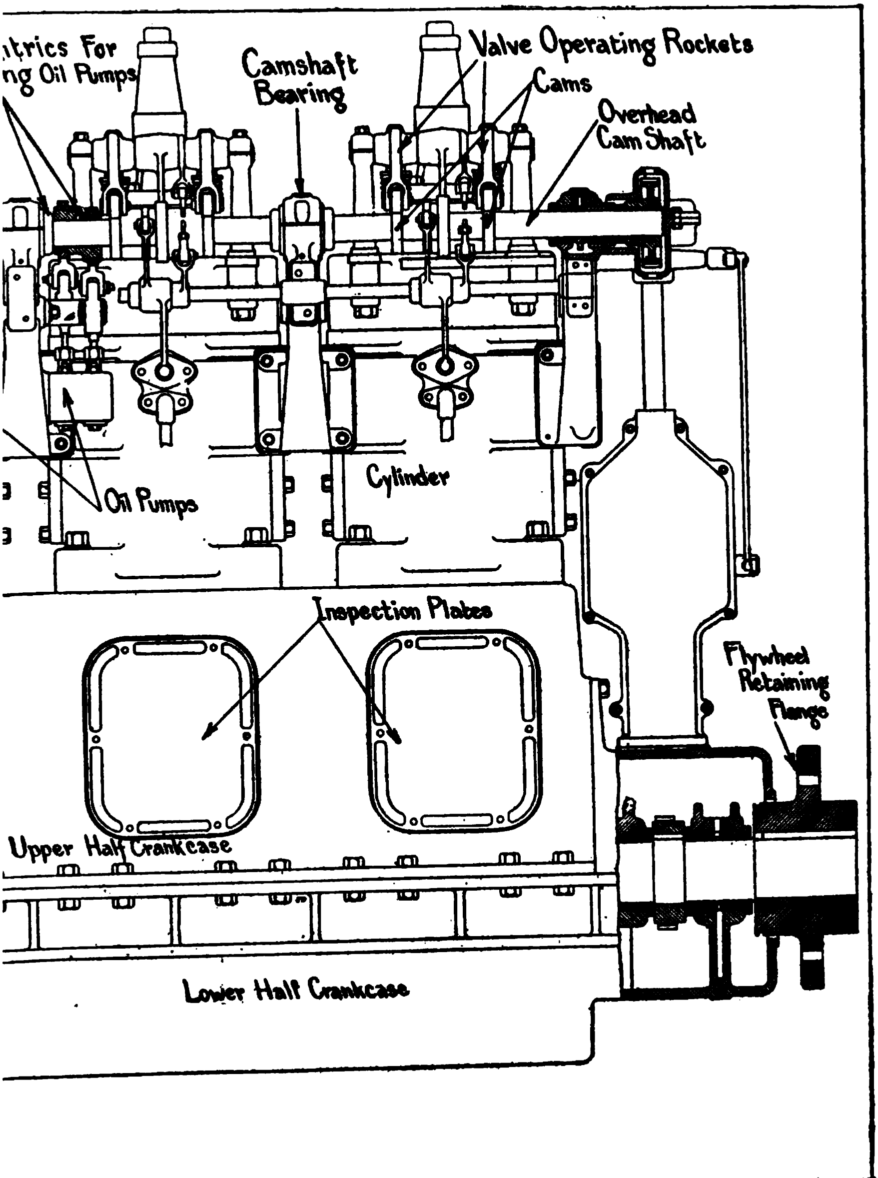


FIG. 329.—Part sectional view of the Gardner-I



…iesel four-cylinder motor adapted for marine service.

cylinder capacity a larger area of combustion chamber surface, and therefore greater radiation during maximum heat. On a high-speed engine, however, the increments of heat succeed each other so rapidly that special attention must be paid to the cooling of the combustion chamber, and for this reason the bore-stroke ratio adopted in this first Gardner-Diesel design has probably advantageously influenced the protracted running of the engine. Greater thermal efficiency could have been gained with a long-stroke engine, but it might have spoilt the freedom from heat troubles which this motor has enjoyed. There is also another factor which enters into consideration. Clearance varies with the bore-stroke ratio, and the greater clearance gives a better disposition of combustion space in so far as the distribution of the injected fuel is concerned. On this account, therefore, the shorter stroke suffers by comparison. But when all is weighed and compromised, and when it is not forgotten that reliability is the prime factor of an engine's value, the judiciousness of the 10.8 bore-stroke ratio is apparent in the case of a running motor turning at 500 revolutions per minute. With the larger diameter there is more space for the valves in the crown, and with high engine speeds a considerable difference is made by the size of the valves. Wire-drawing at the inlet means less air for combustion, and a strangled exhaust has an equally bad effect, the combination of the two resulting in a lower mean effective pressure.

An upper and a lower casting constitute the crankcase. The lower piece is bedplate and oil-tray, the outflow of the forced lubrication through the bearings being drained to a sump below the compressor crank, in line with which is the force pump for the oil. The barrel of this pump is cast in the bedplate outside the crankcase, and the plunger is worked by a rocking lever pinned in a boss on the inspection door of the compressor line. It is thus in a very accessible position. A cushion dome is fitted on the delivery side, and the main feed is carried straight along the crankcase to supply the five branches for the main bearings, from which a portion of the oil passes through the drilled crankshaft and rods to the big-ends and gudgeon-ends. On the main feed there is an adjustable bye-pass which enables the oil pressure to be set as desired, by shunting a greater or lesser quantity of the lubricant directly into the pit, and the oil-pressure gauge is handily fitted just above the by-pass valve. This external oil-piping is properly fitted, which is not so generally practiced as it should be; each Tee is formed with a lug for bolting to the crankcase. Both at the compressor end and at the governor end the forced lubrication is used, and no sight feeds or mechanical lubricator are used at all. For the smearing of the cylinder walls the oil thrown from the big-ends is relied upon, and this proves quite satisfactory. The amount so flung on the walls is of course, considerably less than is thrown up by a splash lubrication system, and apparently is just the right quantity in this case, for no trouble occurs through any passage of oil past the rings. Did the arrangements leave anything to be desired, it is sure that an alteration would have speedily been effected. As it is, the absence of special cylinder oiling devices tends towards considerable simplification on an engine of these restricted dimensions. It may be interposed here that the swinging links which operate the oil pump are paired with a similar couple intended to drive the water pump on the opposite side of the crankcase, both sets being attached to the connecting-rod of the compressor. As depicted in the illustrations, the engine is running without the water pump, the circulation being provided from the factory mains.

Through bolts carry the operating stresses from the cylinders to the webs of the main bearings, and the upper portion of the crankcase can therefore be liberally supplied with doors on both sides. One of these carries an oil-filling funnel for the replenishment of the lubricating supply. The casing of the compressor line is formed integrally with the main crankcase, but the governor gearing and mechanism is separately enclosed by sections which, however, bolt up to the main case, and thus virtually become part of the chief oil chamber. Through the inspection doors easy access is obtained to the main bearings and big ends, but there is not sufficient depth to the pit to get the pistons out by these means. The main bearings and big-end bearings are lined with white metal, but a phosphor-bronze bush is employed for the gudgeon eyes.

Through the horizontal disposition of the starting valves the mechanism associated with the cams bears an uncommon aspect. In a certain measure it is reminiscent of some of the high-speed practice of Louis Nobel, and it recalls also an arrangement favored by a couple of American firms. The advantage of it is twofold; it removes from the crown one of the valve-pockets which are so difficult to cast with proper water-spacing all around when the head is of small size, and it makes a better job of the rocker-shaft with steadier rockers on account of the extra width of bearings that can be provided. Against it there is an extra layshaft and the added links which join to the eccentric mounting of the fuel-valve rockers. For the smaller sizes of Diesel engines the balance of advantages seems to lie with the horizontal starting valves. To the main couple of advantages cited above there attach certain corollary advantages. Accessibility profits by the absence of these valves from the crown, not only directly by reason of the lesser valve-gear overhead, but also indirectly by removal of the air-pipe to a less crowded position on the engine. Starting is also accomplished in a more straightforward manner, there being a single lever for getting the engine under way, whereas with the ordinary differential mounting of fuel-valve rockers and starting-rockers, there is on stationary engines almost without exception a separate handle on each cylinder, which entails climbing about. Land engines generally are arranged to start with air admitted to only one-half of the number of cylinders — or in the case of a three-cylinder engine two throws are most often fixed for air power. The single handle is very convenient. It notches in a sector and puts the starting valves into or out of action by the angular movement of the eccentric mountings of the valve-rockers. Simultaneously it throws out of, or into, operation the fuel-valve rockers, this being accomplished by rods joining the bearing of the injector rocker to a collar on the layshaft in each cylinder line.

For the camshaft drive a vertical shaft is stepped at one end of the crankshaft, the lower gearing being housed in a compartment open to the crankchamber whilst the upper gearing is encased. On this shaft is fitted the governor, which also is completely enclosed. This type of governor has been used on the Gardner vertical paraffin engines for some years and has proved most regular. The governing rod has in this case been taken up to a layshaft fitted parallel with the camshaft and acting upon the fulcrums of the pump suction-valve rods. Thus control of the fuel charges is obtained in the ordinary manner by shunting fuel back through the suction-valves on the delivery strokes. The plungers are operated by eccentrics from the camshaft, the group of four being subdivided into pairs for the sake of a symmetrical arrangement on both sides of the intervening camshaft bracket.

On the delivery side of the pumps each feed has an auxiliary check-valve in addition to the principal delivery valve, and there is also a test cock on each line. The fuel injectors are of the orthodox pattern with oil-spreading plates, but in lieu of a single orifice the flame plate has several injection channels spread fanwise to distribute the fuel well over the whole area of the combustion space. A patent packing, having apparently asbestos and graphite as its chief ingredients, has proved thoroughly suitable for the glands of the fuel-valves. Within the small dimensions of the injector casings used on this high-speed engine it is reasonable to suppose that metallic packing would scarcely satisfy the conditions. Owing to the small amount of space available on the cylinder heads the bolting flange of the fuel-valve casings are raised several inches for accessibility sake, as indicated in the drawing of the motor. Between the three valve chests of each cylinder crown there is sufficient space to ensure water passages, the inlet and exhaust valves for this reason being carried slightly outside the periphery of the cylinder bore. A cranked rocker is utilized for the fuel-valve motion, and curved rockers actuate the other overhead valves, the shape of these rockers permitting the camshaft to be kept somewhat lower. Ordinary pillar-supported rocker shafts are provided, and, owing to the horizontal arrangement of the starting-valves, there is sufficient room for wide rocker-bearings, which make for enhanced rigidity. The camshaft itself is being held by amply stiff brackets bolted to the cylinders, and on these same brackets are fitted the bearings for the starting-valve shaft. Grease cups are fitted to the overhead rockers, and ring-oilers are provided for the camshaft bearings.

A compound compressor supplies the air for the blast, the air being cooled on both stages. The low-pressure intake is surmounted by a regulator of uncommon pattern, which consists of a large diameter helical spring fitted with a tightening screw. The air is drawn through the long spiral slit which acts both as a silencer and as a throttle for the tightening of the coil reduces the effective intake area. Both the connecting rods of the main and air compressor cylinders are of somewhat unusual construction as may be noticed in the drawing. Taking the compressor cylinder connecting rod first it will be noticed that a hole is drilled through its entire length; and above, the pin carrying the pump operating stirrups is of greater diameter than below. Oil is fed to the big end of the rod through channels drilled in the crankshaft, shown dotted in the sectional elevation, from where it is led to the small end bearing. Unless the oil-way were enlarged at the point where the pin is carried through the rod it would completely choke the further passage of the oil and so prevent the lubrication of the small end bearing. An unusual form of construction is also used for providing adjustability to the small end brass. Above the eye in the connecting rod is a key-like piece of metal whose inner side is machined to coincide with the radius on the outer surface of the bearing brass, which is held in a rectangular slot cut in the end of the connecting rod. The brass of the bearing is split in the usual manner and packing pieces are interposed between the two halves so that when refitting becomes necessary the operation is simplified because removal of a thin packing strip will allow the two parts to come closer together, and they may then be scraped to fit the gudgeon. After this has been done the set screw in the end of the rod is screwed down on to the special strip of metal which in turn pressed the two halves of the bearing together and down solidly into the rod. A lock nut on the set screw then makes all secure. A similar arrangement is employed on the main connecting rods, but owing to the increased size, two set screws are used to secure the two half bearings into the rod. In the big end of each connecting rod a plug must be fitted to prevent escape of lubricant into the base through the hole that is bored for the passage of the oil along the rod to the small end. To secure the gudgeon to the piston bosses to prevent rotation a key is used, as shown in the illustration, Fig. 329. It will be seen that a flat is milled at each end of the gudgeon and a half round key fitting into a channel in the piston bosses, the flat part being in contact with the flat on the gudgeon. It is held in position by a set screw which presses on its under side and which is locked by a nut making the thread tight in the boss.

A plan view of two of the groups of fuel pumps is given at Fig. 329, and it will be observed that each pump plunger delivers oil to the cylinder next to it and not to the one with whose number it corresponds. By tracing the strokes of fuel plungers and pistons it will be seen that by this arrangement the effect of the governor is brought into operation without the lag that would be incurred if each pump delivered oil to its respective cylinder. Thus the governor limits the power of each impulse immediately after the last has taken place and does not allow an accumulation of greater charges than necessary before cutting down the oil supply. Hence, in conjunction with the governor which is already very sensitive, a very great degree of cyclic regularity is obtainable. A close examination of the high pressure compressor piston at Fig. 329 shows that the end is not made in sections but the rings have to be sprung over the enlarged part before being received in their grooves. No doubt the reason for enclosing the end of the ram by a screw stud is to obtain an end that will allow of a very small clearance while leaving the ram as light as possible. By this method the top may be bored internally until only the necessary thickness of metal is left, and the top may then be closed by the screw, whose head can be of a suitable contour.

AUGSBURG DOUBLE-ACTING HORIZONTAL DIESEL ENGINE

Although for industrial purposes the horizontal double-acting Diesel engine is a new style of power-unit, the type is in reality six years old. It was in 1908 that the Augsburg firm installed in its own power station the first engine of this class, a tandem 600 b.h.p. set, the running of which under service conditions gave proof of the value and practicability of the type. Naturally, more than a few months were necessary to establish conclusively the capacity of the engine for continued satisfaction, and no orders were taken for some time. The first contract then made was for a unit of considerably greater power than the trial set, a 1,600 b.h.p. twin tandem engine being fixed on the order of the municipal authorities of Halle a/S for the central lighting and tramway station of that town. This was put into service early in 1911, since which date Halle has been somewhat of a Mecca for users of large Diesel plants.

In a tandem pair of double-acting cylinders the four combustion chambers enable four power impulses to be obtained per cylinder of four strokes. By setting the cycle of operations in the individual combustion chambers consecutively 180 degrees forward, from the back of the engine to the front, the power impulses can be obtained regularly one to each stroke. This is the timing

practiced in the Augsburg design, that is to state, when the pistons are on a stroke from the back dead-center to the front dead-center then the various operations occurring simultaneously in the combustion chambers counted Nos. 4, 3, 2, 1 from back to front are respectively suction, compression, firing and exhaust. On the return stroke the operations in the same order of cylinders will become compression, firing, exhaust and suction. With a single crank a turning moment is realized equal to that of a four-crank single-acting engine and, if two tandem pairs of double-acting cylinders are combined in a twin set, the turning moment is as uniform as that of an ordinary eight-crank engine. Since as a matter of fact no eight-cylinder four-cycle engines are built for ordinary industrial purposes on account of the expense of constructing the multiplied parts, it can be stated that the twin tandem double-acting engine has a smaller co-efficient of fluctuation that any other four-cycle design. Over the four-cylinder two-cycle pattern it has, however, no superiority in this respect.

In considering the frame of the machine the relation between the stresses in the driving parts of the tandem double-acting set and those in the single-acting single-cylinder four-cycle engine of one-quarter of the power should be observed, as the stresses due to the drive are, except for the inertia effects, quantitatively the same, the increase of power in the tandem set being due to the utilization for power of the idle three strokes of the smaller machine with which comparison is made. The driving parts being thus not stressed in greater magnitude, but only in greater continuity in the tandem set, there is no call for larger dimensions on that account. Actually there are slight differences, but they are not of great importance; the reciprocating parts have a greater inertia and a reversal of driving thrust occurs, this latter, in particular, entailing heavier dimensions for the big-end. The consideration that the stressing is simply made more continuous and not greater, means that the cylinder framing in the two cases is alike. Thus in the tandem double-acting engine the utilization of material is much more efficient from the points of view of weight, production and, consequently, of cost. Each of these factors has its individual significance. Of course, the tandem Augsburg engine has to offset the cost of the closing of one end of one cylinder and the addition of another double-acting cylinder with stuffing-boxes, and the piston rod with the extra piston, as against three more cylinders with their extra framing and driving parts in an ordinary single-acting motor.

Referring to the sectional view of the Augsburg engine at Fig. 330, the manner in which the framing is built up may be observed. There are six sections, the bedplate, the two cylinders, the pedestal trunk between them, the rear pedestal and the rear guide housing. These are all bolted up stiffly in line, but only

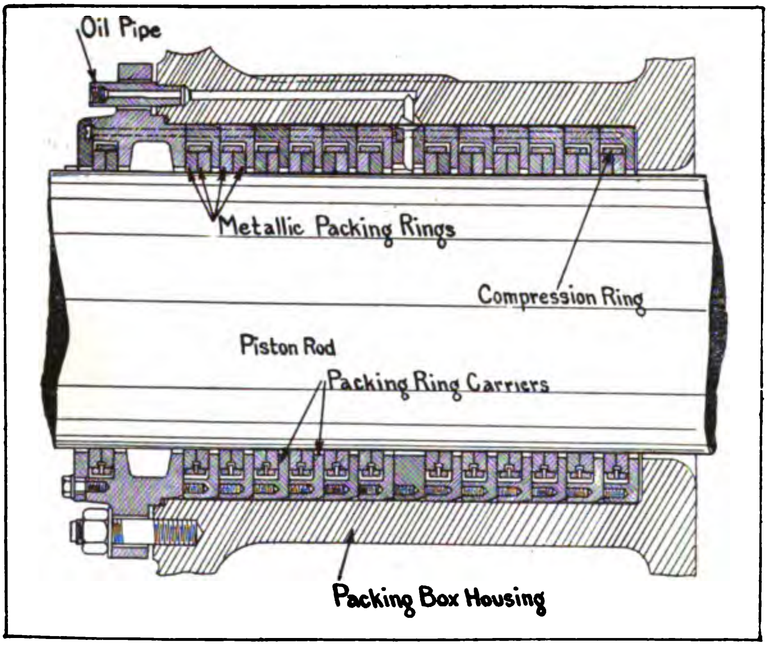


FIG. 331.—Diagram showing construction of metallic piston rod packing used on double-acting Augsburg-Diesel engine.

the bedplate section is fastened down rigidly to a foundation, because this is the only section that takes the drive, the other sections being subjected merely to the stresses arising from the gas operations. The weight of the cylinder framing is borne by two pedestals, one formed with the waist between the cylinders and the other at the rear end, while the feet of these pedestals rest on angles supported by two walls, between which space is afforded in a pit to give access to the valves under the engine.

Erected in this manner the cylinder framing is free to adapt itself longitudinally as it expands or contracts in heating or cooling. Also it is a point for remark that this type of engine is less exacting than any other type of equal power in relation to the nature of the soil on which the foundations are fixed, and sometimes this counts as a great advantage.

In each cylinder are two inlet and two exhaust-valves, one of each behind and in front of the cylinder respectively. To the inlet valves which are above, fresh air is brought through piping from the outside of the building, a provision which saves a lot of noise in the engine-room. With the exhaust valves under the cylinders, as at Fig. 332, the hot gases are led away without passing through the engine-room, which avoids unpleasant heat, and leaves the space round the engine clearer for the attendants. For a tandem set only a single camshaft is used to actuate all the valves, but, of course, on a twin tandem engine there are two camshafts. The position of the shaft along the near side of the engine at the height of the middle of the cylinders is extremely convenient for attendance. At the forward end it is driven by spur gearing off a lower shaft driven by skew-geared from the crankshaft, and it runs at half the engine speed. Supported in five bearings carried by brackets bolted to the cylinders, it has ring oiling with a drip tray under each bearing, while in addition to the valve rods, it actuates the fuel-pumps and the vertical governor shaft and drives the lubricating pumps.

Besides the inlet valve and exhaust valve each cylinder is fitted with two fuel valves, the double fuel injection being needed for a reason which will appear later. This entails, therefore, four cams per cylinder without counting a cam for the starting valve. Actually the two cams for the duplicate injectors are formed into one cam-block, the cam-peak for one injector being in the middle of the block, while that for the other is composed of two narrow pieces abreast of each other on the edges of the block. To the motion rod of the first injector a plain roller is fitted which rides over the central peak, but permits the slotted peak to pass it on both sides. A forked roller, however, is furnished on the rod of the second injector and this lifts from the slotted peak, but bridges over the central peak. In each combustion chamber the duplicate injectors must open simultaneously and the angle between their deliveries in the cylinder is about 120 degrees, but between the roller ends of the motion rods of the valves there is only about 75 degrees, which, therefore, is the angle between the central peak and the slotted peak on the cam circle. This angle is arbitrarily chosen when the design is being got out, but is kept small in order to have a favorable angle of incidence between the cam-peaks and the rollers. In erection it can be adjusted very accurately owing to the separate fitting of the peaks into the cam-block. The motion rods of the injectors are confined by radius links eccentrically mounted on a layshaft at the rear of the cam-shaft. There is a separate layshaft for each cylinder, the reason

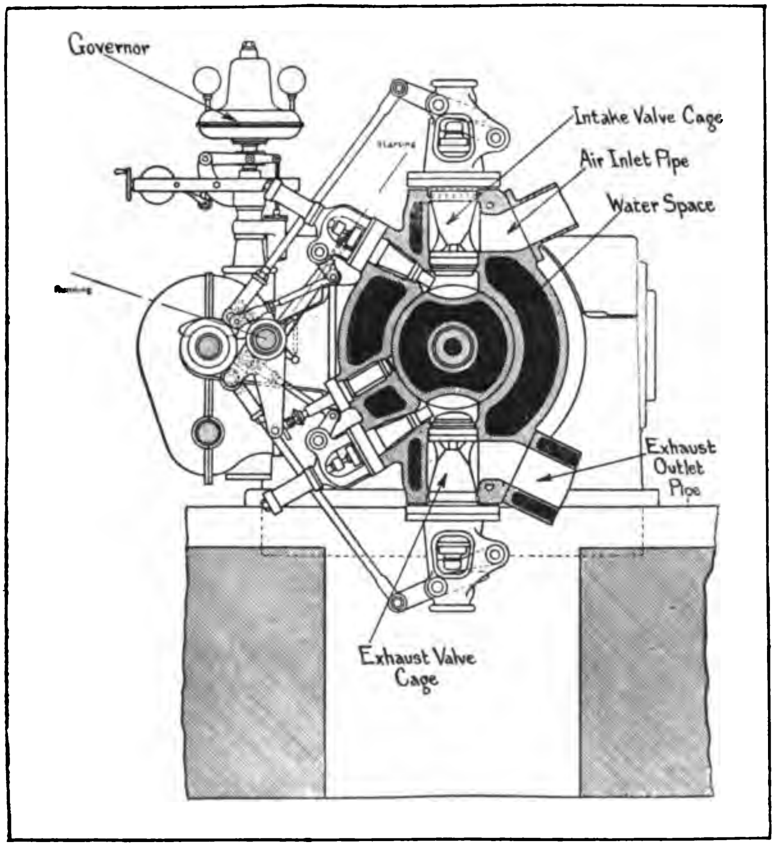


FIG. 332.—Sectional view through valve chamber of Augsburg-Diesel engine.

being merely that the space abreast of the center pedestal of the engine is required for pumps. It entails, however, the provision of separate levers to cut out the fuel-valves of the front and rear cylinders respectively, these levers acting upon the eccentric mountings of the links that confine the roller ends of the fuel-valve rods. Starting-air valves are fitted only into the ends of the forward cylinder, and in this case the control of the fuel valves is disposed differentially with that of the starting-valves, preventing both groups from being in operation at one and the same moment. When the engine is being started the fuel valves of the rear cylinder are thrown in before the air is cut off from the forward cylinder and then, as soon as the engine has picked up on the rear cylinder, the air-valves are cut out and the injection valves thrown in simultaneously on the forward cylinder. In twin tandem sets the usual arrangement is to fit the starting-valves in the forward cylinders of both throws.

At the end of its stroke the piston comes right up to the inner face of the cover, the clearance being very small, and the compression space is the volume of the two pockets formed where the cover is dished out (in correspondence with these pockets the piston is also dished). It is into these pockets that the fuel is injected and, therefore, the piston rod is shielded from the most intense heat of combustion, which is a consideration that has an important influence on the stuffing-boxes. The distinctive character of the combustion chamber and of the injection system is due to this very consideration. It is the most weighty factor in the satisfactory running of the engine, and in its patents covering the fuel injection and combustion arrangements of this design the Augsburg firm possesses considerable advantages over competitors. The packing of the cylindrical joint of the cover is done with a band of some bronze alloy, and a big water space is available round the stuffing-box, so that, with proper supervision, this part of the engine gives no trouble whatever.

These stuffing-boxes have no other duty to perform than to keep the piston rod gland from blowing, as none of the weight of the piston nor of the rod is taken by them, the entire mass of the moving parts being borne by the cross-head guide and slipper guides. The actual design of stuffing-box employed by the Augsburg firm we are unable to illustrate, but it follows very closely

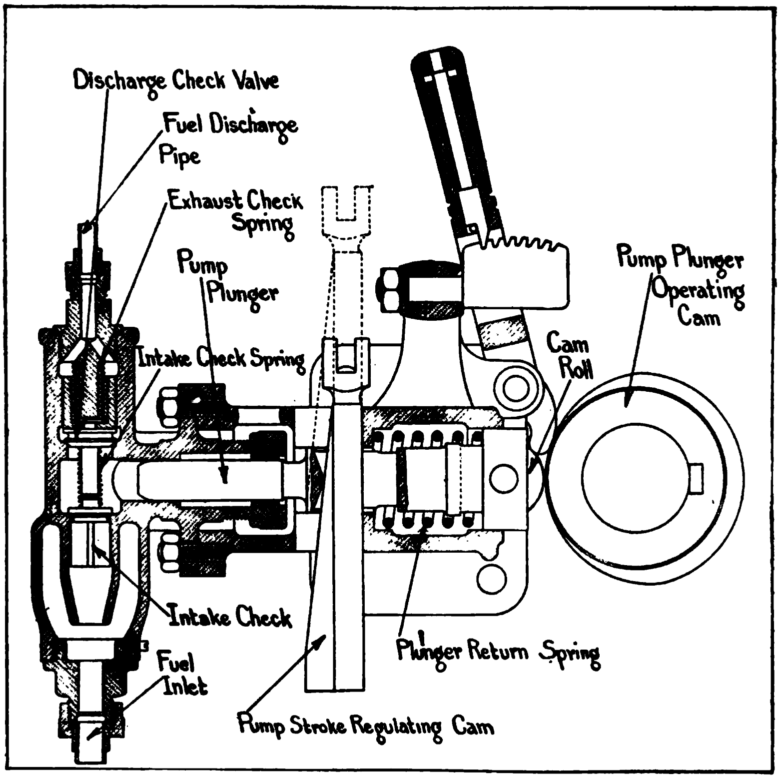


FIG. 333.—Sectional view showing construction of fuel supply pump on the Koerting-Diesel engine.

the arrangement shown in Fig. 331, which illustrates the stuffing-box of a double-acting engine constructed by the allied Nuremberg firm, which has remained tight after continuous runs of 120 hours, though it is nothing more than an adaptation of the ordinary piston packing. There are a number of sections containing triple rings with a central device for lubricating, and success depends almost entirely upon maintaining a continuous flow of oil.

Relative to the advantages of the various Diesel engine types, there is much to be said in favor of either the vertical or horizontal forms. There is scope for the cheap production of electric power by Diesel engines of the horizontal type; and in such cases, where for the most part single- or double-cylinder engines would be used, the horizontal type would run more smoothly, owing to the frame re-actions being taken up in a heavier mass with a lower mass-center. The relative shallowness of the foundation for the horizontal engine is also advantageous to such installations. The cost of production plays an important part in the comparison. Given definite manufacturing conditions, i. e., quality of material, workmanship, and finish being fixed, then a factory equipped for the work can turn out the horizontal type at a price twenty per cent. or thirty per cent. below that of the vertical type. The higher of these figures is given on the authority of a firm which is producing both types. It is evident that this reduction in price

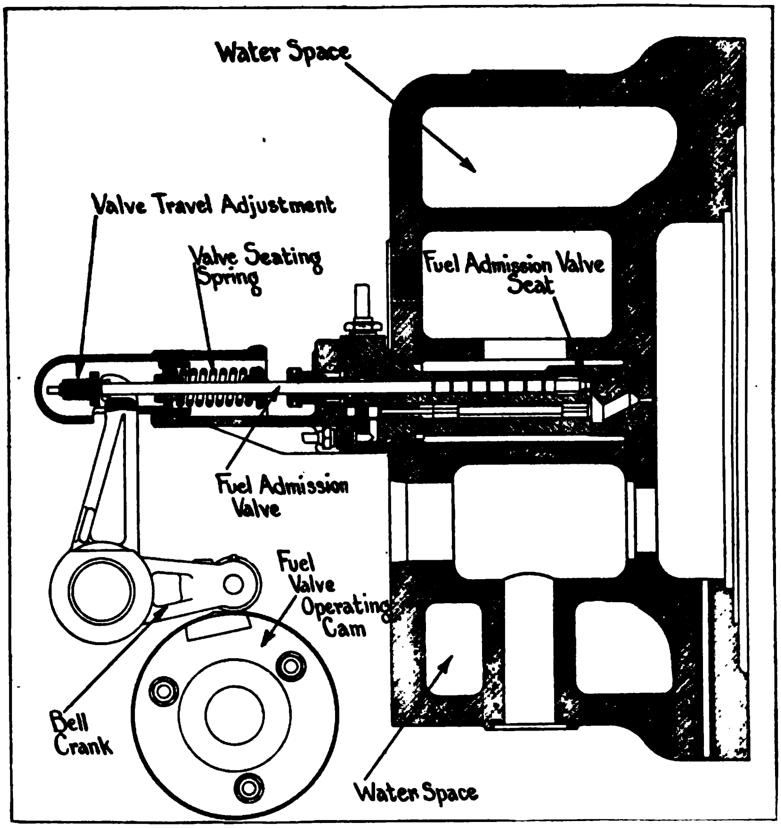


FIG. 334.—Details of the Koerting-Diesel fuel injection valve and method of operation.

will extend the market for Diesel engines considerably, against which it is frequently alleged that the capital cost is too high. Of course, there are horizontal engines being produced which are more expensive than certain of the vertical type, but difference in the quality of the work is responsible entirely for that seeming negation of the fundamentally cheaper construction of the horizontal design.

In respect of ground space the vertical engine is less exacting. In towns and cities this aspect of power installations is often of considerable importance, not necessarily on account of the cost of ground, but more generally perhaps owing to the impossibility of obtaining ground for extension. Beyond these circumstances the question of ground space is seldom weighty. For high speeds the vertical type is also practically the only one to be considered, but in nearly every other instance there is a direct challenge between the two types. In the larger horizontal engines, built on the tandem double-acting or tandem opposed-piston style, there is an advantage of crank effort which no constructor of vertical engines would think of attempting to obtain. An equivalent torque can be realized by multiplying the number of cylinders in the vertical engine, but that expedient entails a loss of the economy of production which the horizontal constructors achieve on a single crank-throw. Thus from the smallest to the highest powers the merits of the horizontal type are most surely not to be despised.

The most interesting detail of construction to the designer familiar with current gas engine practice when considering the Diesel design is undoubtedly that of the fuel supply system. The construction of a fuel pump employed on the Koerting horizontal Diesel engine is clearly outlined at Fig. 333 with important parts shown. A typical fuel admission valve, also of Koerting design, is shown at Fig. 334. The method of varying the degree of valve lift and the method of operation by cam and bell crank, as well as the location of the injector valve assembly in the cylinder, is clearly shown in drawing.

CHAPTER XVIII

FARM MOTORS — GAS TRACTOR POWER PLANTS — ELECTRIC PLANTS

Before the advent of the internal combustion motor it was difficult for the farmer or other user of small power plants to realize the advantages of mechanical power on account of the complication of the steam-engine and boiler outfits then used for power and the skill required to operate them. Attempts were made to utilize the horse by means of the tread mill but this was a very clumsy method of securing the limited energy of the animal. The water-wheel had but restricted application because water is not always available and the supply often fails in dry weather. The wind mills depending, as they do, upon uncertain climatic variations, and changes in atmospheric conditions, also were objectionable because of the unreliable nature of the motive force. It might be available at a time that power was needed but if there was no wind, it made no difference how urgent the demand was for power as it could not be obtained.

The ideal power for the agriculturist must be ready for use at any time, and should not cost anything to maintain except when in use. It should be easy to operate and portable, not subject to or depending on weather conditions and should be available at any desired point indoors or out. The source of power must also be reliable, safe in use and possessing a limited fire risk. The gasoline-engine is the one form of power that best answers these various demands. Its flexibility and simplicity permit of adapting it to many different conditions. The operator of a gas or gasoline-engine needs no greater training than required to operate any of the farm machines. Gasoline-motors for farm use can be obtained in one-fourth horse-power sizes for light work incidental to the household or in as high power or as great capacity as conditions demand. The farm engines of to-day work properly in zero weather or excessive heat and function reliably no matter what the thermometer registers.

When applied to the traction engine the gasoline-engine is capable of furnishing the power for ploughing, harrowing and seeding all in one operation. This work is being done by the square mile instead of the acre as formerly and is being accomplished better, quicker and cheaper than horse-power can do it. Gas traction engines are harvesting the grain when fields would be too soft to use the ordinary horse-drawn binder and under conditions where the steam tractor would be helpless. At the harvesting it will furnish power for threshing, and then it may be employed to convey the produce to market. It is capable of loading hay in the fields and then placing it in stacks or unloading at the barns. The working-life of the gasoline or kerosene-engine is measured in periods of 24-hour days instead of eight or ten hours or less as is true of horse flesh. The gasoline engine may be employed in irrigating fields and by furnishing power for spraying, it is saving much fruit that would otherwise be consumed by parasites and fungus growth. The gasoline-engine is cutting the wood in the forests, hauling the logs to the mills and furnishing the power to transform these into lumber. It is hard to mention any phase of farm life or any process incidental to agricultural work that cannot be performed more cheaply and quicker with the aid of the internal combustion motor than with any other form of power.

For farm work most of the engines govern on the hit-and-miss system, that is, the engine will take an explosion and will then run idle until the speed falls sufficiently, when the explosions will again occur until the speed has risen to the proper point. This cutting out or missing of the explosions is obtained in a variety of ways by the different makers, some holding the exhaust valve open so that, instead of drawing in a charge of fuel and air on the suction stroke, inert exhaust gases are drawn in instead. In others, the fuel alone is cut off and a charge of air drawn in, compressed and then exhausted. It is understood, of course, that these operations are controlled entirely by the governor. Theoretically each explosion occurs in the cylinder at its maximum efficiency under proper working conditions so that for the lighter loads the hit-and-miss engine is much more economical of fuel than the throttling type. However, for close regulation and in the larger sizes the throttling engines are used exclusively. In most hit-and-miss engines the low tension make-and-break system of ignition is used, this being connected to the governor in such a manner that on the idle strokes its operation is also cut out.

Farm engines are usually made in simple one-cylinder forms when used for stationary power and may be of either the vertical or horizontal pattern. Most all farm engines are provided with a substantial base by which they may be easily supported on any stationary or portable foundation. When more power is needed than can be obtained from a single cylinder it is possible to use two-cylinder motors of the heavy duty type in either the vertical

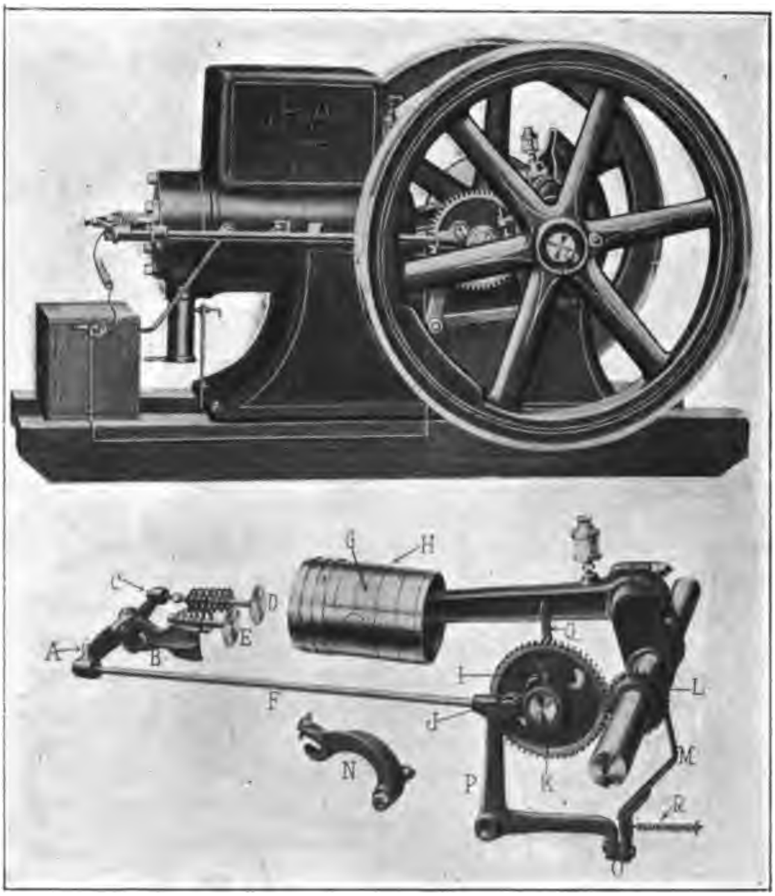


Fig. 335.—The Gray farm motor and important internal parts.

or horizontal pattern. The power plants used for the best gas tractors are generally of the four-cylinder form.

A light stationary engine of Gray design is shown at the top of Fig. 335, while its important internal parts are outlined at the bottom of the illustration. It will be noticed that the engine is of the horizontal pattern and is mounted on a very substantial base of cast iron which in turn is securely attached to a pair of hard wood timbers or skids. The power plant as outlined is complete inasmuch as it carries its ignition, cooling, lubrication and fuel supply system on the same base. The cylinder is lubricated by a sight feed oiler while a similar device is mounted on the connecting rod and is used to supply lubricant to the crank pin bearings. The engine is of the hopper cooled pattern and has an automatic inlet valve. The exhaust valve is operated by a simple rocker arm that is worked by a cam through the medium of a push rod and lever.

The longitudinal sectional view of this engine given at Fig. 336 shows clearly the internal construction and arrangement of parts. It will be noticed that the gasoline tank is enclosed inside of the engine base and that a large cast iron hopper is cast integrally with the water jacket in order to hold an adequate amount of cooling water. The method by which the sight feed oiler attached to the cylinder serves to lubricate the cylinder walls, piston and wrist pin bearing of the connecting rod is clearly indicated. The location of the auxiliary exhaust port at the bottom of the cylinder which is uncovered by the piston when that member reaches the end of its power stroke is also outlined. The direct gas passage provided by the valve-in-the-head construction may also be ascertained.

The horizontal engines of the simple form, and many of the vertical ones as well, are provided with two flywheels, one at each end of the crankshaft. The weight is thus divided between two members and the crankshaft is not stressed as much as it would be if a single large flywheel member, weighing as much as the two, was mounted at but one end of the crankshaft. The part sectional view of the horizontal cylinder of the I. H. C. engine shown at Fig. 337 shows clearly the application of the valve-in-the-head principle and the hopper cooling system often employed on light or medium-powered farm motors.

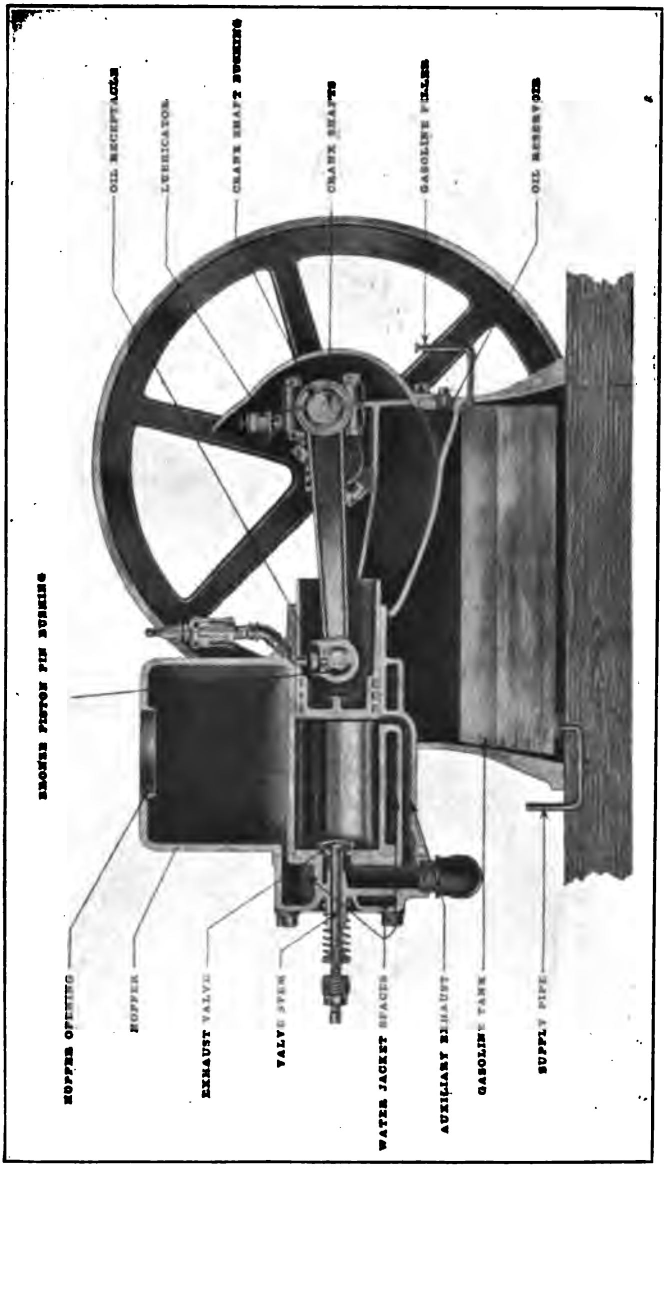


FIG. 336.—Sectional view showing internal construction of the Gray farm motor.

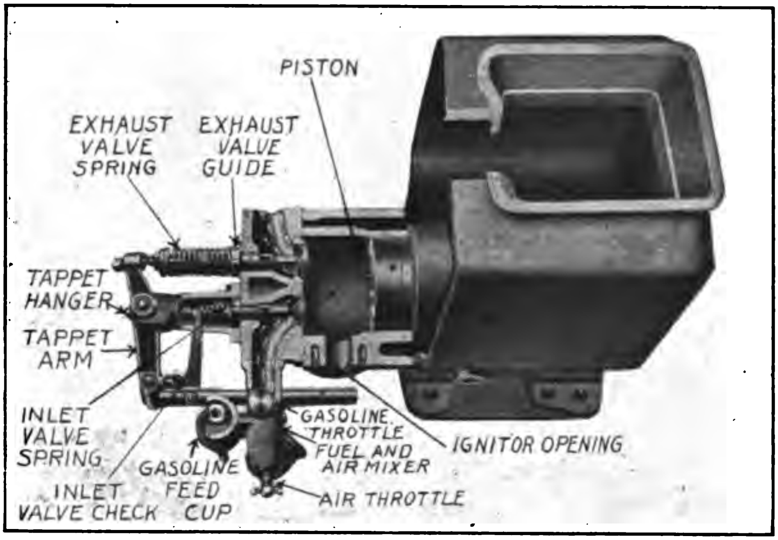


Fig. 337.—Part sectional view of hopper cooled farm motor cylinder with valves in head.

The construction of a vertical cylinder farm motor is clearly shown at Fig. 338. The view at A shows the general arrangement of the parts very clearly, such as the application of the two flywheels, the pulley for delivering power attached at one end of the crankshaft and the governor for regulating the engine speed at the other end. The view at B with one of the flywheels removed shows the application of the plunger pump for circulating the cooling water through the cylinder jacket and of the smaller plunger pump attached to the engine base employed for drawing fuel from the container which may be placed at any convenient point adjacent the engine. The internal construction of a vertical engine of the form outlined is clearly shown at Fig. 339.

When more power is needed than can be obtained by the large simple power plants, it is possible to secure engines of fairly large capacity of the form shown at Fig. 340. The one at A is a two-cylinder vertical type while that at B is a two-cylinder horizontal opposed form. The opposed engine usually runs with less vibration than the vertical type depicted at A, though when used in stationary installations the engines are generally provided with heavy flywheels and are secured to a firm base, so that the vibration of either type at the normal operating speed, even when working at the full load, is practically negligible.

When either of the types of engines shown at Fig. 340 are used it is necessary to provide a force pump to promote the flow of circulating water through the jackets. In the simple hopper system, such as indicated at Fig. 336, the cooling water is kept in motion by the thermal displacement and the heat absorbed from the engine cylinder is radiated to the air by the exposed surface of liquid at the top of the hopper and by the metal sides of the hopper as well. The hopper system is not used on large engines on account of the amount of water that would be necessary to insure proper cooling. When used in stationary

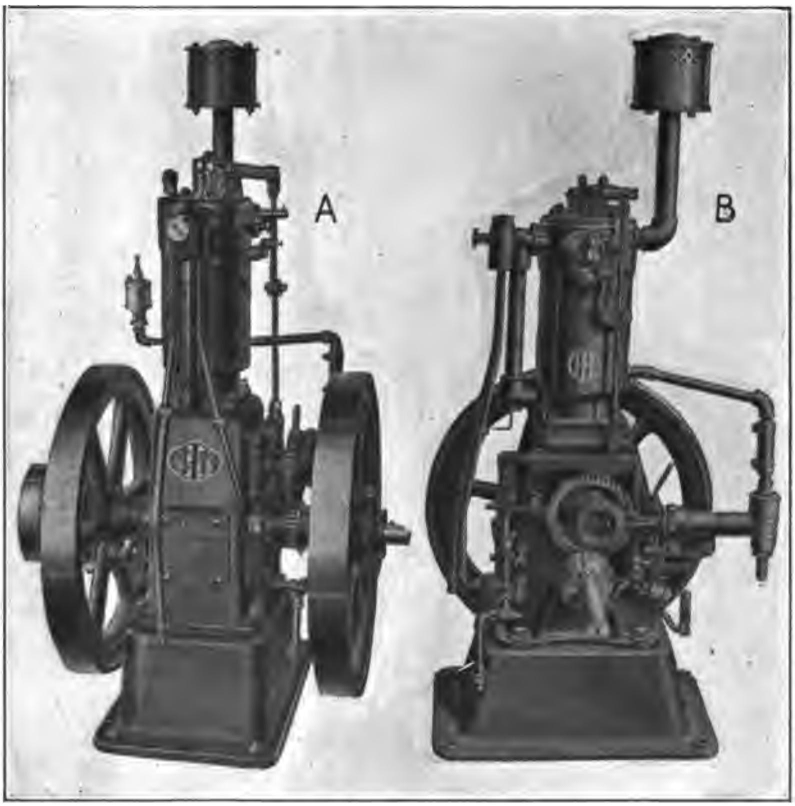


Fig. 338.—The I. H. C. single-cylinder vertical farm motor.

