VALVES AND VALVE GEARS

VOLUME I—STEAM ENGINES AND STEAM TURBINES
VOLUME II—GASOLINE, GAS, AND OIL ENGINES

BY
FRANKLIN DeRONDE FURMAN, M.E.
Professor of Mechanism and Machine Design
at Stevens Institute of Technology
Member of American Society of Mechanical Engineers

VOLUME II

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PREFACE TO VOLUME II

THE part of this work on Valves and Valve Gears which relates to internal combustion engines is printed as a separate volume in order to meet the requirements of varying courses of study, and the needs of those generally who are interested only in gasoline, gas, or oil engines. This will enable those who wish to study particular cases to do so without being encumbered with literature relating to the steam engine when only matter concerning the internal combustion engine is desired, or vice versa.

Those who wish to make a general study of the methods of operating the several types of internal combustion engines should first study Sections III and IV of Volume I of this work. In these sections it is shown that seven fundamental valve forms underlie all valves in general use, and that six fundamental mechanisms, with their variations, underlie all valve gears whether applied to internal combustion engines or to steam engines. An understanding of how the whole subject of valves and valve gears for all prime movers can be reduced to so few fundamental forms should materially aid in obtaining a grasp of the subject as it is applied, in the present volume, to practical gasoline, gas, and oil engines.

The preface given in Volume I applies to this work on Valves and Valve Gears as a whole, and the comments and acknowledgments there made refer to this second volume as well as to the first. It is, therefore, reproduced on the following pages.

F. DeR. FURMAN

HOBOKEN, N. J., June, 1915.

PREFACE

THIS work has grown from a set of mimeographed notes which were first issued by the author in 1903. Three years ago, in order to serve their purpose better, these notes were issued privately in book form. This book referred chiefly to steam engines and briefly to steam turbines. In the present edition the matter relating to present-day reciprocating steam engines has been considerably increased, the steam turbine has been treated much more completely, and a separate volume has been added on the several types of the internal combustion engine. In all of these editions the subject of valves and valve gears has been treated from the standpoint of mechanism, rather than from that of power, and the chief aim has been to tell in particular, instead of in general, just how the engine or motor is regulated; also to tell how the valves and valve gears may be laid out, with due regard for the laws of mechanism, to give desired control of the steam or gas or other operating agent.

A feature of the present edition is a collection of all types of practical prime-mover valves into a few (seven) fundamental forms, as illustrated on pages 51 and 52, and a grouping of six fundamental types of mechanism from which all practical valve-gear constructions may be formed. These are stated on page 55. As the work grew it became evident that there would not be sufficient time for a student in any prescribed four-year engineering course to study even a minor fraction of all the successful and characteristic valve gears for the steam and internal combustion engines of the present day. The fundamental groups of valves and valve gears referred to above were then prepared with a view to having the student learn them thoroughly, and then take up at least one application of each group as it is applied in the practical valves and gears described in the book. The remaining cases constitute a ready, and it is hoped a reliable, technical reference for students and for all who may be interested in the various phases of the subject.

In order to facilitate the use of this book as a text, it has been divided into many paragraphs which are consecutively numbered in each section. An endeavor has been made to devote each paragraph to either a purely technical or purely descriptive phase of the subject in hand, so that the instructor may readily select and assign paragraphs to bring out either fundamental or applied material to suit any course of study.

The new material which has been added on the subject of the reciprocating steam engine in the present edition includes the semi-plug and high-pressure piston valves of the American Balance Valve Co., the Rice & Sargent-Corliss engine, including the Rites and Sargent governors and directions for setting valves and valve gear, Nordberg valves and valve gears, J. T. Marshall valve gear, Sulzer steam engine valve gear, Williamson steering gear, and several types of the uniflow steam engine. The Curtis "steam-actuated" and "mechanical" gears, the Westinghouse "direct" and "steam-operated" gears, and the De Laval gear have been added to the section on steam turbine. All of the material on the gasoline, gas, and oil engines is newly prepared. The method of using the sinusoidal diagram in laying out the sleeve valve in connection with the Lyons-Knight engine, and of analyzing the kinematic action of revolving engines such as the Gnome and Gyro, are original, although it is quite possible that the manufacturers have laid down work of a similar nature in the development of their respective engines.

The features of the older edition which have been retained in the present work include: the order of presentation of the topics; the numerical marking of the lines of the valve diagrams, in the order of their construction, at the beginning of the course, thus requiring synthetic as well as analytic study; the formula for determining exactly the steam lap from the Zeuner diagram when port opening, lead, and cut-off are given; the introduction of preliminary free-hand problems before taking up the regular drafting-table problems; the combining of the valve ellipse with the steam engine indicator card to determine the steam and exhaust laps, steam- and exhaust-port openings and lead while the engine is in service, or without removing the steam chest cover; the method of determining the width of the cut-off blocks for the Meyer valve; the Corliss valve-gear design; and the condensed arrangement of Auchincloss's method of design of a link motion.

In addition to the above, instances occur throughout the work where the author has been enabled to add to or rearrange the work of others, as a result of considering the subject principally from the point of view of mechanism. In most cases the information that has been made use of as a basis for the present work has been gathered from a wide range of books, periodicals, and conversations, and by far the larger part of it all may be found scattered in duplicate in various forms of record. There are instances, however, where definite credit is due to originators of construction, method, or arrangement, and in all such cases the present writer has cheerfully given such credit in the body of the work where the references occur, so far as he has known that credit is due.

Much of the material in this work has been arranged after extended visits to drafting rooms in which the work in valve gears was being carried on in a practical way, and it is believed that the methods here presented will be found to agree fairly well with general practice. In writing up the descriptions of the practical forms of valves and valve gears the author has received numerous courtesies from manufacturers which are hereby acknowledged. It has been the rule to have the manufacturer of the valve or gear or engine described to pass finally on the accuracy of the description and of the illustrations that appear in this work. Every illustration has been newly prepared for this edition. In order to avoid the use of subscripts in numbering the illustrations and paragraphs as the work of preparation proceeded, and as changes and additions were made, the author laid out the work originally by leaving ten numbers free at the end of each section. It will be found that these numbers have been all used at the end of some of the sections and that none have been used at the end of other sections. The page numbers, however, are in consecutive order throughout the book.

In concluding, I wish to record my appreciation of the assistance and the suggestions that have been received from time to time from Dr. D. S. Jacobus, who was formerly Professor of Experimental Engineering at Stevens Institute of Technology, and from my colleagues, Professors F. L. Pryor, R. M. Anderson, and W. R. Halliday, and Messrs. C. E. Hedden and S. H. Lott.

F. DeR. Furman.

Hoboken, N. J., March, 1915.

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VOLUME II

General Considerations

498. Valves and valve gears suitable for gasoline, gas, and oil engines will be considered in this volume. The valves in nearly all cases are single flat-disk or poppet valves. The slide valves used in reciprocating steam engines are generally not suitable for internal combustion engines, due to the extremely high temperatures and pressures with consequent possible irregular expansions of adjoining parts and scoring of valve and valve-containing surfaces. Without the slide valve, and with different timing problems from those involved in the steam engine, the ordinary forms of valve gear used on the latter engine cease to be practical on the internal combustion engine. On these accounts the valve gears for gasoline, gas, and oil engines have become largely a matter of cams, for these may, in general, be so formed and so placed as to give any desired motion at any desired time and for any period of time. Before proceeding further with descriptions of valves and valve gears, several paragraphs will be devoted to brief general considerations relating to gasoline, gas, and oil engines.

499. The operation of gasoline, gas, and oil engines depends on the combustion of fuel within the cylinder. In gasoline and gas engines particularly the combustion is so very rapid as to amount to an explosion, while in the Diesel type of oil engine the combustion takes place during an appreciable period of time. In both cases the valves and valve gears proper regulate the quantity or quality, or both, of the fuel, while auxiliary means are employed in igniting the fuel, these auxiliaries frequently containing a mechanical device that will regulate the time of ignition. Other mechanical devices used in connection with gasoline engines and light-oil engines are vaporizers or carburetors which permit the liquid to be vaporized and mixed with air, or, as it may also be stated, permit the air to be enriched by the gasoline vapor to such a degree as to form an explosive mixture.

SECTION IX.—GENERAL CHARACTERISTICS OF INTERNAL COMBUSTION ENGINES

Methods of Operation

500. The general distinctions between gasoline, gas, and oil engines, briefly stated, are:

The gasoline engine uses gasoline vapor.

The gas engine uses, in general, a fuel gas that is obtained by the burning of coal or other solid fuel in gas producers specially built for the purpose; and in sections where available, natural gas or the gaseous by-product of the blast furnace is used.

The oil engine uses heavy oils which are highly atomized as they are introduced into the engine cylinder.

Other vapors than gasoline, such as kerosene, naphtha, and benzine of the petroleum group, benzol, which is distilled from coal, alcohol, etc., have been and may be used. Engines using kerosene in the atomized form are classed with the oil engines.

501. All of the above forms of fuel engines are known as internal combustion engines, and the operation in all cases depends on a proper mixture of the fuel with atmospheric air both as to quantity and pressure. In both the gasoline and gas engines and in many forms of kerosene engines the fuel and the air are drawn into the cylinder together by the suction of the piston, or are forced in under slight pressure, and *this mixture* is then compressed by the piston and finally, at about the maximum point of compression, it is ignited by an electric spark or by a hot tube or surface or flame. In some forms of oil engines pure air only is admitted into the cylinder during the suction stroke or intake, and when it is compressed the fuel spray is injected by pressure and the mixture thus formed is ignited generally by a hot tube or surface. In the Diesel type of oil engine the characteristic intake feature is that the pure air, which is all that is admitted during the suction stroke, is compressed to the exceptionally high pressure of 450 to 500 pounds per square inch. This high compression of pure air generates a temperature of about 1000° F., which is entirely sufficient to ignite the oil spray when it is introduced, without the aid of any hot tube or specially heated surface or spark of any kind.

Classification of Internal Combustion Engines

502. All internal combustion engines are often referred to simply as gas engines, the basis for the term lying in the fact that at the instant of ignition all fuel charges are in a gaseous or atomized form regardless of whether pure gas, gasoline, or oil are the commercial

forms of the fuel used. It is safer, in general, to adopt the nature of the fuel as a basis for naming the engine, and we then have gas, gasoline, kerosene, and oil engines, etc. In Europe the gasoline engine is generally referred to as the petrol engine.

503. A more scientific classification of internal combustion engines would be based on the condition of the fuel mixture during the period of ignition, rather than on the nature of the fuel itself at any phase, the engines being classified, in studying heat efficiency, as:

Explosion engines if the ignition takes place at constant volume.

Pressure engines if the ignition takes place at constant pressure.

Constant temperature engines if the ignition takes place at constant temperature.

504. Practically all internal combustion engines of the present day have ignition at constant volume, a distinct exception being the Diesel oil engine, in which the ignition under some forms of construction and conditions is practically at constant temperature, but in most cases it takes place at practically constant pressure. Whichever characteristic is shown during ignition depends practically on the quality of the fuel and the rate of feeding it, and theoretically upon a proper relation between temperature, pressure, and volume. The Brayton engine, which will be referred to later, was built on the constant pressure principle, but it is not now manufactured.

Four-Stroke and Two-Stroke Engines

505. All gasoline, gas, and oil engines may be and are built to run on either one of two distinct principles of operation and construction. The first and most efficient form of construction is the one which is widely known as the "four-cycle" engine, although the designation of "four-stroke cycle" is also used. This latter term is more descriptive but is somewhat cumbersome, and the shorter term four-stroke engine will, therefore, be used. This type of engine, which is illustrated in Fig. 313, operates on the Beau de Rochas cycle, or the Otto cycle, the former being the name of the man who first set forth the principles of the heat cycle involved, in 1862, and the latter being the first to successfully apply these principles in an engine about fifteen years later.

The second form of construction is generally known as the two-cycle engine, although here again the term two-stroke cycle better expresses the meaning for the reason that it avoids confusion of mechanical cycles and heat cycles. The term two-cycle or two-stroke really means that there are two strokes of the piston for one heat cycle of the engine, or for one explosion in the engine cylinder. Two different forms of two-stroke engines are shown in Figs. 314 and 315.

506. In the four-stroke engine the piston makes four complete strokes while the engine passes through one heat cycle, as may be seen by examining Fig. 313. On the first down stroke, the fuel mixture is admitted through the mechanically operated valve in the intake pipe; on the first up stroke the gas is compressed; at or near the top of the stroke it is ignited and then combustion and expansion take place during the second down stroke; and on the second up stroke the burned gases are exhausted through the mechanically operated valve in the exhaust passageway and the heat cycle is completed. It follows, in this type of engine, that there is only one power stroke in every four piston strokes and that the engine must make two full revolutions for each driving impulse. In order to have a driving impulse on the engine shaft for each stroke of the piston, as is obtained in the ordinary single-cylinder steam engine, it is necessary to have four cylinders of the four-stroke internal combustion type side by side on the same shaft. The valves shown at F and E are independently operated by cams at the desired times, which are not generally at the exact top and bottom of the stroke, as is shown, for simplicity, in Fig. 313. In some four-stroke engines, only air is admitted on the first down stroke, the fuel being injected through a separate valve or spraying device at or near the end of the first up stroke.

Two-Stroke Two-Port and Two-Stroke Three-Port Engines

507. In the two-stroke engine there are two forms of construction known respectively as the "two-cycle two-port" engine and the "two-cycle three-port" engine. The former, Fig. 314, requires an automatic or check valve, while the latter, Fig. 315, requires no valve at all and, therefore, is sometimes called the valveless engine, on which account it has been referred to, also, as the simplest internal combustion engine.

508. Before taking up a detailed description of two-stroke engines it must be considered in connection with the two-port type, Fig. 314, that as the piston moves from near the bottom of its stroke to the top it acts as a suction pump and automatically lifts the suction valve at G and draws in a charge of the fuel mixture, filling all of the space under the piston, including the crank case. The crank case in the two-stroke engine must be constructed to be air tight when no separate air compressor is used. Although there are no valves at H and K, these ports are tightly closed by the piston

itself as soon as it passes them on the way up. On the down stroke of the piston the gas previously drawn in is compressed in the crank case and in the passageway *F H*.

509. The explanation of the two-stroke engine may now be taken up. As the top of the piston uncovers the exhaust port *K*, Fig. 314, on the down stroke just described, the crank being then at *A R*, the burned gases from the previous explosion are allowed to escape, and an instant later when the crank is at *A S* the port at *H* opens and the compressed gas is forced in, driving before it the bulk of the burned gas remaining in the cylinder. The areas and positions of the admission and exhaust ports at *H* and *K* and the timing angles for exhaust and admission are so designed that little if any of the fresh mixture has time to escape through *K* before the piston is down to the bottom of its stroke and back again ready to close the port *K* and to further compress the fresh charge which is now trapped in the cylinder, to a much higher pressure. As the piston nears the top of the up stroke the highly compressed mixture is ignited. Expansion then takes place during the greater part of the down stroke, when exhaust occurs and the cycle is completed. In this type of engine, then, there is one power stroke in every two piston strokes, or, stated in another way, there is a driving impulse for each revolution.

510. When a two-stroke engine is made with three ports as shown in Fig. 315 at *H*, *K*, and *G*, the piston itself acts as a valve in opening and closing the intake, as well as the compressed mixture admission and the exhaust ports. The action of the three-port type is the same in every particular as the two-port type explained above, except that the period of intake is definitely marked from *A V* to *A W* in the former, while in the latter, Fig. 314, it depends on the suction and begins somewhat later than at the position *A U*, and ends at or near the top of the stroke, depending on the size of the crank case relatively to the piston displacement, the rapidity with which a sufficient vacuum is formed to allow the check valve *G* to be lifted, and the inertia of the gas.

511. In some forms of two-stroke gas and gasoline engines and in oil engines the preliminary compression of the fuel mixture is done in separate compressors instead of in the crank case; and also in some forms of oil engines this preliminary compressed charge is admitted through independently controlled valves at the top of the cylinder instead of through a port, as at *H*. These two expedients give, respectively, more reliable fuel charges and better scavenging of the burned gases, and thus contribute to a marked increased efficiency in the larger sizes of the two-stroke engine.

Pressure Diagrams, Useful Work, Inertia of Gases, Flame Propagation, Scavenging, and Sizes of Valves, Ports, and Cylinders

512. A pressure diagram shows the pressure in the engine at every phase of the heat cycle. An example of such a diagram is illustrated at $J\,L\,M\,N\,O\,Q\,J$ for a four-stroke engine in Fig. 312. In Fig. 313 the admission, compression, explosion, and exhaust strokes are all shown, as stated in paragraph 506, as full strokes and as though each began and ended exactly at the top or bottom of its stroke. Such a simple arrangement would require not only that the ports and valves be extremely large, but that the velocity of the gases be infinite. To allow for the actual time required for flow of gases and for reasonable port and valve sizes the several events, such as admission, compression, etc., must begin approximately as indicated by the crank positions in Fig. 312, although these may vary greatly with different forms of design and proportions.

513. The pressure diagram, Fig. 312, may now be followed, considering that the piston is just completing the exhaust stroke and that the exhaust valve is still open. The exhaust gas is then passing out at E and is a trifle above atmospheric pressure, as is shown by the vertical line passing through J. This line is just above the atmospheric line I. As the piston descends, the exhaust valve closes just after the crank has passed top dead center, and during this very short interval the piston has been moving down a small amount and reducing the exhaust pressure. An instant later the intake valve opens and the pressure line, if it has not already dropped to a point below atmosphere, now does so, and the fuel gas is drawn in at practically constant pressure, as indicated by the line to L.

514. In order to get the maximum amount of fuel into the cylinder, advantage is taken of the inertia of the incoming fuel charge by leaving the intake valve open for a short time after bottom dead center. As the piston rises from the bottom of its stroke the gas pressure rises, crossing the atmospheric line above L and then continuing, after the intake closure, to rise until the desired compression of the charge is reached at M. If the intake valve is closed too late, the compression line $L\,M$ will have a portion coinciding approximately with $Q\,J$ and some of the charge will be driven back through the intake port.

515. If the flame propagation and combustion were instantaneous, the ignition would take place at M and the pressure line would rise directly to N. Since both of these require a small fraction of the stroke time, which is particularly noticeable in large sizes and at

high speeds, it is customary to ignite the charge, when running at full speed, before the dead center, and even then the line $M\ N$ usually has some slant away from the end of the card as it rises. The expansion of the burned gases is shown at $N\ O$. At O the exhaust opens before the piston reaches the bottom, so as to give the burned gases time to escape without using any more of the engine power than is

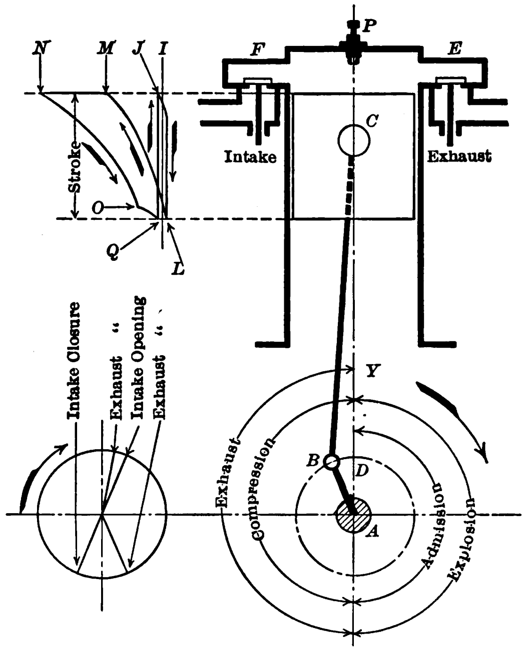


FIG. 312 FIG. 313

FIGS. 312 AND 313.—FOUR-STROKE ENGINE, SHOWING NAMES OF STROKES, PRACTICAL CRANK POSITIONS FOR INTAKE AND EXHAUST, AND PRESSURE DIAGRAM

necessary to drive them out. The pressure necessary for exhaust is considered constant at a point slightly above atmosphere, as shown by $Q\ J$ in Fig. 312.

516. The pressure diagram is also a measure of the work done by the engine, for if the back-pressure ordinates of the diagram, which tend to stop the engine, are subtracted from the forward or driving pressure ordinates and the average or net ordinate found, it may be multiplied by the length of the diagram as representing the length of the piston stroke in feet, and we have the foot-pounds of useful work done during the heat cycle. Knowing the number of heat cycles per minute, which are equal to one-half the number of

revolutions for four-stroke engines, and equal to the number of revolutions for two-stroke engines, we have foot-pounds per minute, or a measure for the horse-power. In a four-stroke engine *one continuous diagram* shows the work necessary: 1st, to drive the exhaust gases out, *Q J*; 2d, to draw the fuel charge in, *J L*; 3d, to compress the fuel charge, *L M*, and these combined must be taken from the work accomplished by the explosive and expansive charge, *N O Q*. These lines show the pressures from which the work may be computed. In addition, the work necessary to operate the valve gears must be deducted from the useful work shown by the diagram to compare it with the valveless two-stroke engine. The pressure diagram of a four-stroke engine is a continuous one, because all the operations of the heat cycle are carried out in the engine cylinder. In practice one can rarely distinguish between the back-pressure and suction lines, *Q J* and *J L* respectively, because an indicator spring which is strong enough to record the explosion pressure is too strong to distinguish between these lines. On account of inertia of the moving parts of an ordinary engine indicator, an optical indicator or manograph is used at speeds above 800 r.p.m. The only moving part of the optical indicator is a small mirror which is moved in one direction by an amount proportioned to the cylinder pressure and in another direction at right angles by an amount proportioned to the piston motion. The mirror projects a ray of light on a ground-glass screen or on a photographic plate.

517. The pressure diagram for a two-stroke engine is made up of two parts because the actual work is done at one place, in the engine cylinder, while the work necessary to get the fuel into the cylinder is done at another place, usually in the crank case in small engines, and in a separate compressor in larger engines. A pressure diagram is shown at *N O Q M N* and *J Y L Z J*, Fig. 315. The former is the engine-cylinder diagram and the latter the crank-case diagram. Explosion occurs at *M N* as explained for the four-stroke diagram, expansion from *N* to *O*, exhaust through the port *K* from *O* to *Q* to *X*, and cylinder compression from *X* to *M*. In the crank-case diagram the intake port *G*, for the phase shown, is open, and as the piston is just finishing its up stroke the fuel is still entering and it is considered that it continues to flow in at the top of the stroke, thus establishing the point *J* a trifle below the atmospheric line *I* as the starting point of the crank-case pressure diagram. As the bottom of the descending piston crosses the intake port *G*, the fuel charge in the crank case is compressed along the line *J Y* until the top of the piston reaches the cylinder-admission port *H*, when the crank-case pressure is released

into the cylinder which is now open to exhaust. The top of the exhaust port is always a little higher than the top of the cylinder-admission port in two-stroke engines.

518. The crank-case pressure, therefore, falls quickly along $Y\,L$ and a corresponding impetus is given to the exhausting burned gas.

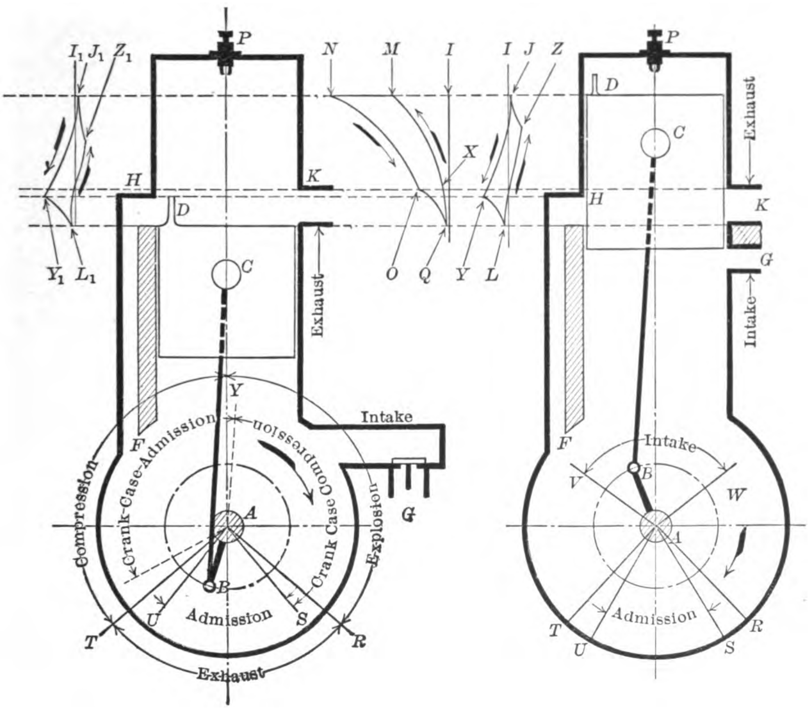


FIG. 314.—TWO-STROKE TWO-PORT ENGINE

FIG. 315.—TWO-STROKE THREE-PORT ENGINE

BOTH FIGURES SHOW CRANK POSITIONS FOR INTAKE AND EXHAUST, AND PRESSURE DIAGRAM

A deflector, D, is placed on the piston head, and when in front of the cylinder-admission port H it drives the entering gas up behind the outgoing gas and thus more completely expels it. This process of expulsion of the burned gas is called scavenging. It will be seen that a close adjustment of port sizes and positions is necessary in designing in order that too much fuel may not escape in this process, or that too much burned gas may not be allowed to remain. In larger two-stroke engines pure air from a separate compressor is sometimes used for scavenging, while the fuel is admitted at a later phase. As the piston rises from the bottom of its stroke the pressure in the crank case may fall slightly until the top of the piston passes the top of the port H when an increasing vacuum will be formed in the crank case, and this will continue until the suction or vacuum line reaches Z. Then the bottom of the piston passes the bottom of the intake port G and the fuel charge rushes into the crank case as shown by line $Z\ J$. From this consideration of the crank-case diagram it may be seen that the crank-case volume must be designed to accord with the piston displacement in order to secure a desired maximum vacuum value. If this value is placed too high there will likely be leakage at the main-shaft bearings if not at the crank-case cover flanges.

519. The pressure diagram for the two-stroke two-port engine will be the same as for the three-port engine so far as the engine cylinder portion $N\ O\ Q\ M$ is concerned. It is not, therefore, repeated in Fig. 314. The crank-case diagram may also remain the same if the crank-case valve at G, Fig. 314, is operated mechanically so as to lift at the same time as the piston uncovers G in Fig. 315. If, however, G, Fig. 314, is an automatic check valve, as it generally is, it will not lift until sufficient vacuum is created in the crank case. This vacuum cannot begin to build up until the top of the piston passes the top of the port H; and some little time after this, depending on the proportions of the engine and the engine speed, the valve will be lifted from its seat as indicated at Z_1 on the vacuum line $L_1\ Z_1$ of the pressure diagram.

Comparison of Power, Speed Variation, Carburetor Construction and Adjustment, and Reversibility in Four-Stroke and Two-Stroke Engines

520. Comparing the four-stroke and the two-stroke engines the latter are at a disadvantage in having shorter times for exhausting and for charging, as may be noted in comparing Figs. 312 to 315. This gives a poorer explosive mixture. The two-stroke motor, however, has the advantage of simplicity of construction, cams, cam rods, and guides being unnecessary in all cases, and springs and valves being unnecessary in the three-port type. The two-stroke engine also has the advantage of a more uniform turning moment per cylinder due to an explosion each revolution. Its revolutions per minute, however, range from two-thirds to seven-eighths of a four-stroke engine. Its power is not double that of the four-stroke cylinder of the same size, as is sometimes assumed, the reason being primarily that there is not so full or so rich a mixture at each explosion as in the four-stroke engine, chiefly because of the larger percentage of burned gas that remains in the explosive charge. A further loss of

power results from escape of live mixture, first through the main-shaft bearings in the crank case and, second, through the exhaust port when the charge is being admitted to the cylinder. These two losses, especially the latter, may be practically avoided in well-proportioned and designed engines. The crank-case clearance volume in two-stroke engines is generally about twice the volumetric displacement of one stroke of the piston, measured at the time when the piston is at the bottom of its stroke.

The two-stroke engine is not capable of a wide and economical variation of speed. It is at its best at or near the higher ranges of two-stroke engine speeds. At low speeds too much time is available for the entering charge and some of it may escape directly into the exhaust. If the mixture is throttled there is not sufficient volume to scavenge the cylinder or to give satisfactory compression and the undue proportion of burned to fresh gas leads to misfiring.

The two-stroke engine is not capable of such high rotative speeds as the four-stroke because at a certain limit there is not sufficient time for admitting and exhausting the gases.

521. The carburetor adjustments for the two-stroke and four-stroke engines are different because the latter on its suction stroke produces a higher vacuum in the cylinder than the former does in the crank case. For this reason the level of the gasoline in the carburetor for the two-stroke engine must be more nearly on a level with the needle-valve opening, and this valve itself should have greater opening than on the four-stroke engine. With such a responsive gasoline supply it is necessary to more carefully control the air supply in a two-stroke engine to prevent back-firing. Back-firing results when the mixture is such that it supports a flaming combustion all the time that the piston is moving down until the admission port H, Fig. 315, is opened, when the flame ignites the fresh charge and explodes it in the passageway $F\ H$ and in the crank case. Since the four-stroke engine is better adapted for varying rotative speeds and is so used, especially in automobile gasoline engines, the carburetor for such an engine must be further considered. It is more complicated than the one for the two-stroke engine, principally because of the fact that a change of air velocity at N in Fig. 320 will not draw off a proportional amount of gasoline through the needle-valve nozzle N. Doubling the air velocity at N increases the gasoline flow more than three times under steady flow. This means that a simple carburetor would supply too little gasoline at low speeds and too much at high speeds to a four-stroke engine. Different

carburetors seek to allow for this by separate or automatic adjustment in which the air openings or gasoline opening or both are changed for varying speeds.

522. The two-stroke engine has an advantage in that it will run backward as well as forward, as may be seen plainly by noting in Fig. 315 that the crank positions for the periods of intake into the crank case, the admission into the cylinder, and the exhaust, are all symmetrical with respect to the engine center line, and, therefore, that the direction of turning is immaterial. The symmetry of the crank positions just mentioned is invariable and must always be so, because the ports are all cast in the cylinder wall and the piston must move equal distances both down and up when uncovering and covering the ports. This is true, of course, only for the ordinary engine in which the cylinder center line passes through the axis of the shaft. Engines with offset cylinders would not run equally well in both directions.

In the four-stroke engine the cams and valves are set to operate at the crank positions shown in Fig. 312, and these, it will be noted, are not symmetrical, and could not, therefore, be the working crank positions for an opposite direction of turning. Small four-stroke engines always turn in one direction, any desired reversal in the drive being obtained by a separate mechanical reversing gear outside of the engine itself. These gears include sliding toothed wheels, planetary wheels, friction disks, etc., and in addition, in motor boats, rotating propeller blades. In large four-stroke engines the engine itself is reversed by building and manipulating the valve gear so that the valves may be made to operate at different times. These times would be approximately those indicated in Fig. 312 if the four crank positions were moved to symmetrical positions across the vertical center line.

Characteristic Features in the Design of Internal Combustion Engines

523. Several considerations relating to characteristic features in the internal combustion engines are:

That the explosion on the piston acts much as a hammer blow, except in the Diesel engines, and must be so treated in proportioning the gudgeon or piston pin, the crank pin, and the engine bearings;

The retardation of the piston in the four-stroke engine is cushioned at the ends of its stroke only once in each four strokes by compression of the gases, thus throwing the retarding effort solely on the pins and bearings most of the time;

In the two-stroke engine the connecting rod is always in compression and the piston is cushioned on each up-stroke;

The piston in both the four-stroke and two-stroke types serves also as the crosshead, and the cylinder as the crosshead guide, while the wear due to working pressure must come always on the same side of the cylinder;

The engine must run at high speeds and either carry heavy flywheels or be built in high-multiple cylinders in order that the infrequent driving efforts (explosions), as compared with the ordinary double-acting reciprocating steam engine, may give a smooth and uniform running of the engine;

The high speeds call for a light-weight construction for the piston and the connecting rod for a given engine power, in order to better balance the engine and reduce vibration;

High speeds contribute to light-weight construction because for a given engine power the stress on the engine parts will be less per impulse as the impulses per minute become greater in number;

The high speeds produce frequent and high sliding or rubbing velocities between the piston and cylinder wall and these in turn would produce rapid wear if the sliding piston did not have such a large sliding surface, thus producing a low number of pounds-per-square-inch bearing pressure, and if lubricating oil were not generously and constantly applied; and finally,

The high temperatures of combustion produce expansion and sometimes distortion of engine parts, including the valve gear. All of the above must be allowed for in internal combustion engine design.

524. Three distinct forms of cylinder-head construction are in use in internal combustion engine design: First, the cylinder with the admission valve on one side and the exhaust valve on the other side of the cylinder. This is known as the T-head and is illustrated in Fig. 341; second, the cylinder with the admission and exhaust valves on the same side of the cylinder, the admission valve usually being directly above or alongside the exhaust. This is known as the inverted L, or simply as the L-head, and is illustrated in Fig. 377; third, the cylinder with the admission and exhaust valves in the cylinder head directly above the piston. Continuing the use of letters to identify the form of cylinder-head construction, this form may appropriately be termed the I-head. It is illustrated in Fig. 362. Each of these forms requires a different consideration in planning the clearance volume of the engine, in the heating and cooling effects, in the timing of the ignition, and in the matter of rapid and effective flame

propagation. These considerations affect the larger sizes of engine cylinder much more than the smaller sizes.

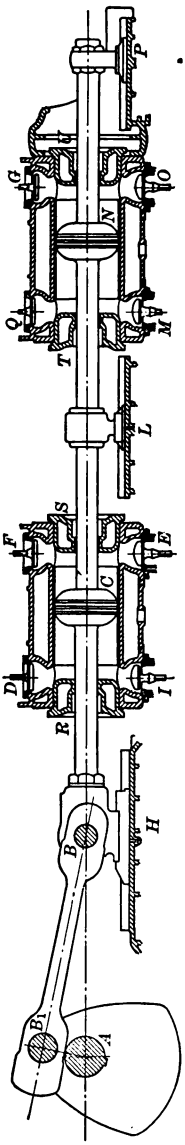


FIG. 316.—ALLIS-CHALMERS DOUBLE-ACTING TANDEM FOUR-STROKE GAS ENGINE

Double-Acting Internal Combustion Engines

525. Double-acting internal combustion engines are manufactured on both the four-stroke and two-stroke plans, the principles remaining the same but the details of construction varying to suit the changed conditions. For example, a double-acting four-stroke engine would differ in its essential features from Fig. 313, in that the connecting rod *C B* would become a piston rod with the end *B* guided by a crosshead, as in a steam engine; the piston *C* would be changed to a more solid form of construction; and another set of valves similar to *F* and *E* and a cover plate would be placed at the lower end of the cylinder. With these changes a working impulse would be obtained during each revolution in a four-stroke cycle engine, while if two such engines were placed tandem, as done on the Nürnberg engine, manufactured by the Allis-Chalmers Co., a working impulse would be obtained on each stroke of a four-stroke engine. Such an engine is shown in Fig. 316.

526. Double-acting two-stroke engines require also that the connecting rod *C B*, Fig. 314, should be changed to a piston rod with a crosshead guide at *B*, and that the piston should be made of a solid or enclosed type of construction. Such an engine is the Koerting "two-cycle double-acting," manufactured at one time by the De La Vergne Machine Co., and illustrated diagrammatically in Fig. 317. The exhaust from each end of the cylinder is effected through a line of exhaust-port openings cast midway in the cylinder. With this construction it will be noted that it is necessary to make the length of the piston equal to the length of the stroke minus the length of the exhaust openings. Instead of first compressing the mixed charge of gas and air in the crank case, as in the single-acting two-stroke engine, the two

elements are separately compressed in auxiliary cylinders and the air charge admitted first to help clear the cylinder of burned gases during exhaust, and later its admission is accompanied by the admission of the gas in a proper proportion, and both are under sufficient pressure to effect their entrance in time for compression on the return stroke. This engine, with a single cylinder, and piston, gives

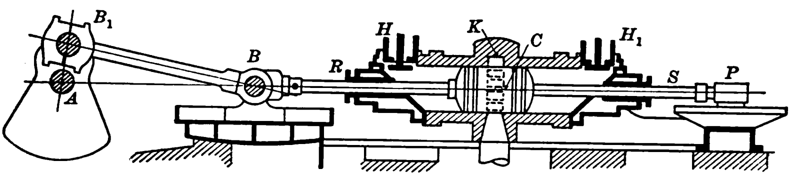


FIG. 317.—DE LA VERGNE-KOERTING DOUBLE-ACTING TWO-STROKE GAS ENGINE

a working impulse on each stroke the same as an ordinary steam engine.