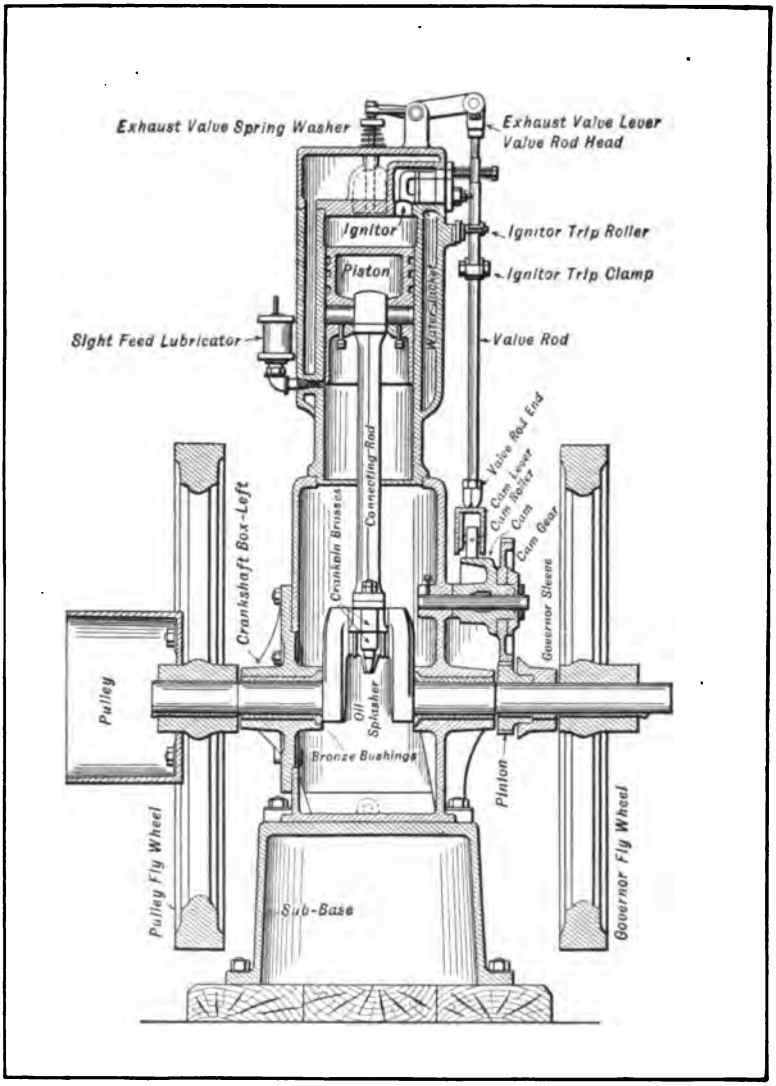


FIG. 339.—Sectional view of I. H. C. vertical cylinder farm motor.

application it is not difficult to have a large mass of water in a suitable container and to pump this through the water jacket by some positive form of pump. The heated water passing out of the top of the water jacket is often directed to a cooling

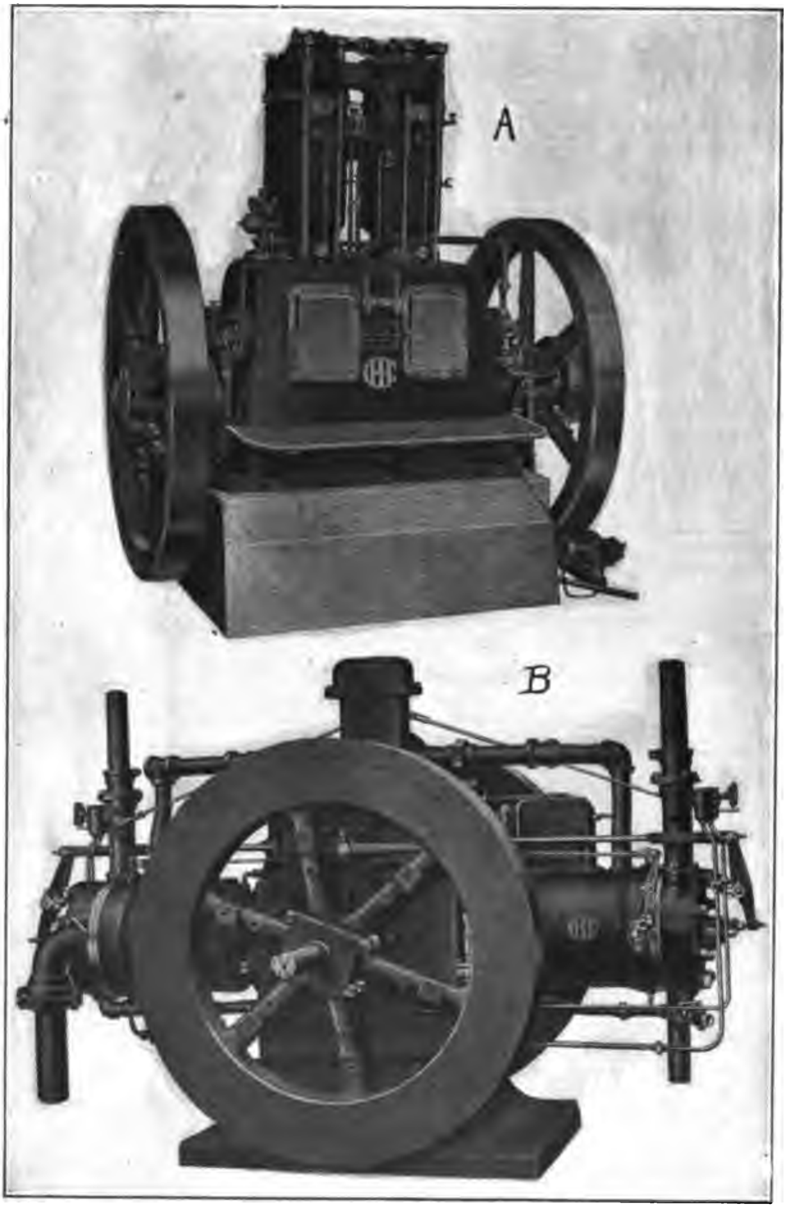


FIG. 340.—Showing two distinct types of two-cylinder farm motors.

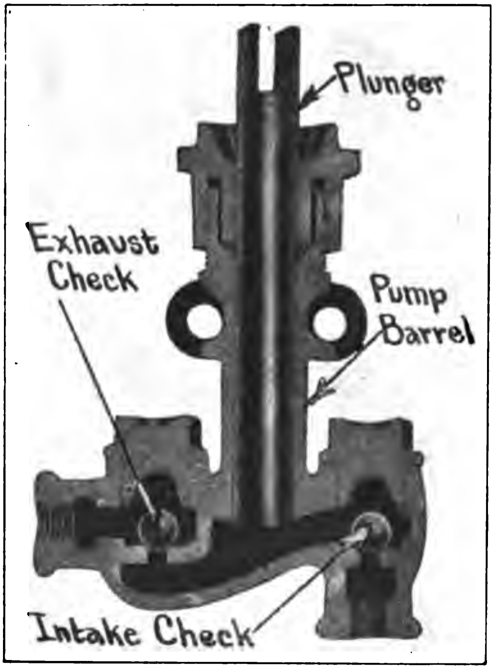


FIG. 341.—The plunger pump used in farm motor cooling systems.

element that consists of a cone of wire gauze or in some cases merely of sheet metal on the apex of which the hot water from the cylinder jacket is discharged. As the water runs down the sides of the cone into the containing tank below it, it is cooled by contact with the air and the cool walls of the metal cone. It is then pumped from the tank back to the water jacket of the engine. Internal combustion engines of large power, such as used for machine shop power plants and for small pumping and electric power generating stations, are often piped directly into the regular municipal water system and the water circulates by virtue of its pressure.

The form of pump at Fig. 341 is the type generally used when it is necessary to lift water from the tank to the vertical cylinder of a medium-size engine. This type of pump is also widely used in marine applications. A plunger reciprocates in the pump barrel, usually being moved up and down by connection with an eccentric driven by the engine. When the plunger goes up, the intake check ball is raised from its seat and water is drawn to fill the space in the pump body. On the return stroke of the plunger, the pressure closes the intake check and opens the exhaust check valve to permit the discharge of water into the engine cylinder jacket. The arrangement of the ball valves is such that one ball will lift only when the plunger goes up while the other leaves its seat only when the plunger is going down. The use of a pump of this nature provides for positive flow of the water.

Farm engines are usually built so that they will operate on other fuels besides gasoline. When employed for stationary use, if natural or artificial gas is available, it is possible to utilize this by the substitution of a very simple gas attachment in place of the usual form of gasoline vaporizer valves. When artificial gas is used the engine should be designed so as to have greater compression because this gas contains less heat units than either natural gas or gasoline vapor, and the gas must be compressed to a greater degree to obtain the same amount of power. Combination gas and gasoline attachments are available and are designed for use in localities where natural gas is available part of the time. They are so designed that natural gas alone may be used as a fuel until the pressure is reduced to such a point that artificial vapor must be supplied. This may be accomplished by opening the gasoline needle valve slightly and allowing the vapor of the liquid fuel to enter with the natural gas in order to produce an explosive mixture in the cylinder, rich enough to explode. If the gas supply fails temporarily the engine may be run on gasoline. Any of the stationary engines designed to use gasoline may be run on mixtures of gasoline and kerosene, on distillate, alcohol, benzol or any other relatively volatile, hydrocarbon liquid.

If proper precautions are taken, kerosene may be used. This requires the use of two tanks, one for gasoline alone and another for the kerosene. These tanks are so arranged in relation to the vaporizing device that the engine may be started on gasoline and when heated up the gasoline supply may be interrupted and the kerosene turned into the vaporizer. The best results are obtained when the vaporizer is provided with a jacket, through which the exhaust gases are deflected before they pass to the muffler.

The marked utility of the internal combustion engine on the farm is well shown by Fig. 342, which shows how the modern gasoline-engine makes possible the performance of many tasks that could only be done by manual labor. The gasoline-engine is in one compartment and drives an overhead line of shafting by belt connection. This shafting drives a grind-stone and a small dynamo for lighting purposes in the same room as the power plant; and various other machines, such as a lathe, drill, or blower for a forge could also be placed in this room if desired. The

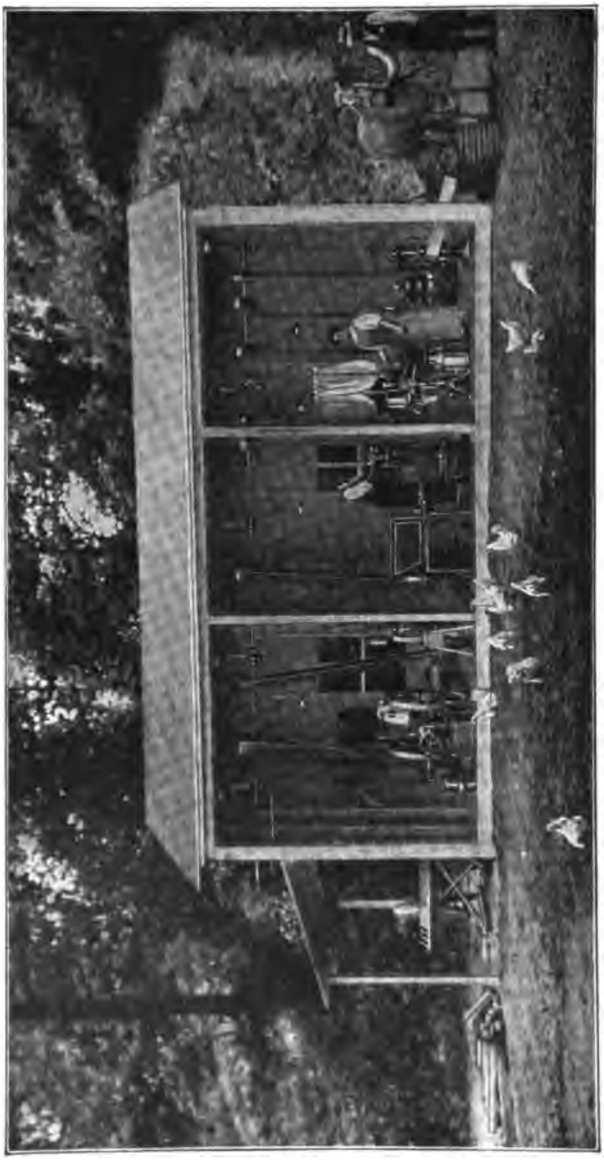


FIG. 342.—The new farm factory made possible by modern gasoline engine. Note wood saw at one end and watering trough at the other.

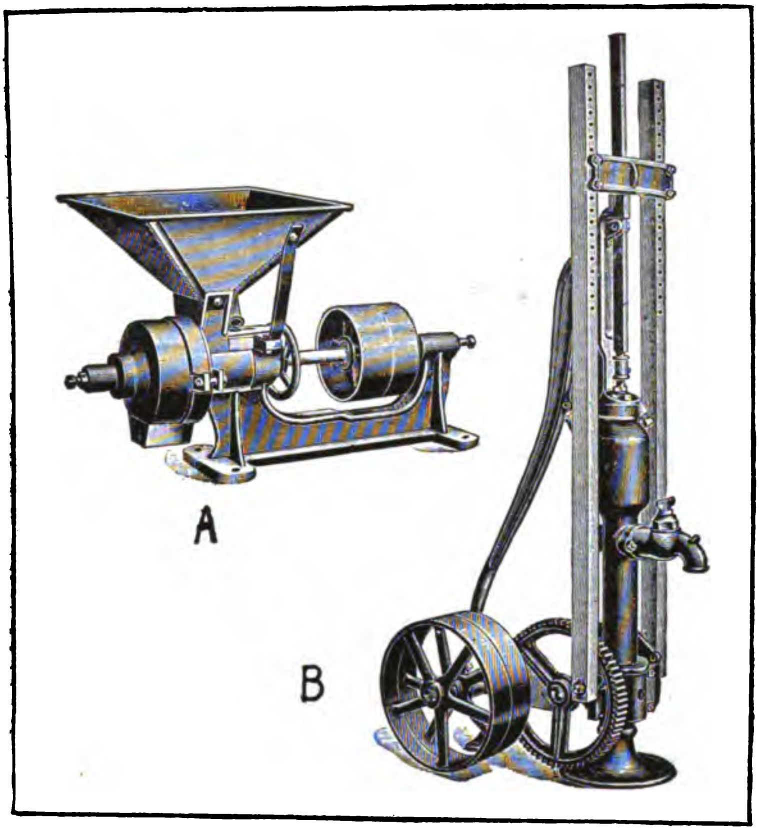


FIG. 343.—Two useful farm machines adapted for motor power.

end of the line shaft extends through the building at the power plant end and is belted to a wood saw, which is under a shed to make it readily accessible for cutting cord wood into short lengths and yet which offers sufficient protection from the weather. In the central compartment of the modern farm factory are feed grinders, ensilage cutters and other tools and machines of this nature. The third compartment is intended for the use and convenience of the lady of the farm and has a power-driven churn, washing-machine and water-pump. The water-pump discharges into a conductor which leads to a trough or tub outside of the building adapted for watering stock.

The arrangement outlined shows some of the possibilities and how the various machines may be grouped conveniently for operation by an engine too large to be readily portable. On most farms, engines of various sizes are used, one of these being of low power and adapted to be moved from one part of the farm to the other, as its power is needed. Many attachments have been designed for use with the gasoline motor. Two of the most useful are shown at Fig. 343. That at A is a grinding mill while at B the usual method of operating a plunger pump by mechanical power is shown. It will be noticed that in event of failure of the engine the pump may be worked by hand with the usual handle. At Fig. 344, a portable saw mill outfit operated by an oil motor is shown. This is mounted on substantial wheels and may

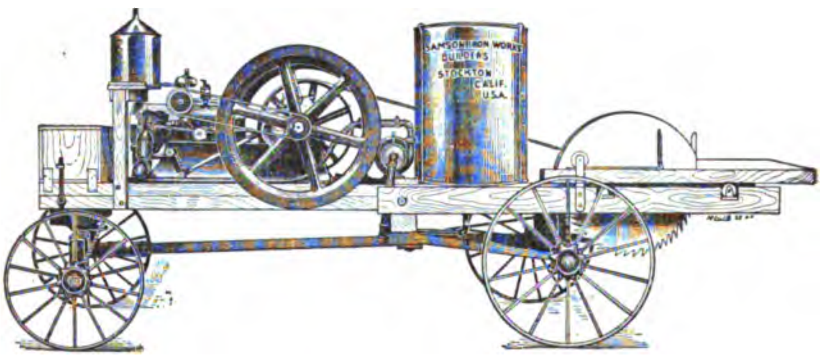


Fig. 344.—Portable oil-motor sawing-rig.

be drawn from one place to another by either animal or other motive power.

The application of a small air-cooled engine to drive a pump employed for spraying is clearly outlined at Fig. 345. The pump and engine are mounted on a common skid or carrying frame, the pump being driven from the engine crankshaft by a small spur pinion which engages with a larger spur pinion attached to the pump crankshaft. This reduction in speed is necessary because the speed of rotation needed to secure efficient power output from the gasoline-engine is too high for proper operation of the pump. It will be noted that a small high-speed fan is mounted at the side of the cylinder to direct a blast of air against the flanges and insure adequate cooling. This fan is driven at high speed by belt connection with the flywheel and turns much faster than the engine crankshaft on account of the relative difference in size between the small pulley on the fan hub and the diameter of the flywheel around which the driving belt passes. It is possible to show hundreds of examples, each one illustrating a different phase of activity and a practical use of the gasoline-engine. These presented are selected merely to show the variety of work this power plant is capable of performing and not because they are unusual or striking in any way.

It is agreed by authorities on agricultural science that there is no point on the farm where power can be used to better advantage or where it is more necessary than in the fields. It is said that more energy is spent in ploughing annually than in all the factories of the world, during the same period. Ploughing has always demanded more expenditure of energy than any other line of farm work and up to the development of the practical steam

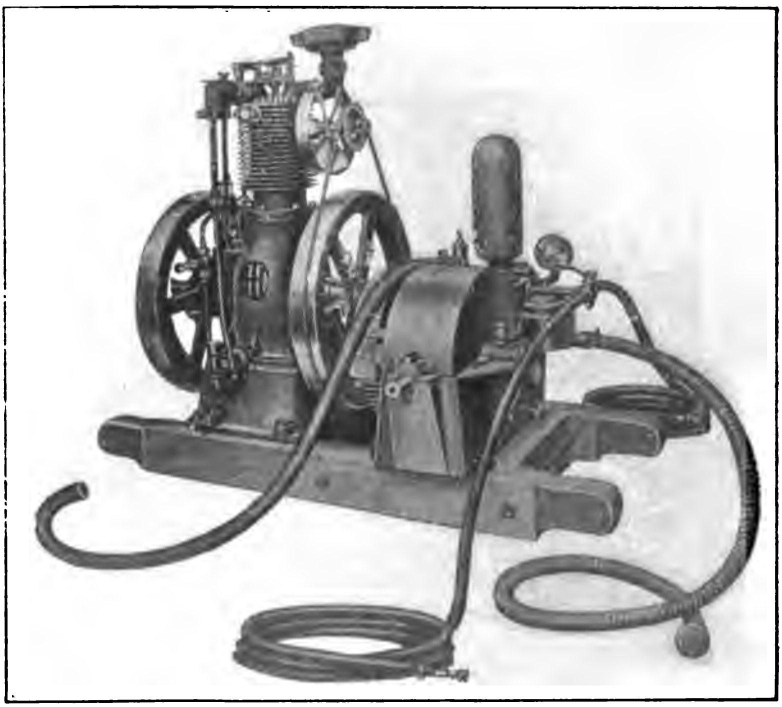


FIG. 345.—Portable spraying outfit using small air-cooled engine to operate the pump.

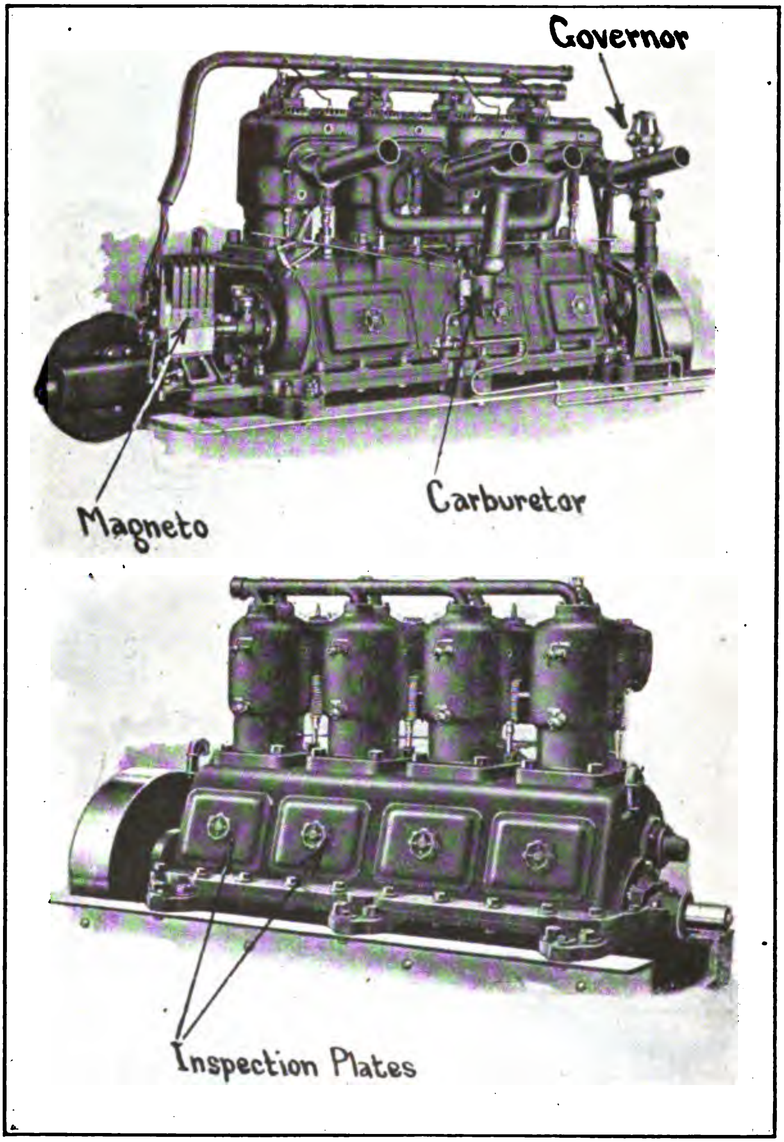


FIG. 346.—Views of the power plant of "Big Four" gas tractor.

traction engine, ploughing had always been done with animal power. While the steam traction engines are practical in every way, they require a skilled and in many States licensed operator to run them successfully. This is not true of the gas traction engine because the nature of the explosive motor makes it easily understood and operated by any one of average intelligence.

The gas tractor has been developed to such a point that it has replaced horses or other animal power on thousands of farms. Animal power is subject to all ills that flesh is heir to and is affected by extremes of temperature, either heat or cold. If the animal is injured or sickens, nature is the only possible repairman and it works slowly. The tractor, when propelled by the internal combustion motor, feels neither heat nor cold. It will operate equally well in summer or winter. If a tractor breaks down, any mechanic of average intelligence, and in many cases the ordinary grade of farm help, may replace worn or broken parts with but slight delay. A tractor can be housed cheaper than the number of horses needed to do the same work can and it requires no care and does not eat or consume energy when not in use as animals do. Most traction engines are provided with a belt pulley so they will furnish power to operate threshers, shredders, shellers, pumps, sawing outfits and any other form of machinery needing power. The usefulness of the tractor is more varied than that of any other farming machine. It is not only the most economical power for ploughing but it operates all machines necessary for raising and harvesting crops with equal economy. These include disc harrows, seeders, drills, packers, binders, etc. It can be used for hauling grain to the elevator, pulling stumps and hauling all kinds of lumber and supplies to the farm.

As the first power plants developed for general farm use were of the simple one and two-cylinder forms, it was but natural that the first gas tractors should utilize the same type of engines so widely used in stationary installations. It was not a difficult matter for the early designer to take a steam tractor running gear and driving mechanism, similar to that with which he was thoroughly familiar and to replace the boiler and steam-engine with one of the simple forms of stationary power plant. At the present time, practically all of the modern gas traction engines are provided with multiple cylinder engines because these are lighter and may be run faster than single cylinder ones of the same power. Regardless of the care taken in designing a single cylinder engine, it is inherently defective in that the power impulses will come in jerks and a very heavy flywheel member or pair of flywheels is needed to equalize the intermittent power strokes. In a multiple cylinder engine, especially of the four-cylinder type, explosions follow each other more closely and the power application is much more even. This means smoother operation and less wear and tear on the mechanism.

Gas tractor power plants may be divided into two types, one termed "the heavy duty," while the other class is termed "the medium duty." In the heavy duty types, all the bearings and working parts have been proportioned unusually large and the engines are usually of the slow-speed forms. While it is believed that the heavier construction of these engines makes for superior

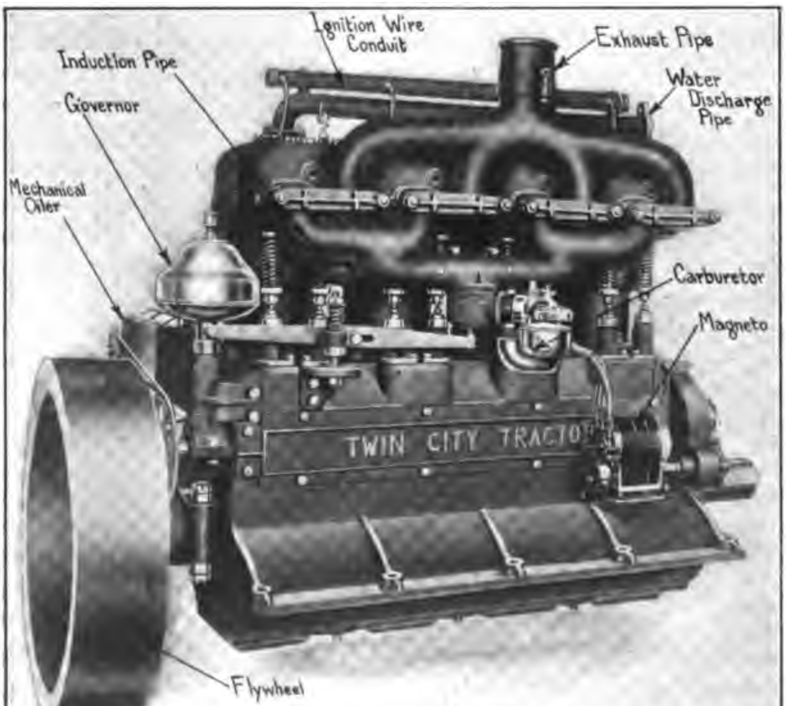


FIG. 347.—The "Twin City" gas tractor power plant.

endurance they usually vibrate more than the faster medium duty engines, and the larger reciprocating and bearing members are subjected to stresses of a magnitude proportional to their size. Heavy duty engines are usually of the horizontal type and have one or two cylinders. The medium duty engines are almost invariably of the four-cylinder, vertical pattern and follow the lines of construction that have proven so thoroughly practical in automobiles and motor boats. In a vertical motor of the four-cylinder pattern the parts may be lighter than in the simple horizontal forms and the same power obtained with smaller

FIG. 348.—Location of power plant on typical gas traction engine.

cylinders at a higher crankshaft speed. The weight of the piston and connecting rods is entirely removed from the walls of the cylinders as these parts practically float in a film of oil and their weight is carried by the easily adjusted main bearings of the engine rather than the cylinders, which can be restored to a true bore, when worn, only at considerable expense.

A marked advantage of the medium duty form of motor in the four-cylinder vertical types is that the vibratory stresses travel in vertical lines and are resisted by the parts best adapted to receive them, as the wheels and axles. In a horizontal motor

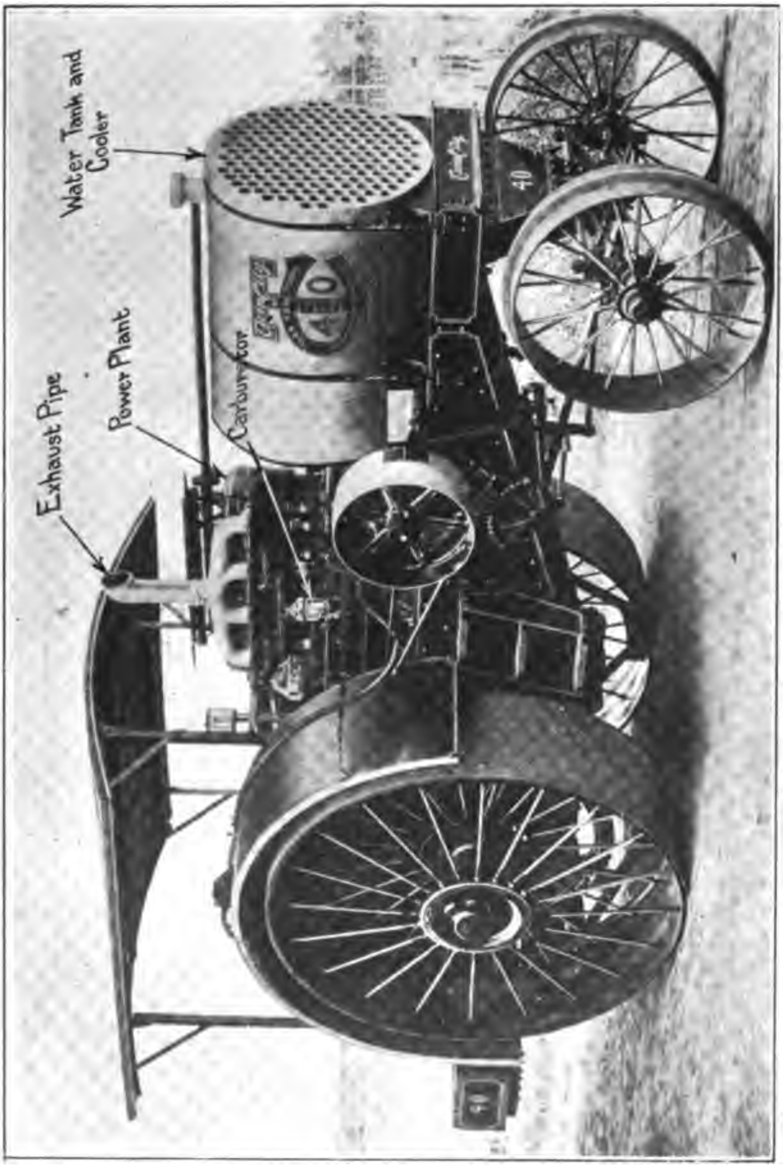


Fig. 349.—Side view of "Twin City 40" gas tractor, showing location of power plant and belt pulley for delivering power to farm machines.

vibration must be absorbed by the frame and gearing and considerable stress is placed on parts not well adapted to resist them. The vertical cylinder motor is usually more accessible than the horizontal engines as the vital parts which need more or less attention such as the valves, spark plugs, ignition wiring, water piping, etc., are more easily reached than in the horizontally disposed power plants.

A four-cylinder engine used on the tractors of the Gas Traction Company is shown at Fig. 346. The cylinders are individual castings of the L head form mounted on a substantial cast iron base provided with liberal-sized inspection plates which permit of ready access to the engine interior. Except for the more substantial proportioning of parts this engine does not differ essentially from the marine engines described in the preceding chapter or the automobile engines illustrated in the following chapter. The design is conventional in all respects, as satisfactory operation is obtained without radical departures from standard practice. The valve side of another very efficient gas tractor power plant is shown at Fig. 347. The location of the carburetor, magneto and governor, the belt drive for the mechanical oiler and the effective method of retaining the inlet and exhaust manifolds are clearly indicated.

The method of installing the gas traction engine in the frame and the relation the power plant bears to the other components of a typical modern form of traction engine is clearly depicted at Figs. 348 and 349. It will be noticed that all parts of the power plant are readily accessible and may be easily reached in event of trouble. The belt pulley from which power may be taken to drive various forms of stationary machines is clearly shown at Fig. 349 just above the top of the frame member and between the power plant and combined water tank and cooler. The view at Fig. 350 is the motor erecting room of a gas tractor factory and demonstrates conclusively that the advantages of this form of power are so well realized in the agricultural industry that a large demand exists which can only be supplied by systematic quantity production.

SMALL ELECTRIC-LIGHTING PLANTS

In the lighting of large dwellings or other buildings, where there

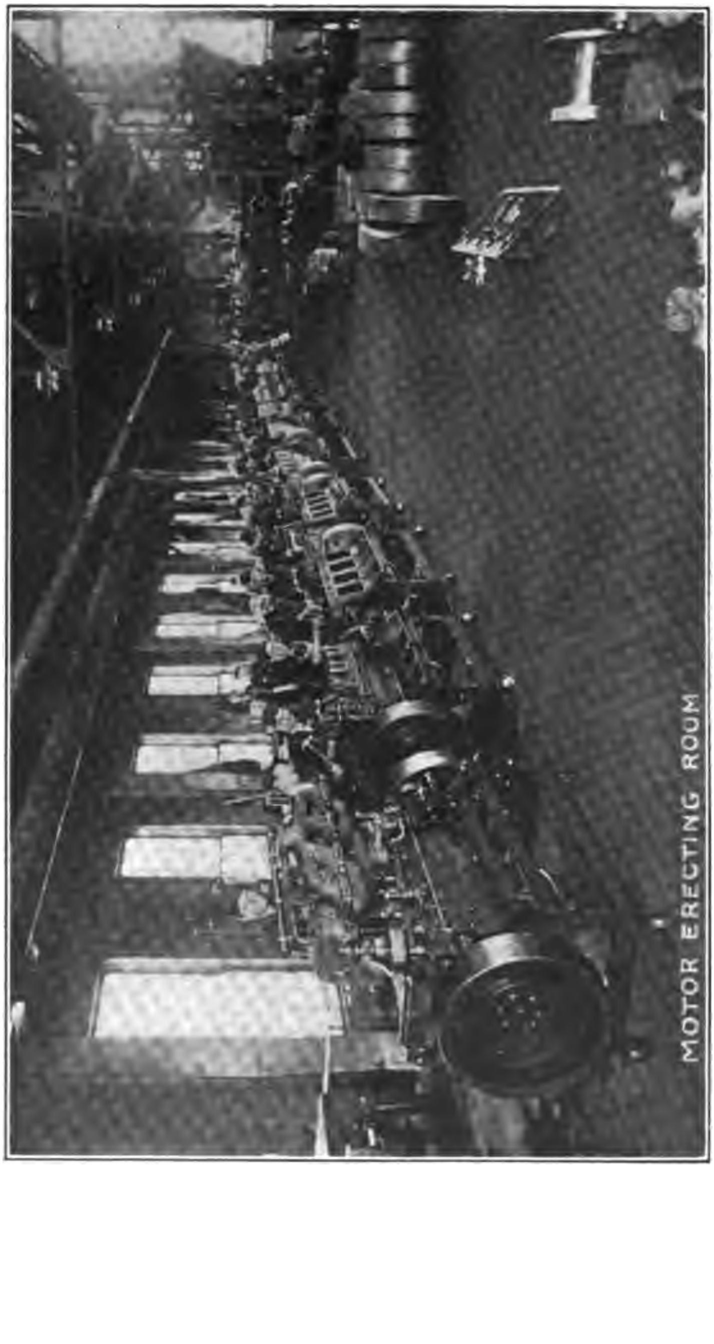


FIG. 350.—View in gas tractor power plant assembling room, showing motors in process.

is no power used for other purposes, the use of gas, gasoline, or oil-engines for operating an electric generator is not only cheaper in running expenses than the steam-engine, but the comparison holds good for the lighting of towns and villages at the usual cost of gas to consumers; but when the generation of producer-gas can be made for such use on the premises of the electric plant and by the same persons that operate the electric plant, the saving in cost of electric-lighting is several-fold less than by direct gas-burning. In many towns where oil producer-gas is used, the cost of material used in making the gas is less than thirty-five cents per

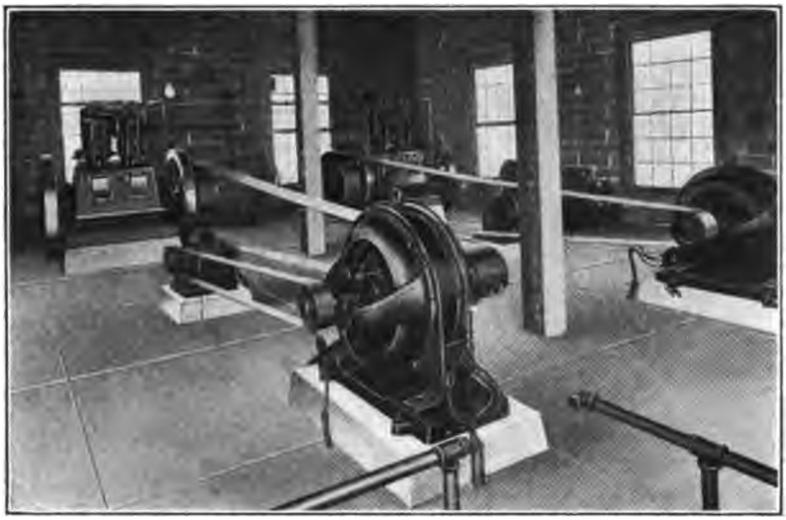


FIG. 351.—Small municipal electric lighting plant using gas-engines as prime movers.

thousand cubic feet of gas produced. In such places the labor of producing the gas for a town of say fifteen hundred inhabitants is from two to three hours per day, and in some towns, as observed by the author, three hours every other day — giving ample time for the same operator to run the electric plant in the evening, or both may be run simultaneously.

When the mere fact of the cost of gas for direct lighting and its cost for producing the same light by its use in a gas-engine to run an electric generator is considered, the difference in favor of electric-lighting in preference to direct gas-lighting is most apparent.

It has been known for some years that for equal light power but about one-half the volume of gas consumed in direct lighting will produce the same amount of candle-power when used in a gas-engine for generating electricity for lighting.

Again, when we leave the realm of a fixed gas and the cost of its producing-plant, the gasoline and oil-engine again come to the rescue of the fuel element for lighting, from an average cost of 7½ cents per hour for 192 candle-power in lights by direct

FIG. 352.—Small lighting plant using I. H. C. gas-engines running on producer gas.

illumination, and 2¼ cents for the same amount of light by the use of illuminating gas consumed in a gas-engine with electric generator, to one cent or less by the gasoline and oil-engine for equal light. In English trials with a Crossley engine of 54 indicated horse-power running a 25½-kilowatt generator (34 electrical horse-power), lighting 400 incandescent lamps (16 candle-power), consumed 1,130 cubic feet illuminating gas per hour, or 2.82 cubic feet of gas per lamp per hour. The gas used for direct lighting was 16 candle-power at 5 cubic feet per hour. Then, if it had been used for direct lighting, it would have produced $\frac{1130}{5} = 226$ — 16-candle-power gas-lights, a little over one-half the amount of the electric light — or the efficiency of the direct light would have been but 56.5 per cent.

To show the difference between running a gas-engine at full or less than full power, the same engine and generator when running with 300 incandescent lamps (16 candle-power) used 840 cubic feet of gas per hour, and $\frac{840}{5} = 168$ — 16 candle-power gas-lights, or 56 per cent. efficiency for direct lighting. When the lamps were cut out to one-half or 200, the consumption of gas was 740 cubic feet per hour, equal to $\frac{740}{5} = 148$ gas-lights, with a direct gas-light efficiency of 74 per cent. — the difference in efficiency being chiefly due to the constant value of the engine and generator friction in its relation to the variable power.

Another trial with a Tangye engine of a maximum 39 indicated horse-power running an 18.36-kilowatt generator (24.61 electrical horse-power), lighting 300 16 candle-power incandescent lamps, consumed 770 cubic feet illuminating gas per hour. With direct lighting, $\frac{770}{5} = 154$ gas-lights (16 candle-power), or an efficiency of 51 per cent. for direct lighting. With 220 incandescent lamps in, 640 cubic feet of gas were consumed per hour, equal to $\frac{640}{5} = 128$ gas-lights and a direct gas-light efficiency of $\frac{128}{220} = 58$ per cent. Again reducing to 100 lamps, 320 cubic feet of gas were used, equal to 64 gas-lights with an efficiency of 64 per cent. for direct gas-lighting.

It will readily be seen by inspection of these figures that the greatest economy in gas-engine power will be found in gauging the size of a gas-engine by the work it is to do when the work is a constant quantity. The investment of local lighting-plants by the use of gas, gasoline, and oil-engines in factories and large buildings has been found a great source of economy as against the direct use of municipal electric current or the direct use of gas. The gasoline or oil-engine makes a most favorable return in economy when used for local lighting as against the prevailing price charged by the operators of large steam-power installations for town and city lighting.

In a trial of eleven days by a 10 horse-power four-cycle gas-engine of the vertical pattern, belted direct to a 150-light direct-current generator making 1,600 revolutions per minute, with

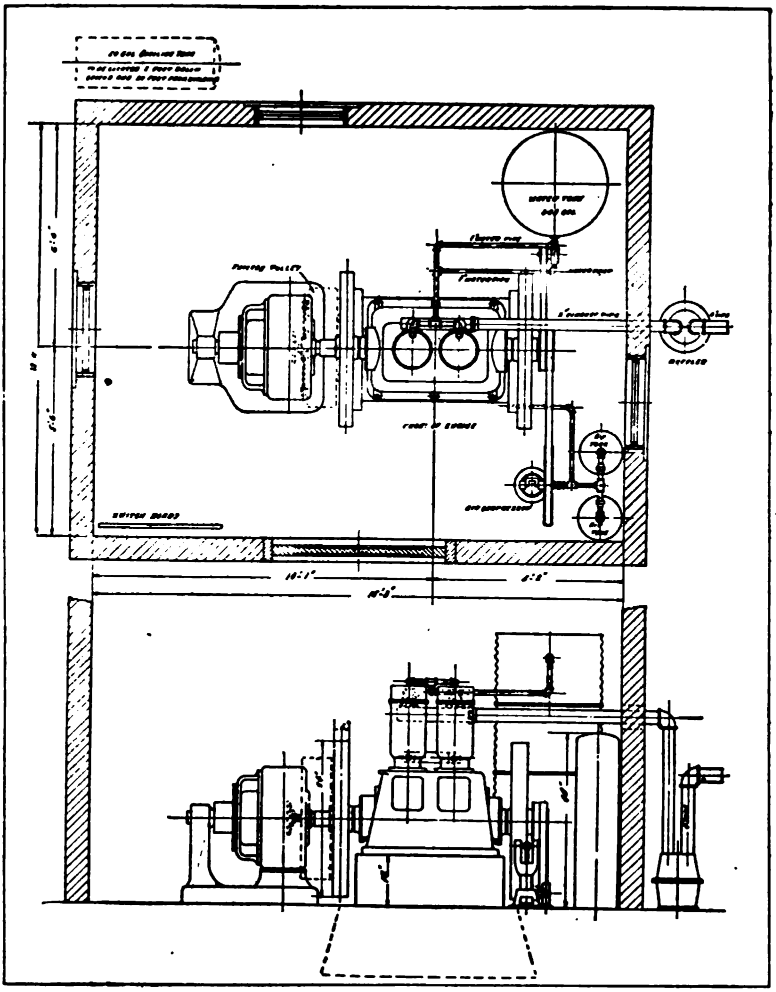


FIG. 353.—Floor plan and side elevation of direct-connected generator outfit using I. H. C. gasoline-engine as source of power.

the current measured by a recording wattmeter, giving a steady current to 90 16 candle-power lamps on a factory circuit, the total cost of gas at $1.50 per 1,000 cubic feet with lubricating oils was $20.16. The kilowatts produced by measure were 239.1 at a cost of .0844 cents per kilowatt. The price of the current by the same measure from the electric company was 20 cents per kilowatt — a saving of 57 per cent. In places where gas is $1 per 1,000 feet, the cost would have been only 5¾ cents per kilowatt.

The last few years have ushered in a most extended use of explosive engines as prime movers for generating the electric current for lighting and the transmission of power. For this purpose the duplex vertical engine and direct-connected multipolar generators are used, from which very favorable results have been obtained. (See Figs. 351 and 352.)

Trials with a 22-brake horse-power two-cylinder vertical engine of the National Meter Co., direct coupled with a 15-kilowatt 6-pole, compound-wound generator, using illuminating gas of 701 thermal units per cubic foot, with engine and generator running at 300 revolutions per minute, are quoted. "The output was 1,312 watts, or equal to 345 lamps of 3.8 watts each — say 16 candle-power, with a total brake horse-power = 22.71. Total consumption of gas per brake horse-power = 17.62 cubic feet. Relative illuminating power of electric light 2.21 as compared with equal consumption by direct gas-lighting. Efficiency of engine 20.6 per cent.; efficiency of generator 83.1 per cent."

Statements of still greater economy for lighting by gas and gasoline-engines, in which claims for from 14 to 16 cubic feet of gas and ⅛ gallon of gasoline per brake horse-power are made for large-sized electric plants, and but a trifle more for smaller sizes. Electric-lighting by the power of the explosive engine is conceded to be economical at all ranges of its power, but with gasoline and oil-vapor the cost of fuel for light drops to less than $\frac{1}{10}$ of a cent per 16 candle-power light per hour.

Electric-lighting plants operated by gas, gasoline, and oil-motors are making rapid advances in the number of units of power, and the small powers of the date of the early edition of this work have gradually advanced to unit instalments of 100, 500, and 750 horse-power in double and triple-cylinder motors, and by duplicating the motor-units, almost any desired installation can be made on the most economical running basis.

CHAPTER XIX

DESIGN AND APPLICATION OF AUTOMOBILE AND MOTOR-CYCLE POWER PLANTS

THERE has been no application of the internal combustion motor which has done more to popularize this form of power plant than their use in automobiles and motor-cycles. The automobile has been a great popular educator and has served to interest thousands of men in mechanics who would not have taken any interest in the mechanical arts previous to the development of the automobile. In order to obtain practical service from a motor-car the owner must familiarize himself with the basic principles of operation whether or not he is mechanically inclined. He must understand the common causes of internal combustion motor failure and must be able to remedy them unaided if he is to become a proficient operator and wishes to take trips away from the regions where machine shops and garages abound. This has resulted in many business and professional men becoming familiar with the use of common tools and ordinary mechanical processes.

Outside of the stationary applications there is perhaps no field of usefulness where the gasoline-engine is of more value than in that of transportation by motor-vehicle. One of the strong features of the gas-engine has always been that it can be designed and constructed in an almost infinite variety of forms. The principles obtaining were such that the engine could be made almost any size within the limits of commercial machine work and have an operative construction. As an example of the wide range of forms in which the internal combustion motor has been made one need look no further than the automobile and motor-cycle industry because in these we find motors ranging from the small, high speed air-cooled types rated at 1½ to 2 horse-power to powerful six cylinder forms of fifty times that rating. We do not find that wide variation in design of power plants in the field of motor-vehicle service as we do in either the stationary or marine application. For the most part motors used in automobiles and motor-cycles follow conventional, well established rules of practice that are not departed from in principle but merely in details and proportions of parts.

The requirements of all forms of motor-vehicle motors are practically the same. They must be light, compact, flexible, powerful and practically vibrationless. The simpler they are in construction the more reliable they are apt to be in practical operation. The automobile industry has been responsible for the development and refinement of the internal combustion motor more than any other field to which it has been applied. It was necessary to obtain the maximum efficiency before this form of motor could be used successfully in self-propelling vehicles. As automobiles and motor-cycles are intended to be operated by the masses it will be evident that the power plants used in either of these conveyances must be designed with a view to simplicity in order that they will receive the proper care from people, for the most part, mechanically inexperienced. The larger engines used in stationary service receive more or less skilled attention as they are almost constantly under the surveillance of competent engineers or mechanics. This is not true of the motor-vehicle engine, which must be made so that it can function for extended periods with minimum attention on the part of the operator.

This meant the development of automatic lubrication systems, and methods of ignition that would be simple yet reliable. The development of the auxiliary groups such as automatic carburetors and cooling methods, mechanical oiling systems and high efficiency ignition appliances made necessary to satisfy the demand of the automobile designer has had a favorable bearing on the entire field of gas-engine construction. As a result of this development many of the marine and stationary engines are provided with automatic lubrication and with positive and practically unfailing ignition systems. Even the designer of farm engines has been able to avail himself of the experience of the automobile designer and has refined and simplified his product from the knowledge gained by those producing gas-engines for automobile propulsion.

When the first automobiles were made, the gasoline-engine in its modern multiple cylinder form was practically unknown. The engines then available as stationary power plants were of the simple one and two-cylinder patterns and it was modifications of these that the early automobile designers incorporated in their crude vehicles. While automobiles with single cylinder engines were formerly very popular there are no cars on the market to-day that use less than two cylinders and these for the most part are of the light cycle-car form employing motor-cycle engines of the V type.