FUEL AND FUEL APPARATUS

527. Before taking up the various types of valves and valve gears proper, an elementary understanding, at least, should be had of the devices used to change over the fuel as it is obtained commercially, into the condition in which it is used at the instant of ignition. In some engines, notably the two-stroke gasoline engine, these devices might be considered as the only valve gear the engine has, aside from the valve action of the working piston itself.

Gas Engine Fuels, Producers, and Gasifiers

528. In gas engines the commercial fuel is coal, coke, wood, peat, or other fuels which are gasified in various forms of gas producers built specially for this purpose. In these gas producers, the gases in the fuel are not only driven off, but the carbon in the fuel itself is gasified, leaving only the ash of the commercial fuel. In addition to producer gas, obtained as described above, illuminating gas, oil gas, coke-oven gas, blast-furnace gas, acetylene gas, and natural gas may all be used directly as gas engine fuel, although in some cases, particularly in blast-furnace gas, it is necessary to introduce a separating or cleansing and "scrubbing" process in order to rid the gas of impurities, such, for example, as dust, foreign vapors of metals or substances which are injurious to the continued operation of the gas engine. The various forms of gas producers, oil gasifiers, and gas cleansers and scrubbers are too numerous and too far removed from

our present subject of valves and valve gears to be satisfactorily included here, and those interested in this branch of the subject will find them quite fully treated in books devoted especially to them, and also in a number of books on gas and oil engines.

Gasoline Carburetors and Mixers

529. In gasoline engines the commercial gasoline must be vaporized and mixed with air in such proportion as will produce an explosive mixture. This vaporizing and mixing is done in different forms of devices known variously as vaporizers, mixers, and carburetors. The carburetor, built on the principle shown in Fig. 320, is most generally used and takes its name from the fact that the pure air which passes through it is carburized by having mixed with it the hydrocarbon vapor of the gasoline.

530. Several methods of carburizing the air which goes into the engine are shown diagrammatically in Figs. 318, 319, and 320. In each of the Figures a vessel is shown containing liquid gasoline *G* as commercially obtained. In Fig. 318 a series of absorbent disks, such as burlap, wicking or other material of a similar nature, are shown at

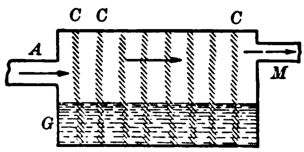


FIG. 318

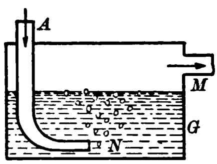


FIG. 319

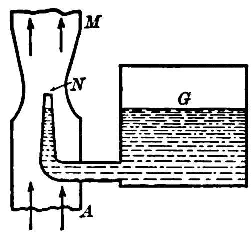


FIG. 320

FIGS. 318–320.—THREE FUNDAMENTAL METHODS FOR SECURING AN EXPLOSIVE MIXTURE OF AIR AND GASOLINE

C C. The pipe *M* connects with the engine cylinder and on the suction stroke pure air is drawn in at *A* and caused to circulate through the absorbent disks which are saturated with the gasoline liquid. Gasoline vaporizes readily when in contact with air under all conditions, and in the present illustration the air is thoroughly impregnated with gasoline vapor and is ready for use in the engine when it emerges at *M*. In Fig. 319 air is drawn through the pipe *A* and it emerges in bubbles at the nozzle *N*. As the bubbles rise through the gasoline they become saturated and carry off the vapor. In Fig. 320 the gasoline holder has a connecting pipe or tube with a nozzle at *N* and the tip of the nozzle is a very slight distance above the level of the gasoline in the main part of the holder. As the engine draws in the fuel mixture on the suction stroke, the fresh air is drawn in at A. It passes the nozzle N at sufficient velocity to draw off the liquid gasoline as vapor with which it mixes and passes into the cylinder ready for ignition.

531. Although gasoline vaporizes readily in contact with air, it is necessary to have a certain ratio between the air and gasoline vapor in order to obtain an explosive mixture. This ratio for some crude oil and for kerosene and gasoline is about fifteen pounds of air for one pound of solid fuel for a true explosive mixture. Owing to varying conditions of chemical composition of the fuel, and of pressure and temperature under which ignition takes place, and the necessity of complete combustion of the engine charge under all conditions,

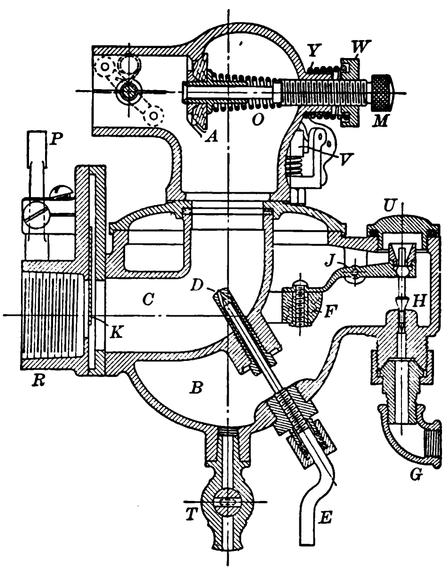


FIG. 321.—SCHEBLER CARBURETOR, MODEL "D"

an excess of air of from 30 to 40 per cent over this ratio is generally used in practice. From the above it will be seen that in order to properly carburize the air used in the engine, it is necessary to add to the simple carburetors shown in Figs. 318–320 a regulating device to control the ratio of air and vapor. To illustrate this, a sketch of a carburetor which has been in service for a number of years in connection with gasoline engines is shown in Fig. 321 and is described in detail in the following paragraph.

Schebler Carburetor

532. In the Schebler carburetor, gasoline from the storage tank enters at *G*, passes up through the opening controlled by the valve *H* and fills the chamber *B*, which surrounds the pipe *C*, until it raises the cork float *F* sufficiently to turn it on its pivot *J* and close the valve *H*, thus automatically shutting off the supply of gasoline when it reaches a certain height in the carburetor. This height is a trifle below the elevation of the needle-valve opening at *D*. The cork float is the shape of the letter U laid in a horizontal position. The engine, on its suction stroke, draws in pure air at *A* down past the needle valve *D*, where it absorbs the gasoline vapor, and the mixture enters the engine through the pipe *R*. The amount of gasoline allowed to enter the mixture is controlled by the needle valve *D*, operated by the handle *E*. The amount of air is controlled by the valve *A*, a certain amount being allowed to pass at all times because of the fact that the air-valve seat is cut away, as shown, for about one-fourth of its circumference. When the engine speeds up and more air is needed, the valve *A* is automatically lifted against the spring *O* by the increased suction due to the higher piston speed. The amount of opening of this valve at high speed is regulated by the compression spring *O* and the adjusting screw *M*. For the purpose of regulating the amount of mixture that goes into the engine a flat-disk throttle valve is introduced at *K*, the speed of the engine being reduced as this valve is closed through the lever at *P*. Sometimes, in starting, a specially "rich" mixture is desired and to obtain this a flushing-pin *V* is provided which pushes down the cork float, opens the valve *H* and allows the liquid to raise its level and overflow through the needle valve *D* into the mixing pipe *C*.

533. There are various types of carburetors, different in details of construction, designed to meet the varying conditions of service. For example, while the carburetor shown in Fig. 321 is used in both four-stroke and two-stroke engines, it is best adapted for the latter service, one of the modifications for the four-stroke engine consisting in keeping the area of the opening at *A* fixed, thus compelling a more nearly constant amount to pass the needle valve *D* under all conditions and allowing any necessary additional air to enter the mixing chamber at *C* after the gasoline has been taken up. This modification makes an entirely different appearing carburetor. Likewise a number of different makers have their own methods of regulating the air and vapor mixture, and some use exhaust gas or hot water to heat the carburetor, but any of them may be readily under-

stood from a good illustration, if the principles just explained and those mentioned in paragraph 521 are taken into account.

Lunkenheimer Gasoline Mixer

534. The gasoline mixer differs greatly in construction from the carburetor. The mixer shown in Fig. 322 is a slight modification of the device as manufactured by the Lunkenheimer Company, and known as a "generator valve." The characteristic feature of this valve is that there is no cork float to control the height of the gasoline. It feeds in directly from the supply pipe designated by the dotted circle *G*, passes through the needle valve at *V* and, when the check valve *S* is lifted on the suction stroke of the engine, mixes with the pure air from *A* and finally enters the cylinder through the opening at *E*. At or near the end of the suction stroke the vacuum which allows the valve *S* to be lifted becomes too weak to continue to hold and the valve, aided by gravity and the non-adjustable compression spring *J*, drops to its seat. The amount of gasoline going into the mixture is regulated by the needle valve, which is set at any desired point and kept there by the flat spring *H* engaging with the milled rim of the hand wheel *K*. The amount of air is controlled by the adjustable stem *D C*, which acts as a stop to the lift of the valve.

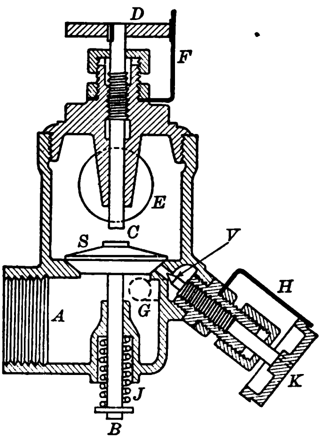


FIG. 322.—LUNKENHEIMER GASOLINE MIXER, OR "GENERATOR VALVE"

Fuel Oils, Vaporizers, and Spraying Nozzles

535. In oil engines the commercial fuel is "fuel oil" or kerosene. Crude oil as it comes from the ground contains a mixture of a large variety of hydrocarbon compounds, the nature of the mixture varying greatly according to locality. The process of obtaining fuel oil, briefly described, is as follows: The crude oil is put through successive stages of distillation. During the first stage highly volatile products are given off at low temperatures, these comprising less than one per cent of the original quantity. Upon increasing the temperature to from 150° to 300° Fahr., gasoline, naphtha and benzine are given off, these comprising from 10 to 15 per cent. A further increase in temperature of distillation yields kerosene and "fuel oil" com-

prising about 50 per cent of the original mixture, the lighter grades of this stage of distillation being the ordinary kerosene lamp oil. The final stages of the distilling process yield lubricating oil, paraffine wax, and solid residue. The "fuel oil" just mentioned, being impractical for lamp oil, and obtainable in large quantities as a by-product, affords at the present time the least expensive of all engine fuels in general use. Crude oil of some special compositions may be used directly as fuel oil, but in general the impurities in the oil and the presence of the lighter entrained oils, such as benzine and gasoline, make the crude oils unsatisfactory for direct use in internal combustion engines.

536. In oil engines the commercial fuel oil or kerosene is injected as a liquid into a heated vaporizer or into a spraying nozzle, both of which open directly into the engine cylinder. In the case of kerosene, a specially constructed carburetor may be used much the same as with gasoline, although the kerosene carburetor has not yet reached the stage of general successful application.

The case in which the liquid fuel is injected into a heated vaporizer through a nozzle is illustrated in Fig. 326, where *A* is the vaporizer heated by an external blow-lamp at starting, and then heated in some cases solely by ignition temperatures when running under normal conditions, or, it may be heated continuously by an external flame. In the Diesel type of engine, a spraying device as illustrated in Fig. 499 is used, through which the liquid oil is forced under high pressure, emerging in a highly atomized condition directly into the engine cylinder where it is ignited by the high temperature of the highly compressed pure air already in the cylinder.

Methods of Igniting Explosives or Burning Mixtures in the Engine Cylinder

537. Having set forth several methods of transforming commercial forms of fuel into gases or vapors which are usable in the internal combustion engine, the next step in natural order is to look into the various methods of igniting the charge when it is compressed in the cylinder and ready for combustion. The different methods are:

(a), Open flame.

(b), Hot tube or surface.

(c), Combination of hot surface and compression temperatures.

(d), Heat due to compression of air only.

(e), Electric spark:

- (e_1), Make-and-break.
- (e_2), Jump spark.
- (e_3), Lodge ignition system.

Open-Flame Ignition

538. The open-flame method was used in the early days of gas engine development but has since been abandoned. As a matter of historical interest, and in order to open up a full practical width of view for the reader, it should be stated, though briefly, that while the idea of applying an open flame to a confined and compressed explosive gas mixture is a simple one, its successful application is both difficult and delicate. A successful method of operating the open flame was devised in connection with the Koerting engine, in which the flame was not snuffed out and in which no appreciable amount of explosive mixture escaped. The device is shown in Fig. 323, where *A C* is a plug with a divergent nozzle with side openings at *B*. *J* is a casing with side openings at *E* and *F*. In the position shown, the engine is compressing its charge and this compression has lifted the plug so that a small amount of the fuel mixture is allowed to escape through the expanding nozzle. The escaping gas is ignited by the flame at *H*. Just as the engine reaches dead center the plunger *K D* is forced down, thus driving the plug down so that *A* seats at *L*. This is done so quickly that flaming combustion continues in the expanding nozzle and spreads out through the openings *B* and *F* as soon as they come into register. The space *G* opens into the engine cylinder, and is filled with the compressed fuel mixture which is thus ignited.

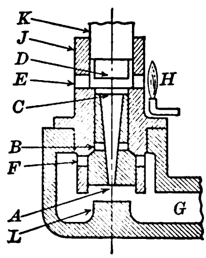


FIG. 323.—OPEN-FLAME IGNITION

Hot-Tube Ignition

539. The hot-tube igniter is illustrated in Fig. 324, where *A* is the engine cylinder, *B G* a tube, closed at the top, to which all mixtures and gases in the cylinder have free access; *E* is a chimney which may be raised or lowered and fixed on the post *H* by means of the set screw *J*; *C* is the extremity of a Bunsen burner, the flame from which heats up the tube *B G* to red heat. After each ignition the tube is completely filled with burned gas which is not entirely replaced, not even at the end of the suction stroke, for the areas of the openings are so designed that there is not sufficient time for diffusion. As compression takes place the fresh mixture forces the residual burned gas to the end of the closed tube, but no explosion follows, even when the

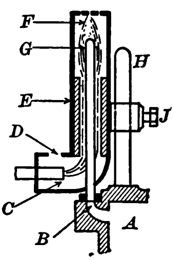


FIG. 324.—HOT-TUBE IGNITION

fresh gas first reaches the red-hot part of the tube, for the velocity of the incoming gas exceeds the velocity of propagation of the flame. This condition continues until at about the end of the stroke when the velocity of flame propagation exceeds the velocity of the incoming gas, and the flame shoots out into the cylinder at A and ignites the charge. An understanding of this action will show the necessity of having the hot zone of the tube $B\,G$ at a proper distance out in order to avoid pre-ignition, and on this account the position of the chimney and the flame is made adjustable as shown.

540. Where a more positive method of timing has been desired a valve located at B and operated by a cam has been introduced, the whole device being designed to hold the valve closed during that period of the cycle when pre-ignition could occur and then to open the valve at the time ignition is desired. This method of ignition is adaptable only for the smaller sizes of engines; on the larger sizes the ignition is not sharp enough, on account of the large volume of gas involved.

Hot-Surface Ignition

541. Hot-surface ignition is shown diagrammatically in Figs. 325 and 326. When the piston E, Fig. 325, compresses the *mixture* which has previously been drawn in through the valve B, the com-

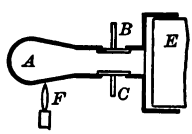


Fig. 325

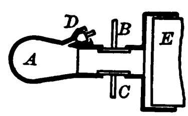


Fig. 326

Figs. 325 and 326.—Hot-Surface Ignition

pression takes place in A and in the space leading to A, and when the compression reaches a certain degree it is ignited by the red-hot walls of A, the red heat being maintained in some cases by a separate burner F. The exhaust valve is at C.

In other cases only air is admitted at B, Fig. 326, and consequently only air is compressed. As air is compressed it becomes heated and advantage of this is taken, the heat due to compression of the air together with the dull red heat of the chamber A being sufficient to ignite the fuel, which is sprayed in separately at D at about the end of each compression stroke. In this case, as shown in Fig. 326, a burner is needed to heat up the chamber A on starting the engine, but after it has been running a few minutes the walls remain hot enough to produce ignition without the aid of a burner. Also, in this case the chamber A is termed a vaporizer, as the hot walls immediately turn the fuel spray entering at D into a vapor.

Ignition Due to Heat of Compression

542. Ignition due solely to the heat of compression requires that pure air only shall be compressed if reliable and economic results are to be obtained. In practice, the pure air is compressed in the cylinder A, Fig. 327, to about 500 pounds per square inch, the corresponding temperature of which is about 1100° F. When ignition is desired the oil fuel is injected through an atomizing nozzle at E by means of cooled compressed air at a higher pressure than that in the cylinder. The heat of the compressed air in the cylinder is sufficient to give a thorough combustion to the complete charge. If a mixture of fuel and air were sufficiently compressed the heat of its own compression would ignite the mixture, but at times which would not be under definite control, and would vary according to the mixture and to the heat and cleanliness of the cylinder walls. In Fig. 327, B is the engine piston, D the air valve, and C the exhaust valve. Ignition by compression only is one of the chief characteristics of the Diesel oil engine.

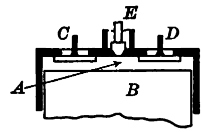


FIG. 327.—IGNITION DUE TO AIR COMPRESSION

Electric-Spark Ignition

543. Ignition by electric spark is accomplished by causing electric current to bridge an air space between two electrodes within the cylinder, as shown in Fig. 328, just as the piston B has compressed the fuel mixture into the space A. E and F are the electrodes, C the fuel valve, D the exhaust valve, and B the piston. Two general methods of causing the electric spark are used, one involving a low-tension or low-voltage current, and known as the make-and-break system of ignition, and the other involving a high-tension or high-voltage current, and known as the jump-spark system.

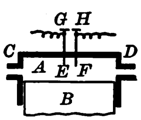


FIG. 328. ELECTRIC-SPARK IGNITION

Make-and-Break Ignition System

544. The make-and-break system is shown in Figs. 329 and 330 in compact diagrammatic form with all the apparatus located in the cylinder head. $H F$ is a fixed and well insulated electrode and the

irregular piece $M\ K\ G\ E\ J$ is the movable electrode. In the position shown the circuit is closed and the current is flowing. When the cam L has turned a few degrees in the direction shown, the spring at M is free to rotate the movable electrode about its axis $G\ E$. The arm $E\ J$ then quickly breaks contact with $H\ F$ whereupon an electric spark flows across the gap and ignites the fuel mixture compressed in A. Q represents a battery of dry or wet cells, and R an intensifying coil or spark coil, as it is more generally called. This coil consists simply of a bundle of wrought iron wires around which is wound a coil of comparatively heavy wire in series with the circuit. The coil acts as inductive resistance, and when the contact is broken at F serves to intensify the pressure and cause a hot spark. It will be noticed that the cam K is designed to keep the two electrodes apart until just before the spark is desired. This is necessary where batteries are used to generate the current which would otherwise be wasted and which might also give irregular ignition if the electrodes were too close and were imperfect or fouled.

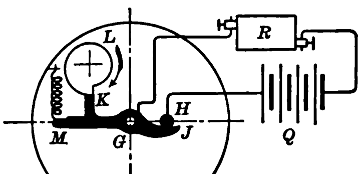


FIG. 329

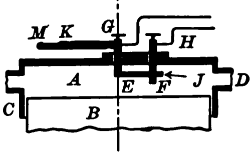


FIG. 330

FIGS. 329 AND 330.—MAKE-AND-BREAK IGNITION WITH BATTERY

545. Where a magneto is used the two electrodes may be designed to remain in contact all of the time, if desired, except the very short period when the spark is wanted, for, in this case there is no current except that generated by the magneto and the mechanism is designed to produce that only at the time needed, as illustrated in Fig. 331. H is the fixed electrode and $M\ G\ J$ the movable electrode exactly, thus far, as in Fig. 329. $O\ R$ is a single rod passing through an opening in the arm $G\ M$ and having a head R, the under side of which, as will be seen later, is to strike against the faced surface at M and thereby rotate the electrode $M\ J$ about G and break the current at $J\ H$. S is a compression spring to keep the electrodes J and H together, except when the spark is desired. $N\ P\ K$ is a double arm

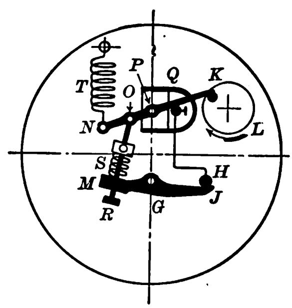


FIG. 331.—MAKE-AND-BREAK IGNITION WITH MAGNETO

rocker keyed to the armature shaft at *P*. *L* is a disk with a pin *K* which swings the rocker through a small angle and then allows it to trip, when it is snapped back to its original position by the heavy spring *T*. The high rotative velocity given the armature, through even this small angle, causes it to cut the lines of force of the magnet *Q* with great rapidity and to generate a strong current, which is carried to the electrode *H* by the wire as shown in the Figure. When the heavy spring snaps the rocker back and generates the current it also pulls back the head *R* of the rod *O R* which presently strikes the arm *M* and breaks the current at *H J* as explained above, thus producing the spark which ignites the compressed mixture in the cylinder.

546. The make-and-break mechanisms described above are located in the cylinder head where it is easy to show the complete illustration of all the parts with reference to the engine cylinder, as indicated in Figs. 329 and 330. A form of make-and-break mechanism that is generally used is shown diagrammatically in Fig. 332. It is built on the front wall of the cylinder near the top. The rod *P Y*, which receives its up and down motion from an eccentric on the engine shaft, carries a rigid arm *Q R*, and this in turn carries a pin on which the double rocker-arm *S R V* is free to swing through a small angle. This double rocker-arm is kept in its normal position with the arm *R V*, vertical by means of a stop block and a small spring located in the arm *R Q*, but not shown in the illustration. As the stem *P Y* is moved up the arm *R V* presses against the sliding disk *L*, which permits the compression spring *M* to act. This causes the bent rocker *K G J* to rotate slightly about the fixed center *G* until the electrode arm *J* comes into contact with the electrode pin *H* inside of the cylinder, thus setting up a current. The bent rocker *K G J* is so formed that the arm *K G* is outside of the cylinder and the arm *G J* inside. The cylindrical part of the rocker between the two arms passes through a stuffing box which also holds the electrode *H*.

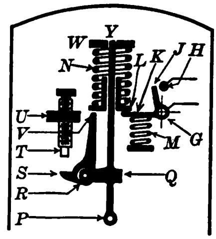


FIG. 332.—DIAGRAMMATIC ARRANGEMENT OF MAKE-AND-BREAK MECHANISM

When the rod *P Y* has moved up a designated distance the toe *R S* of the double rocker-arm strikes a regulation screw at *T*, thereby throwing the supporting arm *R V* from under the disk *L* and permitting it to fall under the influence of the compression spring *N*. As it nears the bottom of its fall, it strikes the arm *K G* and quickly separates the electrodes J and H, thus drawing the ignition spark. The bent rocker $J\ G\ K$ is provided with a stop block, not shown, which keeps it from turning too far.

547. An assembled view of a practical make-and-break mechanism is illustrated in Figs. 333 and 334. The details of construction

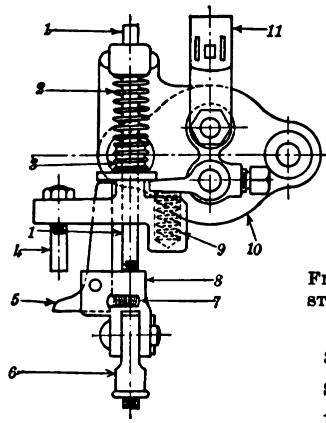


FIG. 333

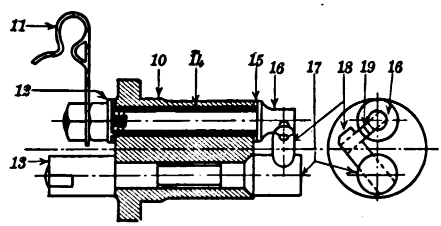


FIG. 334

FIGS. 333 AND 334.—SHOWING DETAILS OF CONSTRUCTION OF MAKE-AND-BREAK IGNITION DEVICE

are varied by different builders and in some cases the electrode *16* is placed in the cylinder head, keeping the rest of the mechanism on the side.

548. The make-and-break devices described above are known also as hammer-break mechanisms because of the hammer-like action exerted by the movable electrode on the fixed electrode, causing rapid wear and fouling. To overcome this, a device known as the "wiper-spark" mechanism is sometimes used, its characteristic action being that the movable electrode is made to continuously revolve about G, Fig. 329, instead of oscillating, and as it revolves a flexible projection on it wipes across the fixed electrode H on the inside of the cylinder, causing a spark as it slides off, thus igniting the compressed mixture. The objection to the wipe-spark mechanism is its rapid wear, and deterioration of spring in the hot cylinder if a spring is used.

549. The advantages of the make-and-break system are low voltage and simple electric apparatus, from four to nine volts being usually ample to operate it. The disadvantages are wear and fouling as already mentioned, and in addition the generally unsatisfactory mechanism required to trip the igniter. These disadvantages increase with high rotative velocities so that most designers have discarded it in high speed work, although it is and has been satisfactorily applied in this connection by different builders, while for ordinary speeds it is largely used.

Jump-Spark Ignition System

550. The jump-spark system of ignition avoids all mechanism entirely and requires only electrical apparatus for its operation. A simple, fundamental, diagrammatic illustration is shown in Fig. 335. The system consists, essentially, of a primary or low-tension electric circuit *A, B, C, D, E, F, G, H, I, J, A* and a secondary or high tension circuit *U, Z, Y, X, W, V, U*. As the shaft *J* revolves the commutator at *I*, attached to it, comes into contact with *H* and closes the primary circuit, thus causing the bundle of fine iron wires at *M* to

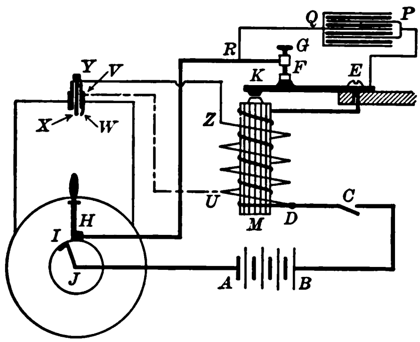


FIG. 335.—JUMP-SPARK METHOD OF IGNITION

become an electro-magnet and setting up a magnetic field, the lines of which go out across the fine wire *U Z* of the secondary circuit. Also the electro-magnet *M* attracts the armature *K* which is fastened to a flat spring *K E*, thus separating the surfaces at *F* and breaking contact in the primary circuit at that point. As soon as the primary circuit is broken the magnetic field surrounding the electro-magnet *M* collapses instantly, rapidly drawing in the magnetic lines of force across the secondary coil *U Z* and thus inducing in this coil a very high potential of several thousand volts. This voltage is sufficient to cause the induced secondary current to flow through the wire *Z Y*, the insulated wire *Y X* in the spark plug, across the air-gap *X W* and, theoretically, through the return wire *V U*, thus completing the secondary circuit. When the secondary current jumps the gap *X W* in the cylinder, a spark is formed which ignites the compressed gas mixture, the timing of this spark being so arranged that it comes just as the piston is at or near the top of its compression stroke.

551. When contact was broken at *F*, as described above, the electro-magnet *M* was demagnetized and the flat spring *E K* served

to draw K away from the magnetic coil M and to place the surfaces at F again in contact, thus re-establishing the primary circuit. A magnetic field immediately builds up about the coil only to collapse again as soon as the coil is strongly enough magnetized to draw in the end K of the spring. As the magnetic field collapses again it induces a new high-tension current in the secondary circuit and causes another spark at $X\ W$. This process is repeated many times while the commutator I is passing once across the brush H, thus obtaining a series of sparks to ignite the gasoline vapor charge in the engine instead of a single spark.

552. In practice the wire represented by the dash-and-dot line $V\ U$ is omitted and the return of the secondary circuit is through the "ground," which is usually the metal of the engine itself, from W to J, and then by the return wire of the primary system through the battery $A\ B$ to D, where the secondary wire is joined to the primary wire.

553. The bundle of wires forming the magnetic coil is covered with a layer of insulating material around which the stouter or heavier insulated copper wire of the primary circuit is wound in several layers. These coils are in turn covered by another layer of insulating material around which the finer insulated wire of the secondary circuit is wound a great many times, each layer of the secondary winding being separated by insulating material to prevent short-circuiting of the high-pressure current. The spring $K\ E$ is known as a "trembler" or "vibrator" or "buzzer." At G is a screw thread regulating the width of the gap at F when contact is broken. The two parts of the spark-plug $Y\ X$ and $W\ V$ are separated by porcelain or other insulating material, which is represented in the Figure by open space.

554. A condenser is shown at $P\ Q$ in a shunted primary circuit. The object of the condenser is to prevent sparking, or at least, excessive sparking, at F, when the magnetic field in the coil M collapses. As stated above, when this occurs a high potential of several thousand volts is induced in the secondary wire which causes the current to jump the gap at $X\ W$, causing a spark; likewise, at the same time, some higher potential than the six to eight volts which are usually furnished by the batteries is induced in the primary winding $D\ E$, and a spark would be caused at F which would rapidly pit the contact points and destroy their usefulness. With the condenser in the circuit as shown, this high potential induced in the primary circuit flows into it, instead of jumping at F, and when F is again closed the condenser releases its charge and helps in sending the

current through the primary circuit. The condenser is built up of a series of tin-foil sheets which are separated by layers of insulating material. The alternating sheets of tin-foil are joined as shown. The condenser is usually small and is placed in a small box together with the magnetic coil, as illustrated in Fig. 336. The box, which is generally known as a spark-coil box, has four terminals or posts, L

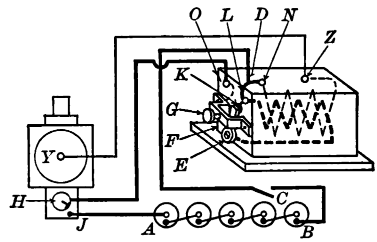


FIG. 336.—SPARK-COIL BOX AND ENGINE CONNECTIONS

and N being the ones from which the primary and secondary coils start, O the one leading to the timer H, and Z the one at the end of the secondary coil and leading to the spark plug. The trembler is indicated at E K. The primary circuit from E to O is through the trembler F, the screw G, the bridge, and the short wire shown by dash lines from the foot of the bridge to O.

Lodge Ignition System

555. The Lodge ignition system uses a high-tension current, but the spark is produced by the discharge of Leyden jars, which give a current of exceptionally high frequency and great heat intensity. Because of the character of this current it is not easily diverted from its prescribed circuit which it maintains despite somewhat defective spark plugs, accumulations of oil or carbon deposits, and even submersion in water. Sir Oliver Lodge demonstrated these points in a lecture which was reported, in part, in *The Engineer*, London, December 8, 1904, and showed further that the strength of the spark was increased rather than diminished by placing a piece of wet blotting paper across the air gap of the spark plug.

556. The characteristic features of the Lodge ignition apparatus are shown in Fig. 337. The wire DE is a primary or low-tension one wound around the core M. The wire UZ is an ordinary secondary or high-tension wire, and the apparatus up to this point is similar to that used in the ordinary jump-spark system illustrated in Fig. 335. The ends of the high-tension wire terminate at the inner coatings of two Leyden jars V and Y. The outer coatings of these jars are

connected with the spark plugs W and X in the engine cylinder C. The Lodge system has been applied principally to large gas engines where it is advantageous to have more than one point of ignition to facilitate the flame propagation in the fuel mixture. Hence two spark plugs are shown in Fig. 337, although one may be used, in which case the wire from V to W would be grounded.

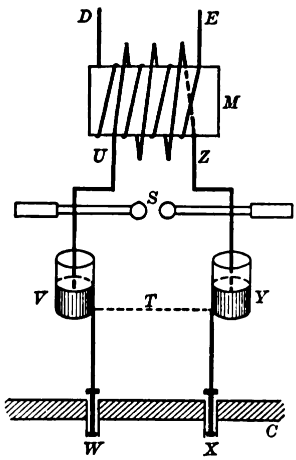


FIG. 337.—LODGE IGNITION SYSTEM

557. When the high-tension current is generated as explained in paragraph 550, in connection with Fig. 335, it flows into the Leyden jars, which, being of small capacity, are quickly charged. As the voltage between U and Z rises the Leyden jars become charged, the circuit between their outer coatings being completed through a partial electrical conductor represented at T. The charge on the Leyden jars is accompanied by potential differences between their inner and outer coatings, which balance the voltage between U and Z, except for the relatively low voltage lost in T. When the voltage rises high enough to break down the air gap S, a spark passes. This spark is a sufficiently good conductor to bring the inner coatings of the Leyden jars to nearly the same potential, and the original potential difference is transferred to the outer coatings. This potential difference existing across the gaps at W and X causes sparks to pass at these points. The consequent discharge has a circuit $SVWXY$ of low resistance and low self-inductance, so that the current rises rapidly to a high value, and then oscillates at high frequency. The partial conductor T does not short-circuit the sparks at W and X because its resistance is so high that a negligible current passes through it during the extremely brief time required for the potential difference to build up across the spark gaps. The sparks at W and X are termed the "B" sparks, and the spark at S is the "A" spark. While the latter is not used in the ignition of the fuel, it is an essential feature, and is useful as an index of the character of the sparks that are being produced at W and X. The knobs at S are adjustable so as to regulate the "A" spark. This explanation of the characteristic electrical action of the Lodge ignition system, together with the mechanical analogy given in the following paragraph, was prepared from descriptions very kindly given to the author by Assistant Pro-

fessor L. A. Hazeltine, M.E., of the Department of Electrical Engineering at Stevens Institute of Technology.

558. A mechanical analogy of the Lodge electrical ignition is of special interest and is illustrated in Fig. 338. The pump at *D* corresponds to the induction coil *M* of Fig. 337, and the pressure in the pump and between the pipes *U* and *Z* to the voltage between the wires *U* and *Z*. The flexible diaphragms *V* and *Y* in the cylinders correspond to the Leyden jars. These are large so that they may yield and accommodate themselves to large pressure and large quantity, whereas the diaphragms at *S*, *W*, and *X* are shown small to indicate that they will break, instead of yield, at a given pressure. The breaking of these small diaphragms corresponds to the jumping of the spark across the air gaps. When the diaphragms at *V* and *Y* are being distended the fluid flows through the contracted area at *T*. When the pressure rises high enough the diaphragm at *S* breaks, equalizing the pressure at its two sides. The distended diaphragms *V* and *Y* now exert a pressure which causes a difference to exist between *M* and *N*, and this difference in pressure immediately breaks the diaphragms at *W* and *X*. The fluid then surges back and forth, or oscillates, in the circuit *SVWXY*. If the fluid in this circuit has small inertia and is subjected to slight friction, its oscillations will be extremely rapid with a corresponding high rate of flow. Of course, to make the analogy complete, one must imagine that the broken diaphragms are automatically replaced. The rapid oscillations of the fluid correspond to the high frequency of the electrical discharge.