The single cylinder engines formerly used were made in two patterns and the cars built in Europe and America were radically

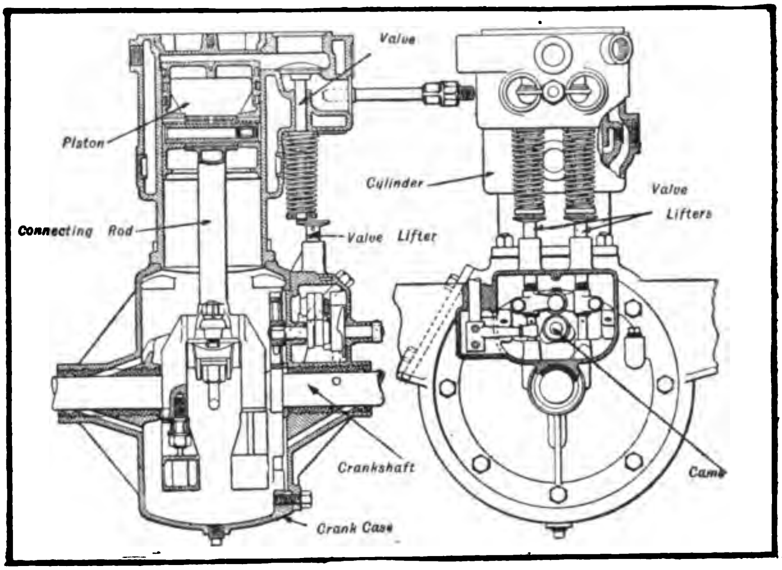


FIG. 354.—Sectional view of Brush runabout motor, a simple single cylinder power plant of the vertical type, designed to operate at high speeds.

different types. The European cars employed small, high speed vertical engines of the form shown at Fig. 354 and in these power was obtained by high rotative speed rather than large piston displacement. In America the trend of construction followed different lines and the early single cylinder cars of domestic construction incorporated large horizontal motors of the form shown at Fig. 355. These power plants did not run at high speed and power was obtained by large piston displacement rather than the rapid sequence of many light explosions. The contention of the European designer was that the small high speed engines did not

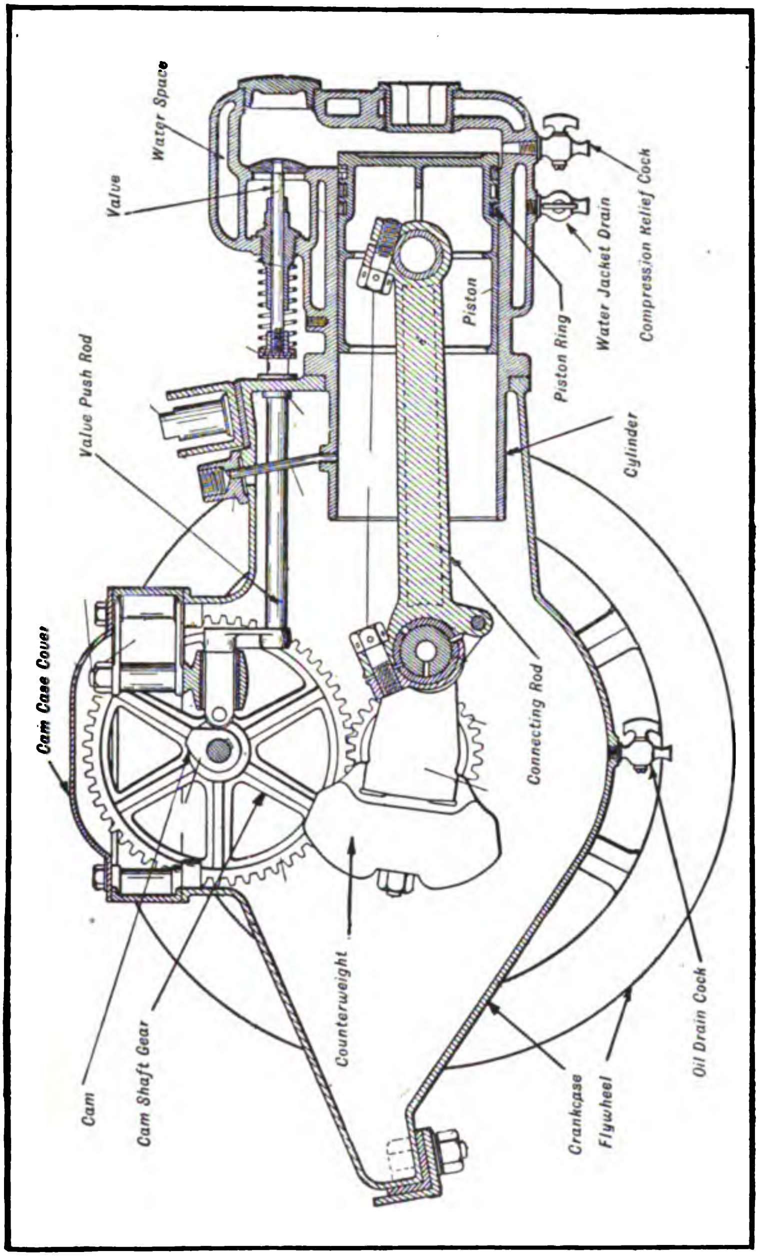


FIG. 355.—Sectional view one-cylinder horizontal engine used on some Reo models, a type that is rapidly being replaced by four-cylinder motors. These motors were operated at moderate speed, and had considerable vibration if speeded up or run slowly.

impose the stresses on the vehicle frame structure that the heavier low speed engines did. The belief of the American designers was that the use of large, low speed power plants permitted one to use heavy duty engines that would have great endurance. There is no question but that the slow speed American power plants were more enduring than the small high speed engines of foreign design. Even at this late date there are many hundreds of

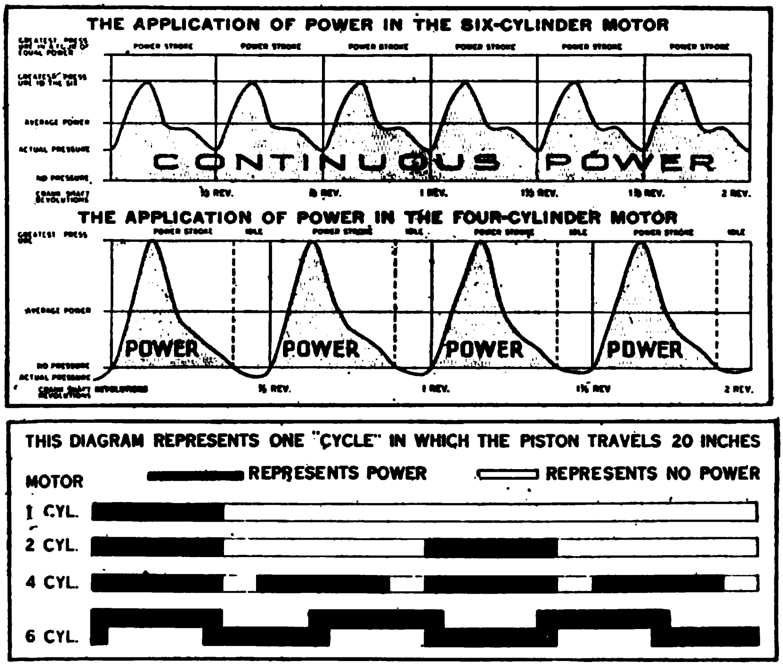


FIG. 356.—Diagrams outlining advantages of multiple cylinder motors, and why they deliver power more evenly than single cylinder types.

domestic single cylinder cars in use that were first put in service eight or ten years ago.

The disadvantages of the single cylinder motor were more marked in connection with motor-vehicle use than in either the stationary or marine applications. The resulting vibration due to the irregular torque had an evil influence on the endurance of the mechanism and also promoted the discomfort of the passengers, especially in the single cylinder engines used for heavy touring cars. It was not long before the advantages of the multiple cylinder construction were generally appreciated and the next improvement was to use two-cylinder opposed motors of the form shown at Fig. 357. After this the four-cylinder forms became the most popular and remain so at the present day for the majority of people though there is growing interest in the smooth running qualities of the six-cylinder power plant.

The reason for the popularity of the four-cylinder and six-cylinder motors can be well understood if reference is made to the diagrams at Fig. 356. At the lower portion we have a series

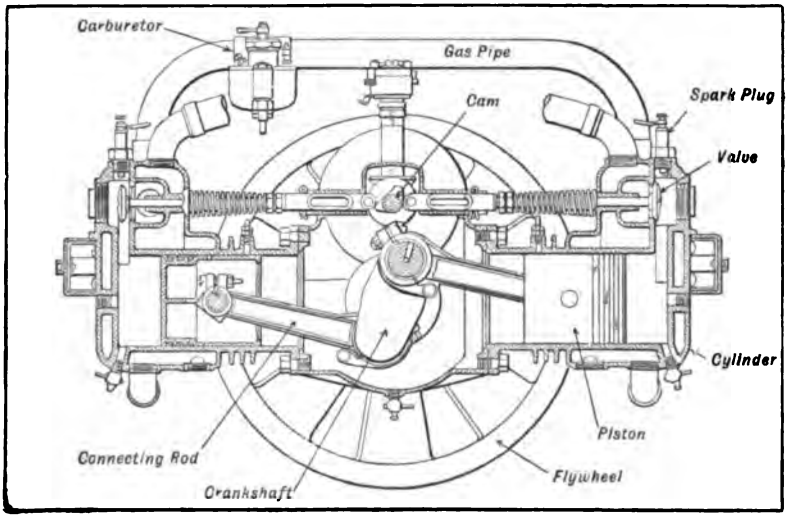


FIG. 357.—Simple form of two-cylinder motor having opposed cylinders; a very popular form of power plant for light service.

of diagrams representing in a graphical manner the useful power impulse in motors ranging from one to six cylinders. It will be observed that with the one-cylinder motor there is one power impulse followed by a long period of no power. In the two-cylinder motor we have two power impulses evenly spaced and the intervals during which no power is delivered from the crankshaft are correspondingly reduced. Even in the four-cylinder diagrams there are periods corresponding to the early opening of the exhaust valve where no useful effort is being exerted on the crankshaft. If we compare the diagrams shown at the top of

Fig. 356, the manner in which the continuous power is obtained from the six-cylinder power plant may be clearly understood. The even torque of a six-cylinder motor is due to an overlapping of the power strokes. When the motor has two or more cylinders it will be apparent that the power strokes may be evenly spaced or separated through the cycle and as the number of explosions increases the power production becomes more continuous and vibration is eliminated.

In the four-cylinder motor, such as shown at Fig. 358, the cranks of the crankshaft are set 180 degrees apart, i. e. two of them are directly opposite the other two, as are the cranks of a bicycle. In a four-cylinder engine each piston is always one complete stroke behind its neighbor. In a six-cylinder, as outlined at Fig. 359, the cranks are set at 120 degrees and not opposite each other and thus in a six the piston in any cylinder is but two-thirds of a stroke behind its predecessor. In a six, therefore, the power of one explosion is not spent before the succeeding power stroke takes hold as on a four-cylinder crankshaft, but it overlaps the preceding power stroke as shown in diagrams at the top and bottom of Fig. 356.

Inasmuch as a motor having less than six cylinders cannot produce continuous power unless of the two-cycle type, it follows that in motors having distinct intervals of no power, a portion of the energy developed by the explosion is being used to drive the motor parts through those portions of the cycle when there is no useful turning effort on the engine crankshaft. Another advantage of a six-cylinder motor is that the diminution in force made possible by the greater number of explosions per unit time not only reduces the vibration but also decreases the wear and tear on the motor parts and prevents untimely depreciation.

To produce equal total horse-power, the cylinders of a six need be but two-thirds as large as the cylinders of a four. This means that the pistons and other reciprocating parts can be lighter and that a better balance obtains than with a four of the same power. It is said that in a single cylinder motor developing 48 horse-power, the piston receives on each power stroke a force equivalent to a hammer blow of 28,800 pounds. In the four-cylinder motor developing the same total horse-power each piston receives a blow of 7,200 pounds. In a six-cylinder engine the pistons will receive a

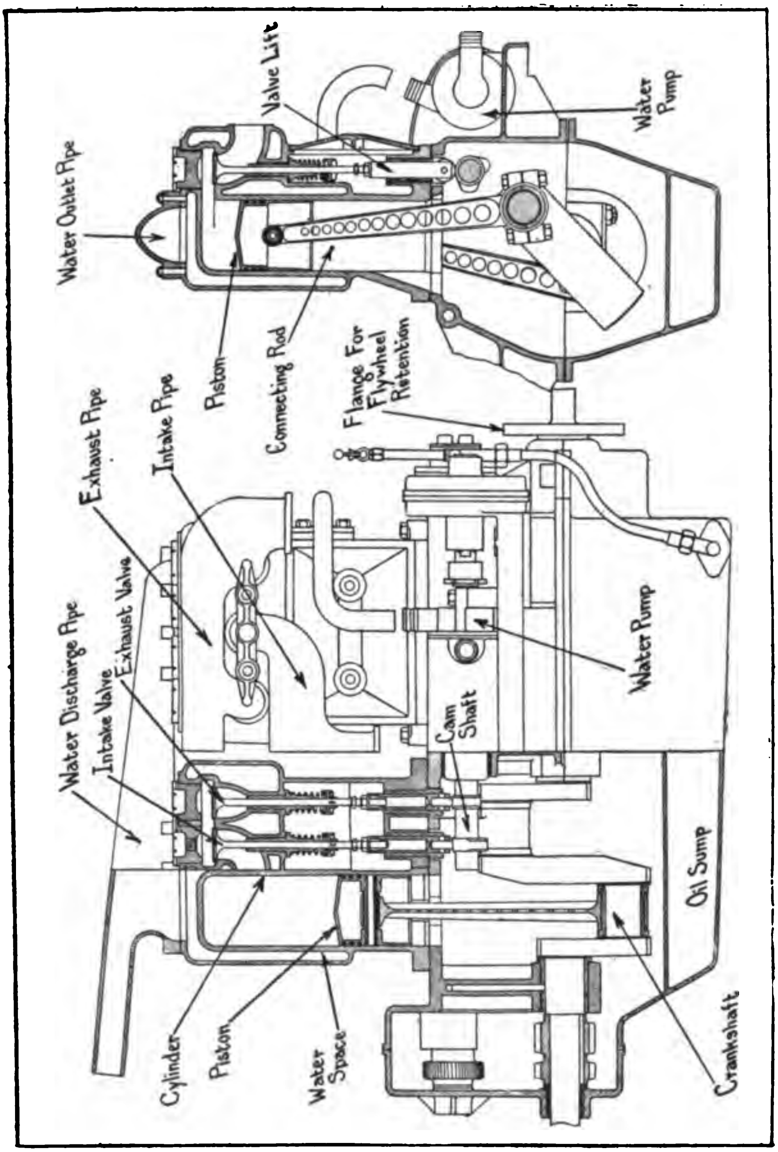


FIG. 358.—Typical four-cylinder automobile power plant of European design.

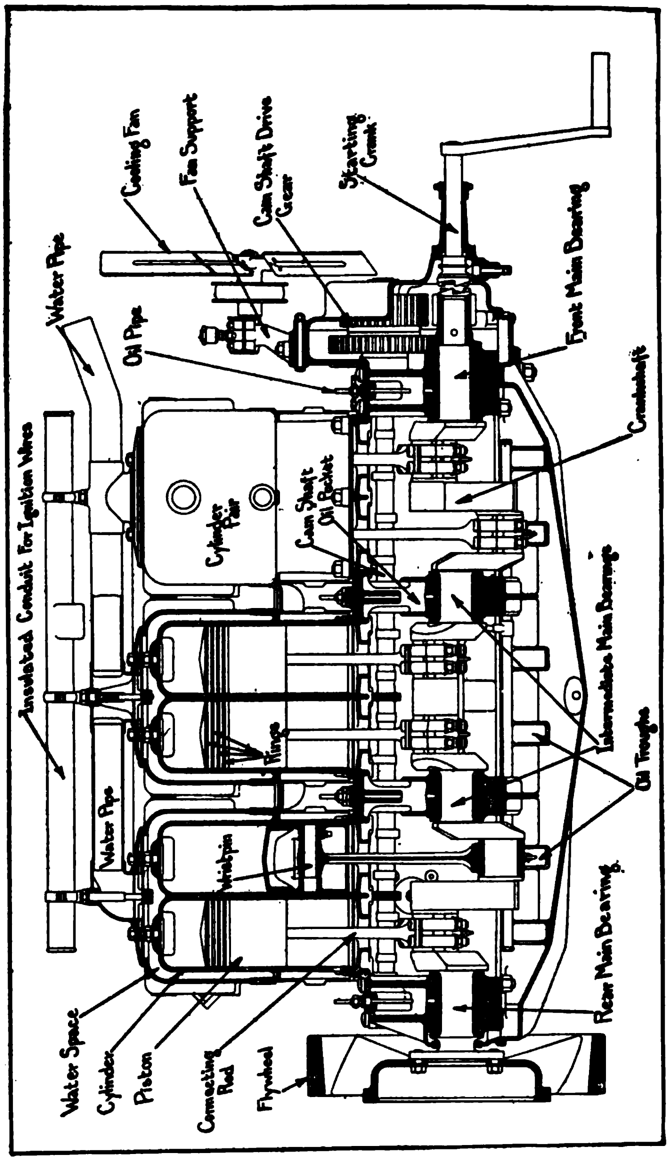


Fig. 359.—Part sectional view, showing arrangement of parts in six-cylinder automobile power plant.

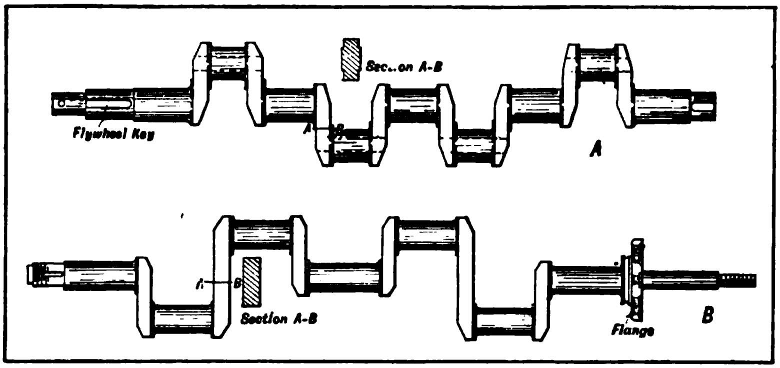


FIG. 360.—Two forms of four-cylinder crankshaft.

A. Five-bearing type with flywheel fastening key at front end. B. Three-bearing type with flange for securing flywheel formed integral.

blow equivalent to but 4,800 lbs. This is brought up as an argument favoring the six-cylinder construction, as it means not only lessened wear on the parts of the motor, such as the connecting rod bearings, crankshaft bearings and on the gears, driving shaft axles and other members of the transmission system but it also means the diminished wear on tires at the traction members on account of the smoother and more uniform power application. The four-cylinder motor, however, has many features of merit and can be so well balanced that its vibration is practically negligible at all excepting extremely high speeds. The six-cylinder engine will undoubtedly be used more and more on cars refined to the highest degree while the four-cylinder engine will remain the ideal power plant on all cars intended for practical every day service.

AUTO MOTOR CRANKSHAFT TYPES

The importance of the crankshaft has been previously considered, and some of its forms have been shown in views of the motors presented in earlier portions of this work. The crankshaft is one of the parts subjected to the greatest strain and extreme care is needed in its construction and design, because practically the entire duty of transmitting the power generated by the motor to the gear-set devolves upon it. Automobile motor crankshafts are usually made of high tensile strength steel of special composition. They may be made in four ways, the most common being a drop or machine forging which is formed approximately to the shape of the finished shaft and in rare instances they may be steel castings. Sometimes they are made from machine forgings, where considerably more machine work is necessary than would be the case where the shaft is formed between dies. Some engineers favor blocking the shaft out of a solid slab of metal and then machining this rough blank to form. In some single-cylinder motors of the enclosed fly-wheel type the crankshaft and fly-wheel are built up as a unit, as at Fig. 362.

The form of the shaft depends on the number of cylinders and the form has material influence on the method of construction.

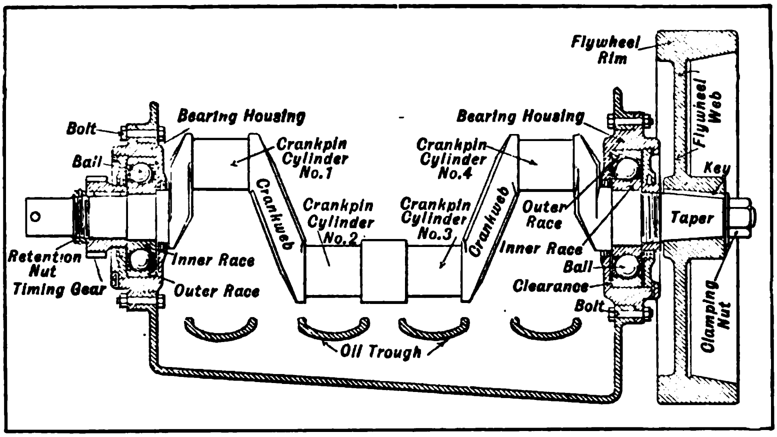


Fig. 361.—Design of four-cylinder crankshaft mounted on two annular ball-bearings.

Note method of flywheel retention by key and taper and bearing housing.

For instance, a one- two- or four-cylinder crankshaft could be made by either of the methods outlined. At the other hand a three- or six-cylinder shaft is best made by the machine forging process.

Crankshafts for four-cylinder engines may have five main bearings as at A, Fig. 360, if the engine employs individually cast cylinders or they may have but three main bearings as shown at B, Fig. 360. When the cylinders of a four-cylinder engine are cast in pairs the three main bearing type of crankshaft is the form generally used. When the four cylinders are cast as a unit or "en bloc" it is possible to employ a crankshaft with but two main bearings.

When this construction is followed the designers may use anti-friction bearings of the annular ball type as outlined at Fig. 361. These bearings have less friction than the plain bushings gener-

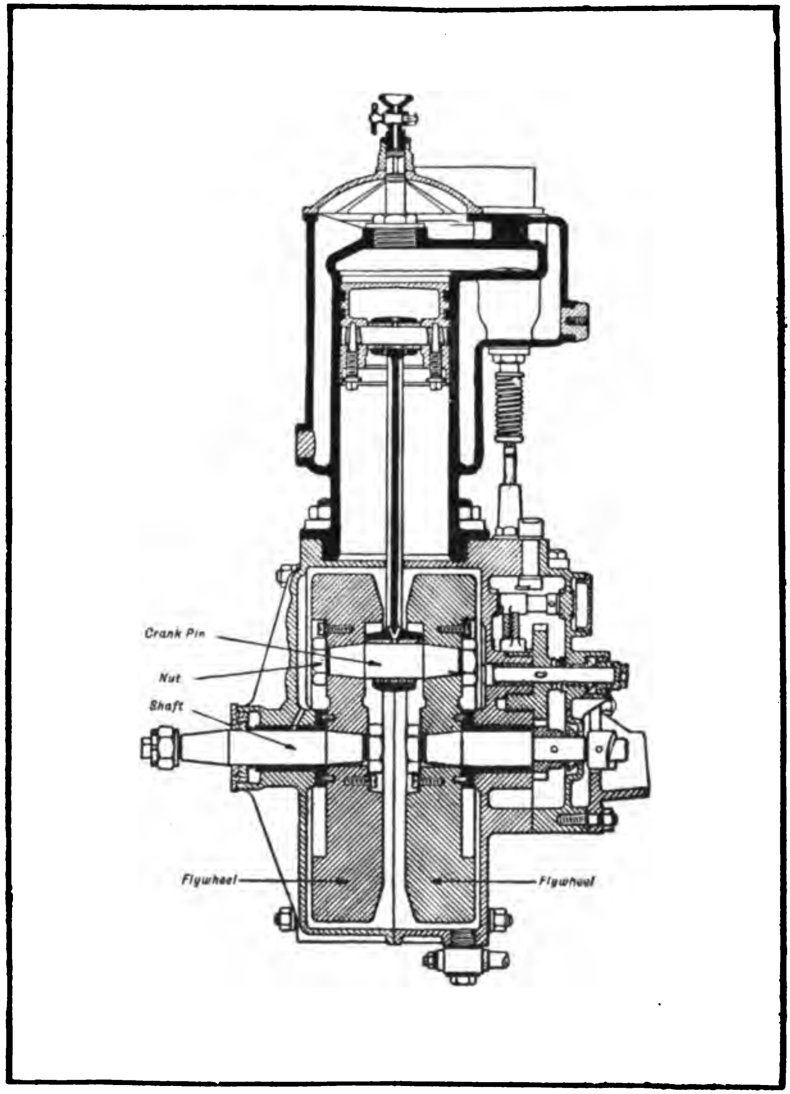


FIG. 362.—Defining built-up crankshaft construction sometimes used in small motors.

ally used and are a refinement in detail which is found usually only on power plants intended for automobiles, motor-cycles or aeroplanes.

SOME NOTES ON AUTO MOTOR DESIGN

Another point of importance in the design of the cylinder and one which has considerable influence upon the power developed, is the shape of the combustion chamber. The endeavor of designers is to obtain maximum power from a cylinder of certain proportions, and the greater energy obtained without increasing piston displacement or fuel consumption the higher the efficiency of the motor. To prevent troubles due to preignition it is necessary that the combustion chamber be made so that there will be no roughness, sharp corners, or edges of metal which may remain incandescent when heated or which will serve to collect carbon deposits by

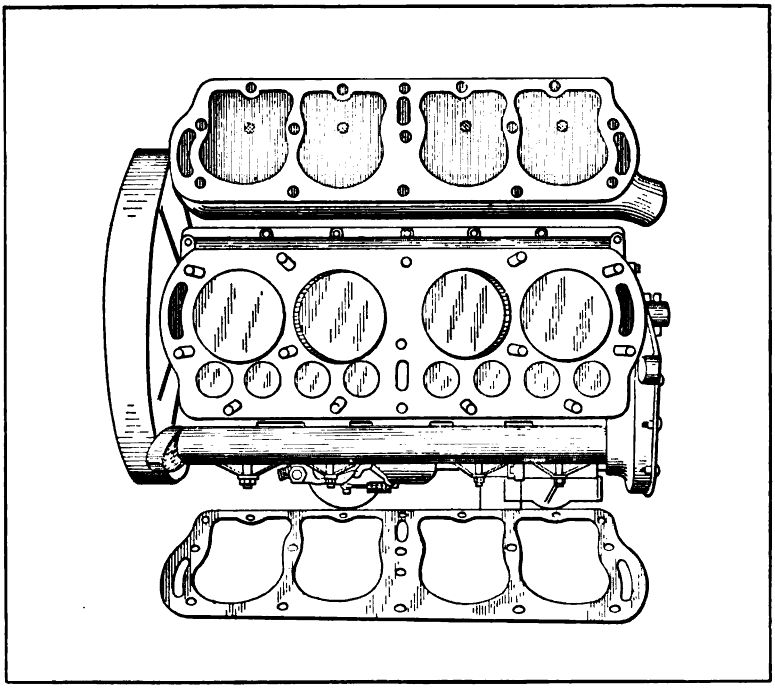


Fig. 363.—Example of four-cylinder block motor having one separately cast head member common to all cylinders. A copper-asbestos gasket is utilized in making a gas and water-tight joint between the parts. Note accessibility of pistons and valves.

providing points of anchorage. With the object of providing an absolutely clean combustion chamber some makers use a separable head unit such as shown at Fig. 363. These permit one to machine the entire interior of the cylinder and combustion chamber and also make ready access to the cylinder interior possible.

A question that has been a vexed one and which has been the subject of considerable controversy is the proper proportion of the bore to the stroke. The early gas-engines had a certain well-defined bore to stroke ratio, as it was usual at that time to make the stroke twice as long as the bore was wide, but this is open to question when high speed is desired. With the development of the present-day motor the stroke or piston travel had been gradually shortened so the relative proportion of bore and stroke had become nearly equal. Of late there seems to be a tendency among designers to return to the proportions which formerly obtained and the stroke is sometimes one and a half or one and three-quarter times the bore.

The disadvantage of short-stroke engines is that they will not pull well at low speeds, though they run with great regularity and smoothness at high velocity. The long-stroke engine is much superior for slow speed work, and it will pull steadily and with increasing power at low speed. It was formerly thought that such engines should never turn more than a moderate number of revolutions in order not to exceed the safe piston speed.

The factor which limits the stroke and makes the speed of rotation so dependent upon the travel of the piston is piston speed. Heretofore, it has been considered desirable not to exceed a speed of one thousand feet per minute, which has been determined to make for greatest efficiency, combined with endurance, by many authorities on design and construction of internal combustion motors. During the past few years there have been instances where engines were giving satisfactory service with piston speed of 1,200 to 1,500 feet per minute. Lubrication is the main factor which determines piston speed, and the higher the rate of piston travel the greater care must be taken to insure proper oiling.

The normal piston speed of automobile motors has increased from year to year, almost since the beginning of the automobile industry, and the rate of increase has been particularly pronounced during the past few years. C. Faroux, in *L'Auto,* devotes an article to this subject. He says that from 1901 to 1914 he has each year determined the average speed of (1) racing motors, (2) touring motors and (3) agricultural and truck motors. In every case there has been a clearly discernible speed increase, and the continuity of this phenomenon in itself bears evidence of its great significance. We reproduce at Fig. 364 a diagram prepared by M. Faroux, which shows the piston speeds attained by motors of the different classes during each year within the period mentioned. The author asserts that the increase in piston speed is the best criterion of the progress made in automobile construction. It so happens that the specific output increases in proportion to the piston speed, and the motor efficiency has become greater in each of the great races since 1901, until in 1913 the race went to the motor which had the highest piston speed.

In a report on gasoline-motors made to the International Automobile Congress at Milan in 1906, it was stated that it was not likely that a speed of 900 to 1,000 feet per minute would ever be exceeded in touring motors and 1,200 to 1,400 feet per minute in racing motors. How far from correct this prognostication was may be judged from the fact that at present manufacturers furnish their customers motors in which a piston speed of 1,800 feet per minute is attained, while in racing motors 2,000 feet per minute is easily exceeded. About the same time some manufacturers ventured the opinion that the piston speed was constant and that only the speed of revolution varied; if a racing motor was doubled in size it would run at half the speed, etc.

In the diagram herewith the upper line shows the piston speed of the winners of the Coupe de l'Auto race each year, and it will be seen that the piston speed increased from 2,070 feet in 1906 to 3,150 feet in 1913. A particularly strong forward move was made in 1910 by the Hispano-Suiza, with the result that there was no further progress the next year and very little the following year. In the A. C. F. Grand Prix races, owing to the easy conditions, the progress achieved in respect to piston speed was slower. However, after an interruption of three years in the running of this race a very important increase in piston speed was shown in 1912. The line representing the piston speed of touring motors is based on data from motors of well-known make whose manu-

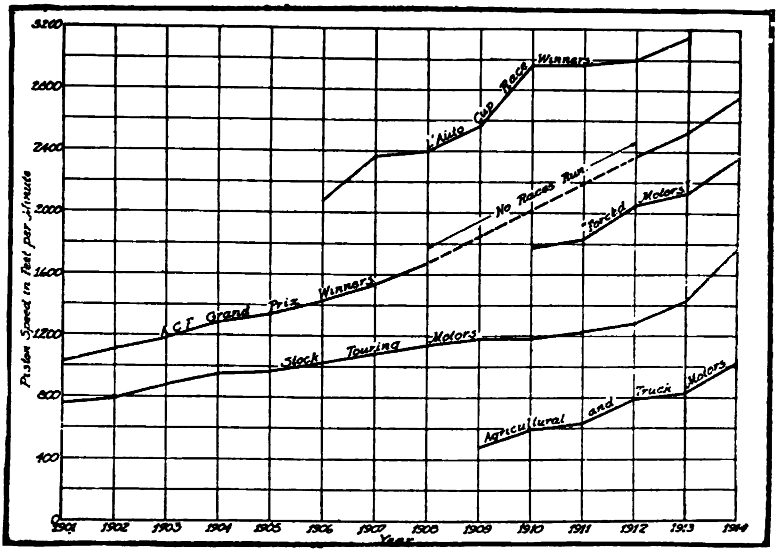


FIG. 364.—Diagram showing increase in piston speed of French racing, touring and truck motors since 1901.

facturers are of a conservative turn of mind. M. Faroux combats the opinion that a high piston speed motor is necessarily a short-lived motor, and says that he can give no better proof of the durability of such a motor than his own personal experience. The past three years he has driven a car equipped with a 3.2x7.2-inch motor a distance of 49,000 miles, and it has never required an overhauling. The car is still capable of a speed of 75 miles per hour on the level, and ascends the Gaillon hill at 59 miles per hour (official time). As shown by the diagram, agricultural and truck motors are operated at very much lower speed than automobile motors; still, the tendency in these motors also is constantly toward higher speeds.

In explaining the principle of operation of the internal combustion engine it was made clear that there were four strokes of the piston necessary to complete the cycle of operation in any one cylinder, and of these but one was a useful or power stroke. The gasoline-engine would not be a practical power producer, especially if made in one- and two-cylinder patterns without some means of equalizing the uneven power generation. Considering first the single-cylinder motor, we find that we have but one explosion every four strokes, and as this represents two revolutions of the crankshaft it will be evident that it is necessary to store up energy by some means in order to carry the crankshaft through the idle strokes. This is accomplished by supplying a heavy wheel which is secured in a positive manner to the crankshaft and which turns with it. When the explosion drives the piston down considerable energy is stored in the fly-wheel rim and it will continue to revolve after the impulse given it has diminished in value to a considerable extent. In fact there is enough energy stored in the fly-wheel of proper weight to carry the piston through all the idle strokes and to equalize the torque produced. This insures an even turning movement and makes for uniform application of power to the mechanism.

The fly-wheel weight is dictated largely by the number of cylinders employed, it being a general rule that the motors having the least number of cylinders require the heaviest fly-wheels. This means that a single-cylinder motor will need a heavier equalizing member than one having a greater number of cylinders and a more even turning moment at the crankshaft. As an example of how the number of cylinders directly affects fly-wheel weight, one may say that if a single-cylinder engine of given power required a fly-wheel of two hundred pounds weight to equalize the power effect, a double-cylinder engine would need one of about one hundred and sixty pounds, a four-cylinder engine would use one weighing but one hundred pounds, while a six-cylinder motor would furnish a uniform torque with a fly-wheel member weighing no more than sixty pounds. Fly-wheel weight is determined by many conditions, some of the important ones being bore of the cylinder, speed of crankshaft rotation, degree of compression, and mode of transmission. It is common practice to provide a fly-wheel somewhat heavier than the actual requirements on multi-cylinder motors of large bore so that these may be more easily started by a person of average strength.

VALVE LOCATION PRACTICE

It has often been said that a chain is no stronger than its weakest link and this is as true of the explosive motor as it is of any other piece of mechanism. Many motors which appeared to be excellently designed and which were well constructed did not prove

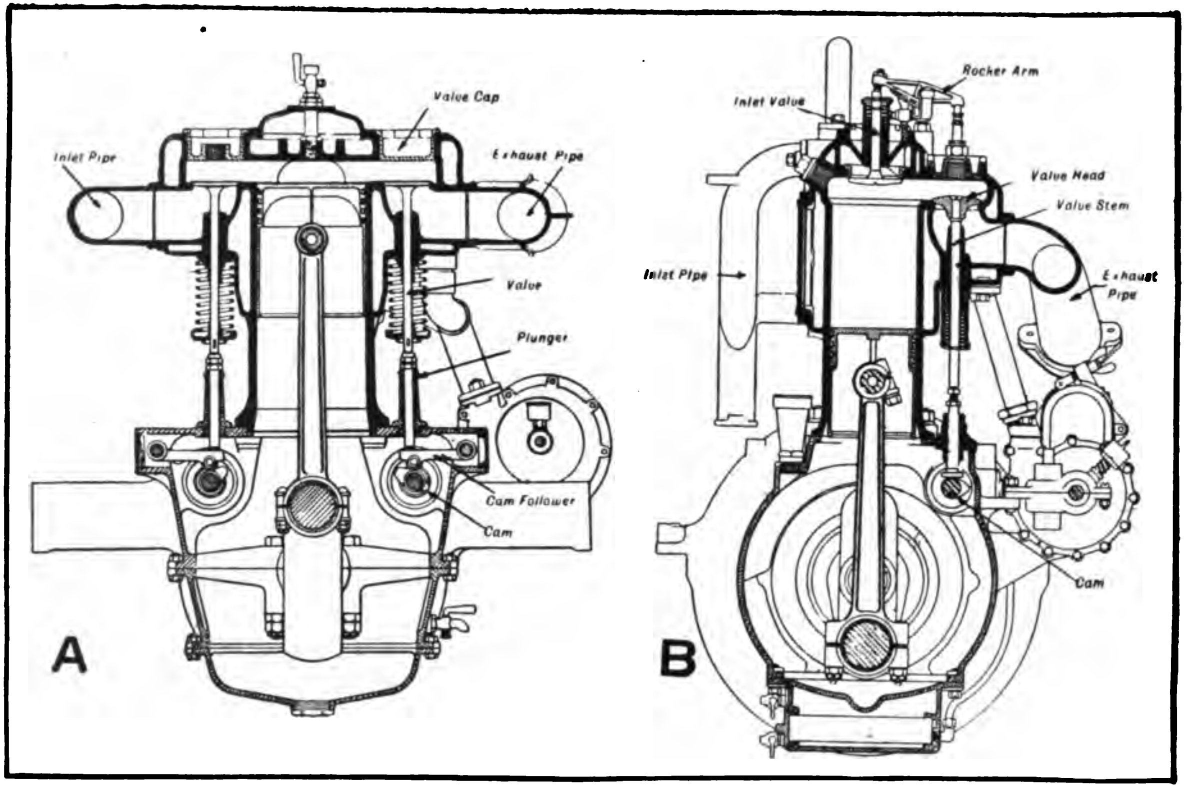


Fig. 365.—Illustrating typical methods of valve installation in internal combustion motors.
A. Valves on opposite sides of T-head cylinder. B. L-head cylinder having intake valve placed directly in the center of the cylinder head.

satisfactory, because some minor detail or part had not been properly considered by the designer. A factor having material bearing upon the efficiency of the internal combustion motor is the location of the valves and the shape of the combustion chamber which is largely influenced by their placing. The fundamental consideration of valve design is that the gases be admitted and discharged from the cylinder as quickly as possible in order that the speed of gas flow will not be impeded and produce back-pressure. This is imperative in obtaining a satisfactory operation in any form of motor. If the inlet passages are constricted the cylinder will not fill with explosive mixture promptly, whereas if the exhaust gases are not fully expelled the part of the inert products of combustion retained dilute the fresh charge, making it slow burning and causing lost power and over-heating. When an engine employs water as a cooling medium this substance will absorb the surplus heat readily, and the effects of over-heating are not noticed as quickly as when air-cooled cylinders are employed. Valve sizes have a decided bearing upon the speed of motors and some valve locations permit the use of larger members than do other positions.

While piston velocity is an important factor in determinations of power output it must be considered from the aspect of the wear produced upon the various parts of the motor. It is evident that engines which run very fast, especially of high power, must be under a greater strain than those operating at lower speeds. The valve-operating mechanism is especially susceptible to the influence of rapid movement, and the slower the engine the longer the parts will wear and the more reliable the valve action.

As will be seen by reference to the accompanying illustrations, there are many ways in which valves may be placed in the cylinder. Each method outlined possesses some point of advantage because all of the types illustrated are used by reputable automobile manufacturers. The method outlined at Fig. 365, A, is widely used and because of its shape the cylinder is known as the "T" form. It is approved for several reasons, the most important being that large valves can be employed and a well-balanced and symmetrical cylinder casting obtained. Two independent cam shafts are needed, one operating the inlet valves, the other the exhaust members. The valve-operating mechanism can be very simple in

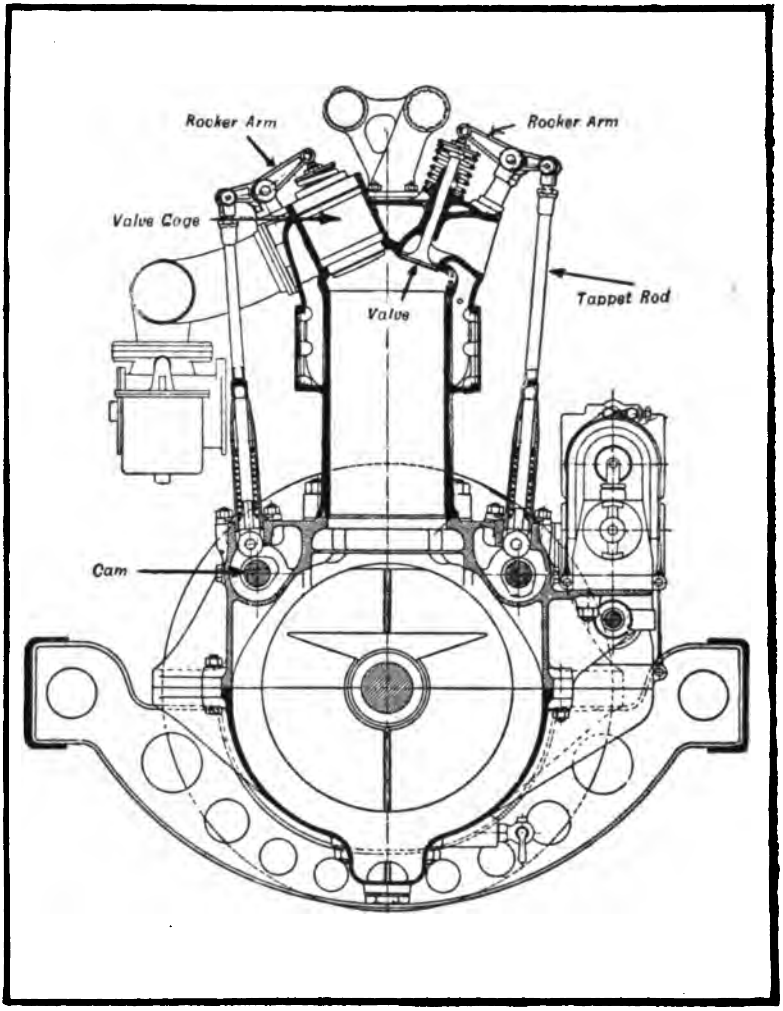


Fig. 366.—Benz racing motor, presented to show method of valve placing so these members open directly into the cylinder head.

form, consisting of a plunger actuated by the cam which transmits the cam motion to the valve stem, raising the valve as the cam follower rides on the point of the cam. Piping may be placed without crowding, and larger manifolds can be fitted than in some other constructions. This has special value, as it permits the use of an adequate discharge pipe on the exhaust side with its obvious advantages.

At the other hand, if considered from a viewpoint of actual heat efficiency, it is theoretically the worst form of combustion chamber. This disadvantage is probably compensated for by uniformity of expansion of the cylinder because of balanced design. The ignition spark plug may be located directly over the inlet valve in the path of the incoming fresh gases, and both valves may be easily removed and inspected by unscrewing the valve caps without taking off the manifolds. At Fig. 365B a section through a typical "L"-shaped cylinder is depicted. It will be evident that where a pocket construction is employed in addition to its faculty for absorbing heat, the passage of gas would be impeded. For example, the inlet gas rushing in through the open valve would impinge sharply upon the valve cap directly over the valve and then must turn at a sharp angle to enter the combustion chamber and then at another sharp angle to fill the cylinders. The same conditions apply to the exhaust gases, though they are reversed. When the valve-in-the-head type of cylinder is employed as at Fig. 366 the only resistance offered the gas is in the manifold. As far as the passage of the gases in and out of the cylinder is concerned ideal conditions obtain. It is claimed that valve-in-the-head motors are more flexible and responsive than other forms but the construction has the disadvantage in that the valves must be opened through a rather complicated system of push rods and rocker arms instead of the simpler and direct plunger which can be used with either the "T" or "L" head cylinders.

VALVE OPERATING MEANS

As previously stated, the method of actuating the valves depends largely on the way they are installed. For example, in the engine shown at Fig. 367 the exhaust valve is carried in the side pocket while the inlet valve is mounted in the head. This permits of the valves being operated from a common cam-shaft but calls for two entirely different methods of valve actuation, as shown at Fig. 368. The inlet valve, which is mounted in the center of the cylinder head, is carried in a readily removable cage so that it may be taken from the cylinder as a unit. As the valve must be pushed in toward the piston to permit the fresh gas to enter the cylinder and as the cam on the camshaft is usually formed to give an upward movement it is necessary to interpose a rocker lever of the first class between the valve operating-cam and the valve-stem. This construction is clearly shown at A and as the cam roller on the valve plunger guide rises on the point of the cam it will push the tappet rod up and the other end of the rocker lever which bears on the inlet valve stem will be pushed down to depress the valve.

The simplicity of the mechanism employed in valve lifting when that member is carried in a side pocket is clearly shown at B.

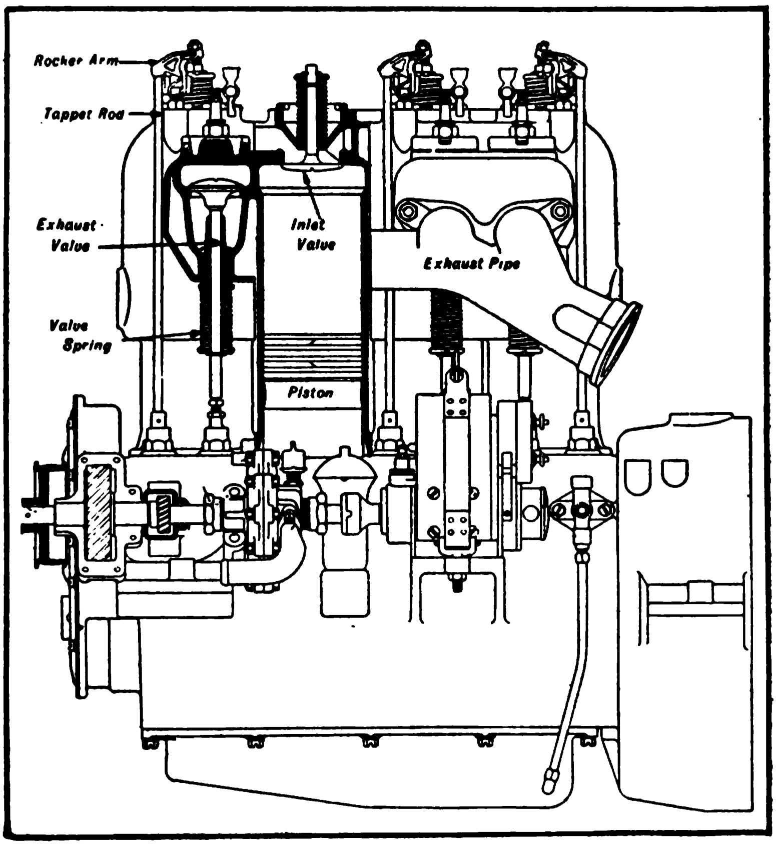


FIG. 367.—Part sectional view of Bergdoll motor, showing placing of valves. The exhaust member is fitted in a side pocket of the L cylinder. The inlet valve is placed directly in the center of the combustion chamber.

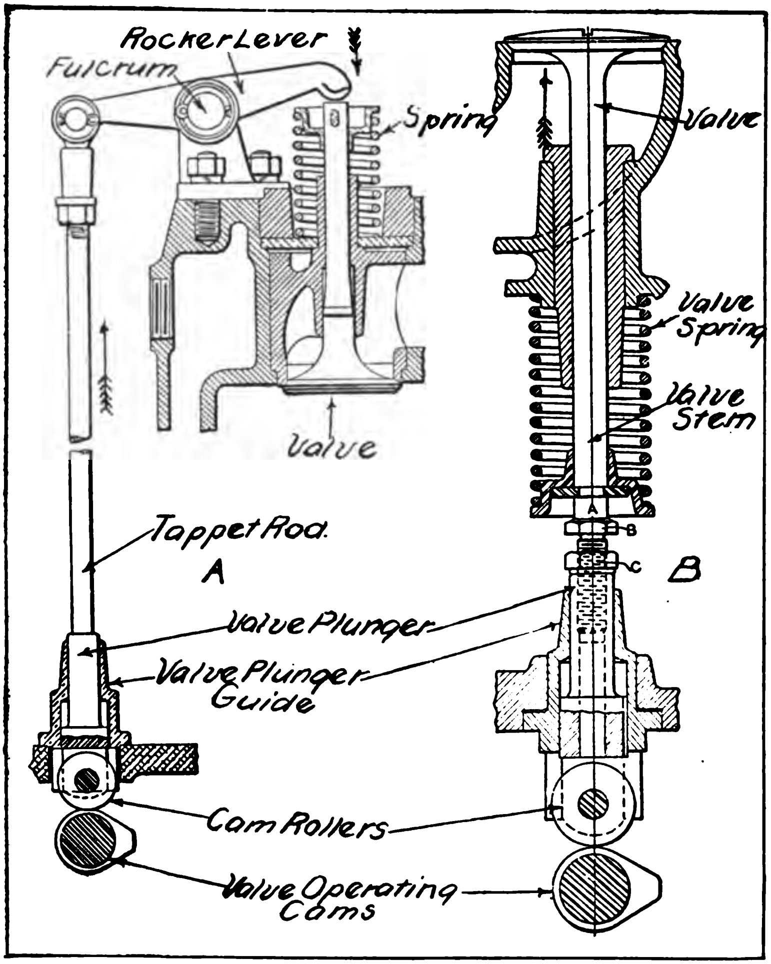


Fig. 368.—Conventional methods of operating internal combustion motor valves.