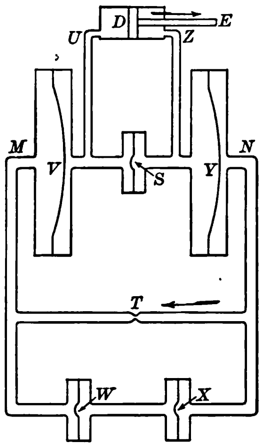


FIG. 338.—MECHANICAL ANALOGY FOR LODGE ELECTRIC JUMP-SPARK SYSTEM

559. The reason that the Lodge ignition system continues to give service despite defective spark plugs and ordinary short circuits is that the sparking current, although it flows initially through the short circuit when it is small, builds up so rapidly and so intensely that the air gap is practically negligible and it jumps at that point, giving the desired hot spark.

The entire apparatus necessary for the production of the Lodge ignition is housed in a wooden box of moderate proportions. The box contains a glass cover to protect it from the visible "A" spark which occurs at S in Fig. 337. The Lodge method of ignition is quite largely used in Europe, especially on larger sizes of internal combustion engines.

Timing the Ignition

560. The timing of the spark is important and provision must be made in the mechanism for regulating it. It might appear that the time to produce the electric igniting spark is just as the piston reaches the top of its compression stroke or very shortly thereafter. In practice it should be produced after dead-center position when starting the engine; but after the engine is started the ignition should not only come earlier but should come slightly before the dead-center position, or in other words, just before the charge is fully compressed, to give best results. The general method of changing the time of the spark is by shifting the commutator brush, as for example in Fig. 335, where the brush H is attached to a handle which is free to rotate about the center J. In the position shown, handle vertical, the commutator I will come in contact with the brush H when the crank is vertical and the combustion will start practically at the top of the stroke. If the handle is turned to the right the two parts will come into contact later and the combustion will start after the piston has moved down a small amount. If the handle is moved to the left of the central position the ignition will start before the end of the up stroke. In some engines, particularly in the larger sizes, some little time is noticeable on an indicator diagram, for flame propagation across the cylinder where the spark plug is on one side, and even from the center to the wall of the cylinder when the spark plug is at the center.

561. In a two-cylinder engine there would be two brushes on the ring of the handle H, and these two brushes would be 180° apart. For more cylinders there would be a brush for each cylinder, all brushes being equally spaced. The engine here considered is of the two-stroke type. In Fig. 335 the commutator and brush are shown, for simplicity, at the engine shaft, and in some makes it is placed there, but in most forms of construction a vertical spindle is rotated by means of bevel wheels from the engine shaft, and the commutator box is placed at the top of the vertical spindle near the top of the engine cylinder, as indicated at H in Fig. 339.

562. Instead of battery cells being used for generating the spark in the electric ignition system, as indicated at A B in Figs. 335 and 336, a storage battery or a magneto may be used in both the jump-spark and make-and-break systems. A special form of construction

of magneto which embodies secondary winding and condenser features may be used directly to generate the high-voltage current in the jump-spark system of ignition without the use of a special coil-box.

High-Tension Distributor

563. Where more than one cylinder is used to develop the required amount of power on the jump-spark system a magneto or a spark-coil box such as is shown in Fig. 336 must be used for each cylinder; or one coil-box may be used, but in this case a "distributor" of the high-tension current must be added to the ignition system. The elementary principle of such a distributor is illustrated at P, Fig. 339. The diagram for this Figure, it will be observed, is identical with Fig. 336, so far as the wiring of the primary circuit is concerned.

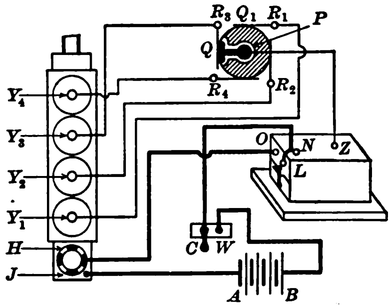


FIG. 339.—HIGH-TENSION DISTRIBUTOR FOR MULTIPLE CYLINDERS

In the secondary circuit, the outgoing wire Z from the high-tension coil goes to the distributor which is constructed so that the commutator Q is in contact with the brushes R_3, R_1, etc., at the time that the spark for ignition is desired. Where multiple cylinders are used it is customary to generate the ignition current by means of a magneto instead of by batteries or cells when the engine is under way. Some forms of magneto construction furnish the current for starting also. Sometimes a storage battery is used, in which case a generator takes the place of the magneto and its current is delivered to the storage battery from which it is drawn off as needed through a switch connection at C.

564. In some engines two spark-plugs are used on each cylinder in order to secure quicker ignition and more complete combustion; also to provide against the power loss of the cylinder should a single spark-plug fail. The general methods of wiring and operation are the same for two plugs as for one. Ignition apparatus for gasoline

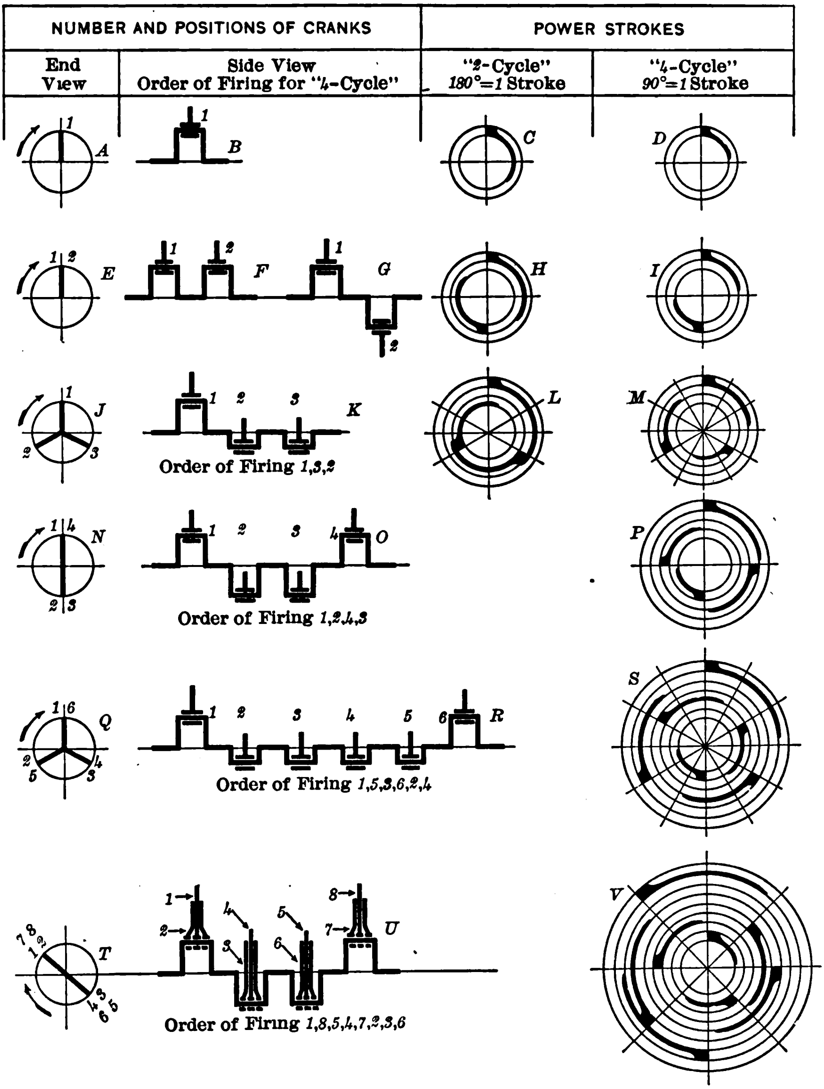


Fig. 340.—Chart Showing Construction of Crank Shafts, Order of Firing and Power Periods for One to Eight Cylinders

engines, particularly, comes in a great many varieties of construction and operation.

Chart Showing Construction of Crank Shafts, Order of Firing and Power Periods for One to Eight Cylinders

565. A chart showing the relative positions of the cranks as usually adopted, the order of firing the cylinders, and the periods during which effort is exerted on the main engine-shaft for single- and multiple-cylindered engines is shown in Fig. 340. It is shown by the heavy curved line in diagram *C* that a single-cylinder two-stroke engine exerts a diminishing rotative effort on the shaft during something less than one-half of the time, while in a four-stroke single-cylinder engine effort is exerted only for a trifle less than one-quarter of the time, as shown at *D*. In order to secure effort on the main shaft for each stroke of the piston, it is necessary to use two cylinders on the two-stroke type as shown at *H*, or to use four cylinders on the four-stroke type as shown at *P*. The six-cylinder four-stroke engine overlaps on each successive pair of power strokes as shown at *S*, the total amount of overlapping being a little more than for the three-cylinder two-stroke engine as shown at *L*. The power diagram at *V* for an eight-cylinder four-stroke engine shows a still greater amount of overlapping of power strokes. The heavy power lines in the above diagrams are made heavy at the initial end to indicate the explosion and are thinned out to represent the expansion. They do not extend the full length of the stroke because in the two-stroke engine the exhaust begins when the crank is from 40° to 55° from the bottom of the stroke, while in the four-stroke engine the exhaust valve opens later than this, as may be noted by referring to Figs. 314 and 312.

Arrangement of Cranks in Multi-Cylinder Engines

566. The arrangement of the cranks depends on the balancing of the reciprocating parts and the type of construction, *i.e.*, whether a two- or four-stroke engine. In a two-cylinder four-stroke engine the two pistons must move up and down together as indicated at *E* and *F* in Fig. 340, in order to secure equal intervals between explosions, thus obtaining a highly unbalanced condition of the elementary reciprocating parts which must be allowed for in the general design of the engine. If, however, the two cylinders are built opposite to each other as indicated at *G*, instead of side by side, the

pistons will be moving in opposite directions and the motions of the two sets of reciprocating parts will balance each other.

567. In a two-cylinder two-stroke engine the cylinders must be set side by side and the two pistons must alternate in their up-and-down motions in order that the explosive or power strokes may follow each other instead of coming together. This accords with the requirements for a mechanical balance of the reciprocating parts and accounts for the large number of two-cylinder two-stroke engines in general use, particularly in low-power marine work.

The cranks for three-cylinder and six-cylinder engines are set at 120° in all cases, in order to secure equal distribution of the power periods, while those for eight-cylinder V four-stroke engines are all in one plane, the same as for the four-cylinder four-stroke engines.

Order of Firing

568. The order of firing the cylinders depends on the relative crank positions and the distribution of the explosive shocks along the length of the crank shaft. In general practice the cranks are set as shown at *J K*, *N O*, *Q R* and *T U* in Fig. 340. The order of firing for the three-cylinder four-stroke engine must be *1*, *3*, *2* with the engine running in the direction shown by the arrow. This will become evident by considering that crank *1* at *J* is just starting its explosive stroke, or its first 180°, and that if cylinder No. *2* should fire next the ensuing power stroke would overlap the first one because crank *2* has only 120° to go. With No. *3* firing after No. *1* there will be an interval of 60° of crank motion between successive explosion strokes. With the same interval of 60° between *3* and *2* and between *2* and *1* and with 180° allowed for each of the three power strokes it will be seen that this order of firing gives three balanced power strokes during a heat cycle which occupies 720° of crank-shaft motion. With a three-cylinder two-stroke engine the order of firing would be 1, 2, 3 for the reason that there must be an explosion each time the crank pin passes the high point of the circle.

569. For the six-cylinder engine the end cranks are generally in unison, likewise the middle cranks and also the intermediate cranks as shown at *Q* and *R*, Fig. 340. With this arrangement, the order of firing is *1*, *5*, *3*, *6*, *2*, *4*, it being correct in this case to have a succeeding firing piston and crank only 120° behind, because the firing or power strokes must overlap where there are six cylinders.

The eight-cylinder V-engine crank shaft has but four crank pins and is identical with the four-cylinder crank shaft in the arrangement of the cranks. In this case of eight cylinders there are two connecting rods attached to each crank pin.

Details of Valve-Gear Mechanism

570. In the following illustrations, typical forms of construction will be shown, generally in diagrammatic form, especially where the details of the actual construction are too complex or involved to make a drawing readily understandable to the student.

The gasoline engine, because of its wide and popular use in automobile and small marine engine work, will be considered first. The four-stroke engine is generally used in automobile and stationary power work, while the two-stroke engine is mostly used in marine work, although there are numerous exceptions in both cases.

571. A diagrammatic illustration of the principles of the mechanical action of the commonest type of four-stroke gasoline engines is shown in Fig. 341. The inlet and exhaust valves at C and T are

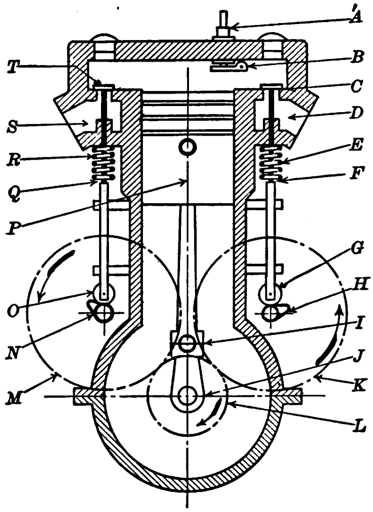


Fig. 341.—Mechanical Detail of Four-Stroke Engine

poppet valves and are operated through the "valve stem" as at $C\ F$, the "valve lift rod" $F\ G$, the "cam roller" G, and the "cam" H. The cam itself is attached to the "cam shaft," to which is also keyed the toothed wheel represented by its pitch circle at K. This wheel is exactly twice the diameter of the toothed wheel L, which is keyed to the main engine or crank shaft. From this it will be seen that the cam shaft turns just half as fast as the main shaft as it must do in the four-stroke engine where the inlet and exhaust valves are required to open only on every other stroke.

Valve-Timing Diagrams

572. In order that the valves may open and close at proper times, a valve-timing diagram should be drawn. Just what are the "proper times" for these events vary in different makes of engines. The fundamental cause for these variations is the inertia of the gases which must be considered in adopting relative sizes of valves and cylinders, speed of rotation, location of valves, etc. The quality of the fuel is also a determining factor. When a valve opens before a dead-center position it is said to have "lead"; and when it opens after dead center it is said to "lag." In the present case it will be considered that the intake valve shall open 15° after top dead center, Fig. 342, and close 20° after bottom dead center; and that the exhaust valve shall open 40° before bottom center and close 10° after top center. In automobile motors the range is approximately as follows: Intake opens 0° to 15° after top center, closes 20° to 45° after bottom center; exhaust opens 40° to 65° before bottom center, and closes 0° to 15° after top center.

573. A valve-timing diagram shows the position of the engine crank at the instants of opening and closing of the valves. Such diagrams are usually made up of radial lines limited by arcs of circles and when used in connection with two-stroke engines there is no confusion, but when applied to four-stroke engines the angles belonging to two different revolutions overlap. A valve-timing diagram limited by circular arcs is shown in Fig. 342. In accordance with the data of the preceding paragraph, the crank position at the instant the intake valve opens is $A B$, the engine turning as indicated by the arrow, and at closure it is $A C$. The exhaust valve opens at $A D$ and closes at $A E$. Although the angle $D A C$ overlaps the "open" angles for both the intake and exhaust valves, the valves themselves are not both open at this time, for it must be remembered that the present illustration is for a four-stroke engine and that the intake arc $B C$ belongs to the first revolution of the crank and the exhaust arc $D E$ to the second revolution.

574. Another form of valve-timing diagram for the four-stroke engine which carries with it a constant reminder that the overlapping angles are in different revolutions, is offered by the adoption of a continuous pseudo-spiral in place of separate circular arcs. The

pseudo-spiral is easy to draw. For example, in Fig. 343, mark off a small square at the shaft center *A*, and project the four sides as shown at *1 C*, *2 D*, etc. With the corner marked *1* as a center and any desired length as *1 B* for a radius draw the 90° arc *B C*. Then with the corner *2* as a center and *2 C* as a radius draw the 90° arc *C D*, and so on, using the corners of the square in succession, until the pseudo-spiral is drawn with one, two, or as many full turns as are desired.

575. Applying the pseudo-spiral in Fig. 344 to the present example of valve-timing diagram it is seen that the continuous curve

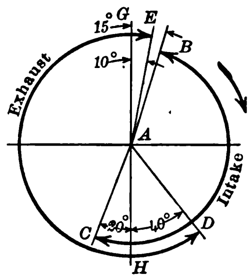


Fig. 342

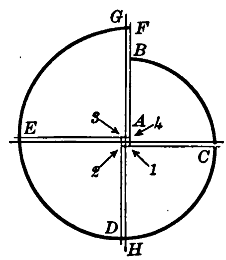


Fig. 343

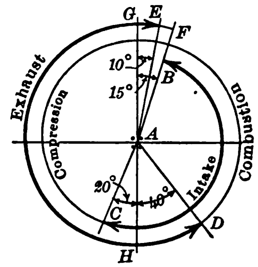


Fig. 344

Figs. 342-344.—Forms of Valve-Timing Diagrams

from *C* to *D* carries with it the suggestion that the crank travels through 300° after the intake valve closes and before the exhaust valve opens. During this period the compression, ignition, combustion, and expansion of the fuel take place. In general it is not necessary in laying out the pseudo-spiral to mark out the small central square and the projecting lines shown in detail in Fig. 343 for, in practical diagrams, the pitch (½ of *B F* in Fig. 344) is small and it is sufficient to make, freehand, four symmetrically placed dots on 45° lines close to *A* as shown, and to use these as centers in drawing the 90° arcs between the imaginary projected lines passing through the dots.

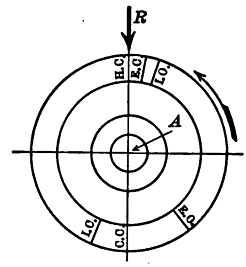


Fig. 345.—Flywheel Marking

Marking the Flywheel

576. From the valve-timing diagram permanent lines are often marked on the flywheel of the engine as a guide in testing the accuracy of the action of the valves. A flywheel is indicated in Fig. 345 on which are

scribed lines as follows: "H. C.," meaning engine on head-end dead center; "E. C.," exhaust valve closes, corresponding to the crank position $A\ E$ in Fig. 344; "I. O.," inlet valve opens; "E. O.," exhaust valve opens; "C. C.," crank-end dead center; "I. C.," inlet valve closes. For a two-cylinder four-stroke engine the same marks would answer for both cylinders; while for a four-cylinder engine the same marks would be duplicated at 180°. In this latter case the present line, marked "E. C.," would be further marked, with suitable abbreviation "for cylinders 1 and 4," while the "E. C." 180° beyond would be marked "for cylinders 2 and 3." Similar markings would be used for the other events.