A. Valve actuated by rocker lever and tappet rod. B. Valve operated by simple plunger resting on the cam.

The end of the valve plunger can bear directly against the end of the valve stem A. In an engine of the T head form such as outlined at Figs. 369 and 370 it is customary to operate the valve by simple push rods working on regular profile cams as in the former illustration which represents the cross section through the Packard poppet

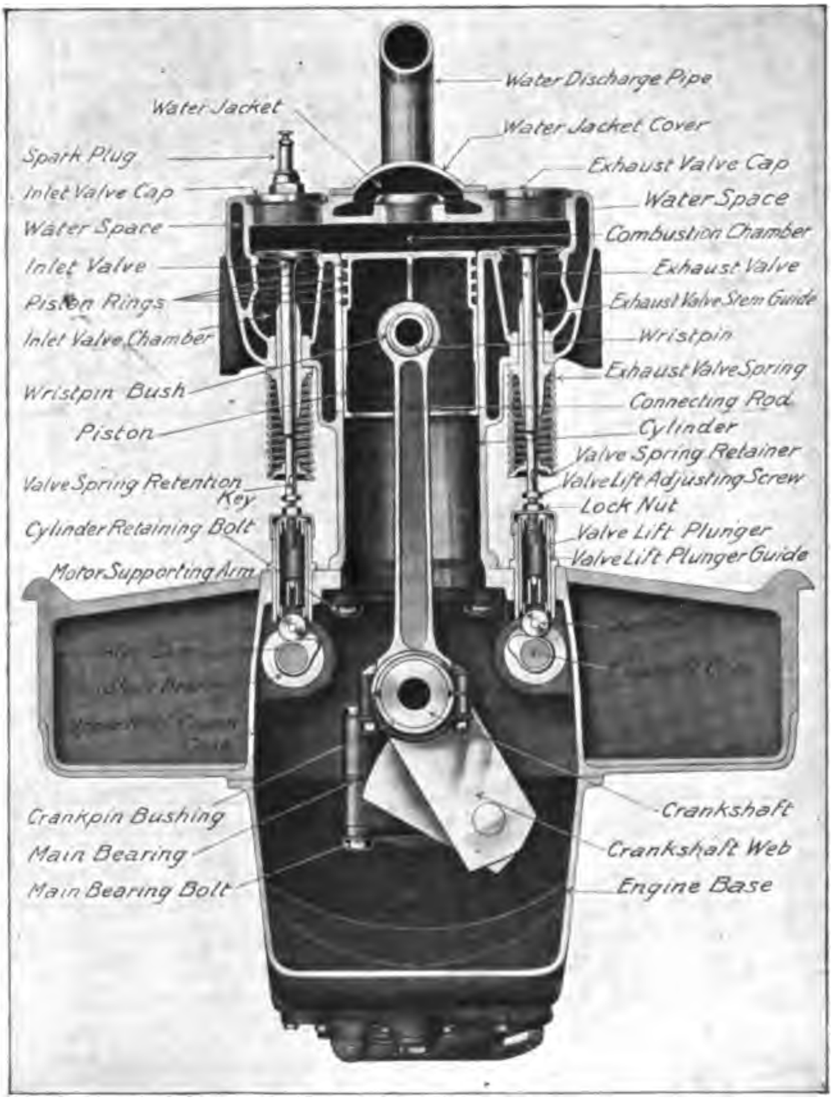


Fig. 369.—End sectional view of typical internal combustion motor of the T head type, showing valve operating mechanism.

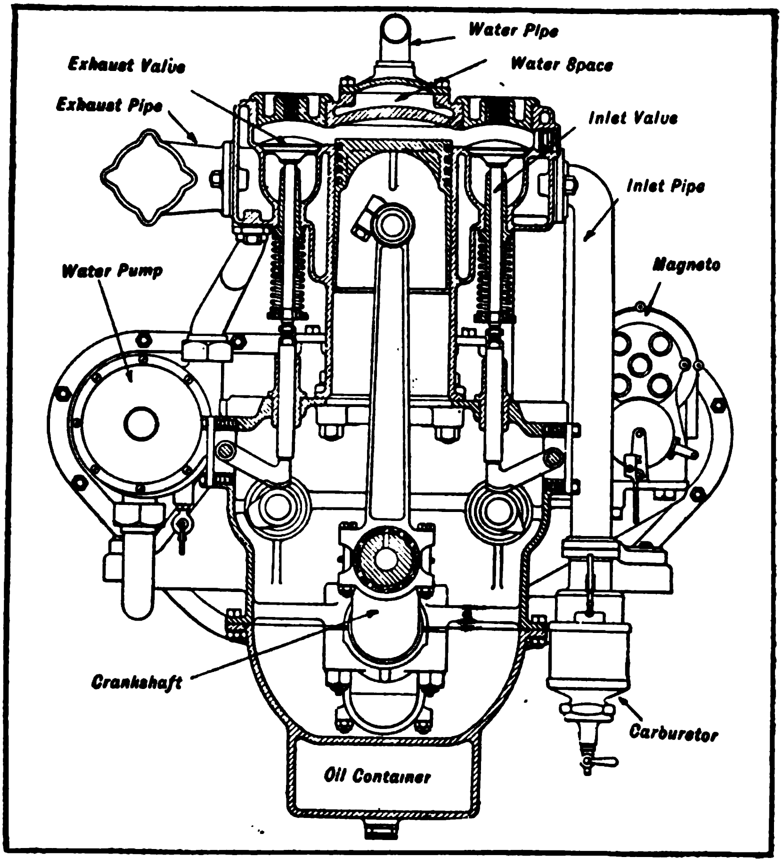


FIG. 370.—Sectional view of rear cylinder of gasoline-engine with important parts indicated.

valve motor. As there is apt to be considerable side thrust on the valve-lift plungers when the cam roller rides on the side of the cam point, some designers make provision to relieve the valve plunger of this thrust by interposing small cam riders between the valve plunger and the cams as shown at Fig. 370. These members take practically all of the side thrust due to the cam operation and the plunger or valve-operating push rod receives only a direct up and down thrust which is not so apt to wear out the valve plunger guide.

Extreme care is taken at the present time by designers of

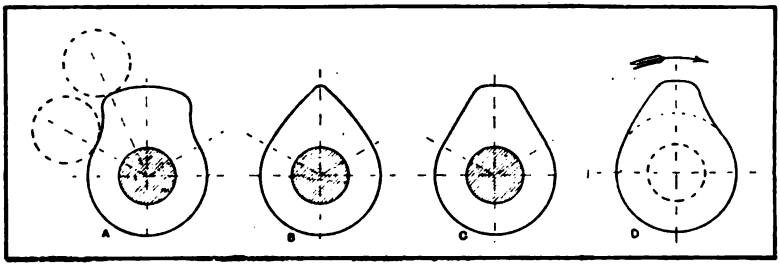


FIG. 371.—Forms of valve-lifting cams generally employed.

A. Cam profile for long dwell and quick lift. B. Typical inlet cam used with mushroom type follower. C. Average form of cam. D. Designed to give quick lift and gradual closing.

poppet valve motors to secure smooth-acting valve-mechanisms that are not apt to be noisy. A number of valve-lifting cams such as generally employed on automobile motors are shown at Fig. 371. That at A shows a profile of a cam that is apt to be noisy in action and which is more suitable for slow-speed motors than high-speed forms. This is designed to give a quick lift and to keep the valve open for a long period of time. This form of cam is generally used for operating exhaust valves. The cam shown at B is a typical inlet-cam well adapted for use with a

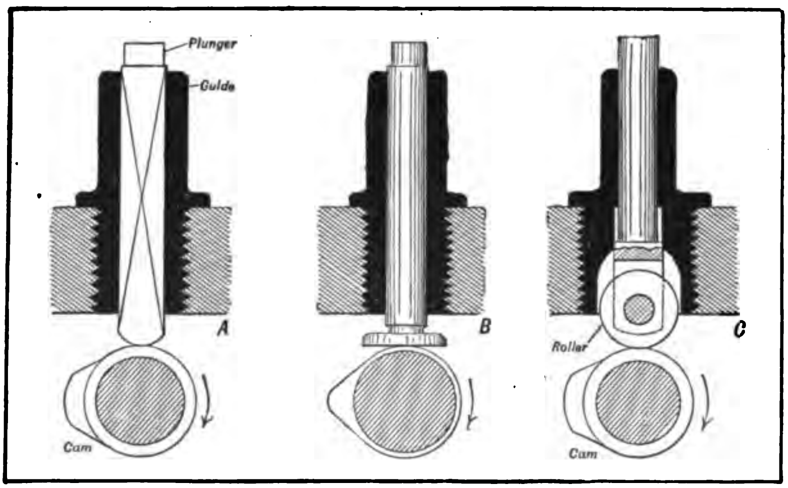


FIG. 372.—Showing principal types of cam followers which have received general application.

mushroom type of cam follower such as outlined at B, Fig. 372. The average form of valve-operating cam is depicted at C. Sometimes a form designed to give a quick lift, and gradual closing such as shown at D may be used, but for the most part the cams that have received the widest application are of the forms shown at B and C.

Another important consideration in the design of an efficient valve-operating mechanism is the form of cam followers used at the lower end of the valve plungers. The design of this member

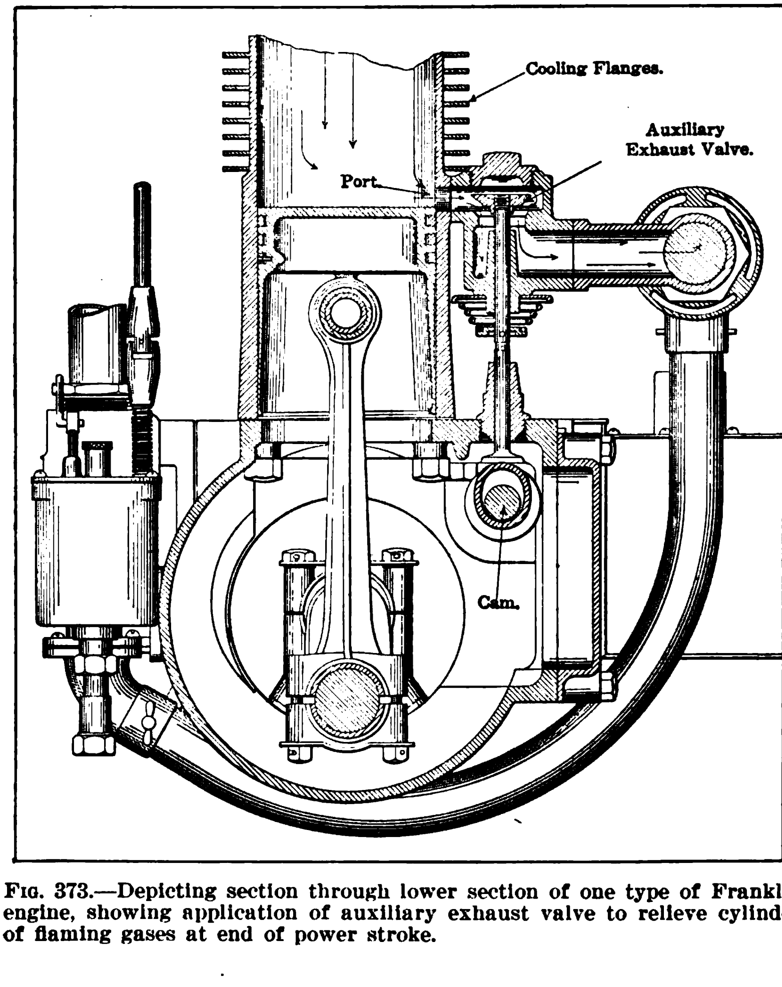


Fig. 373.—Depicting section through lower section of one type of Franklin engine, showing application of auxiliary exhaust valve to relieve cylinder of flaming gases at end of power stroke.

depends to a certain extent upon the form of cam used. A cam with a gradual profile shown at Fig. 372A or Figs. 371B and C will work satisfactorily with a simple rounded end push rod as indicated. The mushroom type cam follower depicted at Fig. 372B also requires valve-lifting cams having gradual profiles. When cams designed for quick lift and having abrupt profiles are used it is necessary to use a roller as a cam rider at the lower end of the push rod as shown at Fig. 372C.

In the chapters on stationary engines the advantages of the auxiliary exhaust port, which is uncovered by the piston when that member reaches the end of its power stroke, have been described at some length. This construction has also been used to some extent on air-cooled motors. In the form shown at Fig. 373 the auxiliary exhaust port is controlled by a poppet valve opened by a suitable cam on the same camshaft operating the main inlet and exhaust valves in the cylinder head. The prompt expulsion of the large quantity of flaming gases made possible by this simple fitting reduces the temperature of the gases discharged through the regular exhaust valve and makes for superior cooling of the cylinder and lessened scaling or warping of the regular exhaust valve head, which must seat properly to retain compression. Even if the lower or auxiliary valve does not seat positively, it is not apt to interfere with engine efficiency.

VALVE TIMING IN AUTO MOTORS

It is in valve timing that the greatest difference of opinion prevails among engineers and it is rare that one will see the same formula in different motors. It is true that the same timing could not be used with motors of different construction, as there are many factors which determine the amount of lead to be given to the valves. The most important of these is the relative size of the valve to the cylinder bore, the speed of rotation it is desired to obtain, the fuel efficiency, the location of the valves, and other factors too numerous to mention.

Most of the readers should be familiar with the cycle of operation of the internal combustion motor of the four-stroke type, and it seems unnecessary to go into detail except to present a review. The first stroke of the piston is one in which a charge of gas is taken into the motor; the second stroke which is in reverse

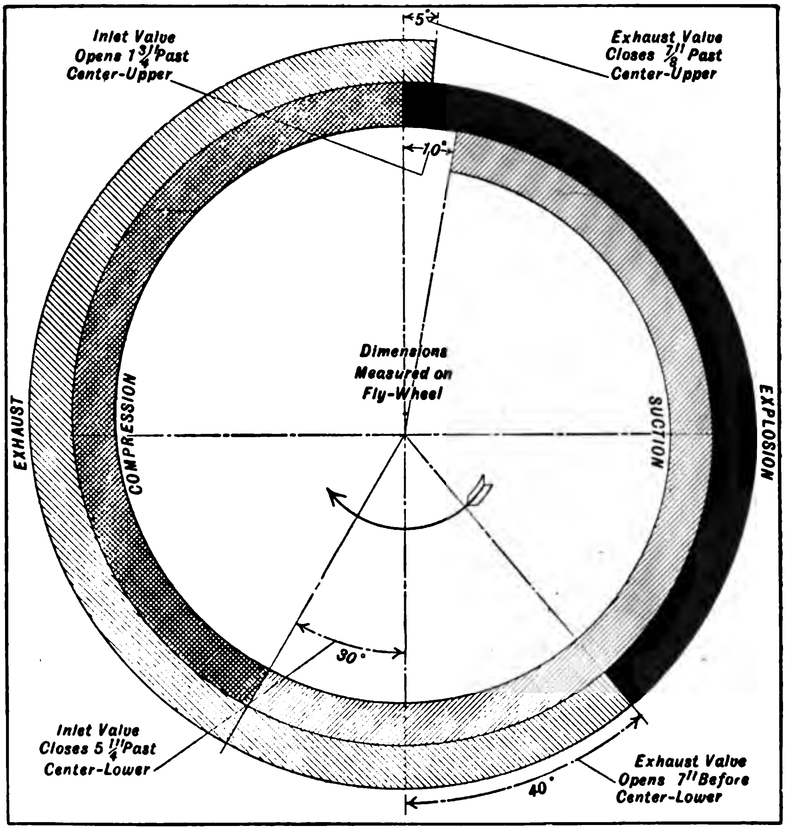


FIG. 374.—Diagram showing actual duration of different strokes in degrees.

direction to the first is a compression stroke, at the end of which the spark takes place, exploding the charge and driving the piston down on the third or expansion stroke, which is in the same direction as the intake stroke, and finally, after the piston has nearly reached the end of the stroke, another valve opens to allow the burned gases to escape, and remains open until the piston has reached the end of the fourth stroke and is in a position to begin the series over again. The ends of the strokes are reached when the piston comes to a stop at either top or bottom of the cylinder and reverses its motion. That point is known as a center and there are two for each cylinder, top and bottom centers, respectively.

All circles may be divided into 360 parts, each of which is known as a degree, and in turn each of these degrees may be again divided into minutes and seconds, though we need not concern ourselves with anything less than the degree. Each stroke of the piston represents 180 degrees travel of the crank, because two strokes represent one complete revolution or three hundred and sixty degrees. The top and bottom centers are therefore separated by 180 degrees. Theoretically each phase of a four-cycle engine begins and ends at a center, though in actual practice the inertia or movement of the gases makes it necessary to allow a lead or lag to the valve, as the case may be. If a valve opens before a center, the distance is called "lead;" if it closes after a center, this distance is known as "lag." The profile of the cams ordinarily used to open or close the valves represents a considerable time in relation to the one hundred and eighty degrees of the stroke, and the area of the passages through which the gases are admitted or exhausted are quite small owing to the necessity of having to open or close the valves at stated times; therefore, to open an adequately large passage for the gases it is necessary to open the valves earlier and close them later than at centers.

That advancing the opening of the exhaust valve was of value was discovered on the early motors and is explained by the necessity of releasing a large amount of gas, the volume of which has been greatly raised by the heat of combustion. When the inlet valves were mechanically operated it was found that allowing them to lag at closing enabled the inspiration of a greater volume of gas. Disregarding the inertia or flow of the gases, opening the exhaust at center would enable one to obtain full value of the expanding gases the entire length of the piston stroke, and it would not be necessary to keep the valve open after the top center, as the reverse stroke would produce a suction effect which might draw some of the inert charge back into the cylinder. On the other hand, giving full consideration to the inertia of the gas, opening the valve before center is reached will provide for quick expulsion of the gases, which have sufficient velocity at the end of the stroke, so that if the valve is allowed to remain open a little longer, the amount of lag varying with the opinions of the designer, the cylinder is cleared in a more thorough manner.

When the factor of retarded opening is considered, without reckoning the inertia of the gases it would appear that if the valve were allowed to remain open after center had passed, say on the closing of the inlet, the piston having reversed its motion would have the effect of expelling part of the fresh charge through the still open valve as it passed inward at its compression stroke. This effect is called blowing back and is often noted with motors where the valve settings are not absolutely correct, or where the valve springs or seats are defective and prevent proper closing.

This factor is not of as much import as might appear, as on closer consideration it will be seen that the movement of the piston as the crank reaches either end of the stroke is less per degree of angular movement than it is when the angle of the connecting rod is greater. Then again a certain length of time is required for the reversal of motion of the piston during which time the crank is in motion but the piston practically at a standstill. If the valves are allowed to remain open during this period, the passage

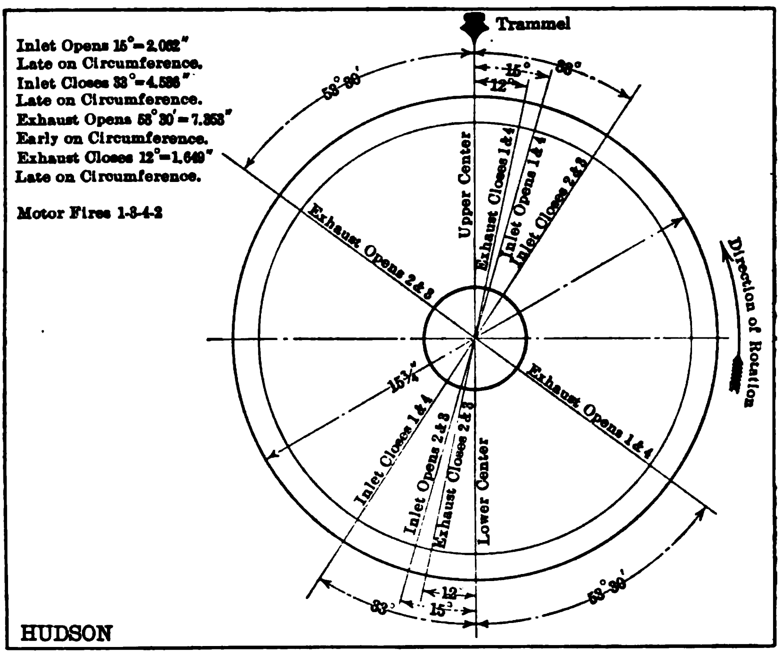


Fig. 375.—Diagram showing method of marking fly-wheel circumference to obtain proper timing of typical four-cylinder motor.

of the gas in or out of the cylinder will be by its own momentum.

The faster a motor turns, all other things being equal, the greater the amount of lead or advance it is necessary to give the opening of the exhaust valve. It is self-evident truth that if the speed of a motor is doubled, it travels twice as many degrees in the time necessary to lower the pressure. As most designers are cognizant of this fact the valves are proportioned accordingly. It is well to consider in this respect that the cam profile has much to do with the manner in which the valve is opened, that is, the lift may be abrupt and the gas allowed to escape in a body, or the opening may be gradual, the gas issuing from the cylinder in thin streams. An analogy may be made with the opening of any bottle which contains liquid highly carbonated. If the cork is removed suddenly the gas escapes with a loud pop, but on the other hand, if the bottle is uncorked gradually, the gas escapes from the receptacle in thin streams around the cork, and passage of the gases to the air is accomplished without noise. While the second plan is not harsh, it is slower than the former, as must be evident.

A point which has been much discussed by engineers is the proper relation of the closing of the exhaust valve and the opening of the inlet. Theoretically they should succeed each other, the exhaust closing at upper dead center and the inlet opening immediately afterward. The reason why a certain amount of lag is given the exhaust closing in practice is that the piston cannot drive the gases out of the cylinder unless they are compressed to a degree in excess of that existing in the manifold or passages, and while toward the end of the stroke this pressure may be feeble, it is nevertheless indispensable. At the end of the piston's stroke, as marked by the upper dead center, this compression still exists, no matter how little it may be, so that if the exhaust valve is closed and the inlet opened immediately afterward, the pressure which exists in the cylinder may retard the entrance of the fresh gas and a certain portion of the inert gas may penetrate into the manifold. As the piston immediately begins to aspirate this may not be serious, but as these gases are drawn back into the cylinder the fresh charge will be diluted and weakened in value. If the spark plug is in a pocket the points may be surrounded by this weak gas, and the explosion will not be nearly as energetic as when the ignition spark takes place in pure mixture.

It is a well-known fact that the exhaust valve should close after dead center and that a certain amount of lag should be given to opening of the inlet. The lag given the closing of the exhaust valve should not be as great as that given the closing of the inlet valve. Assuming that the excess pressure of the exhaust will equal the depression during aspiration, the time necessary to complete the emptying of the cylinder will be proportional to the volume of the gas within it. At the end of the suction stroke the volume of gas contained in the cylinder is equal to the cylindrical volume plus the space of the combustion chamber. At the end

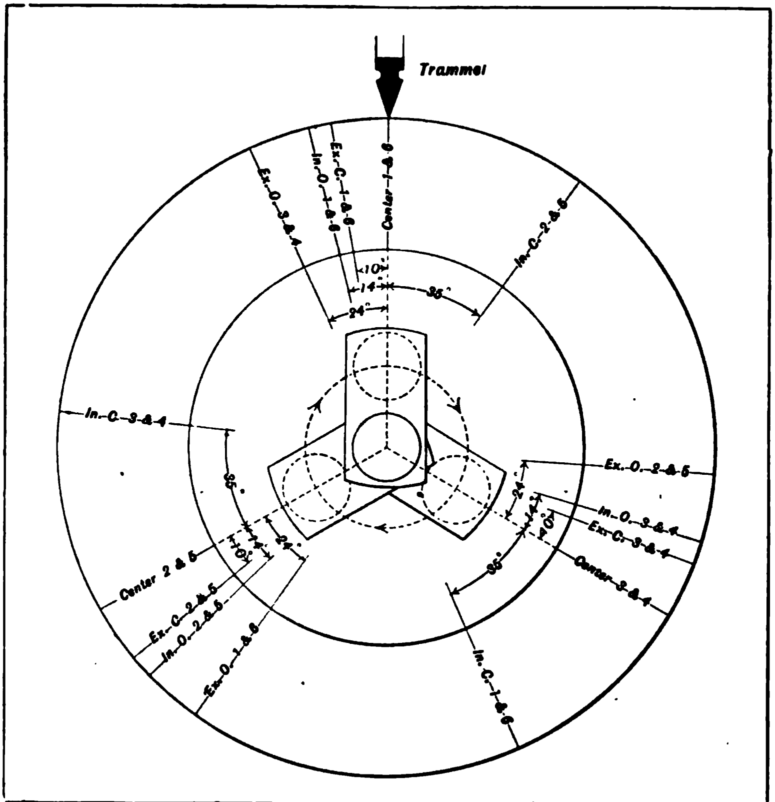


FIG. 376.—Showing method of marking rim of six-cylinder fly-wheel for guiding repairman or motorist to retain correct valve timing.

of the exhaust stroke the volume is but that of the dead space, and from one-third to one-fifth its volume before compression. While it is natural to assume that this excess of burned gas will escape faster than the fresh gas will enter the cylinder, it will be seen that if the inlet valve were allowed to lag twenty degrees, the exhaust valve lag need not be more than five degrees, providing that the capacity of the combustion chamber was such that the gasses occupied one-quarter of their former volume.

It is evident that no absolute rule can be given, as back pressure will vary with the design of the valve passages, the manifolds, and the construction of the muffler. The more direct the opening, the sooner the valve can be closed and the better the cylinder cleared. Ten degrees represents an appreciable angle of the crank and the time required for the crank to cover this angular motion is not inconsiderable and an important quantity of the exhaust may escape, but the piston is still very close to the dead center after the distance has been covered.

Before the inlet valve opens there should be a certain depression in the cylinder, and considerable lag may be allowed before the depression is appreciable. So far as the volume of fresh gas introduced during the admission stroke is concerned, this is determined by the displacement of the piston between the point where the inlet valve opens and the point of closing, assuming that sufficient gas has been inspired so that an equilibrium of pressure has been established between the interior of the cylinder and the outer air. The point of inlet opening varies with different motors. It would appear that a fair amount of lag would be fifteen degrees past top center for the inlet opening, as a certain depression will exist in the cylinder, assuming that the exhaust valve has closed five or ten degrees after center, and at the same time the piston has not gone down far enough on its stroke to materially decrease the amount of gas which will be taken into the cylinder.

As is the case with the other points of opening and closing, there is a wide diversity of practice as relates to closing the inlet valve. Some of the designers close this exactly at bottom center, but this practice cannot be commended, as there is a considerable portion of time, at least ten or fifteen degrees angular motion of the crank, before the piston will commence to travel any extent on its compression stroke. The gases rushing into the cylinder have considerable velocity, and unless an equilibrium is obtained the pressure inside and that of the atmosphere outside, they will continue to rush into the cylinder even after the piston ceases to exert any suction effect.

For this reason if the valve is closed exactly on center, a full charge may not be inspired into the cylinder, though if the time of closing is delayed, this momentum or inertia of the gas will be enough to insure that a maximum charge is taken into the cylinder. The writer considers that nothing will be gained if the valve is allowed to remain open longer than twenty degrees, and an analysis of practice in this respect would seem to confirm this opinion. From that point in the crank movement the piston travel increases and the compressive effect is appreciable, and it would appear that a considerable proportion of the charge might be exhausted into the manifold and carbureter if the valve were allowed to remain open beyond a point corresponding to twenty degrees angular movement of the crank.

TIME OF IGNITION

In this country engineers unite in providing a variable time of ignition, though abroad some difference of opinion is noted on this point. The practice of advancing the time of ignition, when affected electrically, was severely condemned by early makers, these maintaining that it was necessary because of insufficient heat and volume of the spark, and it was thought that advancing ignition was injurious. The engineers of to-day appreciate the fact that the heat of the electric spark, especially when from a mechanical generator of electrical energy, is the only means by which we can obtain practically instantaneous explosion, as required by the operation of motors at high speeds, and for the combustion of large volumes of gas.

One would consider that a motor with a fixed point of ignition was not in every way as desirable as one in which the ignition could be advanced to best meet different requirements, and the writer does not readily perceive any advantage outside of simplicity of control in establishing a fixed point of ignition. In fact, there seems to be some difference of opinion among those designers who favor fixed ignition, and in one case this is located forty-three degrees ahead of center, and in another motor the point is fixed at twenty degrees, so that it may be said that this will vary as much as one hundred per cent. in various forms. This point will vary with different methods of ignition, as well as the location of the spark plug or igniter. The writer favors a variable point of ignition, as this offers advantages which cannot be obtained with fixed ignition, and enables one to best gauge the requirements of the time of firing the charge by conditions of operation from time to time. The range may be as desired, varying from a point after center for starting to one forty-five degrees advanced for

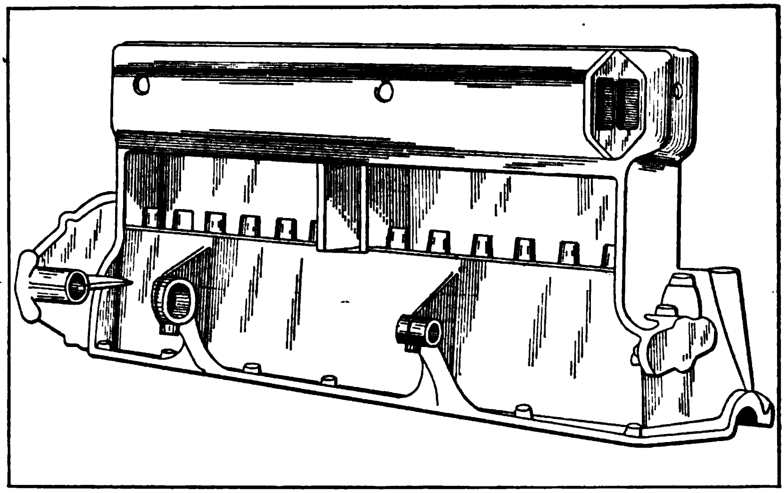


FIG. 377.—Block casting of Everitt "Six," a modern innovation in motor design because six cylinders, upper part of crankcase and inlet and exhaust manifolds are included in one casting.

maximum speed. Then again, flexibility of control is greatly increased when spark time may be varied to suit requirements.

It is obvious by consideration of the foregoing that there can be no arbitrary rules established for timing, because of the many conditions which determine the best times for opening and closing the valves. It is customary to try various settings when a new motor is designed until the most satisfactory points are determined, and the setting which will be very suitable for one motor is not always right for one of different design.

Typical valve timing diagrams are presented at Figs. 374 to

376 inclusive. The first shows the actual duration of the various parts of the cycle of a four-stroke engine in degrees and in inches measured on the special flywheel shown. Diagram Fig. 375 shows the timing marks for a four-cylinder motor, that at Fig. 376 shows the method of marking rim of a six-cylinder motor flywheel.

BLOCK CYLINDER CASTINGS

Considered from a purely theoretical point of view the individual cylinder casting has much in its favor. It is advanced that more uniform cooling is possible than where the cylinders are cast either in pairs or three or four in one casting. More uniform cooling insures that the expansion or change of form due to heating will be more equal. This is an important condition because the cylinder bore must remain true under all conditions of operation. If the heating effect is not uniform, which condition is liable to obtain if metal is not evenly distributed, the cylinder may become distorted by heat and the bore be out of truth. When separate cylinders are used it is possible to make a uniform water space and have the cooling liquid evenly distributed around the cylinder. In multiple cylinder castings this is not always the rule, as in many instances, especially in four-cylinder block motors where compactness is the main feature there is no space between the cylinders for the passage of water. Under such circumstances the cooling effect is not even, and the stresses which obtain because of unequal expansion may distort the cylinder to some extent.

The advantage of casting the cylinders in blocks is that a motor may be much shorter than it would be if individual castings were used. It is admitted that when the cylinders are cast together a more compact, rigid, and stronger power plant is obtained than when cast separately. There is a disadvantage, however, in that if one cylinder becomes damaged it will be necessary to replace the entire unit, which means scrapping three good cylinders because one of the four has failed. When the cylinders are cast separately one need only replace that one that has become damaged. The casting of four cylinders in one unit is made possible by improved foundry methods, and when proper provision is made for holding the cores when the metal is poured and the cylinder casts are good, the construction is one of distinct merit.

When cylinders are cast in block form it is good practice to leave a large opening in the jacket wall which will assist in supporting the core and make for uniform water space. It will be noticed that some castings have a large opening in the side of the cylinder block. These openings are closed after the interior of the casting is thoroughly cleaned of all sand, core wire, etc., by brass, cast-iron or aluminum plates. These also have particular value in that they may be removed after the motor has been in use, thus permitting one to clean out the interior of the water jacket and

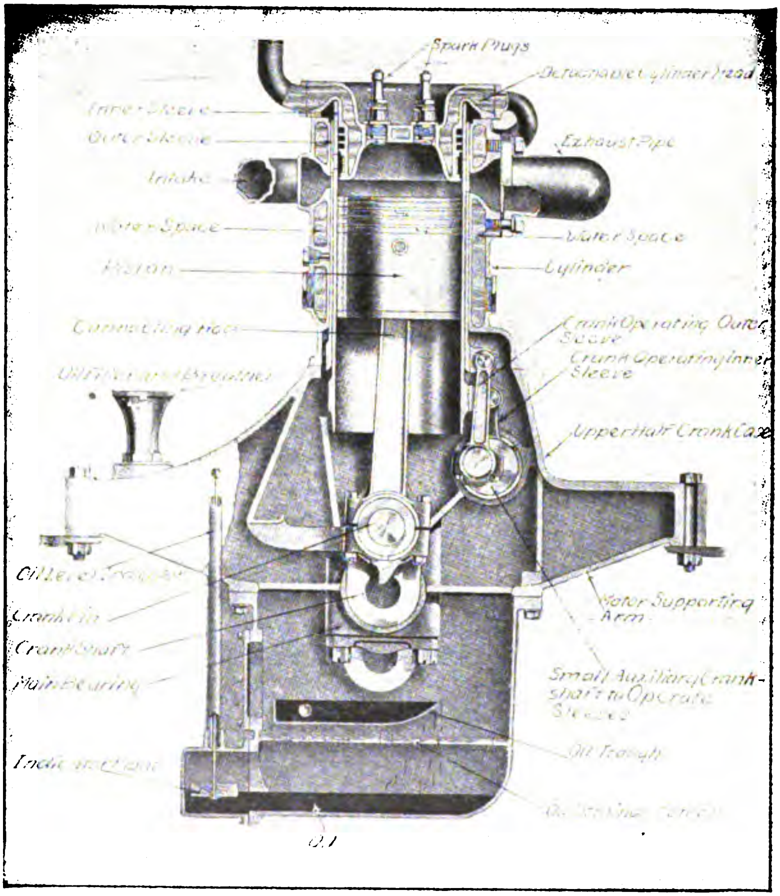


FIG. 378.—End section of Knight engine, showing sliding sleeves which replace poppet valves for controlling gas passages.

dispose of the rust, sediment, and incrustation which is always present after the engine has been in active service for a time.

Among the advantages claimed for the practice of casting cylinders in blocks may be mentioned compactness, lightness, rigidity, simplicity of water piping as well as permitting the use of simple forms of inlet and exhaust manifolds. The light weight is not only due to the reduction of the cylinder mass but because the block construction permits one to lighten the entire motor. The fact that all cylinders are cast together decreases vibration, and as the construction is very rigid, disalignment of working parts is practically eliminated. When inlet and exhaust manifolds are cored in the block casting as is sometimes the case, but one joint is needed on each of these instead of the multiplicity of joints which obtain when the cylinders are individual castings. The water piping is also simplified. In the case of a four-cylinder block motor but two pipes are used; one for the water to enter the cylinder jacket, the other for the cooling liquid to discharge through. A typical block casting in which six cylinders are cast as a unit is shown at Fig. 377. The upper half of the engine crank case and the inlet and exhaust passages are also included in this casting. Larger six-cylinder engines usually have the cylinders cast in pairs or in blocks of three.

THE KNIGHT MOTOR

One of the most recent developments in automobile engines and the most radical departure from the accepted conventional designs is the Knight slide valve motor shown at Figs. 378 and 379, which has many features of merit. The operating principles in this engine do not differ materially from other four-cylinder, four-cycle types, the only difference being in the method of admitting and expelling gases from the cylinder. The illustration as Fig. 378 shows very clearly the difference which exists between the slide valve and the conventional poppet valve motor at Fig. 369. Both of these are the same in general design, except that changes have been made in the power plant to permit the use of reciprocating sleeves.

The Knight motor shown at Fig. 379 has four cylinders cast in pairs. The top of each cylinder has two lateral slots which communicate respectively with the inlet and exhaust pipes. The

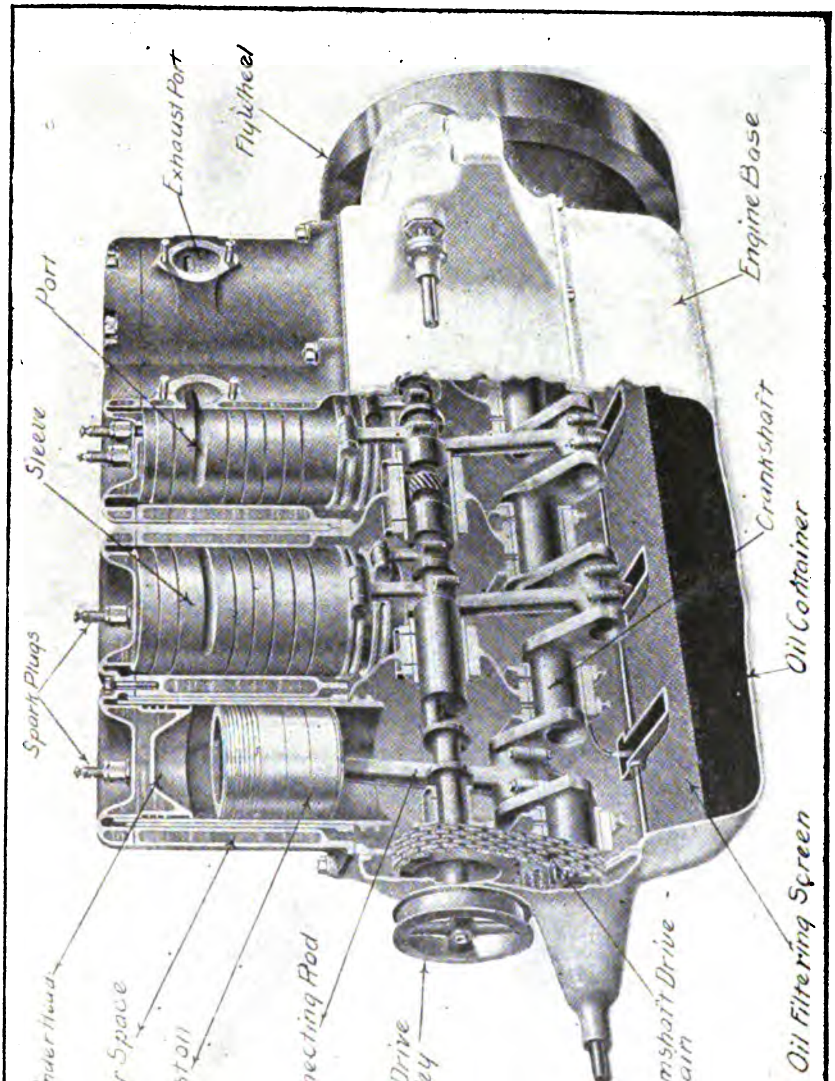


Fig. 379.—Partial side sectional view of Knight sleeve valve engine, showing interior mechanism.

cylinder is water-jacketed, and inside of this member and interposed between it and the piston are two thin, hollow cast-iron cylinders, or sleeves, adapted to be moved up and down by a suitable crankshaft and connecting rod mechanism or eccentrics. These sleeves have large ports which communicate with the orifices in the cylinder wall. They are moved in such a manner that the slots in the cylinder are opened and closed by the reciprocating movement of the sleeves. They are operated by small connecting rods which work from a smaller crankshaft mounted on one side of, and above the main crankshaft, and driven by silent chain gearing. The travel of the sleeves is comparatively small, as their velocity is but one-tenth that of the piston. The openings in the sleeves are so wide that the gases enter and leave the combustion chamber much more easily than they could through ports closed by valves of the conventional type.

The movement of the sleeves (Fig. 380) is such that the ports in the cylinder are closed by one or both sleeves during three-quarters of the cycle of operation, and are kept open during the remaining quarter by a simultaneous lining up of the openings in both sleeves with that in the cylinder. It is claimed that in the Knight motor the absolute constancy of compression makes for uniformity of action because the intervals between the successive explosions are always equal and all of the power strokes have the same strength. It is also advanced that the construction of the Knight motor makes it possible to obtain combustion chambers which are equal in volume, which condition is difficult to attain with the ordinary construction, because of the difficulty met in securing perfect equality of castings. As the cylinders and cylinder heads of the Knight motor are machined to the required dimensions and polished, all combustion chambers will have the same volume. Another advantage is that there will be no projecting particles of metal such as would be present in castings that might remain hot and cause premature explosions. It is also difficult for carbon to adhere to the absolutely smooth walls of the combustion chamber or piston head.

There is very little strain on the parts, and as the wear of the sleeves is negligible the motor action improves with service, because the sleeves become polished and work easier the more they are used. As the sleeves are driven by cranks and connecting rods and not by cams as poppet valves are, they are not liable to go wild at even the highest motor speeds. The ports are opened and closed exactly at the proper time, and the openings or passages for the gas are so large that the motor capacity invariably augments with an increase of velocity up to the limit of rotative speed.

In a comparative test of two similar motors, one with mushroom valves and the other with sleeves, the former developed but twenty-five horse power at 2,000 revolutions per minute, while the sleeve

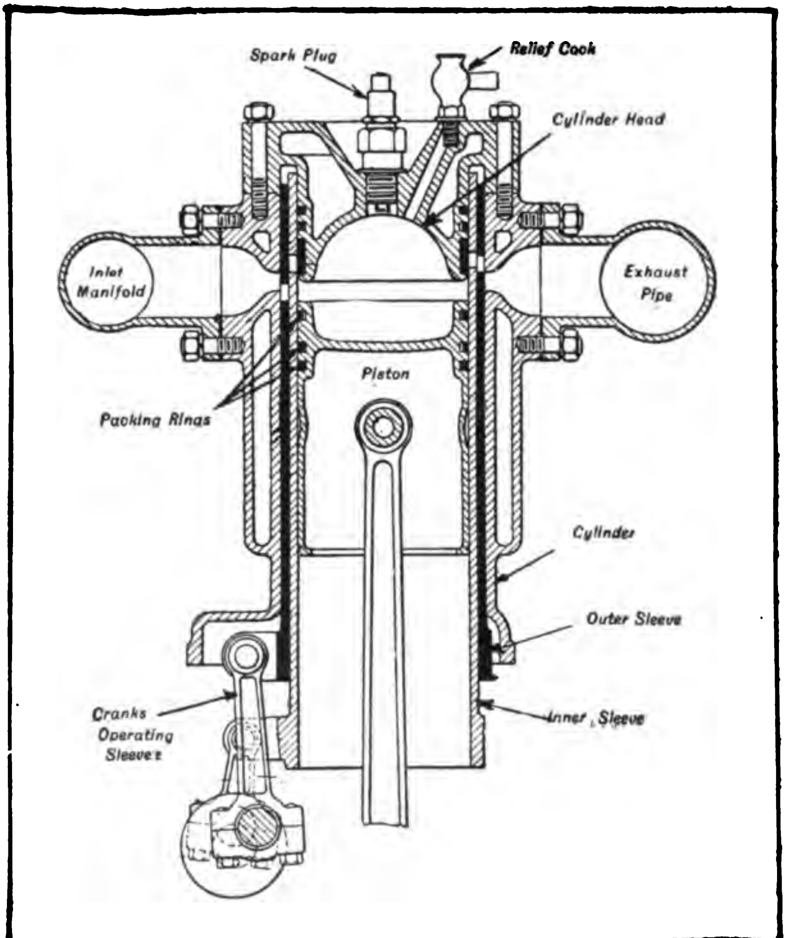


FIG. 380.—Section through cylinder of Knight motor, showing important parts of valve motion.

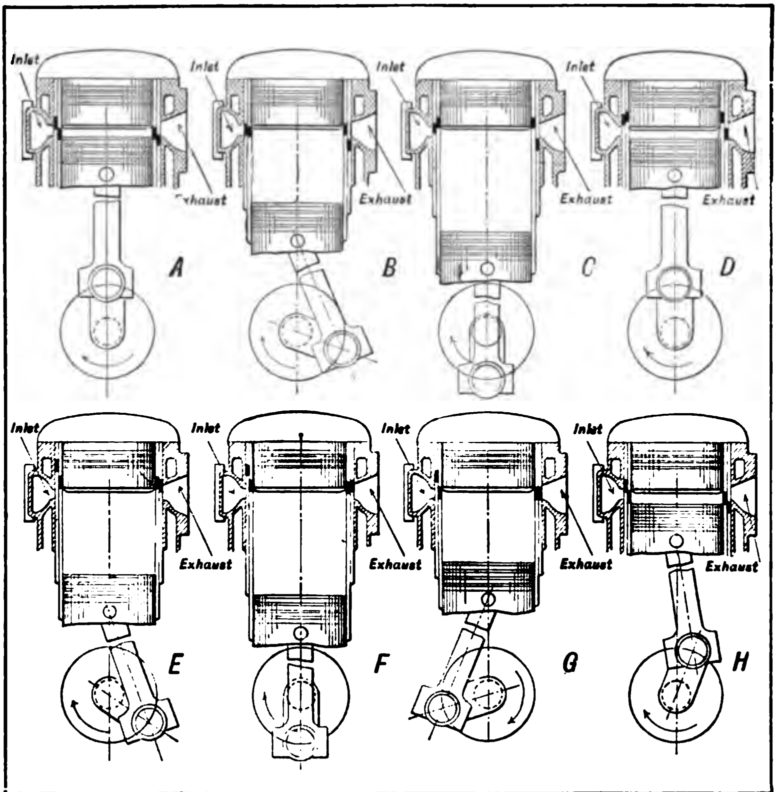


FIG. 381.—Diagram showing relative movement of sleeves and camshaft of Knight type motor.

Note port opening at various piston positions. Shaded portions of sleeves represent ports.

type generated in excess of thirty horse-power under the same conditions. The Knight motor has been subjected to severe tests before adoption in comparison with motors of the poppet valve type. In one of these an engine rated at thirty-eight horse-power which had cylinders of 5-inch bore and stroke developed 55.3 horse-power continuously during a period of 5½ days, or 132 hours. The fuel consumption was but 0.85 pint of gasoline per horse-power hour. The average fuel consumption of the four-cycle type of motor is placed at one pint per horse-power hour. At the completion of this running in test the power plant was installed in a car weighing 4,000 pounds. This was driven over 2,000 miles on Brooklands Motor Track, near London, England, at a speed which averaged forty-three miles per hour. At the completion of this test the motor was replaced on a test stand in the shop where it developed an average of 57.25 horse-power during a run of five hours at 1,200 revolutions per minute. The fuel consumption was reduced to 0.75 pint of gasoline per horse-power hour, and it had gained two horse-power, or about four per cent. by use. This type of valveless motor is considered to be an improvement over the conventional forms, and it is all the more strange when one considers that the height of its development has been reached at a time when all

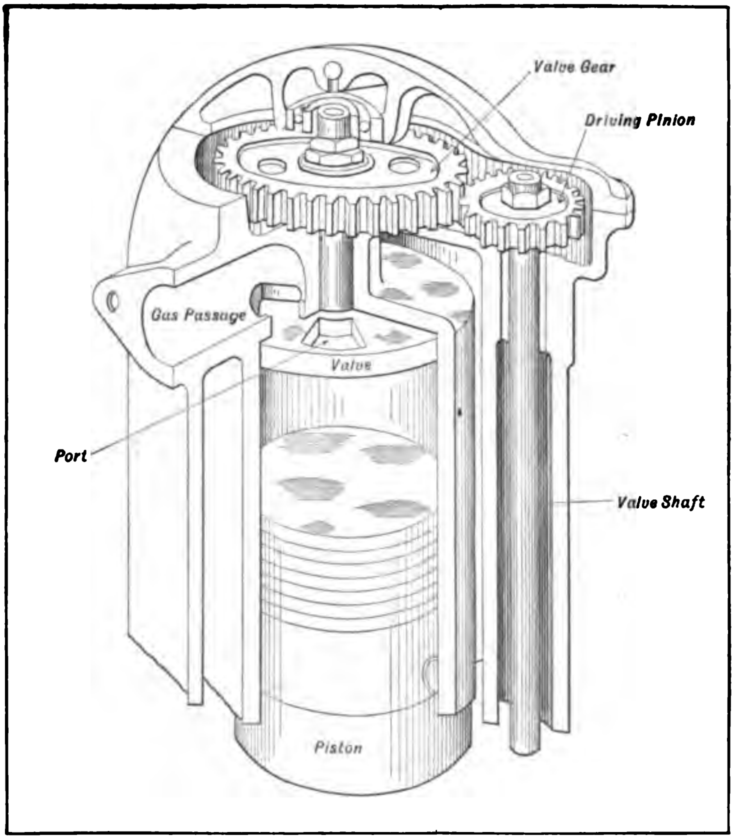


FIG. 382.—Partial section of Reynolds rotary valve motor cylinder, showing method of rotating simple disc valve and ports in cylinder head.

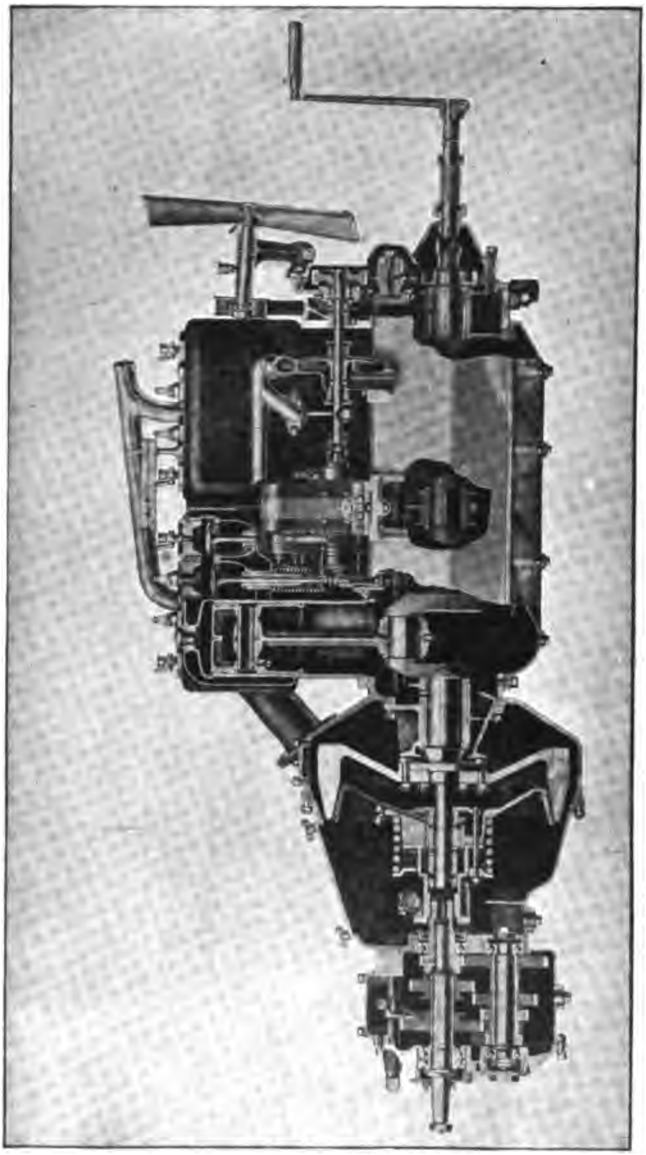


FIG. 383.—Sectional view of unit power plant used on Jackson automobile, showing interior of motor, clutch case, and speed change mechanism.

believed the explosion motor had attained its maximum efficiency. The success attending the use of the Knight motor has promoted great interest in all forms of valveless motors which are being actively experimented with at the present time.

A ROTARY VALVE MOTOR

The Reynolds motor, a sectional view through one of its cylinders being shown at Fig. 382, has not been used to any extent in automobile service but has proven thoroughly practical in marine applications. The valve consists of a flat disc seating directly against the top of the combustion chamber. It is turned by a shaft which extends through a boss on top of the cylinder head and which is driven direct from the crankshaft by gearing at half the motor speed. The valve has a port cut into it of the keystone shape, clearly shown in illustration, this registering successively with openings in the cylinder head. The valve mechanism is said to be very quiet, and as will be seen at Fig. 382 the cylinder is a very compact design. A disadvantage is cited that the force of the explosion keeps the valve disc tight against the seat, this tending to cause considerable resistance to its motion. It is claimed that no difficulty is experienced from this source, and that an oil film is maintained positively between the valve disc and its seat so that it turns with minimum friction.

Various other forms employing rotary valves have been devised, and some of these are said to have been used in a practical way. An American design is known as the Mead. This is a four-cylinder motor with two long cylindrical valves extending along opposite sides of the cylinders in close connection with the combustion chamber. These cylinders have ports cut through them at distances equal to the center line of the cylinders and are suitably spaced so that the ports in the cylinders are uncovered in proper succession. One of the drums serves to control the inlet ports; the other regulates the exhaust openings. The valves are driven at one-quarter crankshaft speed by suitable gearing.

POWER PLANT INSTALLATION

The method of installing the power plant varies on different types of automobiles, though the majority of cars have the engine placed at the extreme front end of the classis as at Fig. 384. In

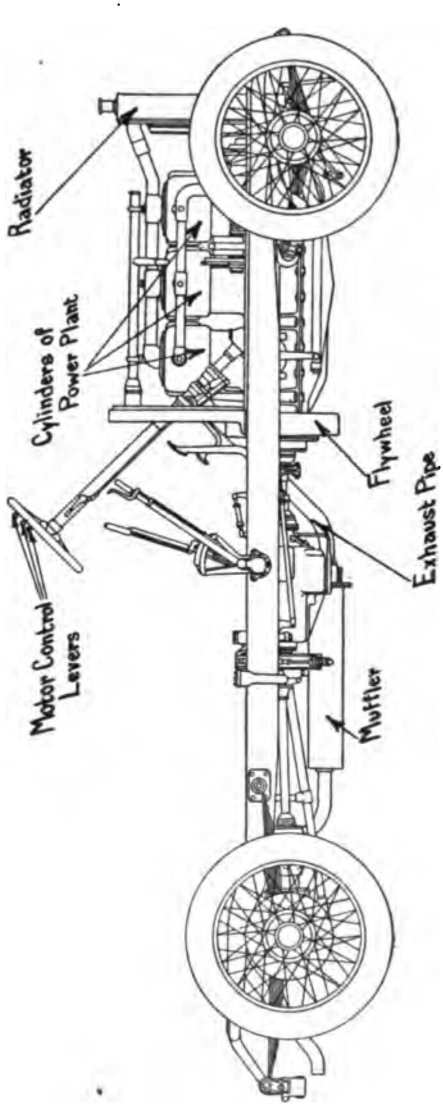


Fig. 384.—Side elevation of typical pleasure car chassis, showing arrangement of power plant, which is representative of conventional practice.

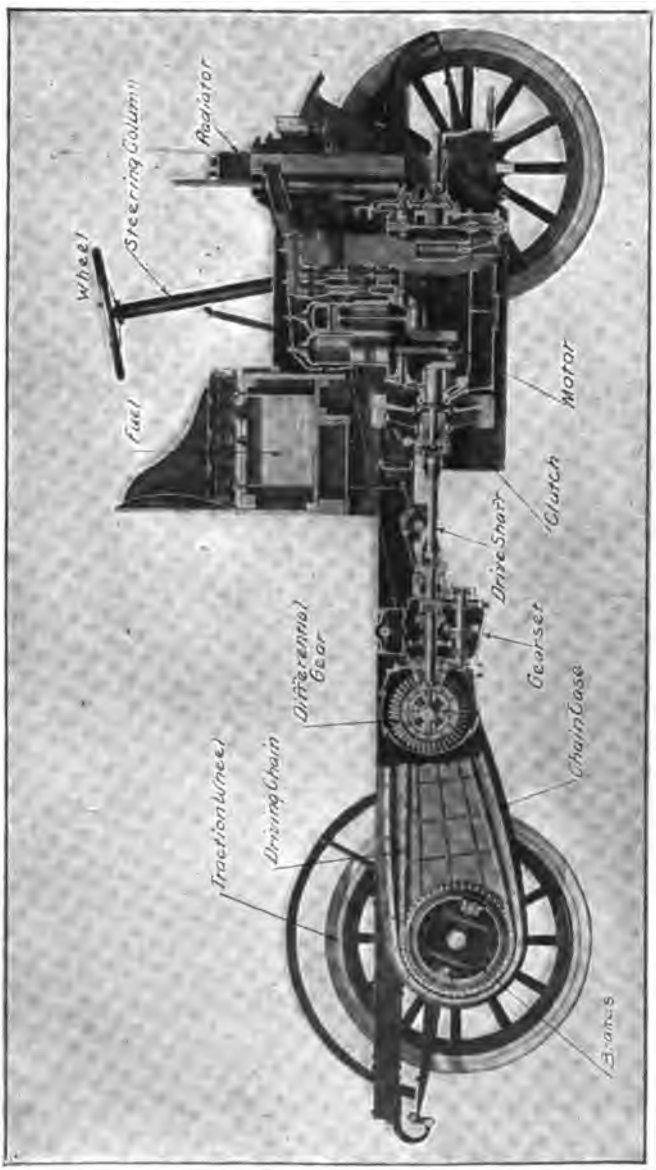


Fig. 385.—Sectional view of Natco truck chassis, outlining disposition of important chassis components and their relation to each other.

some types of cars where single or double motors of the horizontal type are used the motor is placed under the body. This type of construction is nearly obsolete at this time, and is found only on early forms of vehicles and one or two commercial cars.

The power plant is sometimes combined with the clutch and change speed gearing in such a way as to form a unit construction. This method of thus joining the parts is widely used at the present time, and is superior to the other common method where the motor and change speed gears are independent units. Each method has advantages. As will be seen by inspecting Fig. 385, when the gearset and motor are separate the transmission may be removed from the chassis frame without disturbing the power plant, and vice versa. At the other hand it is sometimes difficult to remove one member without having to take the entire unit from the frame.

The unit construction has the advantage of retaining positive alignment of the gearset with the engine indefinitely. This relation between the parts is obtained when they are first assembled and the alignment cannot be changed by any condition of operation after the unit is installed in the frame. This method of mounting also permits the three-point suspension which is very desirable. For instance, the power plant shown at A is supported on four points and the gearset is supported on another series of four points. While the tendency of these members is to brace the frame and prevent disalignment, it is possible on extremely rough roads for the frame distortion to vary the relation of the transmission and engine shaft to some extent. Where a three-point suspension is employed, as outlined at Fig. 383, the frame distortion will not impose stress on the individual members of the power plant because in a rigid unit construction all parts must remain in alignment. The advantages of this design are becoming better appreciated and it is widely used at the present time.

In trucks, as is clearly shown at Figs. 385 and 386, the general method is to mount the engine under the driver's seat in a suitable compartment. Some trucks have the motor mounted, as in pleasure car practice, in front under a bonnet. The advantages of the motor under the seat location may be very well summed up by saying that it permits more loading space and less over all or wheel base for a given carrying capacity. The shorter wheel base vehicle is especially valuable in congested city traffic, because it may be

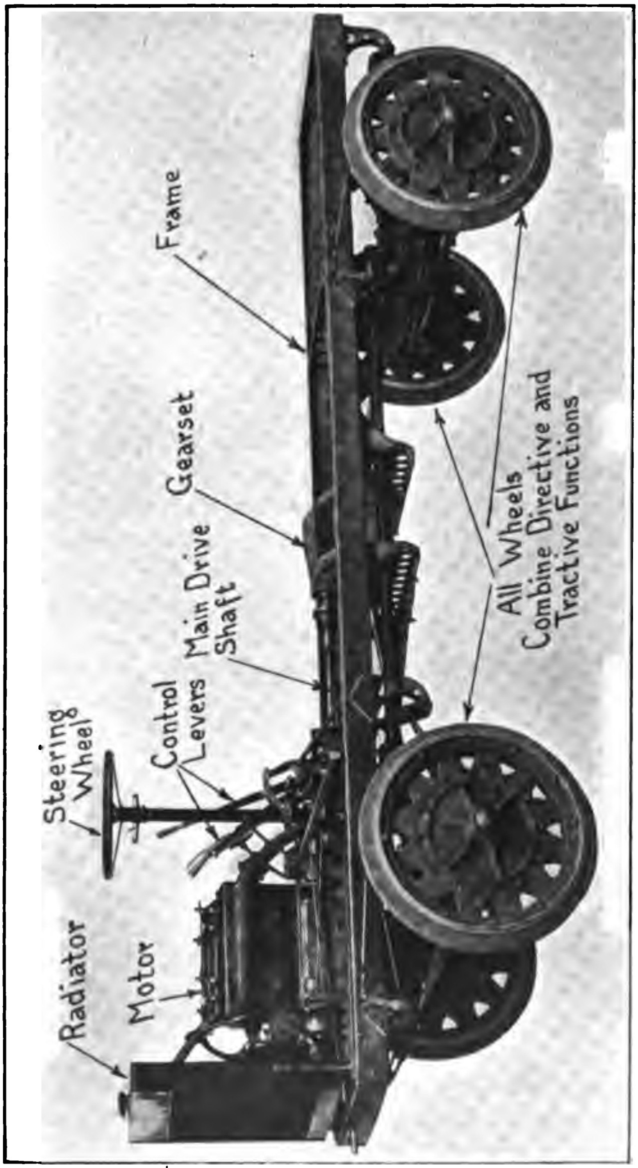


FIG. 386.—Showing location of power plant on the Jeffery four-wheel drive truck chassis.

more easily controlled when driving in narrow thoroughfares, taking corners, or backing up to a loading platform. The main advantage advanced for the motor in front type of commercial vehicle is accessibility of power plant, which may be easily reached by raising the hood. This feature is not lost when the motor is placed under the seat, however, because all average adjustments may be made by raising the floor boards or by opening a hinged door at the side of the motor compartment. Some makers who install the motor under the seat arrange the components in such a manner that they may be removed as units permitting ready access to the motor and making for its prompt removal in event of overhauling or serious accident.

MOTORCYCLE ENGINE DEVELOPMENT

Before the internal combustion engine had been fitted to any form of four wheel motor vehicle it had been applied successfully by Daimler, prior to 1885, to a two-wheel conveyance built on the lines of the bicycle of that period. This proved successful in this application and the crude machine of that time in its essentials became the parent of both the present day automobile and motorcycle. The efforts of the earlier designers was to use the forms of bicycles then available and to transform these into a motorcycle by the addition of a simple power plant consisting of a small air cooled gasoline-engine and its auxiliaries. It was soon learned that the ordinary bicycle frame structure was not strong enough to resist the vibratory stresses imposed by even the small motors of that period, which were of about 2½″ bore and stroke and which were credited with 1½ or 2 horse-power at speeds of from 2·500 to 3,000 revolutions per minute.

The converted cycle frame structure depreciated also from the vibration produced when operated at the speeds mechanical power made possible over the average highway surfaces, which were not altogether favorable to any form of vehicular travel in the days before the army of wheelmen started the agitation for their improvement. As soon as it was found imperative to strengthen the frame structure it was also necessary to increase the motor power provided to handle the augmented weight of the machine and to produce the necessary power for overcoming the resistance imposed by hills and sandy roads. The power of small engines was

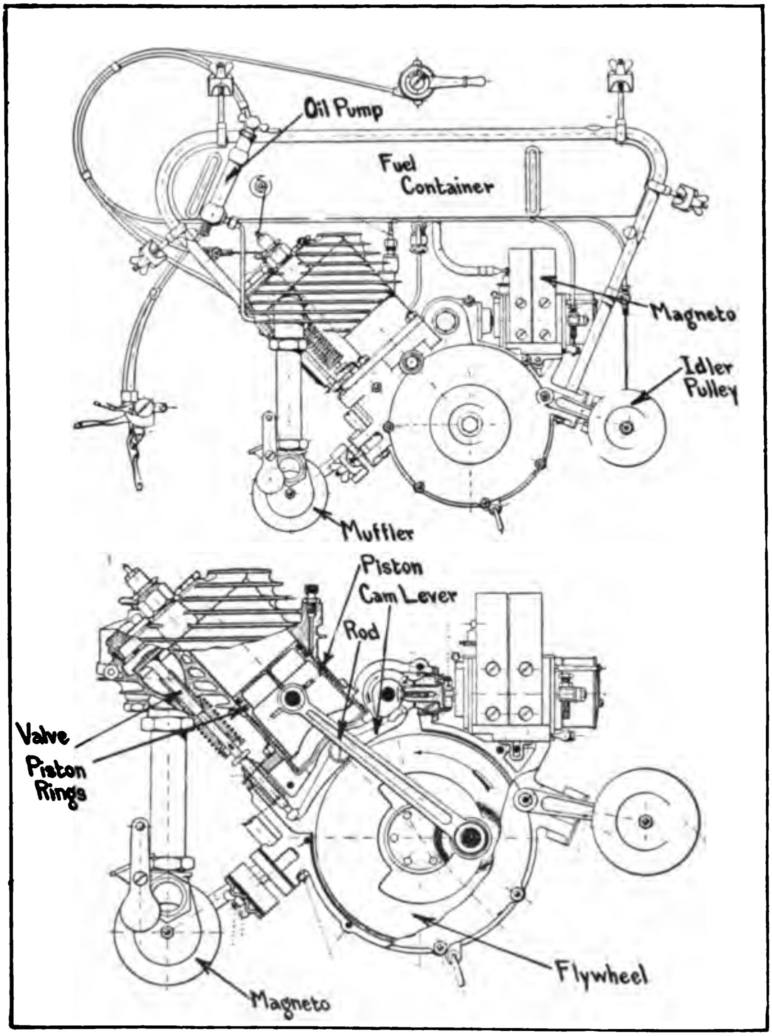


FIG. 387.—View showing construction of Motosacoche light-weight power plant adapted for converting diamond frame bicycle to motorcycle.

gradually augmented until, at the present time, the average motorcycle is provided with a twin cylinder engine rated at four to six times the power rating of the early prime movers.

As an example of one of the light power plants intended to be applied to motorcycle propulsion the reader is asked to study the illustration at Fig. 387. This power plant, which is sold complete, ready for attachment in the frame of the ordinary bicycle, is shown with all the elements in place within the supplementary or sub frame which can be clamped in place to the tubes of the ordinary diamond frame bicycle without any trouble. The power plant consists of a small single cylinder motor of the high speed air-cooled type with the cylinder head inclined toward the steering head and the crank case mounted at a point that will just clear the pedal crank hanger of the bicycle. The fuel is carried in a tank attached to the top bar of the sub frame and is vaporized by a float feed carburetor attached to the inlet valve chamber which is not shown in the illustration, being hidden by the engine cylinder. An oil pump of the plunger type is mounted at the front end of the tank and communicates with a compartment at the front in which lubricating oil is carried. The magneto that furnishes the ignition spark is attached to a suitable bracket above the crank case and as outlined in the lower view at Fig. 387 it is driven from the camshaft by means of bevel gearing.

The internal construction of the motor may be clearly understood by referring to the part sectional view at the lower portion of Fig. 387. A feature of merit is the manner of casting the cylinder flanges so they are horizontally disposed relative to the frame tubes, though at an angle to the cylinder center line. This feature insures that the air flow induced by motion of the cycle will reach all parts of the cylinder head, valve chamber and that portion exposed to the heating effect of the explosion. The method of installing the valve chamber at the front end of the motor so the cooling air currents will strike that portion first is also to be commended. Power is delivered from the engine crankshaft by a pulley adapted to receive a round rawhide belt that passes over a movable idler pulley before it reaches around the large driving pulley attached to the rear wheel rim.

The construction of a typical single cylinder motorcycle engine of American design is clearly shown at Fig. 388. This engine is of the L head form and has both valves mounted at one side of the cylinder. The inlet and exhaust valves, their springs and actuating plungers are duplicates and interchangeable. The method of operating the valve lifters by means of cam riders in the form of bell cranks is clearly outlined in the part sectional view showing

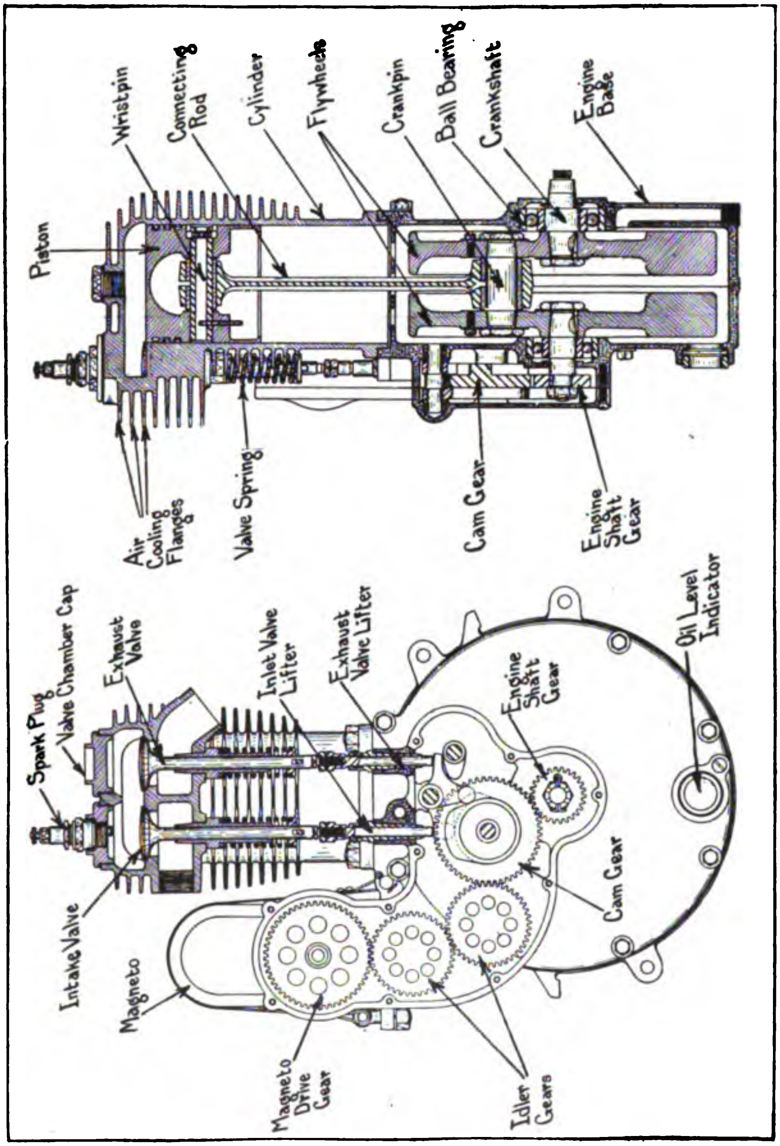


FIG. 388.—Sectional view showing arrangement of interior mechanism of the Reading Standard single cylinder motorcycle engine.

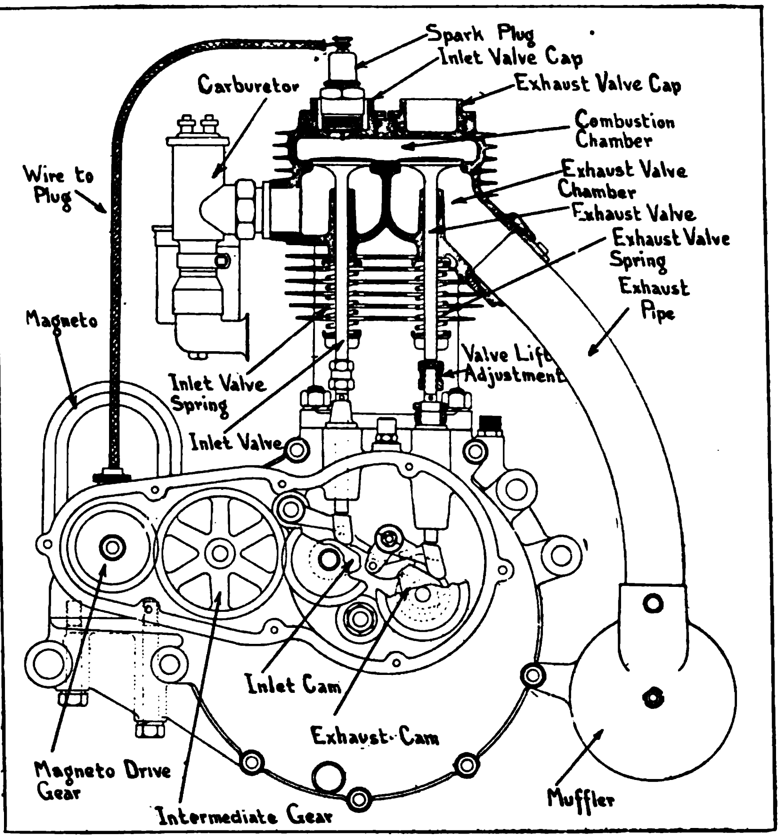


FIG. 389.—Motorcycle power plant of foreign design, showing location of carburetor, muffler, and ignition magneto.

the section through the timing gear case and the valve chamber. As this engine is intended to be mounted in the motorcycle frame with the cylinder vertical the flanges are cast horizontally. Attention is directed to the method of driving the ignition magneto by means of idler gears from the open cam gear. The gear on the engine crankshaft that drives the cam gear has but half its number of teeth, because, as is true of any four-cycle motor, the camshaft must be driven at half the crankshaft speed.

The arrangement of the connecting rod, piston, flywheel assembly and other internal parts is clearly shown in the sectional view at the right of Fig. 388. It will be observed that the crank-

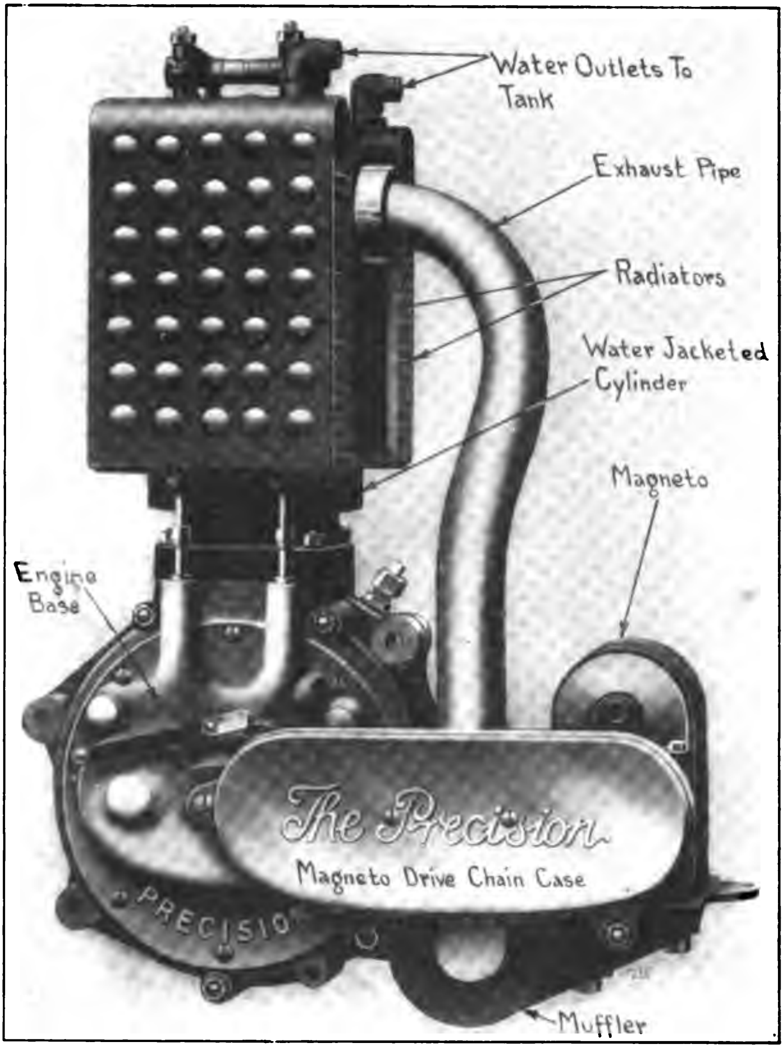


FIG. 390.—Arrangement of water-cooling radiators on motorcycle power plant.

shaft assembly is a built-up form, being comprised of two flywheel members held together by the crank pin. Each flywheel member is provided with one-half of the crankshaft and the entire assembly revolves on ball bearings of the single row annular type. The bearings at the upper and lower end of the connecting rods are bronze bushings. The construction of the engine base, which is made in two halves joined together at the engine center line is made necessary in order to assemble the flywheel member easily. This form of engine is rated at about 4 H.P., which is ample for medium weight machines not intended for racing purposes. Another single cylinder power plant of English design with the important parts clearly outlined is shown at Fig. 389. This engine is sold as a unit with the silencer or muffler attached and the ignition magneto and gas producing carburetor in their proper places. This engine does not differ much structurally from those previously shown.

In America practically all motorcycles that have received commercial application are provided with power plants of the air-cooled form. In Europe, some effort has been made to employ water-cooled engines on motorcycles. While the regular methods of utilizing water for cooling would involve the use of more or less complicated cooling apparatus, such as a radiator, water tank, circulating pump and water piping, in the motor shown at Fig. 390 a very simple form has been obtained that does not call for much added mechanism. The engine is practically the same in general design as the form shown at Fig. 389 except that the cylinder is provided with a water jacket, two sides of which are formed by water-cooling radiators of simple construction. The only piping necessary is two small pipes leading to a compartment of the fuel tank which is reserved to carry about two quarts of water. As the radiators are carried at the side of the cylinder they are directly in communication with the water jacket, are not in the way and are not likely to be injured, should the machine tip over. The spaces or air passages are arranged in such a manner that air must pass through the cooling radiators whenever the motorcycle is in motion. American designers have not considered it necessary to use water-cooling on motorcycles because of the splendid results given by the simpler air-cooling methods, which are really effective on light motors having less than four inches bore and stroke.

It is therefore necessary to utilize two cylinders when more than four or five horse-power is needed in order to keep the cylinder size to the limits where effective air-cooling is obtained.

As is true of any power plant, the use of two cylinders insures a more uniform delivery of power and smoother operation than secured with a single cylinder engine. A popular American twin motor, which is used on the Indian motorcycle, is shown in part section at Fig. 391. It is a combination of two single cylinder engines mounted above a common engine base with the connecting

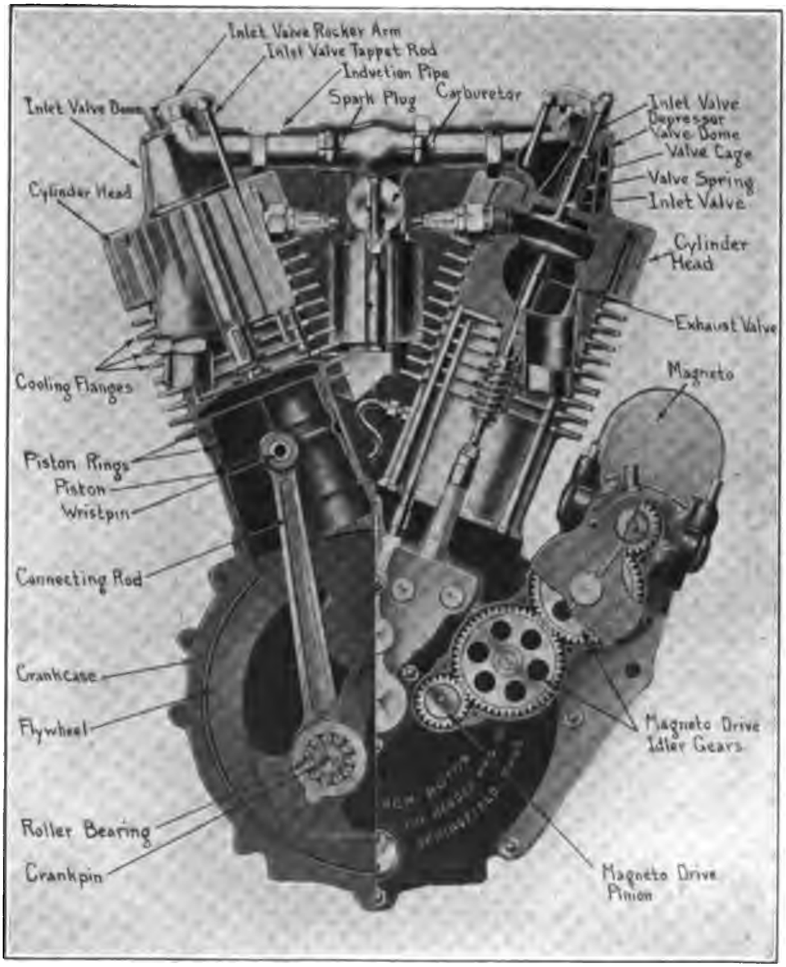


Fig. 391.—Part sectional view defining construction of twin-cylinder V motor used on Indian motorcycle.

rods attached to a common crank pin in the regular pattern fly-wheel assembly. In this motor the inlet valve is located over the exhaust member and is operated by an overhead rocker arm actuated by the usual form of tappet rod from the cam case. The attention of the reader is directed to the use of a roller-bearing at the lower end of the connecting-rod assembly instead of the usual plain bronze bushings. Anti-friction bearings not only have superior endurance but actually consume less power than the plain bushings. Their reliability and effectiveness were thoroughly

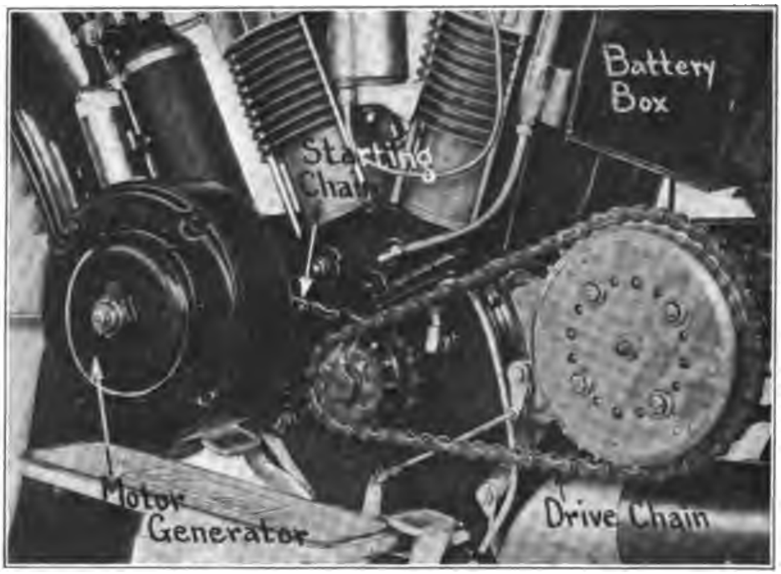


FIG. 392.—Application of electric starting motor to Indian motorcycle power plant.

proven by many months of severe testing in racing engines before they were supplied as standard equipment on the stock models.

A feature of some merit that is incorporated on some models of the Indian motorcycle is an electric self-starting system which makes it possible to turn over the motor crankshaft by means of an electric motor connected to it by a starting chain. The motor derives its current from storage batteries carried under the seat. This apparatus is in the form of a combined motor-dynamo and when the engine is started it becomes a dynamo and is driven

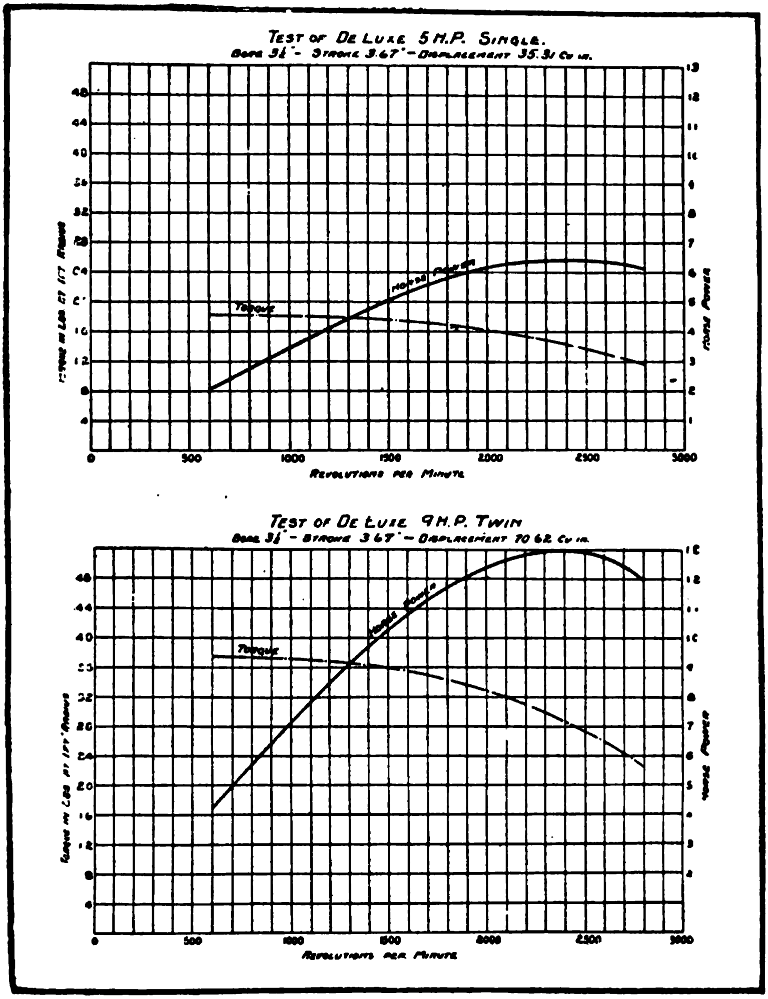


FIG. 393.—Diagram showing horse-power rating of De Luxe motorcycle engines.

from the engine to generate current to keep the storage batteries always properly charged. Automatic regulating means are provided so the batteries will not be over-charged when the engine speed increases. A motorcycle fitted with an apparatus of this kind such as shown at Fig. 396 uses the electric current generator, not only for starting the motor, but for ignition purposes and lighting as well.

The small motors built for motorcycle propulsion deliver considerably more power than their normal rating. Diagrams are shown at Fig. 393 giving horse-power and torque curves obtained by an electric cradle dynamometer which shows the actual brake horse-power delivered at various motor speeds. In the upper diagram it will be apparent that the single cylinder type rated at 5 H.P. and having a bore of $3\frac{1}{2}$ inches and stroke of 3.67 inches

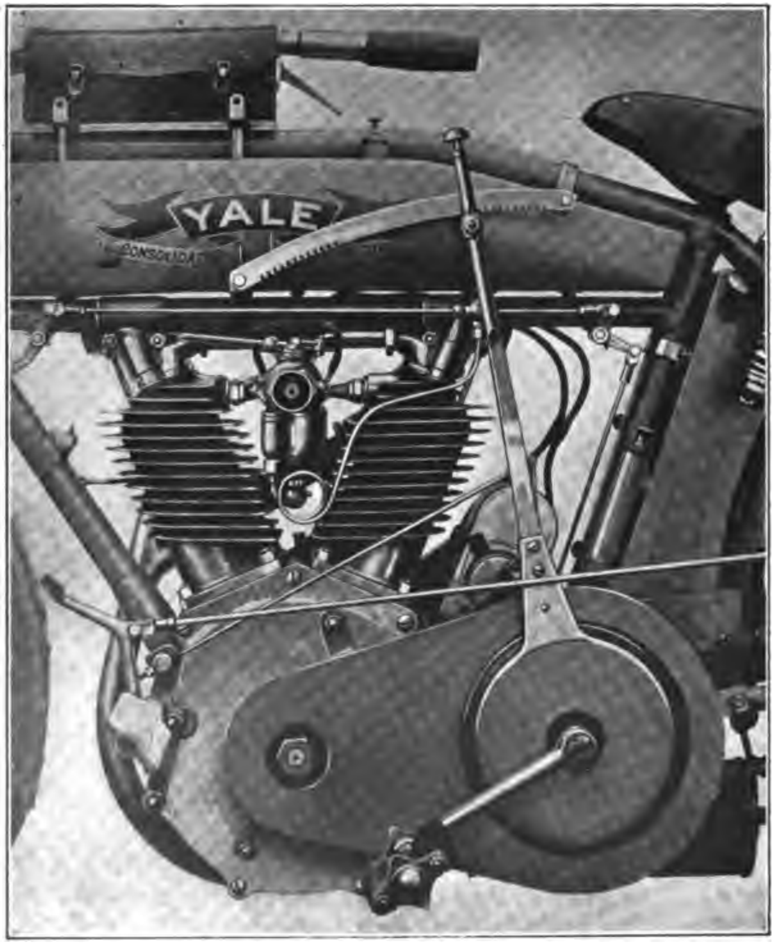


Fig. 394.—Carburetor side of Yale two-cylinder motorcycle power plant, showing method of installation in frame.

will deliver in excess of 6 H.P. at speeds between 2000 and 2800 R.P.M. The displacement of this single cylinder motor is 35.51 cubic inches. In the lower diagram curves showing a test of a twin cylinder motor having a piston displacement of 70.62 cubic inches and rated normally at 9 H.P. shows it will deliver 13 H.P. at a speed of 2400 R.P.M. A single cylinder motor complete with carburetor magneto and spark plugs that will deliver 6 H.P. weighs about 65 lbs., while a 9 H.P. form that actually is capable of developing 13 H.P. weighs not more than 100 lbs. with complete equipment. The average single cylinder power plant will therefore deliver about 1 H.P. for each 10 lbs. weight while the twin cylinder

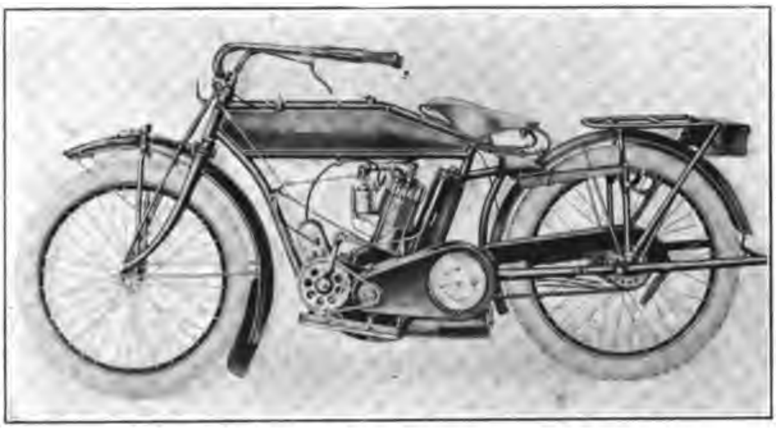


FIG. 395.—Typical single cylinder motorcycle of modern design, incorporating two-speed gear, spring frame, and kick starting arrangement.

forms will give 1 H.P. for each 8 lbs. weight. This ratio of weight to H.P. is lowered only on the extremely light forms of power plants intended for aerial navigation.

The installation of a typical twin cylinder motor in the frame of a representative American motorcycle is clearly shown at Fig. 394. A standard single cylinder machine equipped with a two-speed gear is shown at Fig. 395. A model "de luxe" with electric starting and lighting system, two-speed gear, spring frame and spring fork and with complete equipment that may be really considered the most refined model of American motorcycles is shown at Fig. 396. The motorcycle has been gradually improved,


Horn Switch
Speedometer

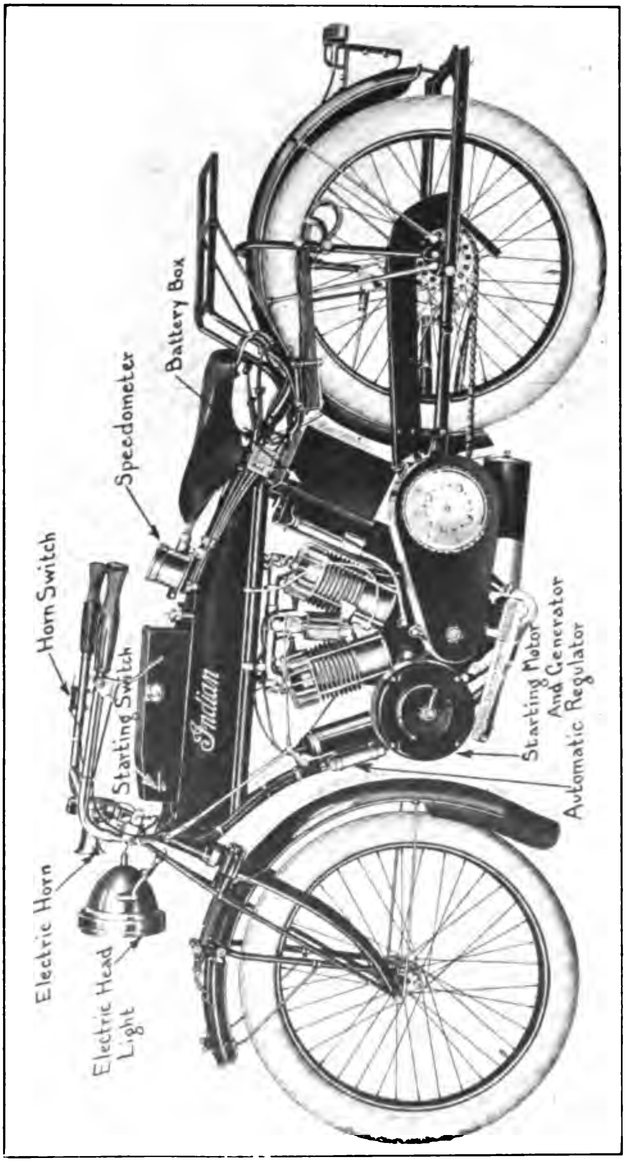


FIG. 396.—View of the Hendee special model Indian motorcycle, which is sold with complete equipment, including electric lighting and starting system.

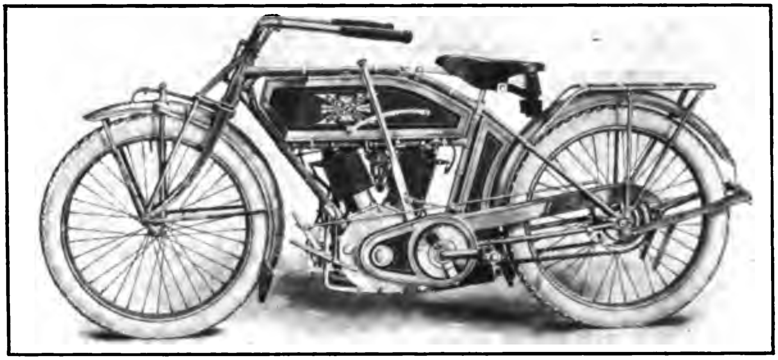


FIG. 397.—The Excelsior Model 7 C. A two-speed, high-powered modern motorcycle.

until at the present time, the only resemblance to a bicycle is evidenced by the use of two wheels, a saddle which the rider straddles, and a pair of handle bars for control of the fork member in which the front wheel is mounted. In actual relation of parts and refinement of detail the motorcycle is practically a two-wheeled

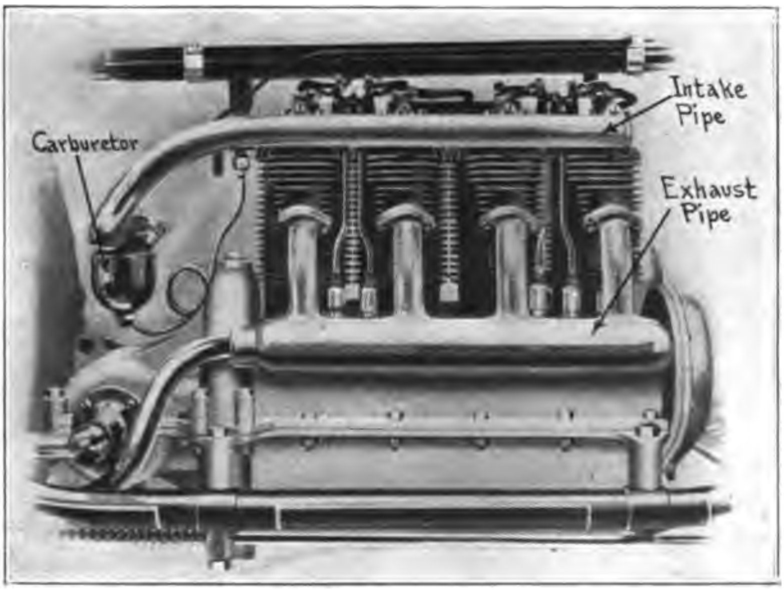


FIG. 398.—Valve side of the Henderson four-cylinder motorcycle power plant.