Adjusting the Valves

577. A setting or adjustment of the valves in accordance with the original design may now be made with the aid of the marks on the flywheels described in the previous paragraph. Usually these marks are on the circumferential surface of the wheel, but are shown in Fig. 345 as being on the side for simplicity of illustration. Fastened to the cylinder wall on the engine center line is an index edge, as represented at R. If the cams and gears are right, and if a very thin piece of paper or "shim" is inserted between the follower rod and the valve stem at Q, Fig. 341, it will be held quite firmly while the crank is at head dead center, but will be released easily when "E. C." reaches R, the flywheel being turned against its regular direction for the present purpose of adjustment. There is always a small amount of clearance between the cam follower rod and the push rod to allow for expansion and for quick opening of the valve. If the shim be now placed at F, it will begin to hold when "I. O." reaches R. Similarly it should be released easily when "I. C." reaches R.

578. In the positions above described, the cams at N and H are just about to cause the follower rods to break contact and to make contact with their respective valve stems as shown in Fig. 341. When the engine is on head-end dead center the rod $O\ Q$ is pressing on the valve stem, but when the crank has moved 10° the cam has turned so that contact at Q ceases. When the crank has turned 15° from head center the cam H has moved the roller G and the push rod sufficiently to take up the clearance at F, and the valve C starts to open.

Cam-Shaft Diagram and Cam Design

579. A cam-shaft diagram such as is shown in Fig. 346 is used in laying out the cam projections. In a cam-shaft diagram the angles

will be just one-half of those in a main shaft timing diagram in four-stroke engines, and, therefore, a simple circular diagram as shown in Fig. 346 is all that is necessary and is readily laid out. It will be seen that the inlet cam *H* of Fig. 341 will have a projection whose working width at the base encompasses 92½°, equal to the angle *B A C* of Fig. 346, and to one-half the angle *B A C* of Fig. 344. The radial height of the projection of the cam will be equal to the desired lift of the valve plus the clearance between valve stem and cam rod. With these limiting points of the cam known the outline of the cam projection may be drawn to some desired contour according to the dictates of one's practical experience. Or, it may be designed so as to give the valve uniform acceleration and retardation while rising and falling, as explained on page 65, Vol. I.; or, the cam may be designed so that there will be uniform pressure between the follower

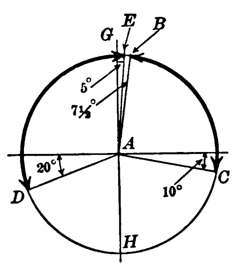


FIG. 346.—CAM SHAFT DIAGRAM

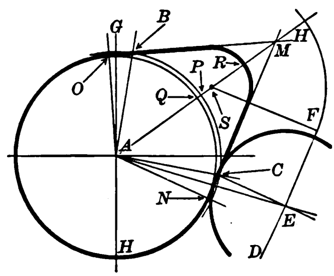


FIG. 347.—A CAM LAYOUT

roller and the cam when under the influence of the compression spring. The design for this latter case requires a special study of cam design, which is too involved to be included here. For the exhaust cam, the working width of the projection in the case here illustrated is seen to be 115°, or equal to the angle *D A E*, Fig. 346, while the working height is the desired lift of the exhaust valve plus the valve-stem clearance.

580. The working and actual widths and heights of cams as referred to in the previous paragraph are illustrated in Fig. 347. The angle *B A C*, equal to 92½°, as in Fig. 346, is laid down and a circular arc *B P C* assumed. On the length of the radius of this arc will depend the pressure angle between roller and cam as referred to in the previous paragraph. Curved or straight lines are then drawn through *C* and *B* so as to meet as at *R*, and also so as to be tangent to some smaller circle, as *N Q O*. These lines may be assumed, or theoretically designed, and also they may be symmetrical or not symmetrical, according to the will of the designer. In the present

case, symmetrical straight lines *M C N* and *M B O* were drawn and the acute angle rounded off with an arc whose center is *S*. Perpendiculars, *A N* and *A O*, were then drawn to these lines. It may now be seen that the cam shaft turns through the angle *O A B* while taking up the clearance or "play" at *F*, Fig. 341, between the follower rod and valve stem; that the valve is being lifted through the working distance *P R* while the cam is turning through the angle *B A M*, and that it is being dropped by the combined action of gravity and compression spring while the cam turns through the angle *M A C*. When the cam has turned through the additional angle *C A N* the full clearance between the cam rod and valve stem has been attained.

581. The path of the center of the roller relatively to the cam is *D E F H*, *E C* being the radius of the roller. The roller is shown in its relative position with respect to the cam for the phase at which the engine valve is just beginning to lift, the roller having already moved out just far enough to overcome the "play" between the valve stem and follower rods. It may also be noted from the Figure that the point of contact *C* between the cam and roller is not on the common line of centers *A E*, but is to one side of it. Failure to lay out the cam curve without proper allowance for this feature will result in a valve motion a little different from that which may have been planned.

582. In order properly to assemble the cams and the gear wheels which operate the cams, it will be evident that the designer must give accurate dimensions and instructions for cutting the keyways or for otherwise fastening the cams and wheels to their respective shafts if the desired timing is to be secured. Furthermore, all cams and wheels should be carefully marked relatively to each other before taking them apart for repair or adjustment work, in order that errors and annoyances may be avoided and time saved on reassembling.

SECTION X.—COMMERCIAL APPLICATIONS OF VARIOUS FORMS OF VALVES AND VALVE GEARS TO GASOLINE ENGINES

MECHANICAL DESIGN OF VALVE GEAR FOR GASOLINE ENGINE USING TWO PISTON VALVES OPERATED BY ECCENTRICS

587. The piston-valve type of gasoline engine is represented in Fig. 356. Where piston valves are used they are generally operated by eccentrics or cranks rather than cams. This restricts the velocity of the piston valve in opening and closing the ports and requires some special planning for eccentric position relatively to crank position when the engine has lead and lag. When the engine

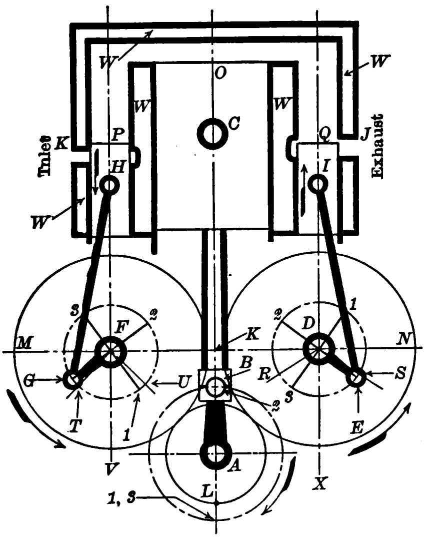


FIG. 356.—GASOLINE ENGINE OPERATED BY TWO PISTON VALVES AND TWO ECCENTRICS

does not have lead or lag, the relation between eccentric and main crank positions is simple and is laid out as follows: Place the main crank on head-end dead center, as at *A B*, Fig. 356. *A L* is the pitch radius of the driving gear wheel and *F M* and *D N* are pitch radii of follower gear wheels which operate eccentric shafts. *F M* and *D N* are each equal to 2 *A L*. Place the eccentrics or cranks *F G* and *D E* in 45° positions as shown approximately in the Figure. The lengths *F G* and *D E* are usually about two-thirds the piston travel so as to obtain large valve travel, larger ports and faster travel at opening and closing. Assume proportions and lengths for pistons and connecting rods so that $F\,G + G\,H + H\,P = D\,E + E\,I + I\,Q = K\,O$. Then, when the cranks are on their 45° positions, mark the top edges of the inlet and outlet ports level with the tops P and Q of the valves. With the valve gear arranged as here described, and with the wheels turning as shown by the arrows it follows that the valve I will be closing the exhaust port J just as the valve H is opening the inlet port K when the engine is on upper dead center. Suction will take place during the first 180° turn of the crank; compression during the second 180° when both ports are closed by the valves; ignition and combustion during the third 180° when the ports are also closed; and exhaust during the fourth 180°. This simple layout, however, does not allow for lag and lead as stated above.

588. To lay out a valve gear which will provide for lead and lag a definite case will be taken from Fig. 342, where the exhaust closure lags 10°, intake lag is 15°, intake closure lags 20°, and exhaust lead is 40°. From Fig. 344 it is seen that the intake valve must be open while the main crank travels 185° (= − 15° + 180° + 20°), or while the eccentric travels 92½°. Make the angle $T\,F\,U$, Fig. 356, equal to 92½° and lay it off symmetrically about the center line $F\,V$. Lay out the angle $T\,F\,G$ equal to one-half the given lag angle of 15°, and G will be the position of the eccentric sheave center when the main crank is at $A\,B$. When $A\,B$ has turned 15°, G will be at T and the port will be just opening. The angle $G\,F\,1$ is 90° and the eccentric sheave center will be at *1* when the main crank has travelled 180° to the position $A\,L$. The angle $1\,F\,U$ is 10°, and U is the position of the eccentric center when the intake port is closed. The numbered positions *2* and *3* show the eccentric positions relatively to the main crank positions of the same number.

589. In laying out the exhaust gear, it is seen from Fig. 344 that the exhaust port must be open for 230° (= 40° + 180° + 10°) and, therefore, that the exhaust eccentric represented by $D\,S$ in Fig. 356 must travel through 115° while the exhaust port is open. Therefore draw the angle $R\,D\,S$ equal to 115° symmetrical with $D\,X$. Then the angle $S\,D\,E$ is made 5° and when E has moved to S the edge Q of the valve will have closed the exhaust port. $E\,D\,3$ equals 90°, and $3\,D\,R$ equals half the exhaust lead angle or 20°. The exhaust eccentric center is at R when exhaust opens and at S when it closes; and it is at the points *1*, *2*, and *3* when the main crank pin B is at positions correspondingly numbered. It will also be observed that when compression of the charge is taking place from *1* to *2* the inlet

piston valve is moving up, necessarily, and that the exhaust piston valve is moving through small distances near the top of its stroke. When compression is completed the up-stroke of the main engine piston is completed and both valves are near the top of their strokes. When ignition takes place the exhaust valve is moving down and inlet valve soon will. In Fig. 356 the piston valves are shown on opposite sides of the cylinder. Practically they are better placed on the same side where the two eccentrics may be mounted on the same shaft and space thus saved. The open spaces marked *W* are for circulating cooling water. The principles here explained are used in modified form as to details of construction on the Hewitt and other engines.

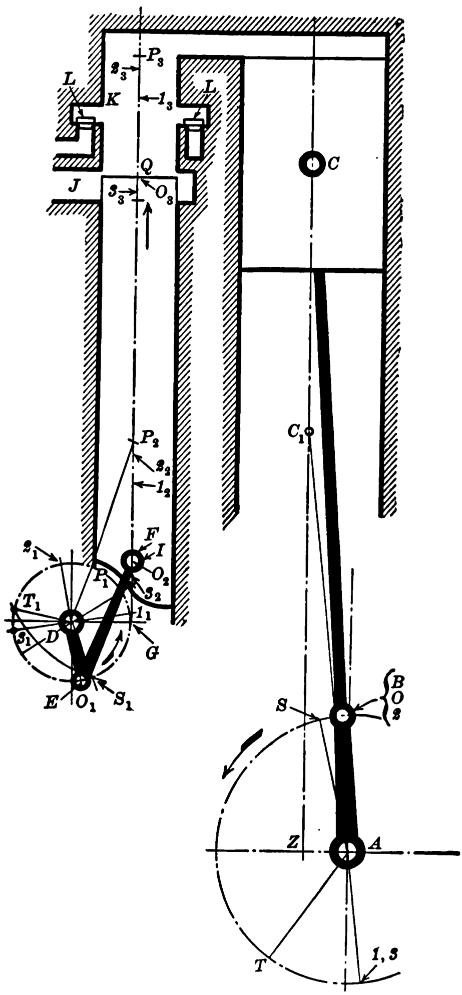


FIG. 357.—GASOLINE ENGINE OPERATED BY SINGLE PISTON VALVE AND ECCENTRIC

LAYOUT FOR GASOLINE ENGINE OPERATED BY SINGLE PISTON VALVE AND ECCENTRIC

590. In order to secure proper timing for both exhaust and inlet ports with only a single piston valve operated by an eccentric, an offset type of construction for both the piston valve and the engine piston has been used. This is illustrated diagrammatically in Fig. 357, where *A* is the main shaft, *B* the crank pin, *B C* the main connecting rod, and *A Z* the offset of the main cylinder. The eccentric shaft is at *D* and is driven from shaft *A* by gearing, not shown, at one-half the speed of *A*. *D E* is the

eccentric radius, $E\,I$ the eccentric connecting rod, and $I\,Q$ is a long piston valve. The exhaust port is at J and the inlet port at K, the latter containing a ring automatic check valve L in some models and in others a positive driven valve. The offset of the piston valve is $D\,G$, equal to the eccentric radius, or nearly so.

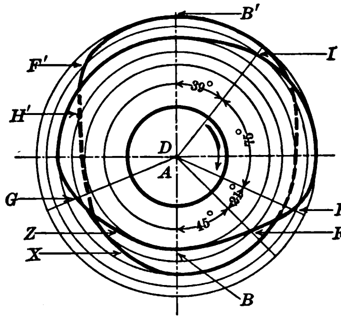
In order to obtain correct timing with such a construction as outlined above, it is necessary to locate a point F on the valve center line such that when an arc of radius $E\,I$ is drawn with it as a center it will cut the circle $E\,G$ in two points S_1 and T_1, which are, say, 115° apart, assuming in this case that the exhaust takes place while the main crank travels 230° as shown in Fig. 342, and also at $T\,1\,S$ in Fig. 357. The point F may be thus found by setting a dividers to 115° on the circle $E\,G$ and setting a compass to a radius $E\,I$ and manipulating both until the desired settings are found.

591. When the crank pin is at B, the main piston is at the top of its stroke, the eccentric center is at E, and the top Q of the piston valve is moving up and is just about to close the exhaust port J. It will close when E reaches S_1, the arc $E\,S_1$ representing the lag at exhaust closure. While the crank pin moves from S to 1 (nearly) suction takes place through the automatic check valve L and it is stopped just before 1 when the edge Q of the piston valve passes the upper edge of the inlet port K. While the crank travels from 1 to 2 compression takes place, and during this period the top of the valve remains near the top of the stroke with but comparatively little motion, moving from 1_3 to P_3 to 2_3. On the explosion stroke, crank pin moving from 2 to 3, the valve moves down from 2_3 to 3_3, protecting the inlet check valve for a period during the strongest part of the explosion pressure, and finally opening the exhaust port J when the eccentric center reaches T_1 and the crank pin reaches T. The distance from T to 3 represents the exhaust lead which is indicated by $H\,D$ in the timing diagram, Fig. 342. By changing either the offsets or the proportions of the mechanical parts, or both, variations in the timing events may be secured. The general forms of piston valve and gear here described are in use on the Acme "slide-valve" gasoline engine.

Detail of Cam Construction for Single Piston-Valve Gasoline Engine

592. Another form of piston valve which is placed in the head end of the cylinder instead of on the side is shown in Figs. 358 and 359. This valve acts also as a head-end cylinder head, while the main

piston acts as a crank-end cylinder head, and the cylinder itself is merely a heavy tube open at both ends. The piston valve is shown in section by the letters *J K* placed at diagonal corners and is op-



AF, Exhaust closes, *AH*, Intake opens, *AI*, Intake closes, *AG*, Exhaust opens.

FIG. 360.—ENLARGED DETAIL OF CAMS SHOWN IN FIG. 358

erated by yoke cams *X* and *Z*. The cams are mounted on a cam shaft running the entire length of the multi-cylinder casting, both the cylinders and the valves being milled out to receive this shaft. The middle cam *Z* acts on a circular roller *Y* which is mounted on a pin *E* set in the piston valve. The cams *X* and X^1 are identical and operate against cross bars or strips *1* which are part of or fastened to the piston valve. An essential feature of the yoke cam construction is that the diametral distance from the edge of one cam *X* to the edge of the other cam *Z* is the same at all places. With this feature in mind it is easy to understand that the piston valve is under positive control all of the time. The intake passageway for the explosive charge is shown at *T* completely surrounding the cylinder, and the width of the intake port in the cylinder is small, as shown

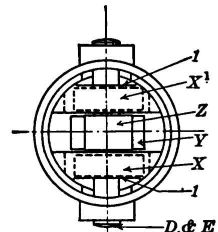


FIG. 359

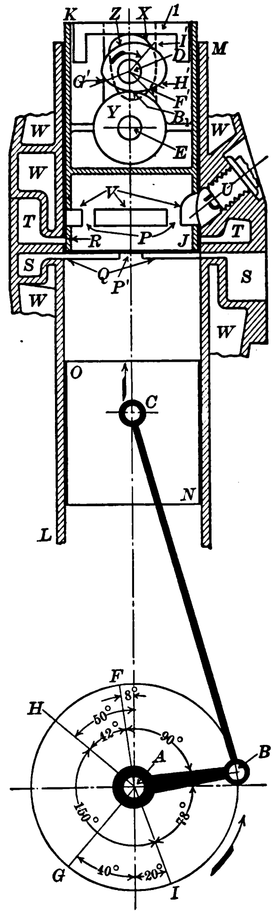


FIG. 358

FIGS. 358 AND 359.—GASOLINE ENGINE CONTROLLED BY CAM-OPERATED PISTON VALVE at R, while the width of the intake port in the piston valve wall is larger, as shown at V, and is further enlarged at U, where the ignition spark plug is situated. The exhaust port Q is also narrow, to keep down valve travel. It also completely surrounds the cylinder except for bridges as at P^1 and empties into the enlarged exhaust passageway S. In the illustration the actual width of the exhaust port, which is represented at Q, was taken ¼ inch, measured on full scale; the bridge between S and T, ½ inch; the inlet port R, ¼ inch; and the distance from the lower edge of the valve to the opening V, 1½ inches. Water-jacket spaces are indicated at W. The cam shaft D is operated positively by chain drive and it turns at one-half the speed of the main shaft A.