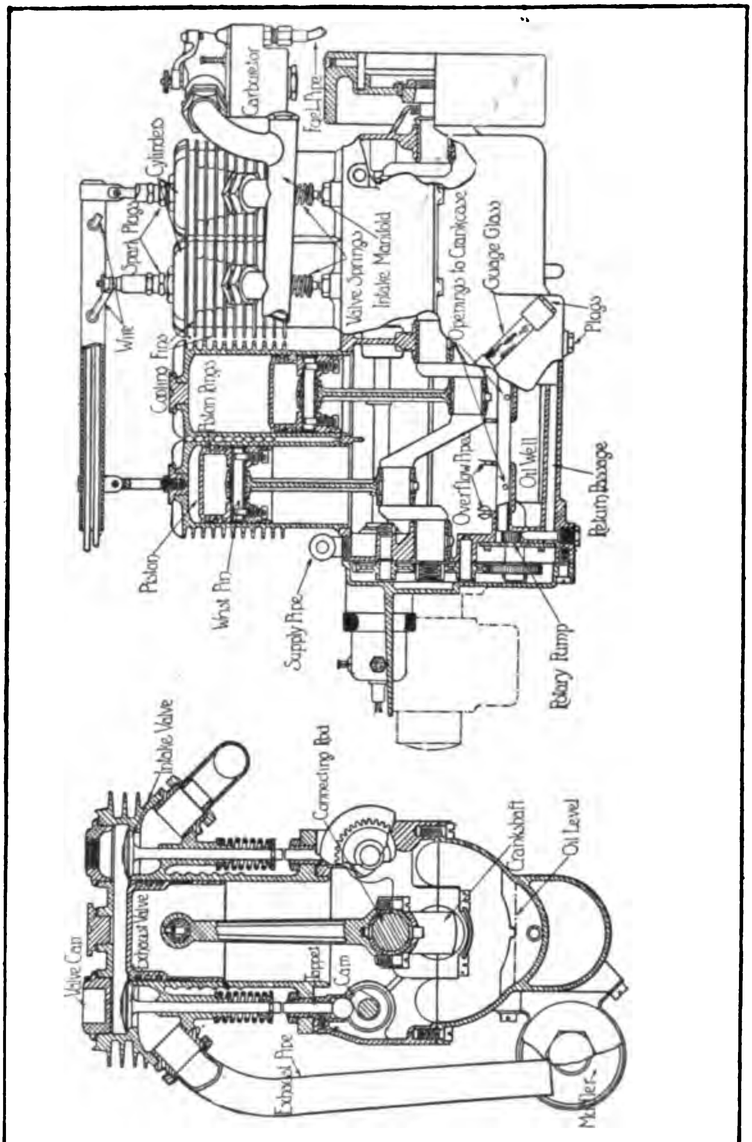


FIG. 399.—Sectional view of Pierce four-cylinder motorcycle power plant.

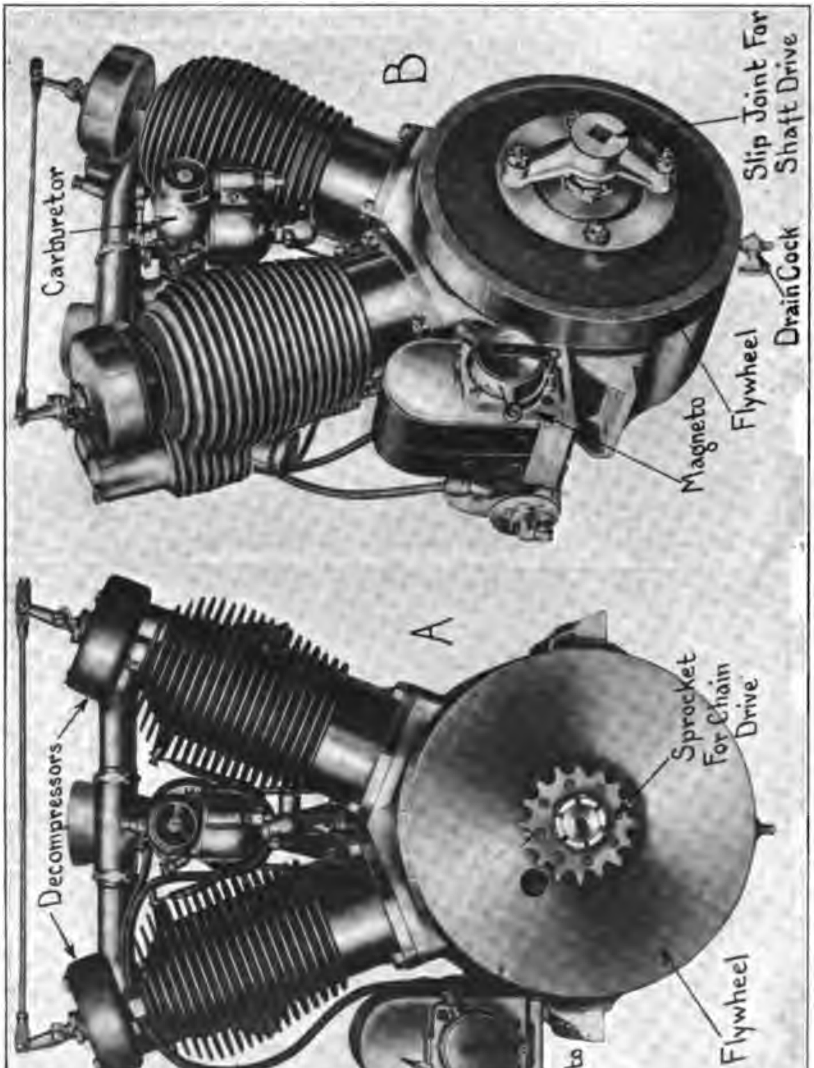


FIG. 400.—Views showing Spacke de Luxe cyclecar power plant.

automobile, inasmuch as the modern forms are provided with supporting springs for both power plant and rider and use free engine clutches, two speeds or other variable speed gears and have a margin of power actually superior to the automobile, in proportion to its weight. A popular twin cylinder motor cycle which incorporates a two-speed planetary gear-set on the countershaft and chain-drive to the rear wheels is shown at Fig. 397.

The good features of the four-cylinder engine, which make that form of such striking value in automobile applications has also resulted in favorable consideration by some motorcycle designers. While the power-producing capacity of the average twin cylinder engine is equal to all practical requirements, it has an element of

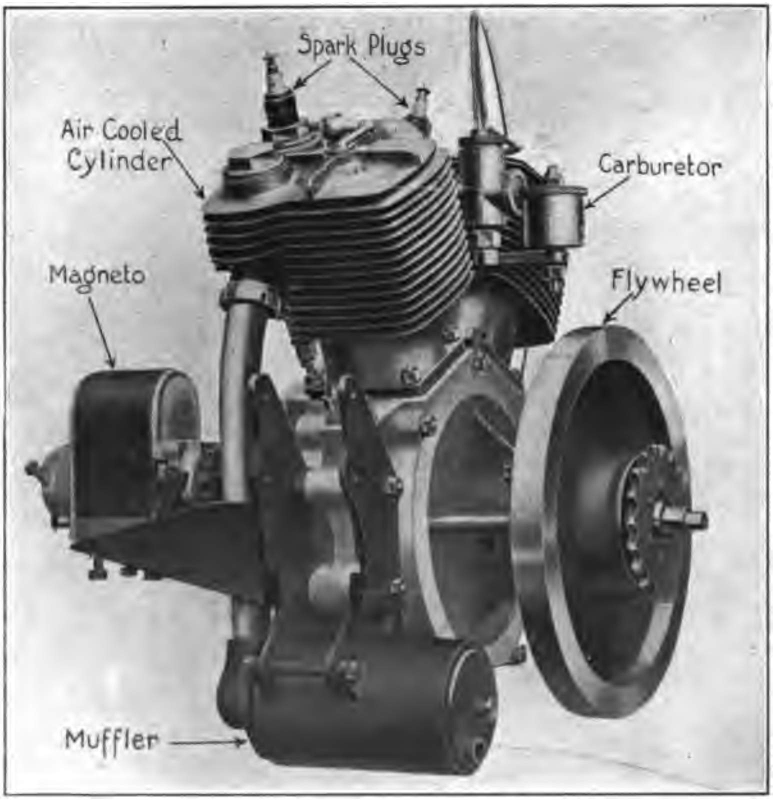


FIG. 401.—The Precision air-cooled cyclecar engine.

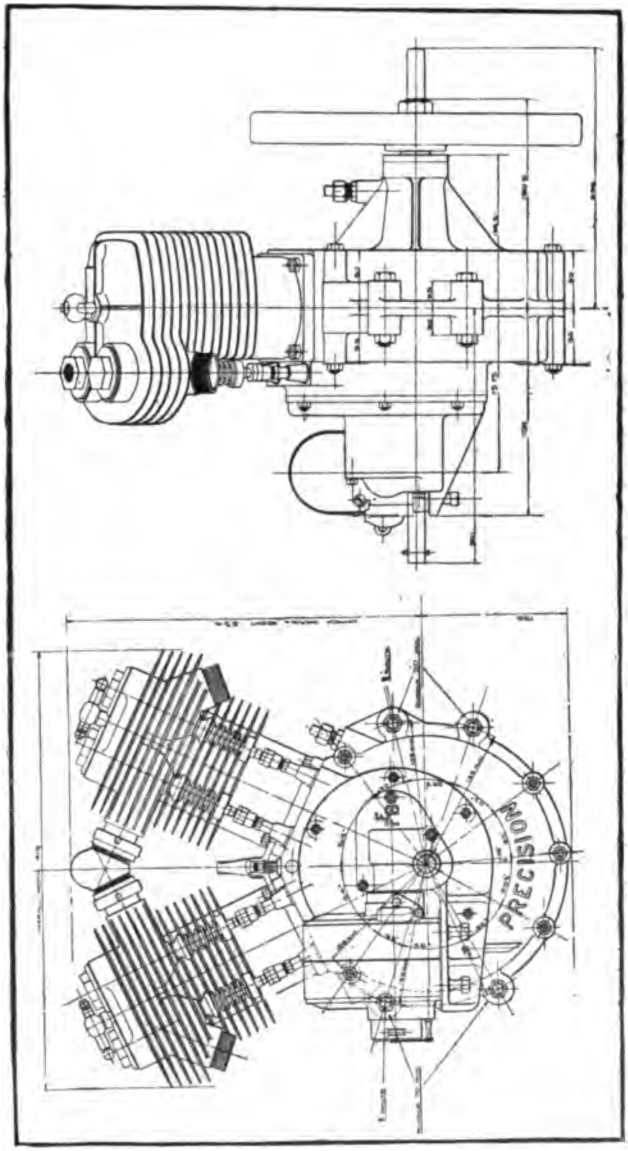


FIG. 402.—Diagrams showing principal dimensions of Precision air-cooled cyclecar power plant.

vibration that is considered objectionable by the fastidious. A properly constructed four-cylinder power plant, especially when it has the small cylinders and relatively light explosions produced in a motorcycle engine, is so nearly vibrationless that it will remove every objection that can be offered against the motorcycle. As will be evident by inspection of Fig. 398, which shows the four-cylinder engine of the Henderson motorcycle and sectional views at Fig. 399 which outline clearly the construction of the four-cylinder power plant used on the Pierce motorcycles, the general features of construction of motorcycle engines of this pattern follow closely the standard features of design so well exemplified in automobile motors.

There has been considerable interest of late in the development of a new form of motor vehicle which is a hybrid between the conventional forms of automobiles and the motorcycle and is called the "cyclecar." In these vehicles many of the parts used on

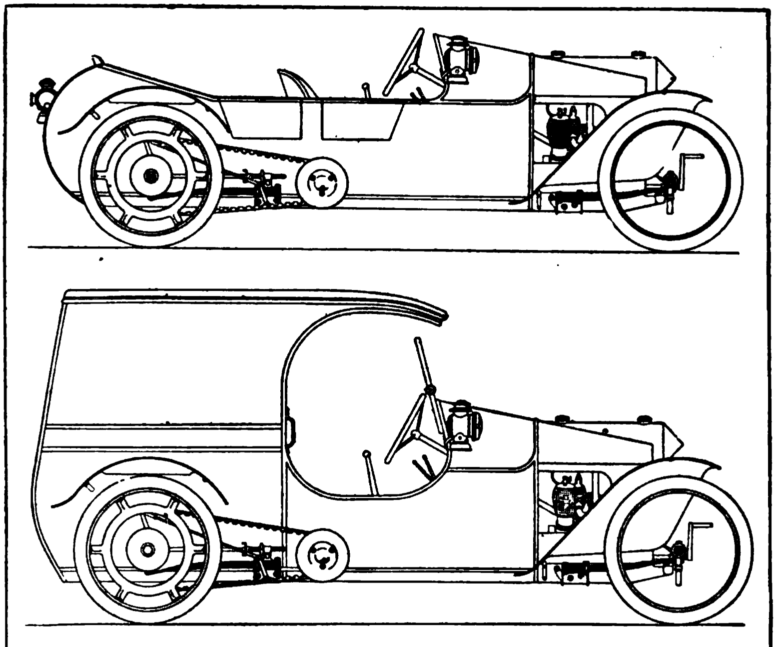


FIG. 403.—The newest form of motor vehicle, known as the cyclecar in pleasure and commercial forms.

motorcycles, such as the power plant, wheels, tires, transmission systems, etc., are retained and a miniature automobile body with conventional spring suspension is provided from the passengers. Cyclecars for the most part are narrow tread, ranging from 36 to 42 inches, whereas the conventional or standard tread of automobiles and horse-drawn vehicles is 56 inches. Cyclecars are built low, and are not much heavier than a motorcycle with side car attachment and provide remarkable speed with maximum economy, as relates to fuel and oil consumption.

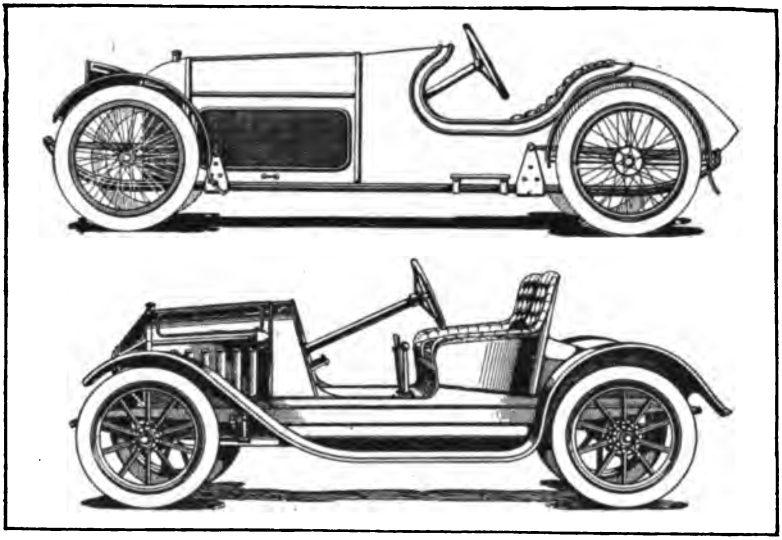


FIG. 404.—Cyclecars of American design following current motorcar practice.

A typical cyclecar engine, which is really a modified form of the Spacke motorcycle power plant, is shown at Fig. 400. This differs from the motorcycle engine in that an external flywheel is provided in addition to the usual flywheel assembly inside the engine base and a cooling fan is mounted at the front of the engine to direct an air-blast around the cylinders. In the engine shown at *A* a sprocket is provided for chain-drive and the engine is supposed to be installed with its crankshaft at right angles to the frame side member. The form shown at B is intended for shaft drive and is installed so the crankshaft is parallel to the side member.

A cyclecar engine of the twin cylinder form and European design is shown at Figs. 401 and 402. In this power plant the cylinders are of the L head form and several modifications are made to make the engine suitable for propulsion of the four-wheel vehicle. The rear bearing is long and is of ample size to take the strain of driving and the weight of the extra flywheel. In this engine the bore is 85 millimeters or 3.34 inches and the stroke is as long as the piston is wide. The cubic capacity is 965 cubic centimeters or 58.8 cubic inches. It will deliver 8 H.P. at its normal speed. The overall dimensions of this power plant can be readily ascertained by referring to Fig. 402, bearing in mind that the dimensions indicated are according to the metric standard. The general appearance of the power plant and the location of the auxiliary fittings such as the spark plugs, carburetor, magneto, muffler, etc., can be readily ascertained from the three-quarter view at Fig. 401.

Representative types of the light cyclecars are shown at Figs. 403 and 404. The one at the top of the illustration Fig. 403 is a two-passenger tandem model of the true cyclecar form, while that below it shows the application of the commercial body to the same chassis. The location of the power plant of the motorcycle type, at the front of the vehicle, the light wire spoke wheels and the V belt drive so generally used on motorcycles may be readily ascertained. Such vehicles do not weigh over 600 lbs. and will carry two people 50 miles on a gallon of gasoline. Modifications of the cyclecar design that favor more conventional automobile forms are outlined at Fig. 404. These are supplied with motors of the four-cylinder water-cooled type and are virtually miniature automobiles and cannot be termed true cyclecars. The drive is by shaft and bevel gear, as in current automobile practice and for the most part the speed-changing and clutching functions are performed by devices that are the same as those used in automobiles except for the reduction in weight and size.

CHAPTER XX

INTERNAL COMBUSTION MOTORS FOR AERIAL NAVIGATION

THE conquest of the air is one of the most stupendous achievements of the ages. Human flight opens the sky to man as a new road, and because it is a road free of all obstructions and leads everywhere, affording the shortest distance to any place, it offers to man the prospect of unlimited freedom. The aircraft promises to span continents like railroads, to bridge seas like ships, to go over mountains and forests like birds, and to quicken and simplify the problems of transportation. While the actual conquest of the air is an accomplishment just being realized in our days, the idea and yearning to conquer the air are old, possibly as old as intellect itself. The myths of different races tell of winged gods and flying men, and show that for ages to fly was the highest conception of the sublime. No other agent is more responsible for sustained flight than the internal combustion motor, and it was only when this form of prime mover had been fully developed that it was possible for man to leave the ground and alight at will, not depending upon the caprices of the winds or lifting power of gases as with the balloon. It is safe to say that the solution of the problem of flight would have been attained many years ago if the proper source of power had been available as all the essential elements of the modern aeroplane and dirigible balloon, other than the power plant, were known to early philosophers and scientists,

Aeronautics is divided into two fundamentally different branches—aviatics and aerostatics. The first comprises all types of aeroplanes and heavier than air flying machines such as the helicopters, kites, etc.; the second includes dirigible balloons, passive balloons and all craft which rise in the air by utilizing the lifting force of gases. Aeroplanes are the only practical form of heavier-than-air machines, as the helicopters (machines intended to be lifted directly into the air by propellers, without the sustain-

FIG. 405.—The Blériot monoplane, a typical single surface machine.

ing effect of planes), and ornithopters, or flapping wing types, have not been thoroughly developed, and in fact, there are so many serious mechanical problems to be solved before either of these types of air craft will function properly that experts express grave doubts regarding the practicability of either. Aeroplanes are divided into two main types—monoplanes or single surface forms, and bi-planes or machines having two sets of lifting surfaces, one suspended over the other. A typical monoplane is shown at Fig. 405, a bi-plane at Fig. 406. The structural differences may be easily understood by comparing the illustrations.

Dirigible balloons are divided into three classes; the rigid, the semi-rigid, and the non-rigid. The rigid has a frame or skeleton

FIG. 406.—The Wright headless biplane, a typical two-surface type.

of either wood or metal inside of the bag, to stiffen it; the semi-rigid is reinforced by a wire net and metal attachments; while the non-rigid is just a bag filled with gas. The aeroplane, more than the dirigible and balloon, stands as the emblem of the conquest of the air. Two reasons for this are that power flight is a real conquest of the air, a real victory over the battling elements; secondly, because the aeroplane, or any flying machine that may follow, brings air travel within the reach of everybody. In practical development, the dirigible will be the steamship of the air, which will render invaluable services of a certain kind, and the aeroplane will be the automobile of the air, to be used by the multitude, perhaps for as many purposes as the automobile is now being used.

ESSENTIAL REQUIREMENTS OF AERIAL MOTORS.

One of the marked features of aircraft development has been the effect it has had upon the refinement and perfection of the internal combustion motor. Without question gasoline-motors intended for aircraft are the nearest to perfection of any other type yet evolved. Because of the peculiar demands imposed upon the aeronautical motor it must possess all the features of reliability, economy and efficiency now present with automobile or marine engines and then must have distinctive points of its own. Owing to the unstable nature of the medium through which it is operated and the fact that heavier-than-air machines can maintain flight only as long as the power plant is functioning properly, an airship motor must be more reliable than any used on either land or water. While a few pounds of metal more or less makes practically no difference in a marine motor and has very little effect upon the speed or hill-climbing ability of an automobile, an airship motor must be as light as it is possible to make it because every pound counts, whether the motor is to be fitted into an aeroplane or in a dirigible balloon.

Airship motors, as a rule, must operate constantly at high speeds in order to obtain a maximum power delivery with a minimum piston displacement. In automobiles, or motorboats, motors are not required to run constantly at their maximum speed. Most aircraft motors must function for extended periods at speed as nearly the maximum possible. Another thing that militates against the aircraft motor is the more or less unsteady foundation to which it is attached. The necessarily light frame work of the aeroplane makes it hard for a motor to perform at maximum efficiency on account of the vibration of its foundation while the craft is in flight. Marine and motorcar engines, while not placed on foundations as firm as those provided for stationary power plants, are installed on bases of much more stability than the light structure of an aeroplane. The aircraft motor, therefore, must be balanced to a nicety and must run steadily under the most unfavorable conditions.

Some of the problems that must be solved by designers of practical aeroplane motors are well summed up in an article by

A. S. Atkinson in a recent number of the "Gas Review," from which the following excerpts are taken:

"The effect of altitude on the motor is one of the little adjustment difficulties that the designers of aeroplanes must consider carefully. The difference in the power developed at sea level and that developed at an ordinary high level may not be very startling, but when the change from a low to a very high level takes place in a short period the effect is striking. The reason for this is that the air becomes more rarefied as we ascend, and the atmospheric pressure is approximately one-half pound for each one thousand feet of altitude, and as the pressure at sea level is 14.7 pounds per square inch, the decrease amounts to about three or four per cent. of this. At an altitude of three thousand feet the decrease would be about ten per cent., and the power developed by an engine at this altitude would be only about ninety per cent. of that developed at sea level.

On this account of change of altitude the builders of motors for airships have to consider the relative effect of a decreased efficiency of the motor on the flying abilities of the machine. A machine reaching an altitude of six thousand feet would thus have its motor efficiency decreased to eighty per cent. of its power development at sea level. At an altitude of five or six thousand feet, the airship plunges into extremely cold air, and while below the mildest summer may prevail, up above the clouds cold, freezing weather makes life miserable for the aviator. This cold has its effect upon the motor, for sudden contraction and expansion of metals subjects all friction parts to severe strains. It may be said that aviators rarely stay long at such high altitudes, but an airship must be constructed so that it can make its way indefinitely through such a cold, rarefied atmosphere, and the builders of engines must consider this in their work. The sharp change from cold to heat and from heat to cold has had the effect time and again of putting motors temporarily out of business. A good many of the accidents in the air have not been entirely and satisfactorily accounted for. All that is known is that the engines stopped, and the aviator was forced to volplane to the earth without the use of any other power than gravitation.

All the tests and experiments with aircraft motors have an influence upon the makers of all other kinds of engines. Manufacturers of marine engines for special speed boats have taken lessons from them in the matter of reducing superfluous weight and increasing the number of revolutions. There is little question but the modern hydroplane owes its speed and development to the tests and experiments carried on in the field of aviation and even the modern high-powered class of motorboats have been improved by their engine designers through a study of aviation motors."

The capacity of light motors designed for aerial work per unit of mass is surprising to those not fully conversant with the possibilities that a thorough knowledge of proportions of parts and the use of special metals developed by the automobile industry make possible. Activity in the development of light motors has been more pronounced in France than in any other country. Some of these motors have been complicated types made light by the skillful proportioning of parts, others are of the refined simpler form modified from current automobile practice. There is a tendency to depart from the freakish or unconventional construction and to adhere more closely to standard forms because it is necessary to have the parts of such size that every quality making for reliability, efficiency and endurance are incorporated in the design. Aeroplane motors range from two cylinders to forms having fourteen and sixteen cylinders and the arrangement of these members varies from the conventional vertical tandem and opposed placing to the V form or the more unusual radial motors having either fixed or rotary cylinders. The weight has been reduced so it is possible to obtain a complete power plant that will not weigh more than three pounds per actual horse-power.

If we give brief consideration to the requirements of the aviator it will be evident that one of the most important is securing maximum power with minimum mass, and it is desirable to conserve all of the good qualities existing in standard automobile motors. These are certainty of operation, good mechanical balance and uniform delivery of power — fundamental conditions which must be attained before a power plant can be considered practical. There are in addition, secondary considerations, none the less desirable, if not absolutely essential. These are minimum consumption of fuel and lubricating oil, which is really a factor of import, for upon the economy depends the capacity and flying

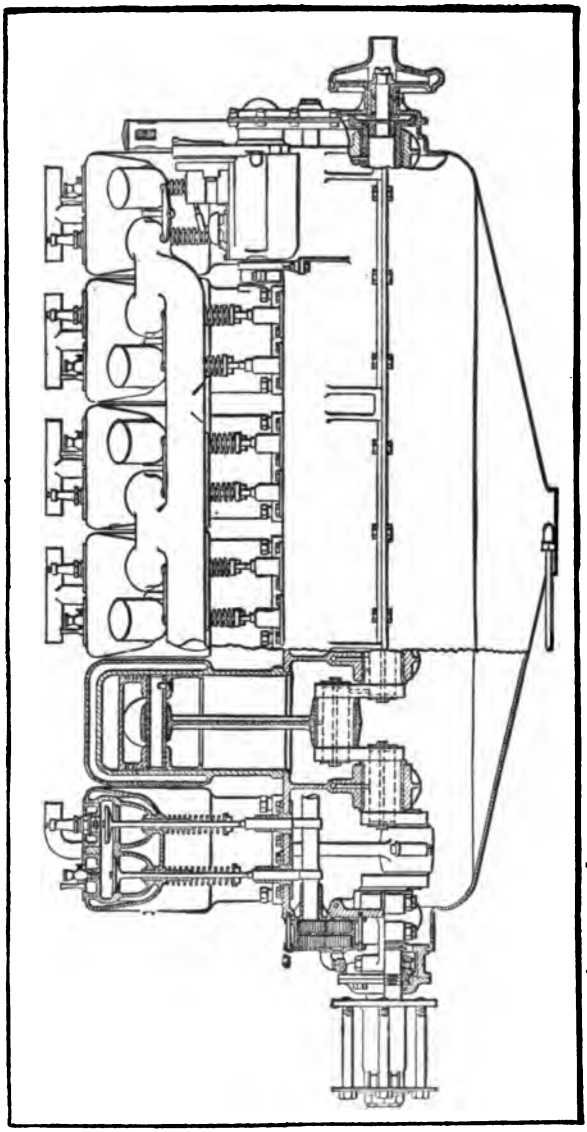


FIG. 407.—Partial section view of the Sturtevant six-cylinder aircraft motor.

radius. As the amount of liquid fuel must be limited the most suitable motor will be that which is powerful and at the same time economical. Another important feature is to secure accessibility of components in order to make easy repair or adjustment of parts possible. It is possible to obtain sufficiently light weight motors without radical departure from established practice. Water cooled motors have been designed that will weigh but five pounds per horse-power and in these forms we have a practical power plant capable of extended operation.

A motor of the four-cycle type having six cylinders in which light weight has been obtained by skillful designing and the use of high tensile strength materials instead of unconventional features is shown at Fig. 407. This is the Sturtevant and is an American design. The six-cylinder form just depicted is rated at 60 H.P. and a four-cylinder form is made rated at 40 H.P. The object in designing this motor was to build one that would be absolutely reliable and one upon which dependence would be placed when making long flights. It is contended that this form of cylinder placing is superior to either the V or radial form because the head resistance is considerably lessened with the tandem cylinder construction. A disadvantage, of course, is that the engine is so long that it will take much more room lengthwise than the V engine or the rotary form of equivalent capacity. The bore of the cylinder and stroke are each 4½ inches, which is somewhat different than that usually prevailing, as in most cases the stroke is longer than the bore is wide. The crankshaft is made of high tensile strength alloy steel and in order to reduce the weight to a minimum the main journals and the crankpins are drilled out. The cylinders are of the L type, having cast iron water jackets formed integrally and with both valves in the same cylinder head extension. This makes for compactness in design and permits the operation of all valves from a common camshaft.

The pistons are cast of semi-steel mixture and are finished by grinding. The piston rings are of cast iron and three are used on each piston. The wrist pins are made of 3½ per cent. nickel steel tubing in order to reduce their weight to a minimum. The inlet and exhaust valves are made of 30 per cent. nickel steel and are interchangeable. The crank-case is of special aluminum mixture that possesses great strength and that is exceptionally light. The connecting rods are made from drop forgings of 3½ per cent. nickel steel and are of the usual I section. The end of the crankshaft which receives the propeller is slightly tapered and carries a flange having six attaching bolts and another flange which slips over the castellated end and between which flanges the propeller is retained in a positive manner.

The lubrication system has been carefully developed with a view of securing one that would be absolutely dependable. By means of a pressure pump driven directly from the end of the camshaft the oil is forced through cored holes in the base and into the hollow crankshaft already described. It is thus distributed to all of the main bearings supporting the crankshaft and also to the connecting rod bearings. The coil thrown off from the rotating connecting rod big ends serves to lubricate the cylinder and the remainder of the interior mechanism. The system is of the constant supply type, as the surplus oil is collected in the pump forming part of the lower portion of the horizontally divided crankcase and is used over and over again while the motor is in operation.

Means are provided for starting the motor in mid-air and this is an advantage of some moment as it will enable an aviator to shut off his motor when he wishes to make a flight under the influence of the attraction of gravity, which is called "volplaning," and yet be able to start it at will without landing. A lifting rod is provided which actuates the exhaust cams and makes it possible to lift the exhaust valves so as to relieve the compression should the motor stop while in flight. With the exhaust valves lifted, the action of the air on the propeller will cause the motor to start atomatically, as the propeller is able to turn over the end of the crankshaft when the resistance to rotation normally present uue to the compression is released.

Successful flights have been made with two-cycle motors, and while they are most popular in marine service there is no question regarding the ability of a two-cycle motor to furnish power enough for flying. The motor shown at Fig. 408 is a light two-cycle Roberts design which will develop 40 H.P. at 1000 R.P.M. It is capable of an extreme speed of 1600 R.P.M. It has four cylinders 4½-inch bore and 5-inch stroke and weighs complete with magneto and carburetor but 195 lbs. It is claimed that neither strength or ability to stand hard service have been sacrificed to any degree; Chrome nickel and Vanadium steel have made possible extremely strong and light connecting rods and crankshaft while magnalium, an extremely light alloy similar to aluminum, has been used in making the engine base. The crank case is designed for high compression and a special mechanically driven rotary intake

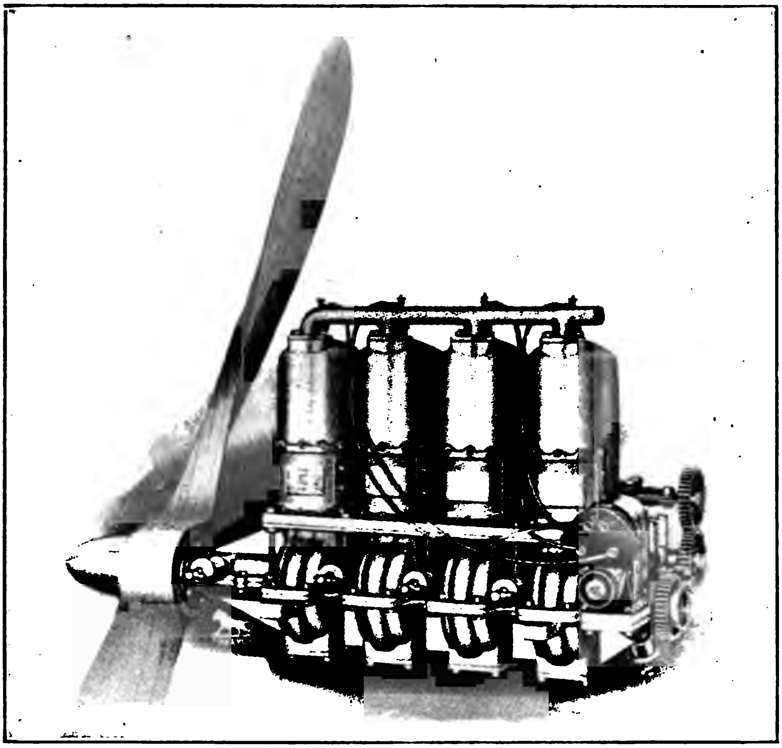


FIG. 408.—The Roberts two-cycle aeronautical motor.

distributor insures prompt charging with gas. The two-cycle engine shown gives the same even turning movement that would be obtained with eight four-cycle cylinders. It is also furnished in the six-cylinder form rated at 60 H.P. and a larger six-cylinder form may be obtained that will deliver 125 H.P. The cylinders in the large motor are 5½-inch bore and 6-inch stroke.

While many methods of increasing the mechanical efficiency of a motor are known to designers, one of the first to be applied to the construction of aeronautical power plants was an endeavor to group the components, which in themselves were not extremely light, into a form that would be considerably lighter than the conventional design. As an example, we may consider those multiple cylinder forms in which the cylinders are disposed around

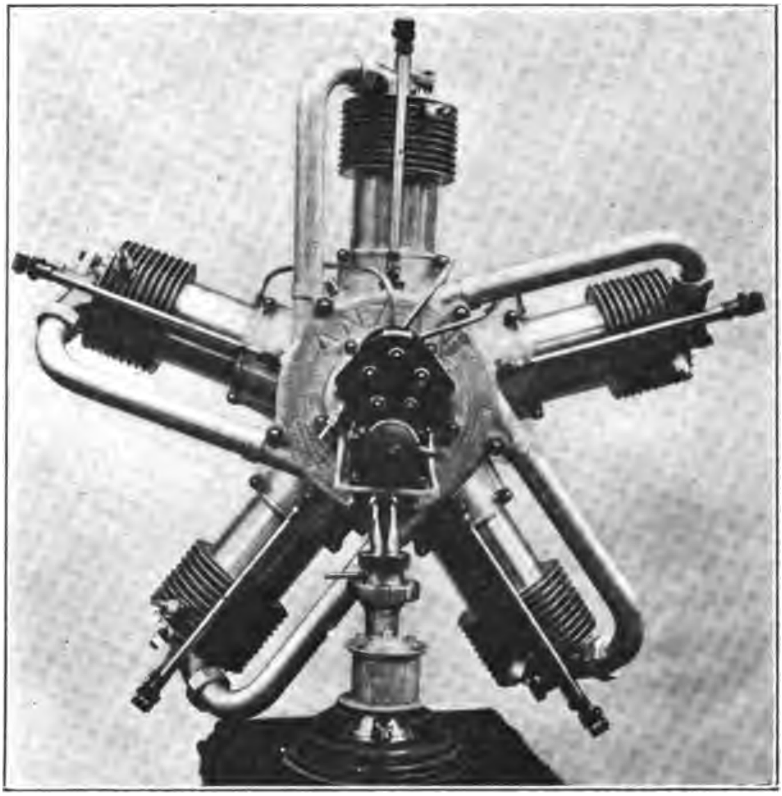


FIG. 409.—Anzani 40-50 horse-power five-cylinder engine.

a short crankcase, either radiating from a common center as at Fig. 409 or of the fan shape shown at Fig. 410. This makes it possible to use a crankcase but slightly larger than that needed for one or two cylinders and it also permits of a corresponding decrease in length of the crankshaft. The weight of the engine is lessened because of the reduction in crankshaft and crankcase weight and the elimination of a number of intermediate bearings and their supporting webs which would be necessary with the usual tandem construction. The revolving form, which is to be described more fully in proper sequence, has a number of advantages, as the cylinders and crankcase rotate about a stationary crankshaft. This makes it possible to lighten the power plant because the revolving cylinders act as their own fly-wheel and make possible adequate cooling without auxiliary apparatus.

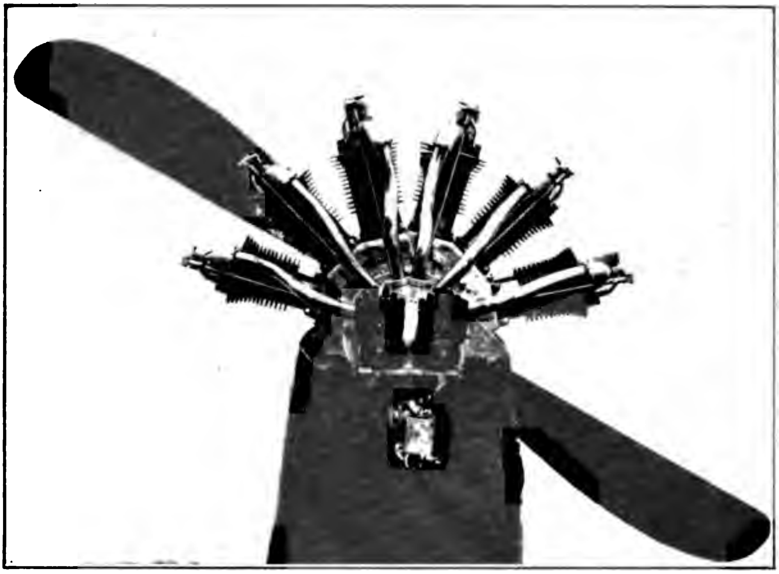


FIG. 410.—Unconventional six-cylinder aircraft motor of Masson design.

FIXED RADIAL CYLINDER ENGINES

In the Anzani form, which is shown at Fig. 409, the crankcase is stationary and a revolving crankshaft is employed as in conventional construction. The cylinders are five in number and the engine develops 40 to 50 H. P. with a weight of 72 kilograms or 158.4 lbs. The cylinders are of the usual air cooled form having cooling flanges only part of the way down the cylinder. The valves are placed directly in the cylinder head and are operated by a common pushrod. Attention is directed to the novel method of installing the carburetor which supplies the mixture to the engine base from which inlet pipes radiate to the various cylinders.

In the form shown at Fig. 410 six cylinders are used, all being placed above the crankshaft center line. This engine is also of the air cooled form and develops 50 H. P. and weighs 105 kilograms or 231 lbs. Two separate magnetos are provided, one of which may be used in event of emergency if the regular ignition device fails. The carburetor is connected to a manifold casting attached

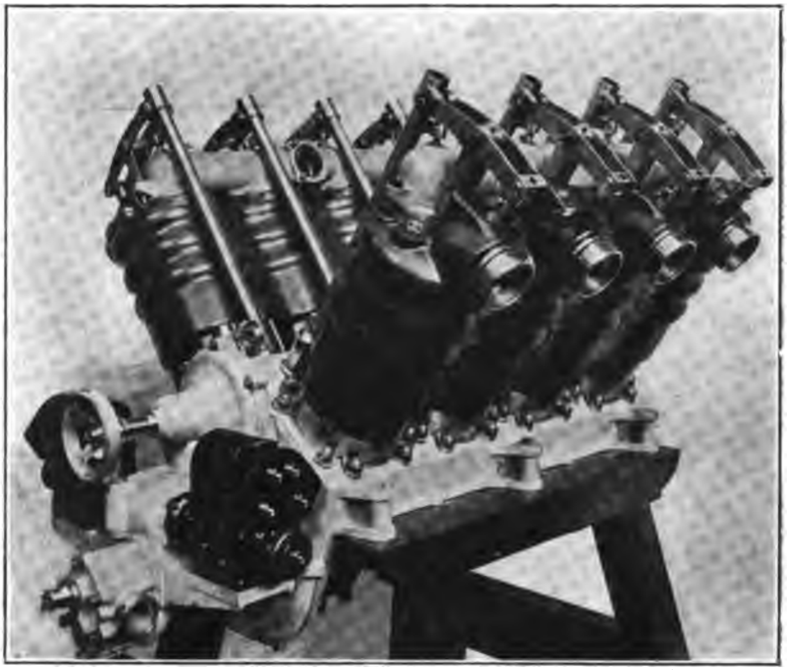


FIG. 411.—The eight-cylinder, 200 horse-power Clerget motor.

to the engine base from which the induction pipes radiate to the various cylinders. The propeller design and size relative to the engine is clearly shown in this view. While flights have been made with both of the engines described, this method of construction is not generally followed and has been almost entirely displaced abroad by the revolving motors or by the more conventional eight cylinder V engines of which that shown at Fig. 411 may be considered a good example.

V TYPE AVIATION MOTORS

This power plant, which is known as the "Clerget," is intended for dirigible balloon propulsion rather than aeroplane use. It is rated at 200 H. P. and weighs 180 kilograms or 396 pounds. Among the novelties of construction in this motor may be mentioned the copper water jackets which are applied by a process of electro-deposition and the two-function valve operating rod. The rod operating the exhaust valve rocker arm has an extension that will

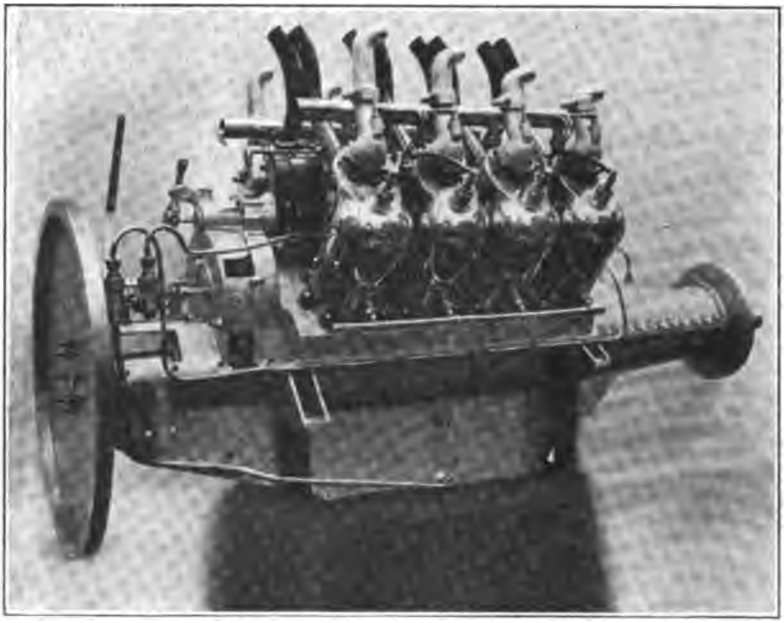


FIG. 412.—The Antoinette eight-cylinder aeroplane engine.

depress the inlet valve when the rod descends. A double profile cam is necessary with this system as when the push rod rides on the profile it depresses the exhaust valve and when it falls into the hollow or depression of the cam it will press the inlet valve down. In this engine two separate magnetos are provided, one for each set of four cylinders.

An 8-cylinder water-cooled motor which has made a number of practical flights when fitted to the Antoinette monoplane is shown at Fig. 412. This motor is rated at 50 H. P. and for a time was one of the most popular of the foreign water cooled forms. One of the striking features of this motor is the use of steel cylinders, which are machined inside and out from drop forgings owing to the difficulty which would obtain of securing steel castings that would be reliable and have thin enough walls. The water jackets are of pure copper, electrically deposited to the steel cylinder forgings. This makes possible jacket walls of excellent heat

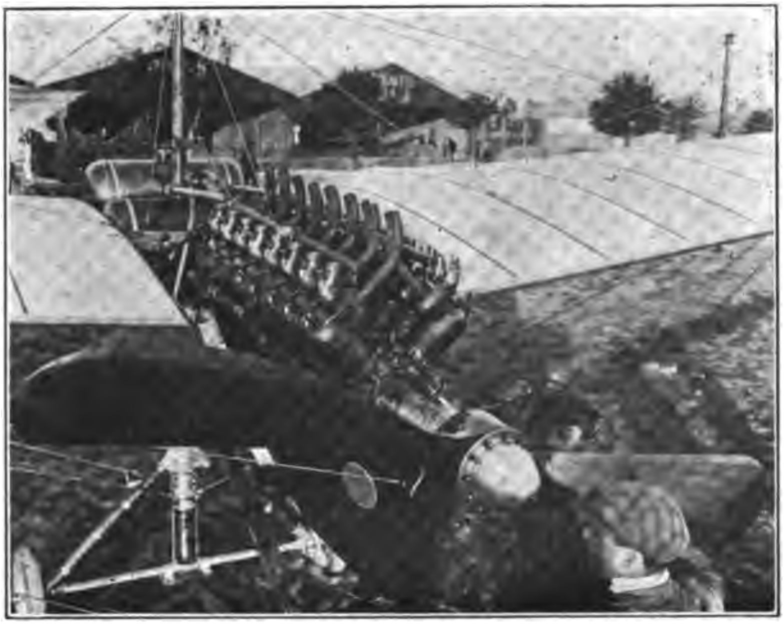


FIG. 413.—Showing method of installation of the Antoinette sixteen-cylinder motor in aeroplane framework.

radiating capacity, of extremely light construction and also insures uniformity of water spaces on each cylinder.

The process of electro deposition, while an expensive one, is not one that offers any great difficulty in manufacturing. After the cylinder is properly machined it is built out with wax to conform to the shape of the copper water jacket desired. This wax has been coated with a conducting film, usually some graphite compound and is placed in the usual form of plating bath. After the copper has been deposited on the wax to the proper thickness, this soft material is melted out by the application of gentle heat and the space that was formerly occupied by the wax becomes the space through which the cooling water circulates.

The cylinders are of the L form and have the inlet valves placed above the exhaust. The inlet valves are automatic, the exhaust are mechanically actuated from a single camshaft, located in the top of the engine base between the cylinders. The exhaust pipes extend upward between the cylinders. The inlet pipes are short vertical elbows to each of which is attached a small copper

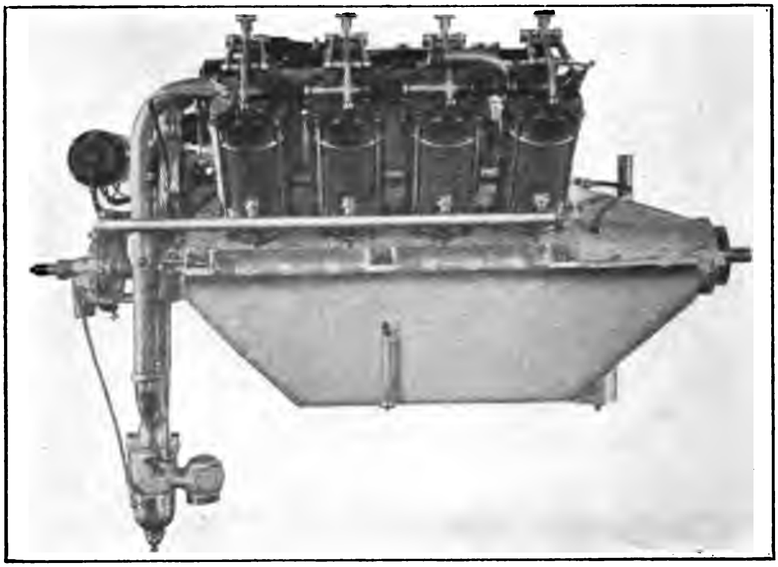


Fig. 414.—Side view of the Curtiss eight-cylinder motor.

tube leading to either one of two plunger pumps that inject the fuel directly into the valve dome. These pumps are actuated by variable throw eccentrics which change the stroke of the pumps to vary the quantities of fuel supplied the engine. It was believed that this method of fuel supply would reduce the liability of carburetion troubles. The motor shown, complete with propeller and condenser or water cooler, weighs about 300 lbs. The Antoinette motor is also made in the 16 cylinder form, which is rated at 100 H. P. This engine complete weighs less than 600 lbs. and is practically two of the eight cylinder units attached together.

The method of installation of this large power plant in the framework of an Antoinette monoplane is clearly outlined at Fig. 413.

One of the most efficient of American aeroplane motors is that designed by Glenn Curtiss and used by him and other aviators in many record breaking flights. This is an eight cylinder V form and is shown at Fig. 414. It is claimed that this motor has an ample margin of power and will fly any standard type of two passenger aeroplane or flying boat. Tests made by the U. S. Navy aviation officers show that with the regular propeller supplied with the motor, a flying boat of the form shown at Fig. 416 with a 600 pound load will fly satisfactorily at 930 R. P. M., which leaves a reserve of 420 R.P.M. available for emergency. The approximate maximum speed of the engine is 1,350 R. P. M. This gives the flying boat a speed range of from 38 to 65 miles per hour. The installation of one of the earliest types of Curtiss engines between the planes of a Curtiss biplane is shown at Fig. 415. This motor has a bore of 4″ and a stroke of 5″. It weighs 320 lbs. and will develop 90 to 100 H.P. It has a speed rating of 250 to 1,500 R.P.M. and the regular propeller will deliver a static thrust of about 600 lbs. It consumes eight gallons of gasoline per hour and a half gallon of oil during the same period.

THE CURTISS FLYING MOTORBOAT

Aviation when first heralded a few years ago opened to imaginative minds visions of man darting through space with the speed of birds, alighting at will on any convenient tree top or mansard roof. After a few accidents there was a great hue and cry for machines of automatic stability, — a demand for aeroplanes that would hang inert in the air with motors stopped, etc., etc. Thousands of men are puzzling over these problems to-day, and have been at it for years.

Before the general public knew anything definite about aviation Mr. Curtiss had been working at its problems and he decided early that it was more practical to have an aeroplane that could fly low, and if possible have something more yielding than the bosom of Mother Earth to alight upon than to attempt to evolve structures to evade natural laws. It was not difficult to arrive at the conclusion that the water highways of the world offered just the courses needed for safe aviation, but it required some years of patient experimenting before he perfected and offered to the world the Curtiss flying motorboat, one of the modern forms of which is shown in flight in Fig. 416.

This machine and others embodying similar principles are destined to take much of the risk from aviation and will do much to further the cause of aerial navigation by popularizing it with sportsmen and others able to enjoy this combination of water and air travel. It will make aviation of every day use, and air flight

FIG. 415.—Outlining installation of power plant in Curtiss biplane.

commonplace. Only those familiar with the principles of constru-tion employed on these up-to-date machines can realize what radical changes have been made. Take the aeroplane of 1910 as designed by Mr. Curtiss and copied in hundreds of cases during the past two years. That machine weighed with motor from 450 to 500 pounds. The flying motorboat of to-day weighs 1,200 to 1,400 pounds. Loaded, the earlier machine weighed less than 600 pounds, while the flying boat fully loaded weighs more than a ton.

With increased weight has come greater speed, strength and reliability.

The hull of this composite air and water craft is wedge-shaped, broad of beam at the bow, tapering to a point at the stern. It measures about 26 feet over-all, though the net length of the boat is 24 feet. The frame-work of the hull is built-up with the integrity of a cantilever bridge. In the bow the boat is ribbed top, bottom and sides. These rib-frames are of ash, spaced on three-inch centers, mortised at the corners, fitted with copper corner-straps, and each frame securely fastened with sixteen copper rivets.

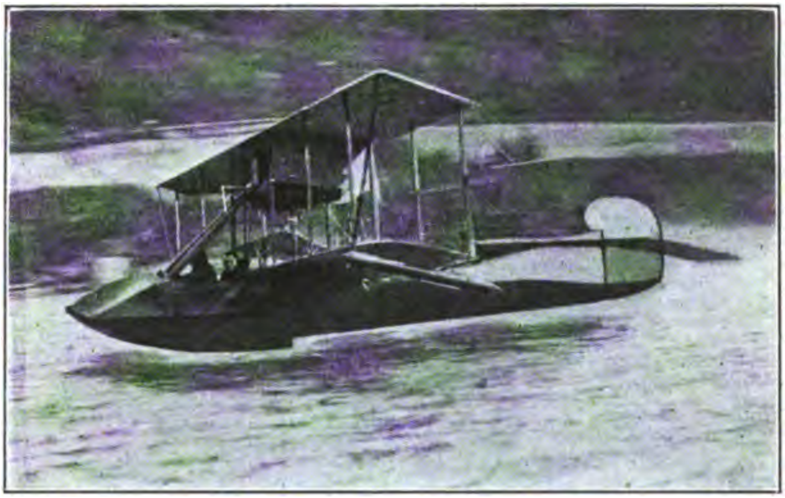


FIG. 416.—The Curtiss flying motorboat leaving the water.

Eight water-tight bulkheads divide the hull laterally so that it is practically unsinkable; for any two of the compartments are sufficient to float the entire machine. The weight of the hull is about 250 pounds, but so strongly is it trussed that if supported at bow and stern by saw-horses it makes a bridge capable of supporting a very heavy load. Longitudinal ribs or battens are spaced on four-inch centers and are of one-inch ash. The bottom is triple sheathed, with, first a planking of 5-16″ mahogany, next a covering of heavy canvas set in marine glue, and an outer planking of ¼″ mahogany, or 16 gauge sheet Duralumin if preferred. This planking is cross laid diagonally and extends back only to the "step" in the hull located about ten feet from the bow.

Stretching across the boat and above the heads of the passengers are the wings. Technically speaking these are termed "planes," but they might with almost equal propriety be called sails, for the flying boat is almost as nearly related to the sailing yacht as to the motorboat. The chief difference between a sail and a wing is that the sail is a passive agent used to gather the energy of active air; while the wing of an aeroplane, to secure sustentation of the craft, must be pushed at high velocity through the air, and thus becomes the active member while the supporting air may be considered as the passive agent.

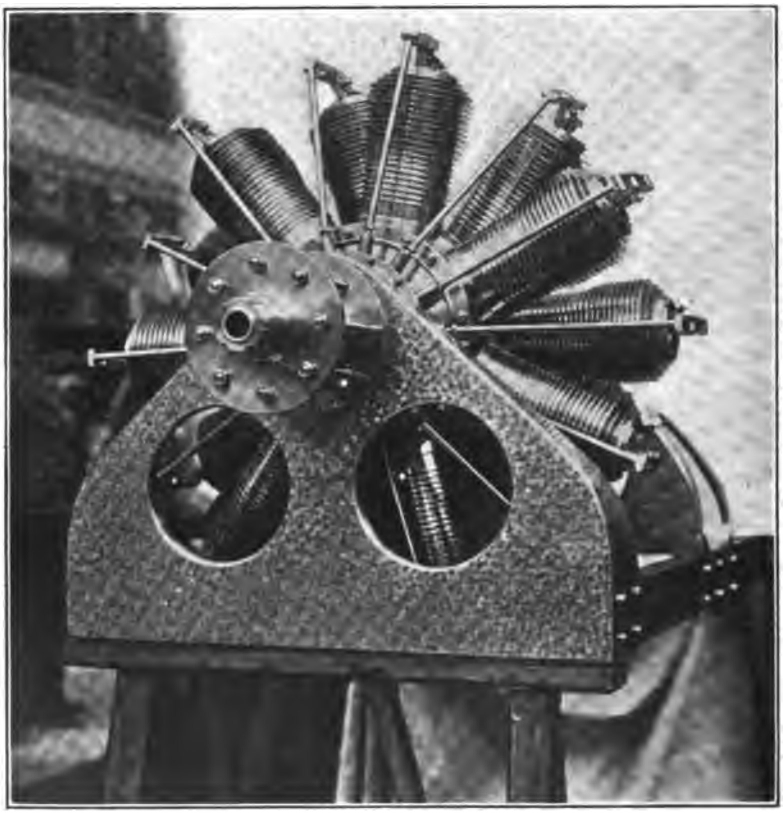


FIG. 417.—The Gnome fourteen-cylinder revolving motor.

Without leaving the water the flying motorboat is capable of a speed of more than fifty miles per hour, or it can be throttled down to a pace slow enough for trolling. Fitted with a special chassis it will travel on land at an equal range of speed, and if run down a beach into the water, the shifting of a lever lifts the wheels up out of the way once the boat is afloat. To the writer's mind it is the first practical solution offered of the problem of safe aerial navigation.

FEATURES OF GNOME MOTOR

It cannot be denied that one of the most widely used of aeroplane motors is the seven-cylinder revolving air-cooled Gnome, made in France. For a total weight of 167 pounds this motor develops 45 to 47 horse-power at 1,000 revolutions, being equal to 3.55 pounds per horse-power, and has proved its reliability by securing many long-distance and endurance records. Quite recently the same engineers have produced a fourteen-cylinder revolving Gnome, having a nominal rating of 100 horse-power, with which world's speed records were broken.

Except in the number of cylinders and a few mechanical details the fourteen-cylinder motor is identical with the seven-cylinder one; fully three-quarters of the parts used by the assemblers would do just as well for one motor as for the other. There is very little in this motor that is common to the standard type of vertical motorcar engine. The fourteen cylinders are mounted radially round a circular crankcase, Figs. 417 and 418; the crankshaft is fixed, and the entire mass of cylinders and crankcase revolves around it. The explosive mixture and the lubricating oil are admitted through the fixed hollow crankshaft, passed into the explosion chamber through an automatic intake valve in the piston head, and the spent gases exhausted through a mechanically-operated valve in the cylinder head. The course of the gases is practically a radial one. A peculiarity of the construction of the motor is that nickel steel is used throughout. Aluminum is employed for the two oil pump housings; the single compression ring for each piston is made of brass; there are three or four brass bushes; gun metal is employed for certain pins — the rest is machined out of chrome nickel steel. The crankcase is practically a steel hoop, the depth depending on whether it has to receive

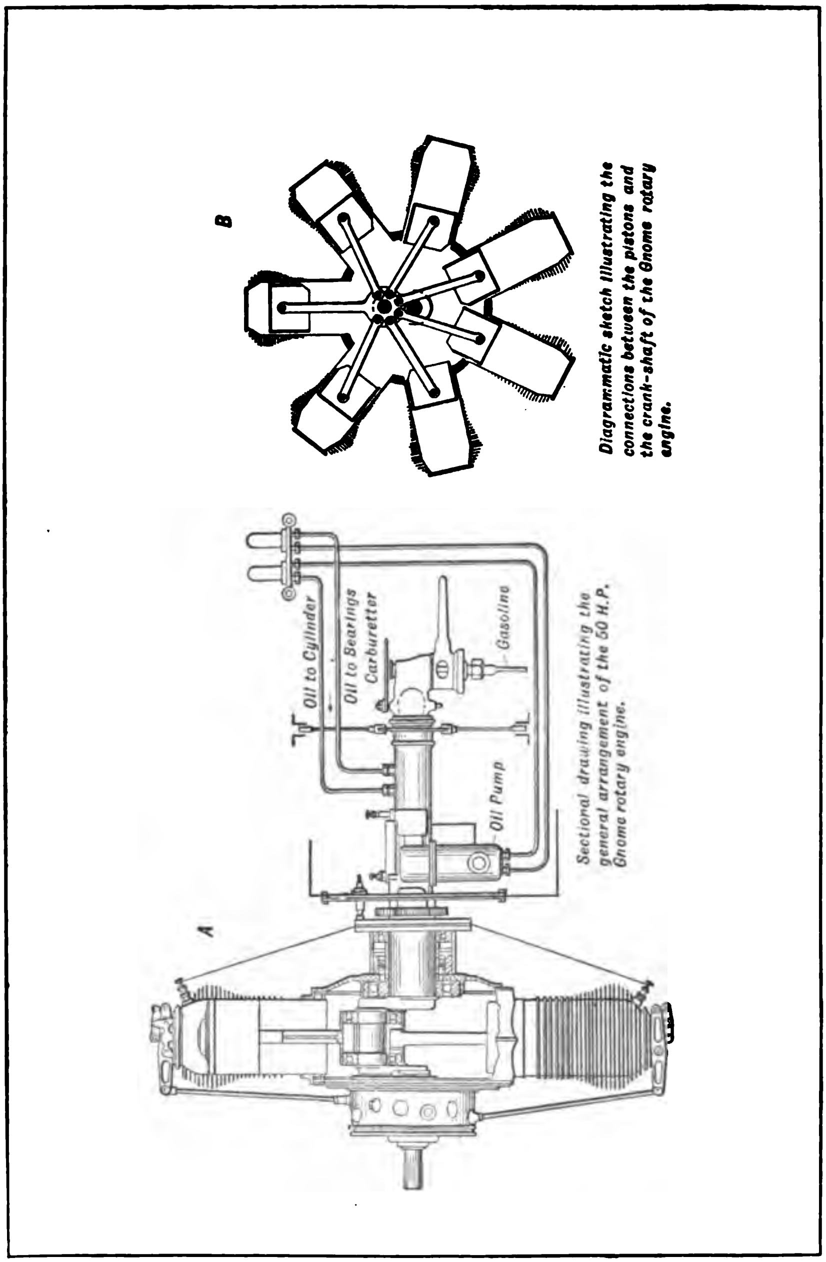


FIG. 418.—Side view of the Gnome seven-cylinder motor at A. B. Diagram showing arrangement of connecting rods.

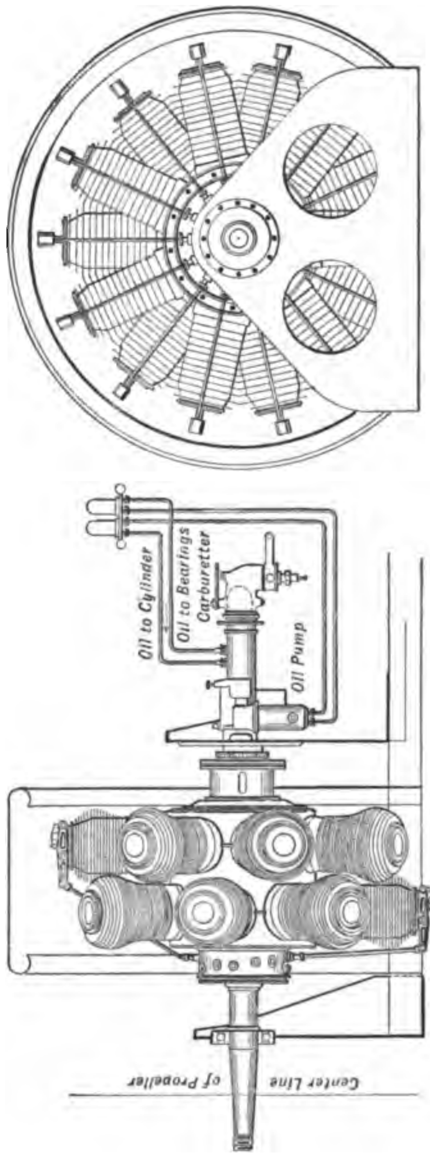


Fig. 418.—C. Assembly view of the fourteen-cylinder Gnome motor.

seven or fourteen cylinders; it has fourteen holes bored as illustrated on its circumference in two distinct planes, and offset in relation one to the other.

The cylinders, which have a bore of $4\frac{3}{16}$ inches and a stroke of $4\frac{7}{16}$ inches, are machined out of the solid bar of steel until the thickness of the walls is only 1.5 millimetres — .05905 inch, or practically $\frac{1}{16}$ inch. Each one has twenty-two fins which gradually taper down as the region of greatest pressure is departed from. In addition to carrying away heat, the fins assist in strengthening the walls of the cylinder. The barrel of the cylinder is slipped into the hole bored for it on the circumference of the crankcase and secured by a locking ring in the nature of a stout compression ring, sprung onto a groove on the base of the cylinder within the crank chamber. On each lateral face of the crank chamber are seven holes, drilled right through the chamber parallel with the crankshaft. Each one of these holes receives a stout locking-pin of such a diameter that it presses against the split rings of two adjacent cylinders; in addition each cylinder is fitted with a key-way.

The exhaust valve is mounted in the cylinder head, Fig. 419, its seating being screwed in by means of a special box spanner. On the fourteen-cylinder model the valve is operated directly by an overhead rocker arm with a gun metal rocker at its extremity coming in contact with the extremity of the valve stem. As in standard motor car practice, the valve is opened under the lift of the vertical push rod, actuated by the cam. The distinctive feature is the use of a four-blade leaf spring with a forked end encircling the valve stems and pressing against a collar on its extremity. On the seven-cylinder model the movement is reversed, the valve being opened on the downward pull of the push rod, this lifting the outer extremity of the main rocker arm, which tips a secondary and smaller rocker arm in direct contact with the extremity of the valve stem. The springs are the same in each case.

The pistons, like the cylinders, are machined out of the solid bar of nickel steel, and have a portion of their wall cut away, as at Fig. 421, so that the two adjacent ones will not come together at the extremity of their stroke. The head of the piston is slightly reduced in diameter and is provided with a groove into which is

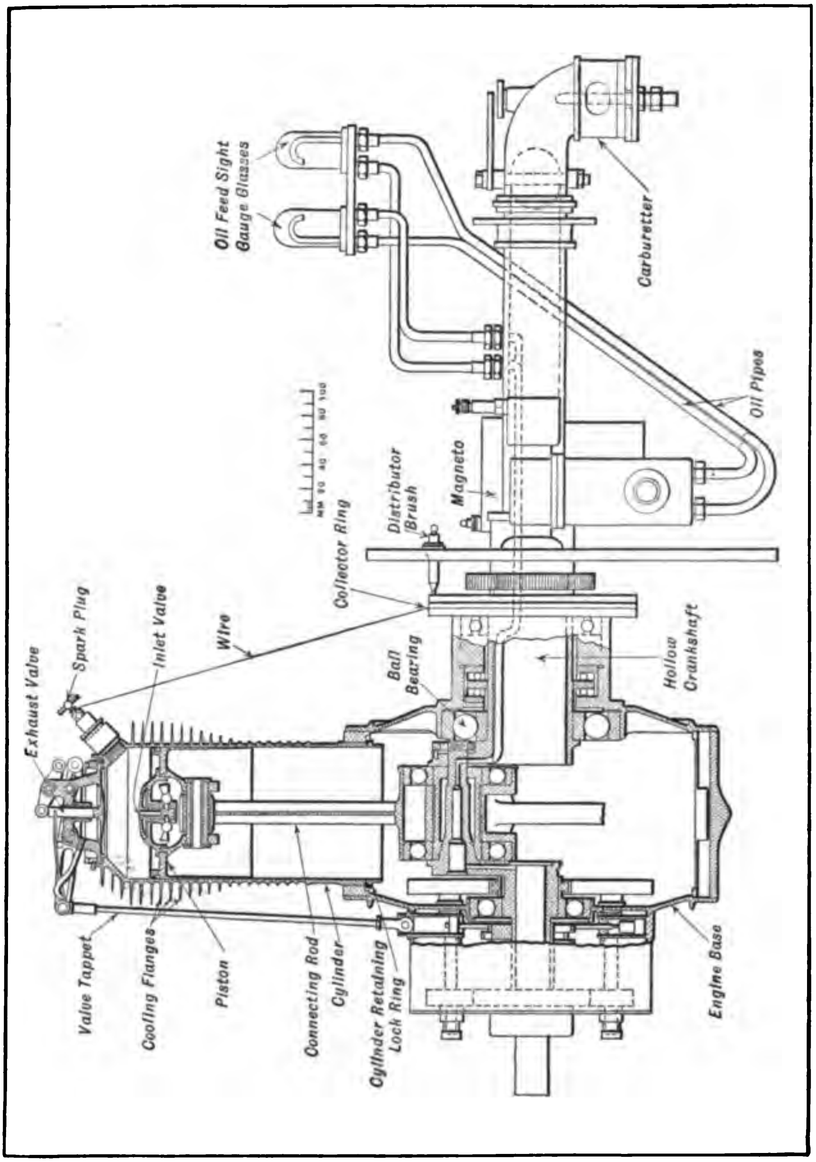


Fig. 419.—Sectional view showing arrangement of internal parts of the Gnome rotating cylinder motor.

fitted a very light L-section brass split ring; back of this ring and carried within the groove is sprung a light steel compression ring, serving to keep the brass ring in expansion. As already mentioned, the intake valves are automatic, and are mounted in the head of the piston. The valve seating is in halves, the lower portion being made to receive the wrist pin and connecting rod, and the upper portion, carrying the valve, being screwed into it. (See Fig. 419.) The spring is composed of four flat blades, with the hollowed stem of the automatic valve passing through their center and their two extremities attached to small levers calculated to give balance against centrifugal force. The springs are naturally within the piston, and are lubricated by splash from the crank chamber. They are a delicate construction, for it is necessary that they shall be accurately balanced so as to have no tendency to fly open under the action of centrifugal force. The intake valve is withdrawn by the use of special tools through the cylinder head, the exhaust valve being first dismounted.

The fourteen-cylinder motor has a two-throw crankshaft with the throws placed at 180 degrees, each one receiving seven connecting rods. The parts are the same as for the seven-cylinder motor, the larger one consisting of two groups placed side by side. For each group of seven cylinders there is one main connecting rod, together with six auxiliary rods. (Fig. 418 B). The main connecting rod, which, like the others, is of H section, has machined with it two L-section rings bored with six holes — 51½ degrees apart to take the six other connecting rods. The cage of the main connecting rod carries two ball races, one on either side, fitting onto the crank-pin and receiving the thrust of the seven connecting rods. The auxiliary connecting rods are secured in position in each case by a hollow steel pin passing through the two rings. It is evident that there is a slightly greater angularity for the six shorter rods, known as auxiliary connecting rods, than for the longer main rods; this does not appear to have any influence on the running of the motor.

Coming to the manner in which the exhaust valves are operated, this at first sight appears to be one of the most complicated parts of the motor, probably because it is one in which standard practice is most widely departed from. Within the cylindrical casing bolted to the rear face of the crankcase are seven, thin flat-faced steel rings, forming female cams (Fig. 421). Across a diameter of each ring is a pair of projecting rods fitting in brass guides and having their extremities terminating in a knuckle eye receiving the adjustable push rods operating the overhead rocker arms of the exhaust valve. The guides are not all in the same plane, the difference being equal to the thickness of the steel rings, the total thickness being practically 2 inches. Within the female cams is a group of seven male cams of the same total thickness as the former and rotating within them. As the boss of the male cam comes into contact with the flattened portion of the ring forming the female cam, the arm is pushed outward and the exhaust valve opened through the medium of the push rod and overhead rocker.

On the face of the crankcase opposite to the valve mechanism is a bolted-on end plate, carrying a pinion for driving the two magnetos and the two oil pumps, and having bolted to it the distributor for the high-tension current. (Fig. 418 C). Each group of seven cylinders has its own magneto and lubricating pump. The two magnetos and the two pumps are mounted on the fixed platform carrying the stationary crankshaft, being driven by the pinion on the revolving crankchamber. The magnetos are geared up in the proportion of 4 to 7. Mounted on the end plate back of the driving pinion are the two high-tension distributor plates, each one with seven brass segments let into it and connection made to the plugs by means of plain brass wire. The wire passes through a hole in the plug and is then wrapped round itself, giving a loose connection.

A good many people doubtless wonder why rotary engines are usually provided with seven cylinders in preference to an even number. It is a matter of even torque, as can easily be understood from the accompanying diagram.

Fig. 420A represents a six-cylinder rotary engine, the radial lines indicating the cylinders. It is possible to fire the charges in two ways, firstly, in rotation, 1, 2, 3, 4, 5, 6, thus having six impulses in one revolution and none in the next; or alternately, 1, 3, 5, 2, 4, 6, in which case the engine will have turned through an equal number of degrees between impulses 1 and 3, and 3 and 5, but a greater number between 5 and 2, even again between 2 and 4, 4 and 6, and a less number between 6 and 1, as will be clearly seen on reference to the diagram. Turning to Fig. 420B which represents a seven-cylinder engine. If the cylinders fire alternately it is obvious that the engine turns through an equal number of degrees between each impulse, thus, 1, 3, 5, 7, 2, 4, 6, 1, 3, etc. Thus supposing the engine to be revolving, the explosion takes place as each alternate cylinder passes, for instance, the point 1 on the diagram, and the ignition is actually operated in this way by a single contact.

The crankshaft of the Gnome, as already explained, is fixed and hollow. For the seven-cylinder motor it has a single throw, and for the fourteen-cylinder model has two throws at 180 degrees. It is of the built-up type, this being necessary on account of the original mounting of the connecting rods. The carburetor is

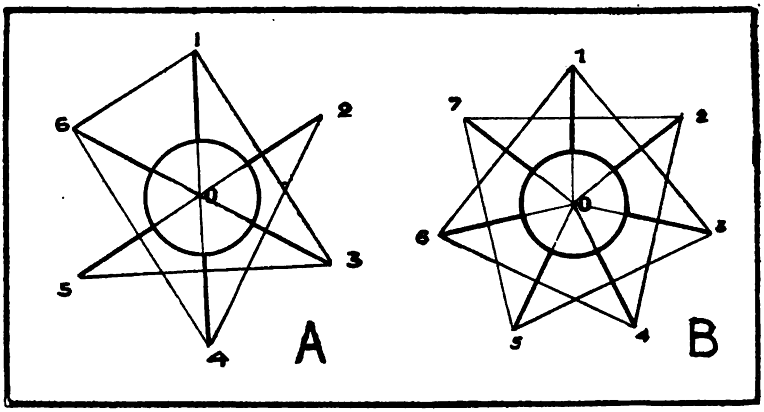


Fig. 420.—Diagrams showing why an odd number of cylinders is best for rotary cylinder motors.

mounted at one end of the stationary crankshaft, and the mixture is drawn in through a valve in the piston as already explained. There is neither float chamber nor jet. In many of the tests made at the factory it is said the fourteen-cylinder motor will run with the extremity of the gasoline pipe pushed into the hollow crankshaft, speed being regulated entirely by increasing or decreasing the flow through the shut-off valve in the base of the tank. Even under these conditions the motor has been throttled down to run at 350 revolutions without misfiring. Its normal speed is 1,000 to 1,200 revolutions a minute. Castor oil is used for lubricating the engine, the oil being injected into the hollow crankshaft through sight-feed fittings by a mechanically operated pump.

Rotary engines are generally associated with the idea of light construction and it is rather an interesting point that is often overlooked in connection with the application of this idea to flight motors, that the reason why rotary engines are popularly supposed to be lighter than the others is because they form their own flywheel, yet on aeroplanes, engines are seldom fitted with a flywheel at all. As a matter of fact the Gnome engine is not so much light because it is a rotary motor, as it is a rotary motor because the design that has been adopted as that most conducive to lightness is also most suited to an engine working in this way. There are two prime factors governing the lightness of an engine, one being the initial design, and the other the quality of the materials employed. The consideration of reducing weight by cutting away metal is a subsidiary method that ought not to play a part in standard practice, however useful it may be in special cases. In the Gnome rotary engine the lightness is entirely due to the initial design and to the materials employed in manufacture. Thus, in the first case, the engine is a radial engine, and has its seven cylinders spaced equally around a crank-chamber that is no wider or rather longer than would be required for any one of the seven cylinders. This shortening of the crank-chamber not only effects a considerable saving of weight on its own account, but there is a corresponding saving in the shafts and other members, the dimensions of which are governed by the size of the crank-chamber. With regard to materials, nothing but steel is used throughout, and most of the metal is forged nickel steel. The beautifully steady running of the engine is largely due to the fact that there are literally no reciprocating parts in the absolute sense, the apparent reciprocation between the pistons and cylinders being solely a relative reciprocation since both travel in circular paths, that of the pistons, however, being eccentric by one-half of the stroke length to that of the cylinder.

The Gnome is a considerable consumer of lubricant, the makers' estimate being 7 pints an hour for the 100 horse-power motor; but in practice this is largely exceeded. The gasoline consumption is given as 300 to 350 grammes per horse-power. The total weight of the fourteen-cylinder motor is 220 pounds without fuel or lubricating oil. Its full power is developed at 1,200 revolutions, and at this speed about 7 to 9 horse-power is lost in overcoming

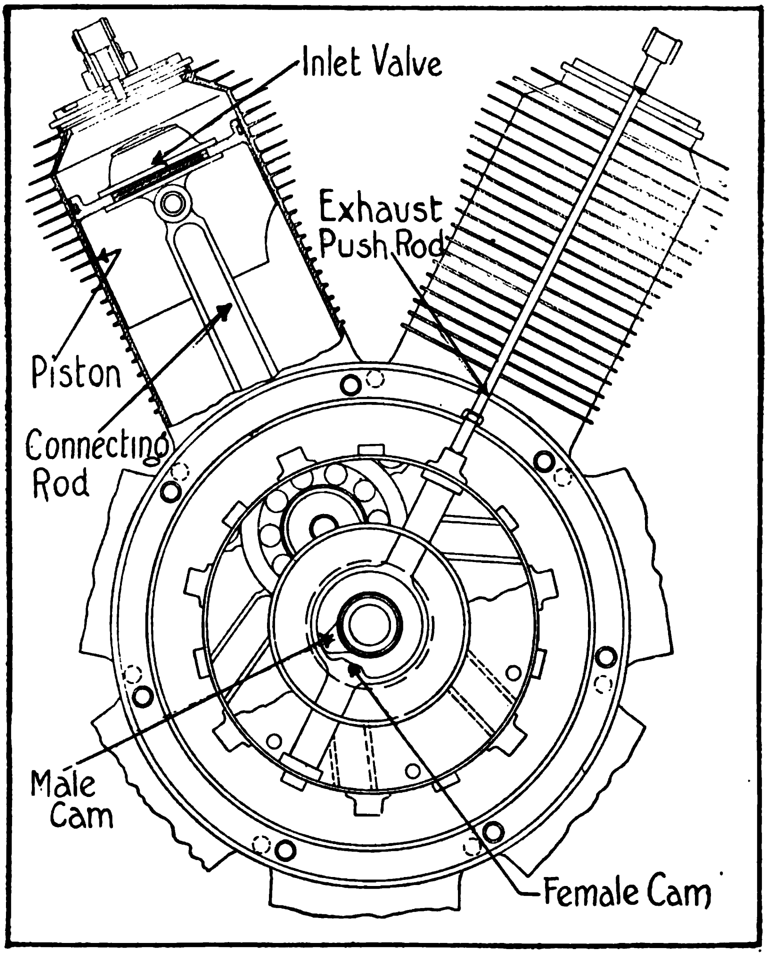


Fig. 421.—Showing cam arrangement of Gnome motors.

air resistance as the cylinders whirl round, and advocates of water-cooling adduce this as evidence that water-cooling is at least equally economical. Cooling has never presented any difficulties, the temperature of the fins immediately after a flight in cold weather rarely exceeding 30 degrees Fahrenheit, or so low that the hand can be placed on the cylinder. Under hot weather conditions the temperature may rise to 150 degrees Fahrenheit, but even this gives a very wide margin of safety.

While the Gnome engine has many advantages, on the other hand, the head resistance offered by a motor of this type is considerable; there is a large waste of lubricating oil due to the centrifugal force which tends to throw the oil away from the cylinders; the gyroscopic effect of the rotary motor is detrimental to the best working of the aeroplane, and moreover it requires about seven per cent. of the total power developed by the motor to drive the revolving cylinders around the shaft. Of necessity, the compression of this type of motor is rather low, and an additional disadvantage manifests itself in the fact that there is as yet no satisfactory way of muffling the rotary type of motor.

The practical mounting of the Gnome seven-cylinder motor in a Bleriot monoplane is clearly shown at Fig. 405 which is typical of the conventional method of installation. In this case, the power plant is at the front end of the frame and a tractor screw is utilized instead of a propeller screw, which is invariably mounted at the rear so it pushes the planes instead of pulling them through the air. At Fig. 422, the installation of a fourteen-cylinder Gnome motor in a Bleriot passenger carrying monoplane is outlined. The location in this case is back of the supporting surfaces. The relation of the various parts of the assembly may be readily ascertained by study of the illustration.

NOTES ON AERIAL PROPELLERS

There is considerable difference in the proportions of screws used in propulsion of marine and aerial craft and while in the former instance, one can consider that standard practice is very well defined, and formulæ available are capable of very broad application, it will be evident that there is a wide difference of opinion among authorities as to the best form of aerial propeller. In marine service the screw operates in a denser medium and therefore the blade area can be very much less than that of an aerial propeller designed to produce the same amount of thrust. This means that with a given horse-power the effective surface must be greater as the resistance of the medium decreases, and impels the conclusion that aerial propellers must be much larger in diameter and have greater blade width than marine screws designed for use in a more resisting fluid. It is for this reason that the researches of those engaged in development of the marine screw are of little utility to aerial experimenters, who have been forced to work on original lines because of the widely different fluids in which the screws are used. As will be seen, rules established by experimentation in a liquid could not be very well applied to a similar appliance designed for use in a gas. As we shall see, however, there is considerable variance in the form of screws which have been applied to the most successful aircraft.

It is a significant fact that the best results have been obtained with two-blade propellers, and the experiments made seem to prove

FIG. 422.—Method of installing Gnome motor in aeroplane.

that the simple type has undoubted advantage over the multi-bladed forms, not only from the viewpoint of efficiency but because of simplicity and lightness as well. It is claimed by those who have made a study of the subject, and who have experimented sufficiently to have ample foundations for any contentions they may make, that the whole question of the efficiency of the two-blade aerial propeller is of diameter and speed; and for any that are of rational design the higher the speed the lower the efficiency, and for any type the larger the diameter the greater the efficiency.

Speaking always of properly designed concave propellers, those of largest diameter and slower speed are always the most efficient, other things being equal. Reduce the diameter and increase the speed and the efficiency will drop, and the falling off at the higher speeds is most remarkable, for while it has ben possible with efficient screws to get thrusts of 40 to 50 pounds per horse-power, the modern aeroplane imposes conditions upon a screw that forces it to work at one-sixth its efficiency, and requires much power to turn it at high speed.

It is on this point that most aerial engineers have erred, and to obtain the constructional advantage of the direct drive, by placing the screw direct upon the end of the engine shaft, as well as a simpler and perhaps a more mechanical power plant placing, have sacrificed the efficiency to be obtained from the average propeller materially, even more than the limits imposed by the design would dictate. Using a single propeller six or seven feet in diameter, running about 1,000 revolutions per minute and requiring 50 horse-power (Fig. 405), as against the two propellers of the Wright machine turning at half that speed and taking but 25 horse-power (Fig. 406), the most efficient of the foreign creations have a lower efficiency of propulsion than the American aeroplanes. It is said that, considered from a purely theoretical aspect, an aeroplane or dirigible balloon should have propellers of very large diameter turning at a comparatively low speed, but as must be evident, the actual diameter must be reduced to conform to the general design, and it is this serious compromise that is the greatest handicap to the present-day aeroplane machine. While Langley demonstrated that one horse-power, properly applied, could carry 200 pounds 40 miles an hour, it is apparent that no machine even remotely approaches this in practice. According to this determination, the Wright machine (Fig. 416) should be supported by a five horse-power engine, or at the most seven horse-power, the added energy being conceded for overcoming the resistance of the necessary supporting members. This would entail propellers of large diameter and pitch turning at very low speeds, but it is evident that they are not possible with the rest of the design. It is the compromise of using the small diameter screw revolving at high speed to suit the present design that has made the use of high-powered motors necessary.

A stand adapted to test all forms of engines and propellers and the pounds thrust of the screw is obtained by a simple arrangement of mechanism. The engine is mounted on a base provided with wheels, which in turn are supported by a track so that the entire power plant may move back and forth as determined by the pull of the screw and the resistance offered by weights supported by the cord to restrain engine movement passing over a pulley or wheel. The amount of weight at the end of the cord may be varied at will, or a simple spring dynamometer, or weighing machine, may be substituted for the scale pan, and the efficiency of the installations may be ascertained by the amount of resistance overcome by the different power plant and screw combinations. When the engine is set in motion the effect of the propeller is to pull the motor forward on the track, while that of the weights is to resist its forward movement. The combination that will lift the most weight is the most efficient and has the greater thrust.

The many theories regarding the principles which govern propeller action may be grouped in either of two classes. To the first may be assigned those which consider the action of the screw upon the medium in which it is submerged, and from the movement of the elastic medium deduce the reaction upon the propeller. To the second class belong the theories which consider only the action of the medium upon the propeller.

The "disc" theory is a notable example of the first class, and considers that the propeller displaces a quantity of the medium in which it turns equal to the propeller diameter, and that given a definite amount of fluid having a certain change of velocity impressed upon it, the reaction resulting can apparently be calculated at once from the known density of the fluid. This method would possess a beautiful simplicity if we knew the exact effect of a propeller upon the fluid it passes through, and if the propeller blades were frictionless. Some authorities have assumed that a screw propeller gave to a column of fluid having a sectional area equal to the disc swept by the propeller a sternward velocity corresponding to the slip, but it would appear that in theories of the first class a change of pressure of the medium in motion is of just as much importance as a mere consideration of change of velocity of the fluid acted upon.

In the "blade" theory, typical of the second class, the face of the propeller blade is treated as if it were made up of a number of small inclined planes advancing through the air, and it is this hypothesis that most authorities seem to favor. As will be obvious if the blade surface were treated as an inclined plane, the medium could be considered as imposing a thrust upon the surface which would vary with the density of the medium and the angle of inclination of the plane as the blade moved through it. Despite the variance of theories it is evident they all bring out the same fact, and that is that rotation of a screw in a suitable medium will produce movement of both screw and fluid in which it is submerged. If the screw is held so that it can move only in a rotary direction the column of fluid it sets in moton only will move. If the screw is operating in an immovable medium, the screw will move in a direction parallel to its longitudinal axis. If both screw and fluid are free to move, the degree of movement will depend upon the "slip" between the screw and the medium in which it works.

The empirical rule that may be followed with advantage in designing either wood or metal propellers having two blades for use in air is as follows: The diameter should be as large as possible compatible with the limits of design; the blade area should be from 10 to 15 per cent. that of the area swept; the pitch should be approximately four-fifths the diameter, and the speed of rotation should be small. As the speed of rotation is increased the diameter must be reduced. Maximum thrust effort will be obtained with large diameter and low speed. The following table, reproduced from a current number of *Machinery,* tabulates some of the details of representative American and foreign aeronautic motors, sizes of propellers used and thrust available.

Comparative Data on Leading Aeronautical Motors.

Name	Rated Horse-Power.	Speed	Developed Horse-Power.	Weight.	Weight per Horse-Power.	Type.	Number of Cylinders	Cycle.	How Cooled.	Gasoline.†	Lubricating Oil.†	Propellers, Diameter.	Propellers, Pitch	Propellers, Thrust.
Sturtevant..	40	1,200	46	200	4.34	Vert.	4	4	Water	4.87	0.303	7' 6"	4' 6"	375
Sturtevant..	60	1,200	69	300	4.34	Vert.	6	4	Water	7.31	0.455	8' 0"	5' 0"	450
Gnome.....	50	1,200	42	167	4.00	Rot.	7	4	Air	4.00	0.375	7' 6"	7' 0"	420
Gnome.....	70	1,200	63	212	3.35	Rot.	7	4	Air	6.00	1.000	8' 0"	8' 0"	500
Renault....	50	1,800*	59.4	420	7.07	V	8	4	Air	.....	......	......	......	...
Renault....	70	1,800*	79.8	453	5.66	V	8	4	Air	.....	......	......	......	...
Roberts....	50	1,200	51	170	3.33	Vert.	4	2	Water	6.00	0.133	7' 6"	5' 3"	320
Roberts....	75	1,200	70	270	3.43	Vert.	6	2	Water	9.00	0.200	8' 6"	5' 6"	480
Curtiss.....	40	1,100	45	175	3.90	Vert.	4	4	Water	4.20	0.750	7' 0"	6' 0"	310
Curtiss.....	75	1,000	78	285	3.65	V	8	4	Water	7.20	1.200	7' 6"	8' 0"	500

* Propeller revolves at one-half engine speed or 900 revolutions per minute.
† Consumption in gallons per hour at developed horse-power.

CHAPTER XXI

PRODUCER GAS AND ITS PRODUCTION

The theory of the formation of this gas is that, by limiting the amount of air admitted to the fire, complete combustion of the coal is not permitted and the supply of oxygen being insufficient, carbon monoxide, CO, is formed instead of carbon dioxide, CO_2, while the steam formed in the vaporizer is led back under the grate and breaks up on striking the incandescent fuel, giving free hydrogen and carbon monoxide, the carbon monoxide and the hydrogen forming the power-gas for the engine.

The average gas, with a good grade of anthracite, should have a heat value of 130 to 140 British thermal units per cubic foot, and the constituents by volume as follows:

Carbon dioxide, CO_2	6 %
Carbon monoxide, CO	24 %
Hydrogen, H	15 %
Nitrogen, N	55 %
Hydrocarbon, CH_4	trace
Oxygen, O	trace

The actual combustion in the producer forms, at the grate, carbon dioxide, which on passing up through the glowing coal above the grate is robbed of one atom of oxygen to supply the coal above, which is getting insufficient air, and becomes in great part CO. The steam, H_2O, which is admitted under the grate, on encountering the glowing mass of coal is broken up into hydrogen and oxygen. The hydrogen passes through the producer as a free gas, while the oxygen unites with the coal to form CO.

Injection of steam under the grate serves four purposes: First. It gives the hydrogen for the actual power-gas. Second. It furnishes oxygen to the fire on breaking up and gives greater freedom from clinkers due to more complete combustion. Third. It keeps the grate cool and prevents the burning out of grate-bars.

Fourth. It is made by the heat of the gas passing from the generator, utilizing this heat which would otherwise be wasted, and bringing the gas to more nearly the temperature required at the engine, where it must be cool. Should the apparatus be of faulty design and more steam be admitted than the fire can break up, the effect will be a deadening of the fire and the diminution of the gas formed and in the end a complete shutting down. On the other hand, if an insufficient amount of steam is provided, the grates will burn out rapidly and the gas, through a lack of hydrogen, will be lacking in power. When anthracite is used, the amount of water transformed should be from 0.8 to 1.2 the weight of the coal.

In the cheaper forms of apparatus the cleaner is often omitted, but an examination of the pipes on such a plant after a month's use will conclusively prove its necessity. Where water is an important factor, the scrubber may be run hot and less water used, but it will be done at the expense of a cleaner which will be forced to remove a greater percentage of dust carried through to it by the uncondensed vapor, so that the sawdust and shavings in the cleaner will require more frequent renewal. Wherever necessary a cooling system can be installed and the water reused after slight filtration. As an example of the efficiency with which the gas is cleaned and dried, an instance may be cited of an installation of Julius Pintsch, at Heusy, in Belgium, which was run for an entire year without any cleaning of either engine or gas-apparatus; the deposit of foreign matter found at the end of the year was inconsequential. The superiority of the suction system over the pressure type of gas producers has been conclusively demonstrated. Originally many objections were raised to the suction type, and the idea of having the engine draw in its charge of gas, thereby making the draft for the fire, was considered impractical. The point was raised that the gas, being at less than atmospheric pressure, would interfere with the satisfactory working of the engine. If it were impossible to regulate the admission of the air with the pressure of the gas this objection would hold true, but this is taken care of by providing a separate air-inlet on the engine which allows the formation of a suitable mixture in all cases. Suction gas-plants are simpler and require less room than pressure plants. They are more economical of fuel and require less attention. They require no separate steam-boilers or large gas-holders and there is no chance for gas to escape into any of the rooms of the building as the whole apparatus is always under slightly less than atmospheric pressure, and any leakage would be of air into the apparatus instead of gas out. A leakage sufficient to bring about a stoppage would have to be very large and could not occur except through some extraordinary accident.

In the pressure type the air necessary to maintain the fire in the gas-generator enters the bed of fuel under pressure, caused by a steam-jet, blower, fan or similar means. Hence the gas passes through the apparatus and reaches the engine under a pressure of two or three inches of water. In the suction type the air required for generating gas is drawn through the bed of fuel and the resulting gas is then drawn through the cooling and cleansing apparatus by the sucking action or partial vacuum created by the engine piston. The pressure system has the advantage of being able to use a greater variety and an inferior quality of fuel than the suction type. Anthracite or bituminous coals, lignite, wood, peat, tan-bark, coke, and charcoal may be successfully gasified in the pressure-type producer. It can also work more satisfactorily when supplying gas to a number of engines from a central producer plant.

In the suction type the character and heat value of the generated gas are essentially the same as from a pressure type of plant. It is of the first importance that good coal be selected, if undue care and interrupted operation are to be avoided. It is best also to install an apparatus of ample capacity for the work desired. The overrated power of these installations has oftentimes caused needless annoyance and expense, besides condemning an apparatus of much merit when intelligently proportioned and rated. To date, the use of bituminous coal is confined to large units. In order to successfully operate a gas-engine, the tar must be removed, which necessitates either an elaborate system of scrubbers and cleaners, or the combustion of the tar in the producer itself. Working on the latter principle, Julius Pintsch has in operation plants for both lignite and bituminous coal, a plant of 400 horse-power working admirably on the latter fuel with the very low consumption of ten ounces per brake horse-power hour. With bituminous coal at $3.00 a ton, this brings the cost per horse-power hour down to $\frac{3}{32}$-cent per horse-power hour, which, barring water-power and natural gas, may be said to be the cheapest form of power yet known.

COKE-OVEN GAS

The coke industry affords an important field for gas-power. Coke by itself represents about 75 per cent. of the best value of the coal coked. The remaining 25 per cent. in the case of the ordinary bee-hive oven is discharged into the atmosphere in the form of products of combustion. The gaseous distillate is practically the same as ordinary retort coal-gas and as such forms a most excellent fuel for power purposes. In the process of coking coal in closed retorts or ovens, the gas obtained is obviously similar to the coal-gas manufactured for illuminating purposes, and contains an average of 39 per cent. of hydrogen, 45 per cent. of hydrocarbons, 5 per cent. of carbon monoxide, and 3 per cent. of carbon dioxides.

For this gas, the gross heat value is 679 British thermal units per cubic foot and the available heat value 560 thermal units. The gas leaves the retorts at a high temperature and carrying a considerable burden of impurities, must be cooled and purified before it is fit for use in an engine-cylinder. Its high hydrogen contents makes it somewhat sensitive and violent; but with reasonably careful adjustment and operation it constitutes a good fuel for use in a gas-engine. In coking one ton of average coking-coal in a retort there are generated from 8,000 to 10,000 cubic feet of gas, carrying from 60 to 100 pounds of tar and 10 to 25 pounds of ammonium sulphate. The tar and sulphate must be extracted and are marketable — their sale value more than covering the cost of their extraction; but generally the gas carries an excess of sulphur and always some dust, and the amount of these must be reduced to a minimum.

Of the total volume of gas only about one-half is required for carrying on the coking process; a balance of 4,000 to 5,000 cubic feet remains available for other purposes, such as illumination or power generation. In other words, in the coking of one ton of coal there become available, and are only too frequently wasted, about 2,500,000 thermal units, sufficient to develop in gas-engines at least 205 effective horse-power hours. Thus for every 11 pounds of coal coked per hour, one effective horse-power is available as a by-product. In the Connellsville district about

300,000 tons of coal are coked per week. The surplus gas from this coal would develop 366,000 effective horse-power continuously. The use of coke-oven gas is one of the results of the perfection of the by-product coke-oven, although the primary object of this form of oven was perhaps as much for the recovery of tar and ammonia as for the waste gases. About half of the gases are, however, used as fuel for heating the ovens themselves for the distillation of the coal charges and the recovery of the gas for this purpose was undoubtedly the primary object sought.

But since only a portion of the gases voided are necessary for heating the ovens, the remainder are available for other uses, and while they have been used as fuel in boilers, it has been found that for the production of power a most efficient use has been to burn them, after purification, in the cylinder of a gas-engine. In showing the adaptability of the waste gases of coke-ovens in gas-engines, and also the magnitude of the power available, it is preferable to sketch briefly the method by which they are generated, and so exhibit their qualities and qualifications for this work.

The possibility of utilizing this source of power may be said to be due to the development to perfection of the by-product coke-oven, though perhaps the contemporaneous development of the gas-engine itself should be counted as an equal factor. It may not be known to all that the operation of coking any coal consists simply in heating it, out of the presence of the atmosphere, so that the volatile matter is distilled off, leaving almost pure carbon or coke as the residual product. The coal is delivered to each oven from a travelling larry, which runs over the top, through spouts, thus delivering the fuel charge comparatively level on top and nearly filling the oven. The heat is supplied to the ovens by the combustion of gas beneath them, the products of combustion passing up through flues in the brick work between each oven. The air used in burning the gas is brought from the outside through a regenerator placed under the ovens, whereby it becomes heated to a high temperature, thus making the temperature of combustion correspondingly higher. The burned gas, after passing through the flues between the ovens, is led through the regenerator. The valve arrangement allows of a transposition of air and burned gas in the regenerators, so that one is being heated by burned gas while the other is giving up its heat to the air used in the combustion. Coke-oven gas is largely in use in England, Germany, and Belgium, and although on limited trials only in the United States, its future extension is apparent, and pipe-line extensions may build up large industries within reasonable distances from the coke-producing centers.

BLAST-FURNACE GAS

The gases from blast-furnaces, heretofore used under boilers for generating steam for power to drive the blowing-engines of the furnaces, is now coming into use for a more direct application of its power by its use in the cylinders of the blowing-engines.

Its limitation to the iron-making districts bars it from general use, but the surplus power above the requirement of the furnace, when used in a gas-engine for the furnace-blast, hot stoves, etc., makes it an available means of profit for distribution to a neighborhood. The approximate analysis of blast-furnace gas is as follows:

Hydrogen, H	5.2 %
Carbon monoxide, CO	26.8 %
Marsh gas, CH_4	1.6 %
Carbon dioxide, CO_2	8.2 %
Oxygen, O	.2 %
Nitrogen, N	58.0 %
	100.0

Heating value 106 British thermal units, and from 80 to 120 cubic feet is required mixed with an equal quantity of air per horse-power per hour.

Blast-furnace gas is found by experience to make an excellent power-gas, as it is not "snappy," therefore permitting of comparatively high compression and consequently high efficiency. The difficulties in cleaning have apparently been overcome and several American engine-builders are prepared to meet the demand for heavy-duty engines of several thousand horse-power capacity. Every iron and steel works operating a blast-furnace establishment should thus become a producer of energy for its own and outside consumption, instead of an augmenter of the smoke nuisance. It is now generally conceded that the blast-furnace gas must be cleaned before use in the gas-engines; if for no other reason than that the cleaning process at the same time reduces its temperature and thus increases its density, thereby increasing the power available from a cylinder of given dimensions. Whether cleaned by transmission through great length of pipe at low velocity, or by contact with sprays or surfaces of water, the temperature is lowered. Cooling and cleaning by the dry or transmission method is not satisfactory, and becomes very costly if a temperature below 120° F. is desired. Nor do conditions of velocity, satisfactory for cooling, permit the settling of the dust, and the finest particles, when dry, require practically absolute rest, which is, of course, impossible. Water cooling and washing is now generally employed.