593. The timing of this piston valve is as follows: In the position shown the piston is moving up on the exhaust stroke and the valve is at rest, as indicated by the fact that the cam curve Z, at B_1, and the cam curve X, at the top, both follow an arc of a circle for a short distance. This is shown more plainly in the enlarged drawing of the cams at B and B' in Fig. 360. In this Figure concentric circles have been drawn ¼ inch apart across the cam curves, in order that the motion produced by the cam may be compared with the port, bridge and valve dimensioning as given in the preceding paragraph. Shortly after the crank passes $A\ B$, Fig. 358, the cam Z begins to extend radially, and it drives the valve down ¼ inch until it closes the port Q, when the crank is at $A\ F$, thereby giving an exhaust closure lag of 8° in this case, as shown in the Figure. When the crank reaches $A\ H$ the valve is lowered ½ inch further and the ports V in the valves are just uncovering the intake ports R in the cylinder. The valve continues down ¼ inch further, and then moves up until, when the crank reaches $A\ I$, the admission is cut off and there is an intake closure lag of 20°. The valve continues up for ¼ inch, until it laps both inlet and exhaust ports by ¼ inch, when it remains stationary, as shown by the circular arc of the cam Z for approximately 90° from the $D\ B'$ position in Fig. 360. During the period that the valve remains stationary the crank turns approximately 180° from the $A\ B$ position, Fig. 358, and the piston is on its compression and combustion strokes. When the main piston is about half way down on its combustion stroke the valve begins to move and it opens the exhaust port Q when the crank reaches $A\ G$, thus giving an exhaust lead of 40°. The valve continues to move up until the exhaust port is full open and it then remains stationary until after the crank reaches $A\ B$ and the cycle is repeated. The cam X is complementary to the cam Z, the diametral distances,

$B\,B'$, $F\,F'$, $H\,H'$, etc., Fig. 360, all being equal. The general features and principles here described are embodied in the Carter Piston Valve Motor, but the actual numerical values used to illustrate this type of valve are entirely independent of the valves used by any manufacturer so far as known.

Franklin Air-Cooled Engine

Compound Poppet Valve Construction

594. A modified form of the poppet valve, as shown at F and E in Fig. 361, is used on the older types of the Franklin air-cooled motor. The intake is at D through the skeleton poppet valve F, which not only moves down through the action of the rocker arm $L\,M$, but it carries with it the solid exhaust poppet valve E as if both were of one solid piece. When it is time for exhaust at K the valve E only is pushed down by rocker $Q\,R$ acting through the exhaust valve stem B. A rigid bracket is shown at T to take the compression of the exhaust valve return spring N_1. N is the return compression spring for the intake valve.

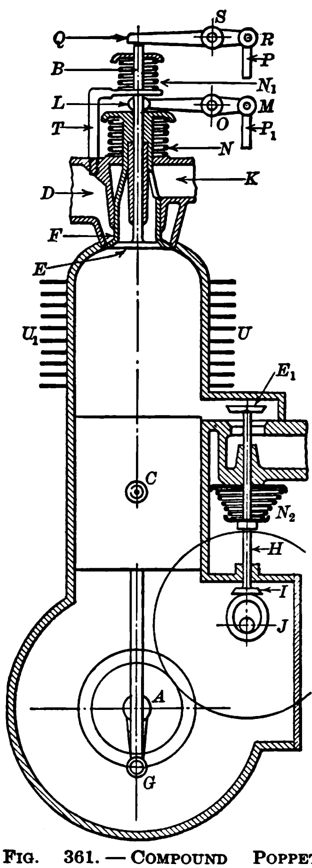


FIG. 361. — COMPOUND POPPET VALVES AND AUXILIARY EXHAUST VALVE ON FOUR-STROKE ENGINE

Auxiliary Exhaust Valve at Bottom of Cylinder in Four-Stroke Engines

595. The placing of an auxiliary exhaust valve at the bottom of the cylinder is characteristic of this older type of four-stroke air-cooled motor. It is unusual so to place an extra exhaust valve in four-stroke engines, but it served a special purpose in the air-cooled type and it has also been and is still used occasionally on the water-cooled type. The purpose is to allow a larger part of the hot exhaust gases to escape quickly, when the smaller portion will expand and will cool on doing so. It is then exhausted through the main ex-

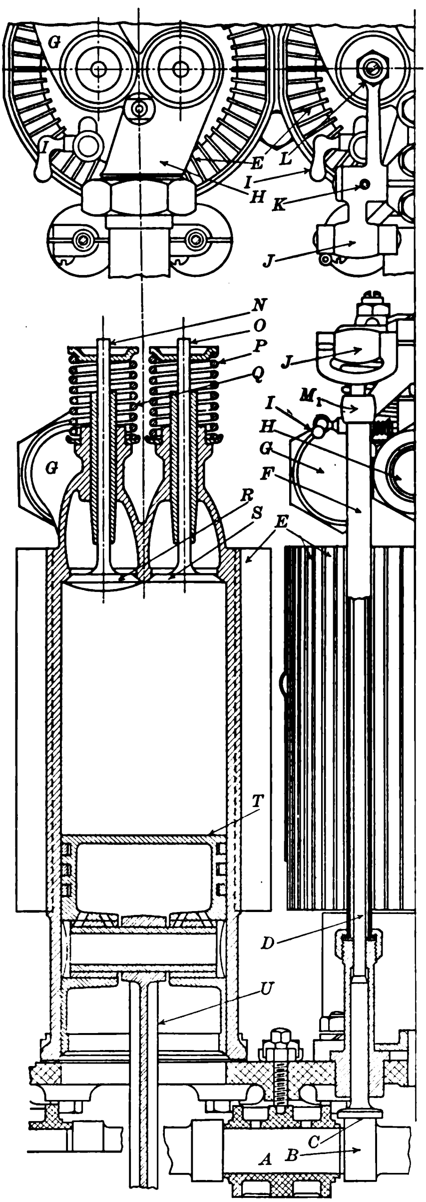


FIG. 362.—FRANKLIN AIR-COOLED ENGINE

haust valve at E, without subjecting the valve parts to the usual high exhaust temperature. This entire action together with the radiating flanges U U_1, which in this type are horizontal, are sufficient to keep the engine cool

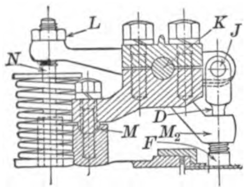


FIG. 363.—DETAIL OF VALVE-ROCKER SUPPORT, TO LESSEN EFFECTS OF EXPANSION

enough for continuous running, without the auxiliary cooling by running water through cored passages such as is necessary in the ordinary gasoline engine. When the engine is running, the main exhaust valve opens a little before the auxiliary one closes. The end of the cam-shaft line is shown at J, and cams with mushroom followers, such as illustrated at I, are distributed along the cam shaft to operate the rocker connecting rods P and P_1.

Separate Admission and Exhaust Valves in Cylinder Head

596. The later type of Franklin air-cooled motor has entirely separate admission and exhaust valves set side by side in the cylinder head as shown at S and R in Fig. 362. The auxiliary exhaust

valve has also been omitted, it having been found that sufficient cooling of the cylinder is obtained by the following method of construction, which is also new: The cylinders, including the cooling flanges E which have been placed vertical instead of horizontal, are surrounded by sheet-metal jackets which open into a chamber which conducts the cooling air from a strong-draft flywheel fan of the centrifugal type to each cylinder which thus obtains fresh cooling air without getting heated air from any previously cooled cylinder. The vertical cooling flanges which serve to increase the rate of heat radiation from the cylinder walls are of steel $1/16$ inch thick, and are fastened in place in the cylinder by being placed in their proper positions in the molds before the cylinder casting is poured. The parts of the engine that are shown are:

A, Cam shaft.
B, Cam.
C, Cam follower disk.
D, Cam follower rod.
E, Radiation flanges.
F, Cam-rod casing.
G, Exhaust.
H, Fuel intake.
I, Priming valve.
J K L, Valve rocker.
N, Exhaust valve-stem.
O, Admission valve-stem.
P, Admission valve spring.
Q, Exhaust valve spring.
R, Exhaust valve.
S, Admission valve.
T, Engine piston.
U, Main connecting rod.

Effect of Expansion of Cylinders and of Valve-Gear Parts on the Valve Timing

597. A fine, but important point in the design of the valve gear for practically all internal combustion engines is strongly exemplified in this new Franklin model. It has to do with the expansion of the cylinder and of various parts of the valve gear when subjected to the high temperatures of combustion and exhaust. This expansion, when exerted on long valve rods, and on cylinders or on other heated parts of the framework which may carry points of support for rocker arms and levers, is often sufficient to entirely change the timing periods for the opening and closing of the valves and to seriously affect the efficiency of the engine, even though clearances are allowed at certain contact joints to mitigate the troubles arising from this cause.

598. A practical example of the designers' problem in taking care of the effects of expansion in the valve gear is pointedly illustrated in the old and new Franklin models. In the former, Fig. 361, the

supports O and S of the rocker arms were fastened only to the cylinder casting. When the cylinder became heated these supports rose in height with the cylinder, and as the push rods P and P_1 did not become heated, the clearance space at the bottom of each of these rods, where they came into pressure contact with the rods from the cam at the proper times, was increased. In the new model, the rocker arm $J\ K\ L$, Figs. 362 and 363, is pivoted at its fulcrum K to a bearing having three points of support, one being an oscillating joint M connected with the cylinder casting, and the other two, M_1 and M_2, being fastened to the supporting tubes F, F, surrounding the push rods D. As the cylinder becomes heated the support at M rises and carries with it the pivot point K, which moves up a proportional amount about the supports M_1 and M_2 which remain stationary because the tubes F are not heated. As K on the rocker arm rises about J as a fulcrum the point L rises proportionally, the whole design being planned to make the rise of L on the rocker equal to the rise at L on the valve stem. This latter rise equals that due to the expansion in the cylinder plus that in the valve stem N. This compensation for expansion permits of much reduced clearance between the push and cam rods.

The Knight Engine

599. The double-sleeve type of valve for gasoline engines was invented by Charles Y. Knight, and patented in the United States in 1912. Although it differs radically from all other valve forms that were used up to that time, it has been adopted and has remained in continued use by several automobile manufacturers both in Europe and in this country. While the Knight sleeve valve is, in effect, a hollow piston-valve type, it has distinguishing structural characteristics, in that it is made to surround the main engine piston instead of being located in a separate valve chest, and in that it is built in two separate cylindrical parts. By using two sleeves, necessary freedom in securing a desired timing of events is secured with the simple use of eccentrics instead of with cams. A single sleeve may be used if a cam or other suitable mechanism is designed to operate it. Mr. Knight's first sleeve-valve patent, taken out in 1910, was for a single-sleeve valve. Several single-sleeve valve engines are now being built both in this country and abroad.

600. The fundamental features of the design of a double-sleeve valve motor, together with necessary and useful diagrams, are shown in their elementary form in Figs. 364–367, while a more com-

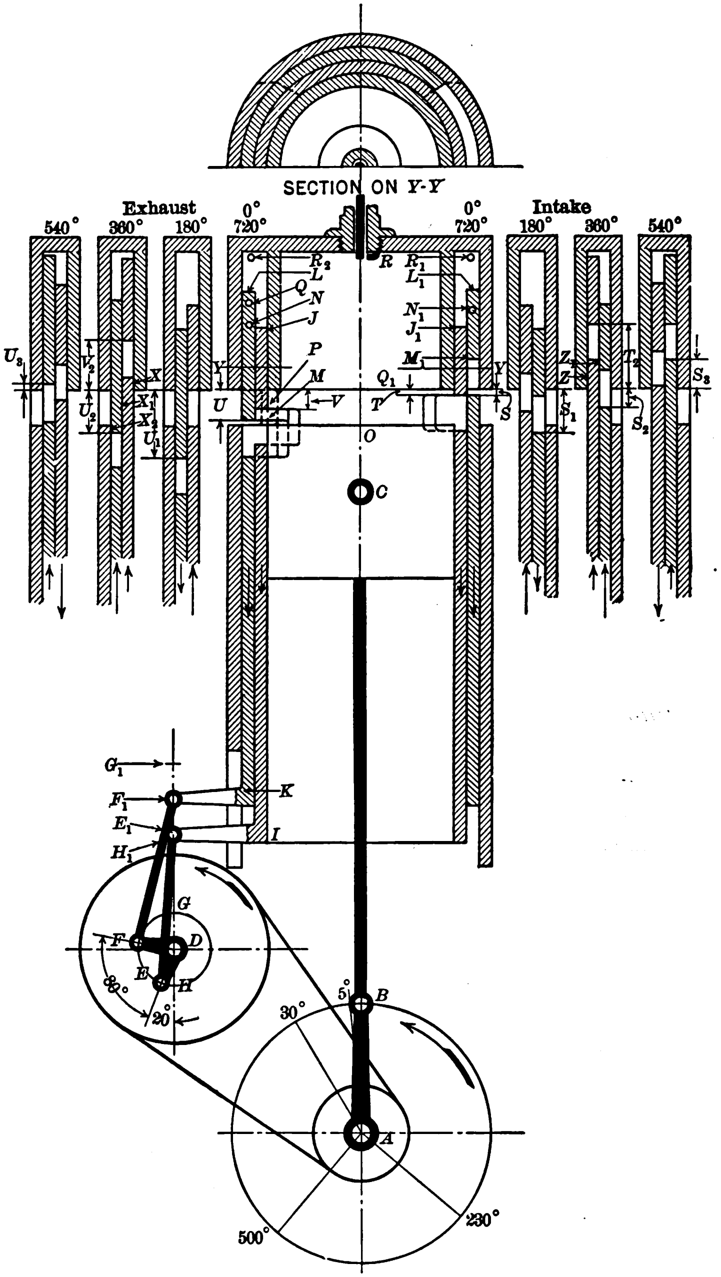


FIG. 364.—SHOWING DEAD-CENTER PHASES OF ELEMENTARY DOUBLE-SLEEVE-VALVE GASOLINE ENGINE

plicated but more practical application as used in the Lyons-Knight engine is illustrated in Figs. 368–370. In Fig. 364, the main engine shaft is shown at A, and the sleeve-valve eccentric shaft at D, the latter being driven at one-half the speed of the main shaft by means of chain and sprocket wheels. The arms $D\,E$ and $D\,F$ represent eccentrics which drive the inner sleeve $I\,J$ and the outer sleeve $K\,L$ up and down through the distance $H\,G$ by means of the connecting rods and sleeve arms as represented in the Figure.

The Designing of Double-Sleeve Valves

601. The first step in the design of the sleeves, after calculating the widths of the ports in the cylinder wall and in the sleeves, is to determine the relative positions of these port openings in their respective parts, in order to secure best results as to timing, velocity of opening and closing, and sealing. By sealing is meant the amounts of overlapping of the sleeve ports between themselves and between them and the main cylinder wall and cylinder head ring in order to prevent leakage. In this elementary illustration the port widths in the cylinder wall and in both sleeves will be taken the same both for intake and exhaust, and the bottom edge of the port in the cylinder wall will be taken at the same elevation as the piston head O at the top of its stroke.

The Use of Superposed Sinusoidal Valve Diagrams

602. In order to locate the positions of the ports in the sleeves to best advantage for timing, for velocities, for sealing and for the angle of advance $E\,D\,F$ of the crank $D\,E$ ahead of $D\,F$, a double superposed sinusoidal diagram is most useful, and, in fact, is essential if a quick effective study is to be made of all the many possible combinations. Such a diagram is represented in Fig. 366. If the sleeves had harmonic motion in their travels up and down, such as would be obtained by the use of "infinite connecting rods" at $E\,E_1$ and $F\,F_1$, Fig. 364, the wave curves would be true sinusoids and could be readily constructed. Since, however, finite connecting rods are used in practice the sleeve motions are irregular, being slower on the lower than on the upper parts of the strokes as shown by comparing the values $H_1\,3$ and $3\,G_1$, Fig. 365.

603. The practical curve showing the sleeve motion is constructed in Fig. 365, where $D\,G$ is the eccentric radius, $G\,G_1$ the connecting rod length, $H_1\,G_1$ the full motion of the sleeves, and $H_1\,A$ any desired convenient length for the "sinusoidal" diagram. Divide $H_1\,A$ into any convenient number of equal parts and erect ordi-

nates; divide the eccentric circle $H\ G$ into the same number of equal parts as at *1, 2, 3*... and find the corresponding positions of the crosshead end of the connecting rod. From the latter points project lines over to the corresponding ordinates and draw the "sinusoidal" curve $H_1\ B\ A$ through the intersecting points. In this elementary illustration the two connecting rods operating the two sleeves are of the same length, and consequently this one "sinusoidal" curve will be enough, but in practice the connecting rods are of different lengths on account of structural considerations and "sinusoidal" curves must be constructed for each rod.

Having obtained the curve $H_1\ B\ A$, trace it in ink on a piece of tracing cloth or tracing paper and extend it on each side, that is, beyond H_1 and beyond A. On the same tracing cloth draw another exactly similar curve parallel to the one already drawn and at a distance from it equal to the port width in the outer sleeve. Draw also, in ink, fine vertical and horizontal reference lines through the lowest and highest points of the "sinusoidal" curve. This piece of tracing cloth should now be marked, "For the intake port, outer sleeve." Other and similarly made sinusoidal diagrams should be made, in general, for the intake port inner sleeve, exhaust port outer sleeve, and exhaust port inner sleeve, but in this problem all ports are taken to be equal in width and both sleeves to have the same travel and, therefore, there will be but two pieces of tracing cloth used, one for the outer sleeve and one for the inner sleeve, and the sinusoidal curves and reference lines on each will be identical.