For the gas delivered at the top of a blast-furnace, consisting of the products of combustion and partial combustion of coke, and the decomposed moisture and volatile contents of the charge, the average volumetric composition is:

Hydrogen	2.25 %
Hydrocarbons	.25 %
Carbon monoxide	24.5 %
Carbon dioxide	.12 %
Nitrogen	62. %

Gross heat for this gas is 92.5 British thermal units per cubic foot and available heat 86 heat units. This gas leaves the furnace top at a temperature of about 400° F. and carrying a considerable burden of dust and moisture. It must be cooled, cleaned, and dried before it is in a condition fit for use in an engine cylinder. The heat value of blast-furnace gas lies chiefly in its carbon monoxide, the proportion of hydrogen being very low; the gas is therefore neither sensitive nor violent, will safely permit a high compression, and as a result its ignition is sure and its efficiency high in spite of its low heat value.

For each ton pig-iron output, the average blast-furnace delivers about 10,500 pounds of gas at its top. In other words, the gas delivered by a blast-furnace weighs 4.7 times as much as the pig-iron it produces. The volume of such gas at 62° Fah. and 30 inches of mercury, equivalent to a weight of 10,500 pounds, is 131,000 cubic feet. Thus, per ton of pig-iron produced, there are delivered by the furnace 11,266,000 net thermal units. A portion of this gas is utilized to heat the blast for the furnace to a temperature of about 1,200° Fah., but a surplus of 76,000 to 77,000 cubic feet, or, say, 6,580,000 heat units per ton of pig may be safely figured upon.

As has been stated, the gas, as it leaves the blast-furnace top, is hot, dirty, and wet, and must be cooled, cleaned, and dried. A typical mode of procedure is to pass the entire volume of gas through a dust-catcher, the area of which is proportioned so that the gas travels at a low velocity. In this dust-catcher the major portion of the heavy dust settles out, and the gas temperature is reduced by radiation. As a rule, the gas passes directly from here to the stoves and boilers. If the gas-mains are long and of ample diameter, a further considerable quantity of dust settles out in them, and where water is scarce and space available, a multiplication of dry-dust catchers or long, large mains with dust-pockets affords an efficient means at low operating cost for all but the final drying and cleaning. But where an ample supply of cold water can be obtained, the cooling and cleaning of the gas becomes simpler and all the gas — whether for stoves, boilers, or gas-engines — should be washed by passing either through vertical tanks or horizontal pipes against fine sprays of water. The gas for gas-engines must be still further cleaned and dried, and various means can be employed for this purpose: coke-scrubbers with steam-jets, lattice-work with water-curtains, or centrifugals with water-injection, these to be followed by filters consisting of layers of excelsior or sawdust, or followed by water-separators. Provision of a gas-holder is always desirable, but its capacity per gas-engine horse-power may be varied to suit the blast-furnace plant — the greater the number of furnaces, the smaller may be the gas-holder. A satisfactory gas-cleaning installation in a plant whose space will not permit more than a fractional cooling by direct radiation, consists of vertical tanks set in water-seal catch-basins followed by centrifugals with water-injection.

PRODUCER GAS FOR MARINE PROPULSION

Experiments are in progress for utilizing producer gas for launch, yacht, and ship service, not only for economy over fluid fuels now in use, but for safety from the occasional disasters due to the use of the highly volatile fluids. Trials of marine engines driven by producer gas now being made in Germany and in England, may make a further and more extended use of the explosive motor for marine propulsion. It is claimed that the additional weight of engine, producer-plant, and coal will be but slightly increased beyond the present equipment of marine motors of the explosive type and far less than for steam-driven motors.

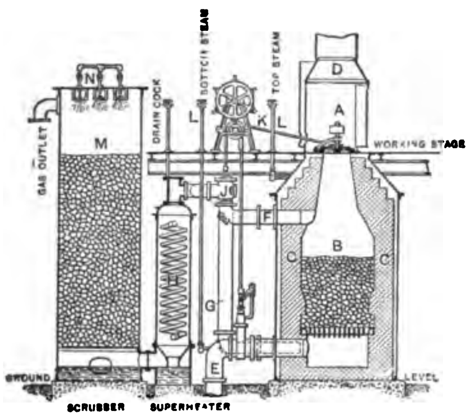


FIG. 423.—Gas producer.

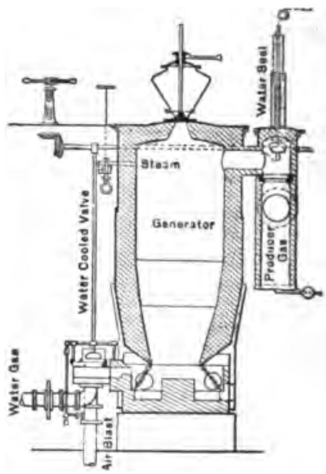


FIG. 424.—Gas generator.

PRODUCER GAS GENERATORS

This gas, like its congeners—water gas, Dowson gas, suction or aspirated gas, and Mond gas — is made by distilling by heat and steam or air, bituminous or anthracite coal in a closed furnace, using the heat generated by their partial combustion for producing the chemical reaction resulting in a permanent gas of varying constituents due to the different methods of operating the generating furnaces. In Fig. 423 is illustrated a producer gas generator in which A is a swing or lift-door for feeding coke, anthracite, or bituminous coal to the furnace B, and for blowing up. C, fire-brick walls of the furnace. E, air-inlet for heating the furnace of the generator. F and G, gas blow-off pipe, interchangeable to reverse the gas-blow. J, valve that automatically closes when A is opened. L, L. steam-pipes for alternating the steam-blow. H, superheating coil for heating the steam by the hot gases passing to the scrubber M. N, sprinkler. K, wheel and drum for simultaneously opening and closing the valves, J and G, and the blast-door A. The initial firing produces CO_2 with air alone, and an addition of hydrogen when steam is blown alternating with air. The air-blast raises the heat of the furnace to a high temperature; when the air is shut off and steam turned into the furnace, it is

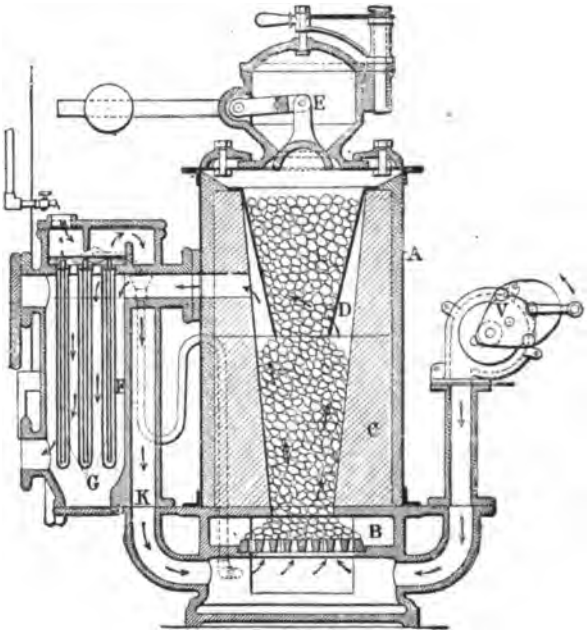


FIG. 425.—Gas and steam generator.

forced into contact with the surface of the hot coal and becomes dissociated, the oxygen uniting with the carbon, forming carbonic monoxide CO, setting hydrogen free. This product is technically termed water gas. While the non-use of steam or the mixed use of steam and air in the after-blow produces the various grades of gases and their respective heat values, all producer gases, but termed technically water gas, semi-water gas, Mond gas, and suction or inspirator gas, are later detailed as to analysis and heat value. Fig. 424 illustrates a simple gas-generator of the

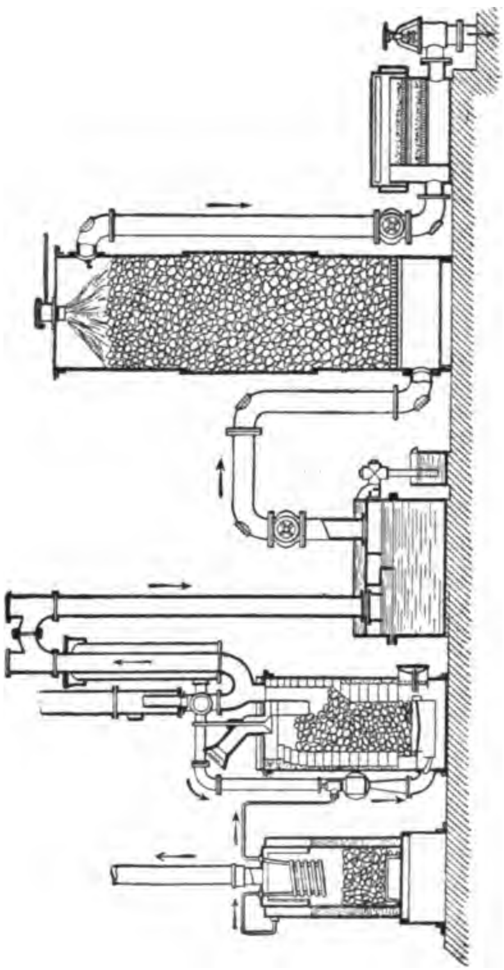


FIG. 426.—Purified producer gas plant.

Lowe type, an iron cylinder lined with fire-brick. Air is blown in at the bottom for heating the coal or coke. Then steam is blown in at the top, passing through the hot fuel, and discharged at the bottom as water gas. Fuel is fed through the hopper at the top. By reversing the blowing by steam and air, producer gas is made and discharged through the side pipe at the right. This simple generator is only suitable for anthracite or coke-fuel.

In Fig. 425 is illustrated a gas and steam generator of Belgian design. A magazine-furnace with a double-valve hopper for charging the magazine. The steam-generator consists of a number of drop-tubes closed at the bottom, each with a central water-feed tube of smaller size. The drop-tubes are screwed into the bottom plate of the steam chamber, which has a partition to separate the water-inlet from the steam compartment, from which the steam is drawn through the small pipe to the ash-pit beneath the grate. The blower at the right is for starting the fire. The air is drawn in for continuing the combustion through the pipe K, by the suction of the motor.

Fig. 426 represents a very complete producer gas generator of German type, in which steam is generated in a double-shell boiler at the left in the cut, superheated in a coil over the fire, and then passed through the combined air and steam inlet to the converter, the incoming air being heated in the jacket of the outgoing gas-pipe. The blower is not shown. To the right of the converter is a tar-box and waste-siphon. In addition to the usual scrubber, a lime-purifier is used to eliminate any sulphurous gases passing the scrubber.

In Fig. 427 is illustrated the German producer gas plant of Julius Pintsch. This producer was simple in construction and operation, required little attention, and gave a brake horse-power hour in small units on one pound of Belgium anthracite. Four years' practical experience with this plant brought many improvements and the construction of the present Pintsch suction gas-plant is as follows: In Fig. 427, A is a blower, furnishing draught for starting the fire and raising the heat in the generator to the proper temperature for the production of gas; B, the generator, equipped with a grate on which the coal is burned, a hopper H, which allows charging during operation, a window-valve for inspection of the fire, and fire-doors for poking down;

C is a vaporizer fitted with a small tubular boiler for the generation of steam, and a relief-pipe or chimney for use when the engine is not running; D, a scrubber consisting of a coke-tower with a water-spray for washing the gas; E, a cleaner containing wooden trays covered with wood shavings or sawdust through which the gas is filtered, giving up the last of its dirt and dust; F, a governor or pressure-equalizer for maintaining a steady pressure throughout the apparatus. To operate the plant, a fire is lighted in the generator and a small amount of coal added, the blower being run until the fire is burning strongly with the relief-valve R open. After ten to fifteen minutes' blowing, the fire is

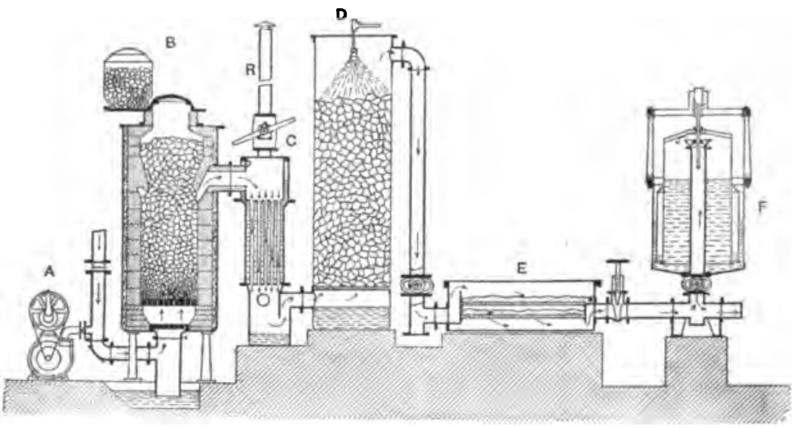


Fig. 427.—Pintsch producer gas plant.

sufficiently hot to give off gas; the relief-valve is then closed and the gas allowed to pass through the apparatus, the blower being kept running at slower speed until the gas burns freely at a test-cock beside the engine. The engine is then started, the blower stopped, and the formation of gas becomes automatic; the suction stroke of the engine furnishes the draft through the fire. In ordinary practice, the fire is left burning overnight with limited draft and only a few minutes' blowing is required to brighten up the fire in the morning. The generator should be kept full of coal and the fire kept clean and bright. Since the apparatus is always under a slight vacuum, the fire-doors can be opened at any time for cleaning out the fire.

The vaporizer is built in three sections, the upper being simply a chamber connected with the relief-pipe or chimney; the middle, a small tubular boiler, and the base section acting as a cleaning-pot and water-seal when the engine is not running. By the passage of the hot gases coming from the generator through the flues of the boiler, the gas is cooled and steam is generated which is passed back under the grate. The cleaning-pot or bottom section collects the heaviest dust and dirt coming over with the gas. By the admission of water to the cleaning-pot on shutting down, the rest of the apparatus is water-sealed and the gas therein kept intact for starting up again. The scrubber should never feel more than warm to the hand, otherwise steam will pass through it to the apparatus beyond, carrying with it a considerably greater percentage of dust, and the gas will not cool when it reaches the engine. The gas must reach the engine cool or the charge taken in will be a charge of expanded and rarefied gas and will not carry sufficient energy. In the cleaner, the gas gives up the last of its dust and moisture and emerges cool, clean and dry.

The apparent simplicity of the suction gas producer has led to the introduction of plants in which the chemical and scientific sides of the problem have been entirely disregarded. Cheapness of first cost has been sought rather than economy of operation, the arrangements for cleaning the gas being in almost every case insufficient, so that the whole installation requires frequent cleaning. The dirt thus allowed to pass through with the gas fouls the valves and cylinder of the engine, causing a leaky piston and rapid deterioration of the moving parts.

In Fig. 428 is illustrated a suction gas-plant of the Crossley type. Besides producers of the pressure type, for use with either anthracite or bituminous coal, Messrs. Crossley make a special feature of their suction gas producer plant, which consists of the producer proper, coke-scrubbers, and an expansion-box. The construction of the principal parts is shown in the cross section, which is largely self-explanatory; the engine draws air and steam through the fuel in the producer generating the gas, which passes through the scrubbers on its way to the engine. The steam is raised by the waste heat of the producer from water surrounding the bell of the feeding hopper, and is superheated before entering the furnace. The hopper holds sufficient fuel to last for four hours without attention, the operation of the plant being automatic. The notable features of this producer plant is the water-jacketed magazine-bell which acts as the steam-generator, air and steam mixing chamber at the top of the generator, and the double-chambered scrubber, in which the gas and water flow in one direction, depositing the ash, tar, and dust in the hydraulic box, while the contrary currents in the compartment further clear the gas from sulphurous gas and ammonia. The friction of the gas is also partially eliminated by

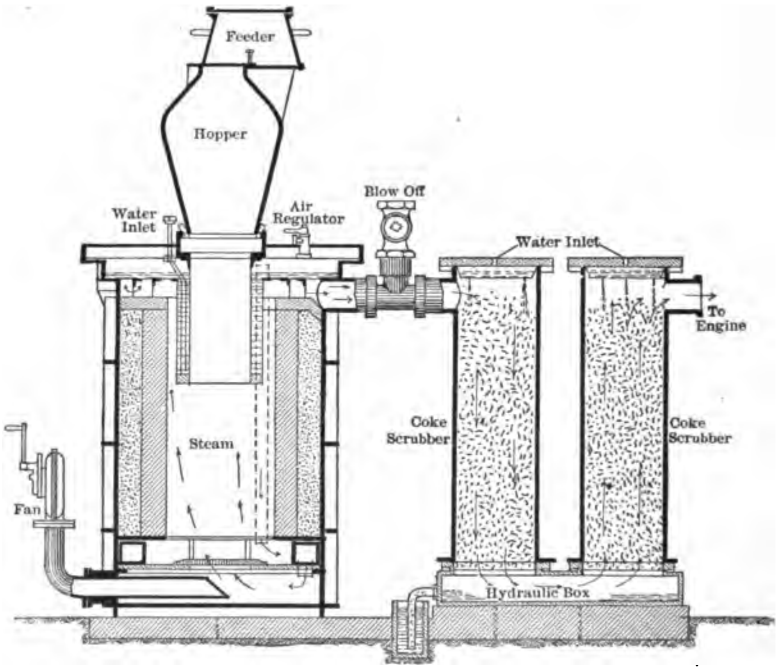


FIG. 428.—Crossley suction gas producer.

passing with the water current through one-half of the length of an equal single scrubber, besides being a convenience in compactness of the plant. It is claimed that there is considerable economy of fuel with the statement that the consumption of anthracite at full load is from 0.65 to 0.85 pound per brake horse-power hour, and of water 1 gallon for all purposes. The plant is made for outputs up to 300 brake horse-power, the largest size occupying a space of 21 feet 6 inches by 15 feet by 19 feet high.

In Fig. 429 is illustrated the Mond gas-generator, which is briefly described as follows:

The cheapest bituminous slack obtainable is mechanically deposited in hoppers above the producers. From this it is discharged into the producer-bell, where the heating of the slack takes place, and the products of distillation pass down into the hot zone of fuel before joining the bulk of the gas leaving the producer. The hot zone destroys the tar and converts it into a fixed gas, and prepares the slack for descent into the body of the producer, where it is acted upon by an air-blast which has been saturated with moisture and water superheated before contact with the fuel. The hot gas and undecomposed steam leaving the producer pass first through a tubular regenerator in the opposite direction to the incoming blast. An exchange of heat takes place, and the blast is still further heated by passing down the annular space between the two shells of the producer on its way to the fire-grate; then the hot products from the producer are further passed through a "washer," which is a large, rectangular, wrought-iron chamber with side-lutes; and here they meet a water-spray thrown up by revolving dashers, which have blades skimming up the surface of the water contained in the washer. The intimate contact thus secured causes the steam and gas to be cooled down to about 194° Fah., and by the formation of more steam tending to saturate the

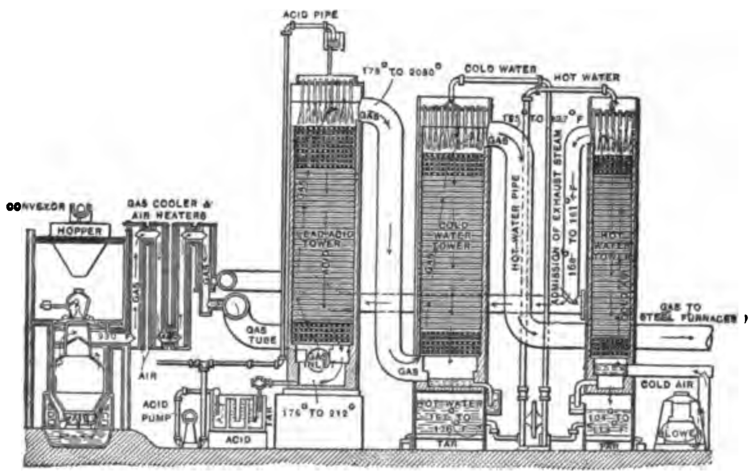


FIG. 429.—The Mond gas generator.

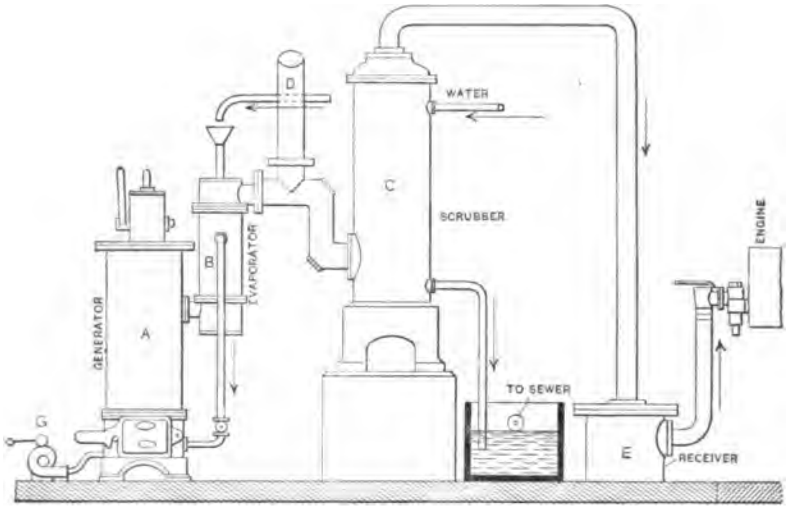


FIG. 430.—Suction or aspirator gas plant.

gas with water-vapor at this temperature, then passing upward through a lead-lined tower, filled with tile to present a large surface, the producer gas meets a downward flow of acid liquor, circulated by pumps, containing sulphate of ammonia with about four per cent. excess of free sulphuric acid.

Combination of the ammonia of the gas with the free acid takes place, giving still more sulphate of ammonia, so that, to make the process continuous, some sulphate liquor is constantly withdrawn from circulation and evaporated to yield solid sulphate of ammonia, and some free acid is constantly added to the liquor circulating through the tower. The gas, being now freed of its ammonia, is conducted into a gas-cooling tower, where it meets a downward flow of cold water, thus further cooling and cleaning it before it passes to the various furnaces and gas-engines in which it is used.

Fig. 430 illustrates a suction or aspirator gas plant and connection with a gas-engine in which A is the generator proper, where the combustion takes place. The gas produced passes in to the evaporator B, the interior of which is filled with small vertical tubes through which the hot gases pass while water trickles over their outer surfaces, cooling the gas and at the same time evaporating the water, which, mingling with the air, also drawn in at the top, is carried into the generator A. The evaporator is provided with an overflow for the water which is not thus evaporated. From the cooler, or evaporator, the gas passes to the scrubber C, which is simply a shell filled with coke through which the water passes downward against the ascending current of the gas, the water being discharged to the sewer from the collecting tank at the bottom, while the gas passes to the receiver E. The coke and the water retain not only the entrained dust, but the ammonia and other chemical impurities of the gas. The receiver E may be replaced to advantage by a small gas-holder with water-seal and top section suspended by a very elastic spring, to neutralize the jumping action of the engine-piston.

In order to start the generator, the small hand-blower G is employed, by the aid of which sufficient air is introduced to ignite the bed of fuel. The gas at first formed, which is not suitable for use in the engine, is allowed to escape to the atmosphere through the escape-pipe D. Some fifteen or twenty minutes after the generator has been ignited, the pipe D may be closed and the engine started. The aspiration by the engine itself commences, little by little the normal condition is established, and in from one-quarter to one-half an hour the gas becomes sufficiently rich to take care of the motor under full load.

In Fig. 431 we illustrate the Nagel suction gas plant. The suction gas producer plant consists of a producer, an evaporator, an overflow water-pot, a scrubber, and an equalizer. The producer is lined with fire-bricks. By the sucking action of the engine a mixture of air and steam is drawn through the burning fuel, whereby the producer gas is generated. The producer is provided with a hopper through which fuel can be filled into the producer without interfering with the working of the engine. The cleaning of the grate may be performed during the regular work. The gas leaving the producer heats up the evaporator and causes a formation of steam which goes under the grate together with the necessary amount of air. From the producer the gas goes through the scrubber, in which it is cooled and purified from the dust and tar. From the scrubber it goes through a small equalizer to the engine. Before starting the engine the fuel in the producer has to be heated up by means of a small hand-blower *a*, attached to same, until the fuel is burning well. For this about ten minutes are required. When this point is reached the hand-blower is stopped and the engine started in the usual way.

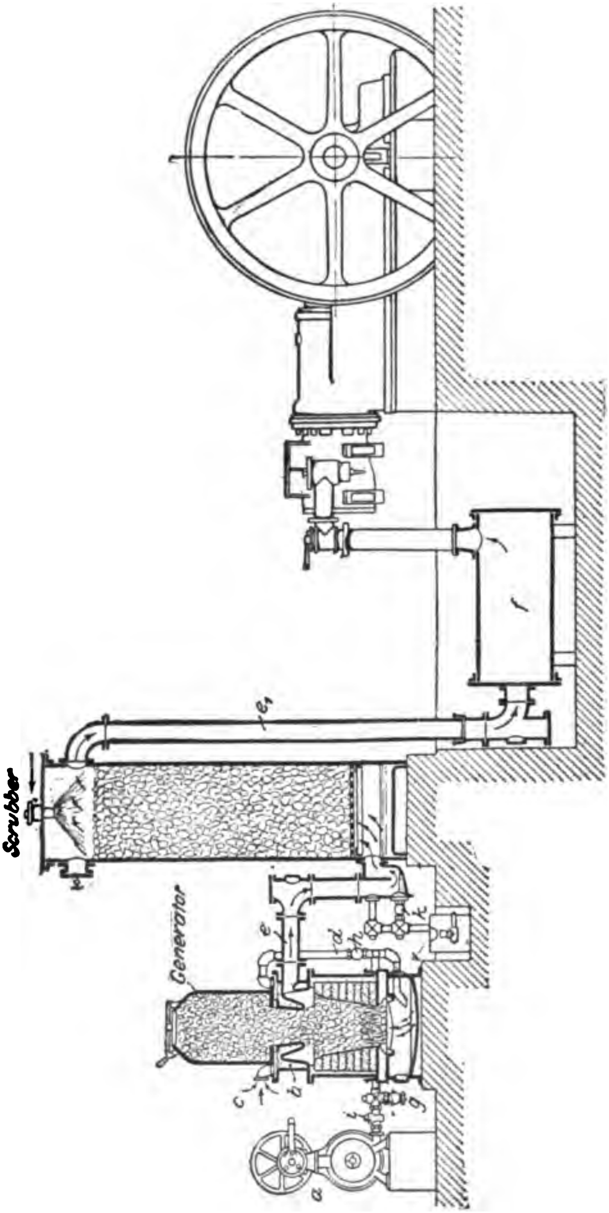


FIG. 431.—Nagel suction gas producer plant, built in units from 5 to 300 horse-power.

The engine then draws, by its own sucking action, the necessary amount of air and is producing its own power-gas. The air is entering at *c* and goes through the evaporator *b*. Here it is saturated with steam and the mixture of air and steam passes through pipe *d* under the grate of the producer, through the fuel, and then through pipe *e* to the scrubber; from here through pipe e_1 to the equalizing tank *f*, which is directly connected with the engine. *k* is the overflow and tar-box. The gas-making process continues as long as the engine is running, but as soon as the engine is stopped the gas-making is also stopped.

The cut shows a sectional elevation of a 25 horse-power plant. The plants up to this size are provided with a sufficiently large fuel-hopper so as to contain fuel for the working-day and to avoid the necessity of recharging the fuel during the working-hours. The sizes above 25 horse-power are provided with a bell-hopper, and the sizes about 75 horse-power have, instead of a water-jacket evaporator, an independent evaporator. These producer gas plants can be used equally well on board of boats in connection with producer gas marine engines. Anthracite, charcoal, or coke can be used for generating gas in the suction gas producer. It will take, according to the ash content, 1 to 1¼ pounds of anthracite or charcoal, or 1¼ to 1⅓ pounds of coke for developing 1 horse-power per hour. With anthracite (pea) at $5.00 per ton, 1 horse-power for 24 hours will cost from 6 to 8 cents. This is about one-sixth the cost of illuminating gas-power (at a price of 75 cents per 1,000 cubic feet of illuminating gas) or one-eighth the cost of gasoline at a price of 16 cents per gallon).

SUCTION OR ASPIRATOR GAS

In the above-described gas producer the boiler and gas holder, two troublesome adjuncts, are dispensed with and their cost, care, and room made a saving clause in the generation of power. In this apparatus the gas is produced directly by the suction or aspiration of the motor in such quantities as required for immediate use. In the use of this gas, an open fire in the generator, to give the draught of the motor as free from obstruction or friction as possible, is desirable, such as derived from coke or clean anthracite coal. The average composition of this gas from coke of 13.240 heat units per pound, consists of:

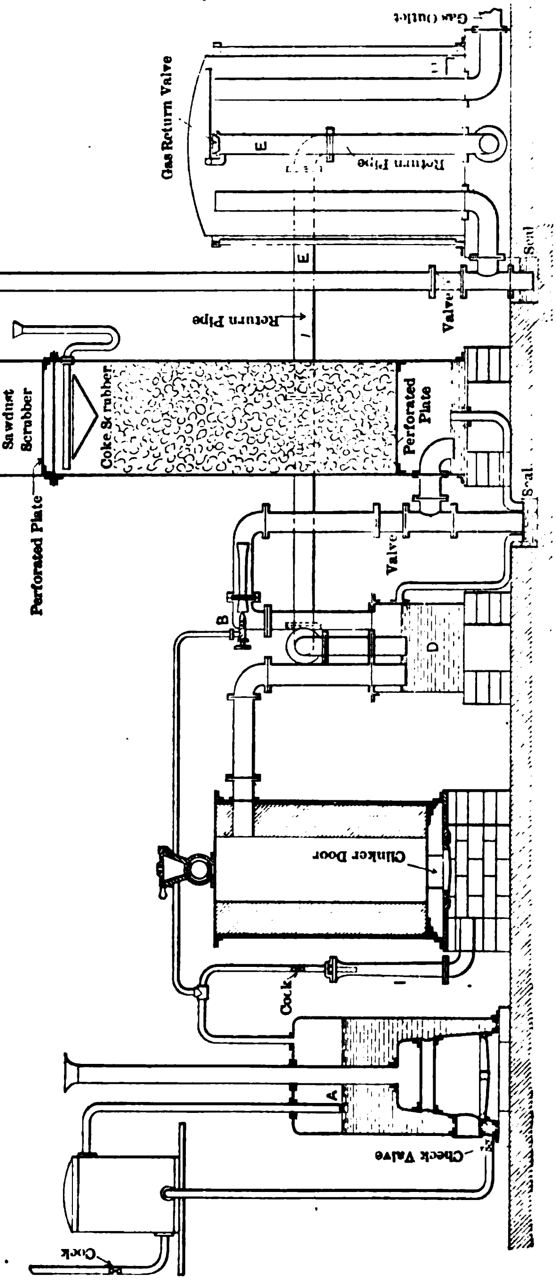


FIG. 432.—The Wile automatic-pressure producer gas plant.

Hydrogen, H	7.0 %
Monoxide of carbon, CO	27.6 %
Methane or Marsh gas, CH_4	2.0 %
Carbonic acid, CO_2	4.8 %
Nitrogen, N	58.6 %
	100.0

One cubic foot weighs 0.0748 pound and density 0.93 (air 1) with a heating value of about 135 British thermal units per cubic foot.

The volumes of air and gas in the charging mixture are proportionately as their heat-unit values; so that, practically, with the low combustible value of this gas, but 1.25 parts of air to 1 part gas is required for perfect combustion. This requires a like portion of the inlet-ports and supply-pipes and their change to these proportions in motors built for illuminating and other high thermal gases and vapors. The size of the motor for a given horse-power is also subject to the heat value of the combustible used for power. Hence a gas-engine of given dimensions, using illuminating gas of 700 heat units per cubic foot and in proportions of 6 air to 1 gas, will represent a power of $\frac{700}{7} = 100$ heat units per cubic foot of the mixture fed to the engine; while with suction-gas of 135 heat units, the power will be represented by the charging mixture, 1¼ air to 1 gas $= \frac{135}{2.25} = 60$ heat units per cubic foot of the mixture fed to the engine. These differences should represent inversely the relative volumes of the cylinders for equal power.

In Fig. 432 we illustrate an automatic-pressure producer-plant as built by the Wile Power Gas Company, Rochester, N. Y. The automatic producers represent a considerable advance in the producer gas industry, combining the best features of ordinary suction and pressure producers.

An important feature of the automatic type is that the producer is under suction while the gas is supplied to the engine under a constant pressure of a few inches of water in a small regulating gas-receiver. The producer is fitted with a regulator which automatically controls the amount of gas generated and at the same time ensures a uniform quality of gas which is essential for the steady working of any gas-engine. This producer uses any class of fuel which is available and makes the gas automatically as it is required.

When the demand ceases, the aspirator, instead of drawing air and steam through the fuel-bed and generating new gas, circulates the gas already made. As only the amount of steam and air enter the fire which is necessary for making gas, the fire in the producer has a uniform temperature and only gas of uniform quality is made. Pressure gas plants, the main characteristics of which are steam-boiler and gas-holder, which can also be used for power or heating, or both, obtain their draught by means of steam raised to a pressure of about 40 pounds in a small steam-boiler, which is led through an injector placed at I (Fig. 432), and enters the generator mixed with air and making the gas as above described, which then passes through the hydraulic seal-box and the scrubbers to the gas-holder. This position of injector for making gas is very satisfactory when the load is constant, but difficulty is experienced in making gas of uniform quality under varying loads, and to meet this demand an improved pressure-plant has been designed, in which the injector is placed at B (Fig. 432), above the water seal-box D, and a return pipe E comes from the gas-holder to the seal-box D. It will be recognized that with the injector at I gas will constantly be manufactured unless provision is made for cutting off the steam-jet when the gas-holder is full and no further gas is required. This is commonly done by a chain arrangement which runs from the gas-holder to the injector and comes into action when the gas-holder is at its top position. This stoppage of the blast tends to cool the fire, and as the gas-holder falls, the steam-jet will again come into action at full force, and a further cooling will take place, due to the impingement of a full blast of steam. These wide variations of blast lead to such variations in the temperature of the furnace that at times operations must be stopped, so as to blow up the fire, the gas-holder shut off, and the poor gas made thrown away. A large gas-holder which the engine can draw upon to keep going is therefore necessary, and also the constant attention of a man.

In the plant shown in Fig. 432 the injector, with its forty pounds of steam pressure placed at B, is always acting on the water-seal D, and owing to the fact that the return-pipe E leads back to the seal, the injector is either acting upon the gas-holder when the gas-holder is at its top position and the gas return-valve open, or is acting upon the generator when the valve is shut and the gas-holder down. The tendency of the injector is to act on the gas-holder, as there is less resistance to the pipe than from the generator. Steam and air at atmospheric pressure are led through the saturator I into the open ash-pit, and the mixture can only enter the generator when the injector is drawing upon it, and only in the quantity required. An even temperature of fire in the generator is obtained, and a uniform quality of gas is made automatically with varying loads. The gas return-valve is opened by the catch H when the gas-holder is in its top position, and the gas is then constantly recirculated from the gas-holder to hydraulic box and through the cleaning apparatus. The steam at B aids greatly in cleaning the gas. With this plant the gas-holder, now a regulator, is continually moving slightly up and down near its top position.

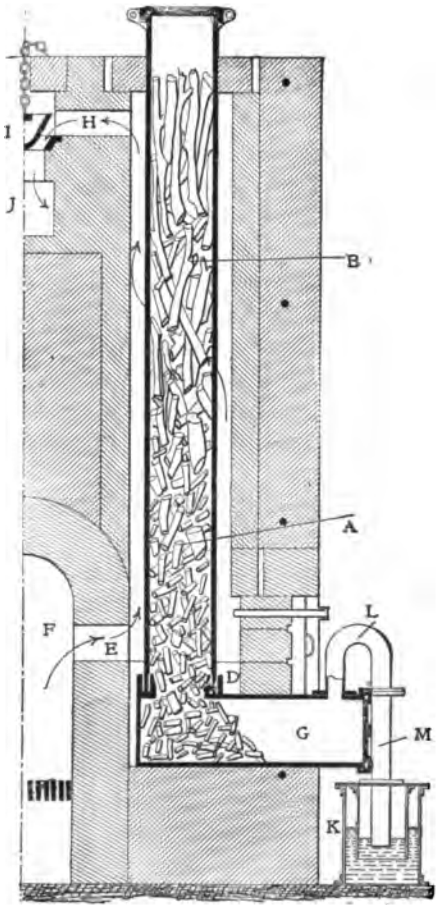


Fig. 433.—Riche distillation producer.

In Fig. 433 we illustrate a wood-fuel gas producer, the design of M. Roché, Paris, France, which brings out the possibilities of utilization of saw-mill waste, slabs, and saw-dust, and the waste of wood-working mills for the production of power gas. It consists of a central furnace in which the fuel charge is burned and which is surrounded by a series of retorts. The fuel used is wood or wood-waste matter, and the products of combustion in the furnace F pass through the flue E and around the retort B. Fuel is fed to the upper part of this retort, which is sealed, and the gas is distilled off by the high temperature maintained. The only exit of the retort is at the bottom, and in travelling down through the retort the gases pass through the lower bed of fuel, which is at a very high temperature, being practically in a state of incandescence, Any condensable gases or vapors in this part of the retort are broken up and fixed so that the gases which pass through the U-shaped pipe L to the holder K are in the condition of permanent gases. When wood is used as fuel the composition of these gases is about 18 per cent. carbonic acid, 22 per cent. carbon monoxide, 15 per cent. methane, and 45 per cent. hydrogen. The calorific value of the gas is about 346 British thermal units per cubic foot. While this is quite high it should be remembered that it is generated by distillation, and is therefore free from nitrogen, which usually forms about 50 per cent. of the volume of producer gas made by combustion, and it also contains a larger proportion of hydrogen. The products of combustion in the furnace F, after circling around the retort, pass out the upper flue H, through the opening in the damper I and out the exhaust-passage J.

In Fig. 434 is illustrated the suction gas plant of the Fairbanks-Morse Company, Chicago, Ill. The plant consists of generator A, a scrubber B, a gas-tank or receiver C, and the economizer or vaporizer D. The generator is fitted with stationary cast-iron grates, and lined with fire-brick up to the gas-outlet. It is surmounted by a coal-hopper or charging reservoir of large capacity, which reduces the frequency of charges. Poke-holes are so located in the top of generator as to permit ramming down any clinker which may collect by the use of inferior grades of fuel. Upon leaving the producer the gases pass through the vaporizer or economizer D, which is constructed with large gas-passages for the purpose of avoiding the objectionable clogging which results with the use of a multiplicity of small tubes of the vertical tubular-boiler construction. From the vaporizer the gas passes to a combined three-way and relief-valve, this valve being so constructed as to either vent to the atmosphere or through the scrubber, and also to serve as an automatic safety valve which will vent gas to the atmosphere in case any excess pressure should accumulate in the system for any reason. Passing from the relief-valve the gas enters the lower part of the scrubber, which is built of unusual height, thereby cleaning the gas thoroughly before its passage to the purifier or engine.

The scrubber is provided with cast-iron grates and a water-pocket in its base, and filled full of coke. A spray-valve or nozzle is located in the center of top of the scrubber and is of such design as to permit carrying full water pressure at the valve itself and control the amount of spray by adjustment of the nozzle. From the scrubber the gases are taken out at the top to prevent carrying an unnecessary amount of moisture to the engine. The gas passes next to a gas-tank or receiver C, which serves to condense any moisture or by-products present in the gas and carry them down

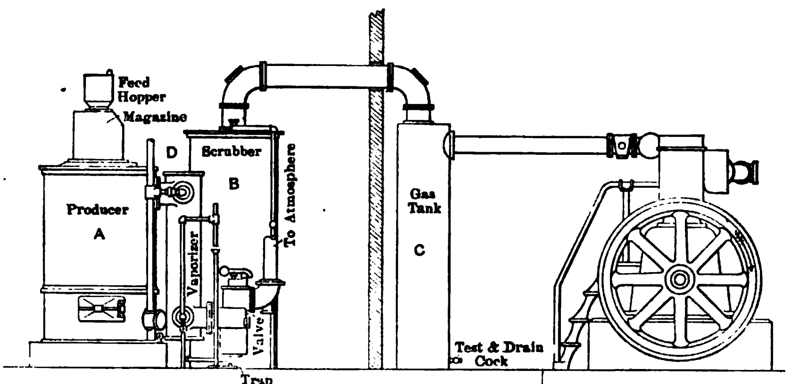


FIG. 434.—Suction producer gas plant.

its side to its base, which is provided with hand-hole openings and cleaning facilities. This enlargement of pipe, or receiver as it is called, also provides sufficient storage of gas immediately adjacent to the engine-cylinder to insure always a full cylinder-mixture and also produces a relatively steady draft through the producer and constant action of the fire. Test-cocks are provided at the three-way valve mentioned and also in the base of the gas-receiver, which make it possible to determine the value of the gas before any attempt is made to put the engine in service. A saw-dust purifier is furnished and installed between the scrubber and engine, whenever the character of the fuel to be used in the producer is of such nature as to make additional cleaning necessary.

Considerable attention has been given to the detail of design with a view to facilitating inspection and cleaning of the various parts and insuring continuous and economical service. All principal piping connections are flange-fitted, having elbows provided with hand-holes to permit of cleaning in both directions. All principal water connections have T's or crosses for the same purpose, and cleaning doors and openings of liberal dimensions are provided in each one of the members. These suction-plants are built in units of from 21 to 150 horse-power each and installed for powers as large as desired. For plants larger than 150 horsepower two or more units are furnished and so piped as to make engines and producers completely interchangeable. Plants of various sizes have been installed which operate continuously 24 hours per day, six days per week, and endurance tests have been conducted which demonstrate that a producer gas installation is in every respect as dependable as the best laid-out steam-plant.

In Fig. 435 we illustrate a German suction gas producer plant of the magazine-generator type, with some peculiarities worthy of note. Reference to the diagram, which represents a section through the plant, will make the matter clear. A is the generator, which is a cylinder of wrought or cast iron with a fire-brick lining. A^1 is a small hand-fan which is attached to the producer, and which is used for starting purposes. B is the vaporizer, consisting of a grilled pipe passing through a water-jacket as shown. Its function is to vaporize the small quantity of water required in the generator. C is a coke-scrubber for cleaning the gas, and D is a gas-box fixed close to the engine. The fire is lighted in the fire-box by means of oil-waste and ordinary kindling. Anthracite coal or coke is put into the generator through the hopper — the fire-box door is closed, the valve E is opened, and the fan A^1 is started. While the fire is being blown up, the smoke and hot gases — which resemble those from a smith's forge — pass through the vaporizer B and escape to atmosphere through the valve E. The passage of these gases heats the vaporizer and forms water-vapor, which is drawn into the bottom of the generator. After about six minutes the gas is tested by a small pet-cock. As it improves in quality the valve E is gradually closed, and the gas is driven through the scrubber, where it meets a stream of water from the rose 1, and so to the engine. There is another test-cock at this point, and as soon as the gas is considered rich enough the valve

E is entirely closed, and the engine is started. The vessels J, J are water-seals for collecting the surplus water from the scrubber.

It is a good practice, where electricity is available, to couple the small blowing fan directly to the spindle of a small electric motor. This is very useful, the cost is small, and it saves labor and gives the engine-driver time to oil up and look round his plant and engine before starting up. It also enables the driver to brighten up his fire from time to time when he is standing by during the dinner-hour or at any other time. For this latter reason the by-pass pipe to atmosphere which is used when the engine is not at work should be made as high as is conveniently possible, so as to

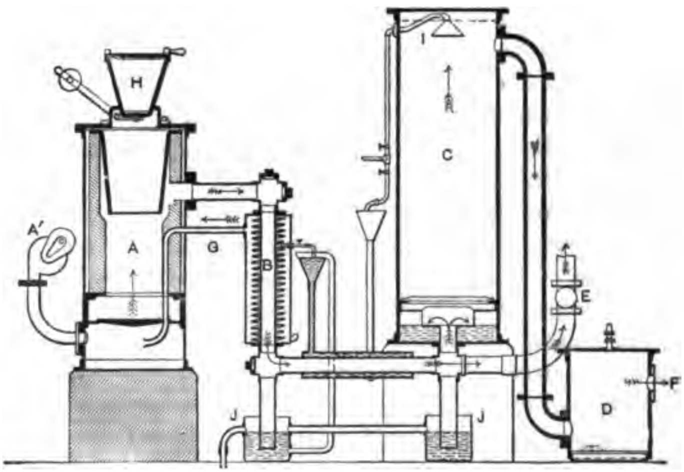


Fig. 435.—Sectional view of suction gas plant.

create a draught and keep the fire alight during the dinner-hour. At some tests made with one of these suction-plants, burnable gas was being produced seven minutes after the fire in the generator was lighted, and the engine was working on its load three minutes later. This may have been exceptional, and as a general rule 15 to 20 minutes from cold is ample for starting purposes. With these plants it is desirable to have a fairly large capacity in the generator-hopper, so that stoking need be less frequent and so that the coal can be warmed and dried before it actually comes into contact with the fire. It is also desirable to have a fairly large capacity in the generator, so that if the fire is dirty, or the coal contains shale, the production of gas does not suffer. The consumption of anthracite coal in a suction-plant is one pound per brake horse-power per hour, but a brake horse-power hour has been obtained on test from 0.6 of a pound of coal, and it seems probable that in future the consumption will be considerably below one pound.

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1. Pressure Systems.—All pressure systems must be located in a special building or buildings approved for the purpose and at such distance from other buildings as not to constitute an exposure thereto.

2. Suction Systems.—(*a*) A suction gas producer of approved make, having a maximum capacity not exceeding 250 horse-power, may be located inside the building, provided the apparatus for producing and preparing the gas is installed in a separate, enclosed, well-ventilated, fire-proof room, with standard doors at all communicating openings.

The installation of gas producers in cellars, basements, or any other places where artificial light will be necessary for their operation, is considered hazardous, and will not be permitted except by special permission of the underwriters having jurisdiction.

(*b*) The smoke and vent-pipe shall, where practicable, be carried above the roof of the building in which the apparatus is contained, and adjoining buildings, and when buildings are too high to make this practicable, the pipe shall end at least ten feet from any wall. Such smoke or vent-pipes shall not pass through floors, roofs, or partitions, nor shall they, under any circumstances, be connected into chimneys or flues.

(*c*) Platforms used in connection with generators must be of metal. Metal cans must be used for ashes.

(*d*) The producer and apparatus connected therewith shall be safely set on a solidly built foundation of brick, stone, or cement.

(*e*) While the plant is not in operation the connection between the generator and scrubber must be closed, and the connection between the producer and vent-pipe opened, so that the products of combustion can be carried into the open air. This must be accomplished by means of a mechanical arrangement which will prevent one operation without the other.

(*f*) The producer must have sufficient mechanical strength to successfully resist all strains to which it will be subjected in practice.

(*g*) Wire gauze, not larger than sixty mesh or its equivalent, must be used in the test-pipe outlet in the engine-room.

(*h*) If illuminating or other pressure gas is used as an alternative supply, the connections must be so arranged as to make the mixing of the two gases, or the use of both at the same time impossible.

(*i*) The opening for admitting fuel shall be provided with some charging device so that no considerable quantity of air can be admitted while charging.

(*k*) The apparatus must have name-plate giving the name of the device, capacity, and name of maker.

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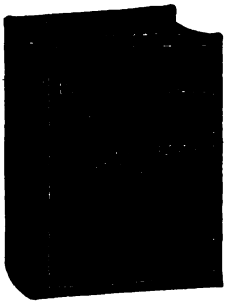

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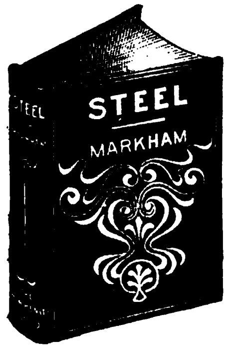

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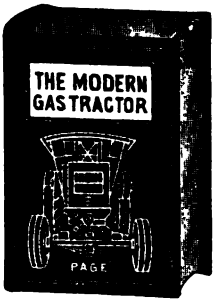

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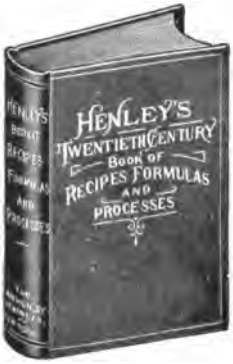

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