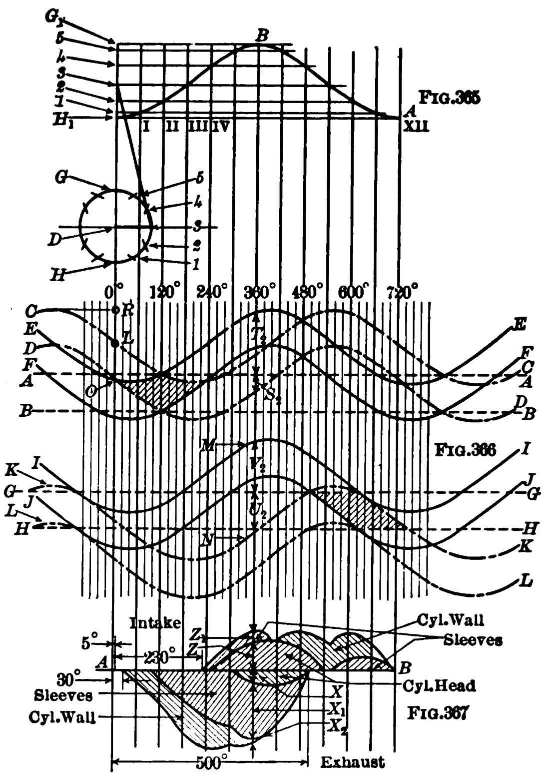
604. On a separate and fixed piece of paper draw two straight parallel lines in ink, placing them a distance apart equal to the port width in the cylinder wall. These lines are shown at $A\ A$ and $B\ B$, Fig. 366. Draw another parallel line, representing the bottom edge, Q_1 (Fig. 364), of the cylinder head ring which acts as an inside cut-off edge and as a sealing surface. In this problem, the edge Q_1 of the ring is at the same elevation as the top edge of the port in the cylinder and the new parallel line representing Q_1 will, therefore, coincide with $A\ A$ in Fig. 366. On one of the horizontal lines thus drawn mark off a length equal to one wave length, equal to $H_1\ A$, Fig. 365, of the sinusoidal curve. This length will represent two complete turns of the engine crank, or 720°, and should be divided into a number of equal parts. The greater the number of parts taken, the more accurate will be the readings of the timing angles to be taken later from this chart. In this illustration, Fig. 366, 36 parts are used.

605. In order to apply the fixed chart just drawn and the two tracings, it is necessary to have some desired timing action in mind,

and in this problem we will endeavor to obtain the conditions represented in the timing diagram of Fig. 342, and stated in Col. A, below.

	Col. A Desired Timing Action	Col. B Timing Action Obtained
Exhaust closes	10°	30°
Intake begins	15°	5°
Intake ends	200°	230°
Exhaust begins	500°	500°

606. By placing first the tracing paper marked "For outer sleeve" over the fixed chart and then the tracing marked "For inner sleeve"



Figs. 365–367.—Showing Sinusoidal and Sealing Diagrams for Elementary Double-Sleeve-Valve Engine

over the first tracing, and manipulating the two tracings with respect to each other and also with respect to the fixed chart, all possible timing actions may be studied, and one of them adopted. The upper half of Fig. 366 shows a final adjustment of the tracings, the dash-and-dot "sinusoids" representing the motion diagram of the outer sleeve port, and the solid line curves the inner sleeve port, while the straight dash lines represent the fixed port in the engine cylinder wall. The cross-sectioned area represents the period of intake, while any ordinate in this area shows the amount that the port is open at the corresponding crank position. The intake port, it will be seen from the point O, which is one-quarter of the first 20° space, begins to open when the main crank is at 5°, is full open at 120°, and closes at 230°.

607. With the final adjustment, obtained as in the preceding paragraph, it will be noticed that the inner sleeve reaches its lowest position at 40° and the outer sleeve at 200°. This is a difference of 160° for main crank positions, but since the eccentric shaft runs at one-half the engine speed the eccentric arms will have to be set with 80° between them. This 80° is the angle of advance by which the inner eccentric precedes the outer eccentric, and is shown by the angle $E\,D\,F$ in Fig. 364.

608. The two tracings for the exhaust opening are next superposed and various adjustments tried *always with the same angle of advance as obtained above,* until the most satisfactory combination is found as in the lower part of Fig. 366. The sectioned part of the chart represents the exhaust period which begins at 500°, is maximum at 620°, and closes at 30°. The results obtained in this and in the two preceding paragraphs are tabulated above in Col. B. They do not agree with the timing action given in Col. A on account of the elementary data here used. By adopting a more complicated construction the desired action may be obtained, as pointed out in paragraphs 619 to 623.

609. The relative settings of the sleeves, the cylinder, and the eccentrics may now be drawn for any particular phase by taking proper values from the chart in Fig. 366. In Fig. 364, assume the crank $A\,B$, the connecting rod $B\,C$, and the diameter and length of the piston. Draw the cylinder intake and exhaust ports with the lower edges flush with the top of the piston when in its highest position in order to agree with the assumption made in paragraph 601, and finally take the eccentric shaft in any convenient position approximately at D so that the same length connecting rods $F\,F_1$ and $E\,E_1$ may be used as described in paragraph 602.

610. To locate the outer sleeve intake port for the 0° position, measure on the 0° ordinate of the chart, Fig. 366, the distance bebetween the straight line A and the "sinusoidal" curve D and lay off this distance at S in Fig. 364. To find the outer sleeve port posi-

tion at the 180° phase again take the distance between the same two lines of the chart, but on the 180° ordinate and lay off this distance at S_1. Similarly the outer sleeve port positions are determined for the 360° and 540° phases.

611. To locate the inner sleeve intake port for the 0° position, measure on the 0° ordinate of the chart the distance between the lines A and E and lay it off at T in Fig. 364. The values of T and S are nearly the same, T being a trifle larger. To locate the port at the 180° phase, measure the distance between the same lines, but on the 180° ordinate, and do the same at the 360° and 540° positions. The value at the 360° position is shown at T_2 on both the chart and the engine diagram.

612. The position of the exhaust port in the outer sleeve relatively to the cylinder port at the 0° phase is found by measuring the distance between the lines G and K on the 0° ordinate and laying it down at U in Fig. 364. To find the positions of the same port at the 180°, 360°, and 540° phases, take the measurements on these ordinates between the same lines G and K.

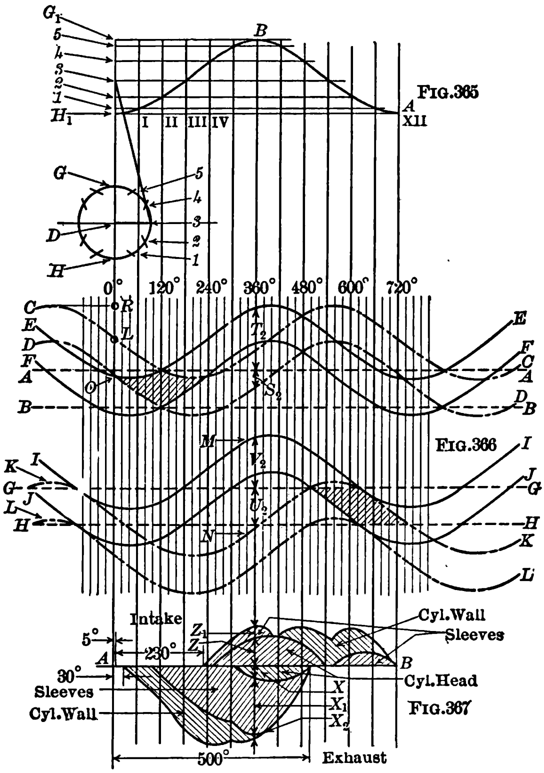
613. The exhaust port in the inner sleeve is located by measuring the distance between the lines G and I on the 0° ordinate in Fig. 366, and laying it off at V in Fig. 364. Similarly, distances are taken on the 180°, 360°, and 540° ordinates and the values laid off at these and other desired points. The values at 360° for both the inner and outer sleeves are indicated at U_2 and V_2 in Figs. 366 and 364.

Determining Sealing Laps on the Sleeves and the Position of the Cylinder Head

614. The upper extremities of the sleeves, or the distances above the ports must be of such length that they will not expose the adjacent port openings at any phase in their motions up and down. To take care of this, it is necessary to observe the maximum distance the upper edge of the intake port of the inner sleeve moves above the upper edge of the intake port of the outer sleeve. This distance is found on the 340° ordinate of Fig. 366 between the lines E and C, and it is laid off at $M_1 N_1$ in Fig. 364. An arbitrary distance, $N_1 L_1$, is then taken for a lapping edge. The same operation is then repeated, starting with the maximum distance the intake port in the outer sleeve moves above the intake port in the inner sleeve. In this problem the values found for the inner sleeve will be identical with those found for the outer sleeve for the intake side.

The highest point reached by the top of the outer sleeve is at R_1, the distance $L_1 R_1$ being taken from the 0° ordinate as shown at $L R$ in Fig. 366.

615. The top edge of the exhaust port in the outer sleeve moves a maximum distance, $M N$, below the top edge of the exhaust port in the inner sleeve as shown at the 340° ordinate between the lines K and I in Fig. 366. Allowing $N Q$, Fig. 364, for overlap, ($= N_1 L_1$), Q would be the highest necessary edge of the outer sleeve so far



FIGS. 365–367 (Duplicates).—SHOWING SINUSOIDAL AND SEALING DIAGRAMS FOR ELEMENTARY DOUBLE-SLEEVE-VALVE ENGINE

as exhaust sealing purposes are concerned. If, however, it is desired to finish the outer sleeve square across, $Q L$ would be further added so that L and L_1 would be in the same plane.

The distance that the upper edge of the exhaust port in the inner sleeve moves below the upper edge of the exhaust port in the outer sleeve is zero, as shown by the tangency of the "sinusoids" K and I in Fig. 366, and, therefore, only a small distance equal to the overlap, $N_1 L_1$ would be necessary above P on the inner sleeve. But more is added, not only to overlap the cylinder-head ring above Q_1, but so as to square off this sleeve also and make J at the same elevation as J_1. It is now found that R_1 is the highest point reached by either of the sleeves and, after leaving a small clearance $R_1 R$, the elevation of the cylinder head with reference to the cylinder ports is determined. In this problem it so happens, when allowance for the height of sleeves above sleeve ports is made with reference to the sleeve ports only, that there is ample allowance for height of sleeves above the fixed cylinder ports also, but with some combinations of data it may happen that the same operations for heights of sleeves as above would have to be made with reference to the cylinder wall ports in addition.

Locating the Sleeve Arms

616. The lengths of the valves below the valve ports and the positions of the arms $E_1 I$ and $F_1 K$ are next to be determined. It will be assumed that the necessary lengths of these valves for proper bearing pressures and lubrication have been determined and that the resulting connecting rod lengths are $E E_1$ and $F F_1$, as already taken at the outset. Assuming the position D for the eccentric shaft, the strokes $H_1 G_1$ for crossheads may be laid down. For the zero position of the main crank the outer sleeve is found to be down from its topmost position $R L$, Fig. 366, and this distance is laid off at $G_1 F_1$ in Fig. 364. Similarly, the inner sleeve is found to be down the distance $R O$, and this is laid off at $G_1 E_1$. The arc $E F$, found by drawing in the connecting-rod positions, should then be equal to 80° as determined in paragraph 607.

The Drawing of the Sealing Diagram

617. A sealing diagram which shows the aggregate of the overlapping edges which tend to prevent leakage, is useful, and is shown in Fig. 367. Any total ordinate included in the cross-sectioned area above $A B$ shows the amount of sealing of the intake port at that particular crank position. There is no sectioning from 5° to 230°, and consequently the intake passageway is open. At 360°, for example, when the explosion has just taken place or is about to take place, the bottom port edge of the inside sleeve is above the cylinder head ring by the amount that the "sinusoid" F is above the line A in Fig. 366, and this value is laid off at Z in Fig. 367, and at Z in Fig. 364. At the same time, the bottom edge of the inner sleeve port

is above the top edge of the outer sleeve port by the amount of the distance between the curves F and C, and this is laid off at Z_1 in both Figs. 367 and 364. Therefore, the seal for the intake ports at the 360° crank position, or top of compression stroke, is $Z + Z_1$.

618. The exhaust seal at the 360° crank position is made up of three distinct lappings. First, the amount the lower port edge of the inner sleeve is above the edge of the cylinder head ring, as represented by the distance between the curve J and the line G in Fig. 366, and shown at X in Figs. 367 and 364; second, by the amount the lower port edge of the inner sleeve is above the top port edge of outer sleeve, as measured between the curves J and K, and laid off at X_1; third, by the amount the upper port edge of the outer sleeve is below the bottom port edge of the cylinder wall port, as measured by the distance between the curve K and the straight line H and laid off at X_2 in Figs. 367 and 364.

Illustrations from the Working Drawings of the Lyons-Knight Sleeve-Valve Engine

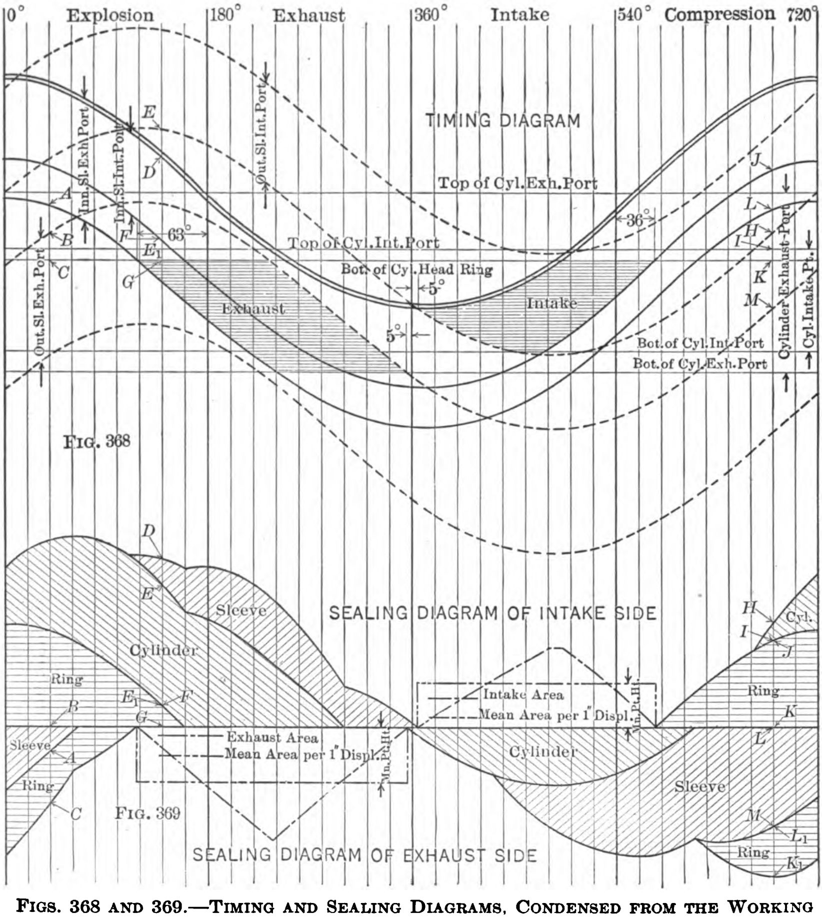
619. The problem of laying out a double-sleeve valve is now complete in its elementary form, as stated at the outset. Such an elementary arrangement, however, may not give the best timing events, and in order that a practical case may be studied, the timing diagram and layout for a $4\frac{1}{2} \times 5\frac{1}{2}$ Lyons-Knight sleeve-valve engine are given in Figs. 368 to 370. In this practical case, the cylinder intake and exhaust ports, and also the sleeve intake and exhaust ports all vary in width. Furthermore, the intake starts with the opening of the sleeve ports while they are in front of the cylinder wall intake port, instead of with the opening of the outside sleeve port and the cylinder wall port, as in the simple case. This is shown in Fig. 368 by the curves E and D intersecting at 365°, where they meet exactly at the center of the cylinder intake port. Other practical considerations are that the piston should not travel over the sleeve port as it will scrape off the oil, which will go out with the exhaust, causing smoke and loss. The ends R and S of the valve-sleeve eccentric rods, Fig. 370, are set to the right of a vertical line through U, the axis of the eccentric shaft. The actual dimensions of the several ports are:

Cylinder wall intake port.....	$\frac{9}{16}''$	Cylinder wall exhaust port...	$1''$
Outer sleeve intake port......	$\frac{9}{16}''$	Outer sleeve exhaust port....	$\frac{11}{16}''$
Inner sleeve intake port.......	$.448''$	Inner sleeve exhaust port.....	$\frac{11}{16}''$

Bottom of cylinder head ring above bottom of cylinder exhaust port..... $.63''$

The travel for each sleeve is 1¼ inches.

620. If the "sinusoidal" curves which may be obtained with the above data are now laid off on pieces of tracing cloth, as directed in the explanation of the elementray problem and trial positions of the superposed tracings studied, a timing action will be obtained similar



FIGS. 368 AND 369.—TIMING AND SEALING DIAGRAMS, CONDENSED FROM THE WORKING DRAWINGS OF THE LYONS-KNIGHT ENGINE

to that shown in Fig. 368, when the crank driving the inner sleeve has an angle of advance of 60° over the crank driving the outer sleeve. This angle of advance appears doubled, as 120°, in the timing diagram, as may be readily noted by observing that the four solid sinusoidal curves for the edges of the ports in the inner sleeve all have their crests on the 0° ordinate, while the crests of similar curves for the outer sleeve are all on the 120° ordinate. The angle of advance in its true size of 60° is shown at *V U T* in Fig. 370. At the phase there illustrated, the inner sleeve is at the bottom of its stroke, while the outer sleeve is moving down. It will be noted that the two sleeve connecting rods *T R* and *V S* are of different lengths and that this will cause the dotted "sinusoidal" curves to be slightly different from the solid ones in Fig. 368, the former set approximating more nearly to the form of the true sinusoid. The timing points from the diagram, Fig. 368, may now be read off as follows:

Exhaust opens	117°	Intake opens	365°
Exhaust closes	355°	Intake closes	576°

621. The several parts of the engine, some of which have already been mentioned, are shown by reference letters in Fig. 370, as follows:

A, Cylinder head.
B, Cylinder head packing rings.
C, Cylinder head ring acting as opening and closing edge.
D, Cylinder head inside packing rings.
E, Outer sleeve intake port.
F, Inner sleeve intake port.
G, Cylinder intake port.
H, Intake passageway.
I, Piston packing rings.
J, Cylinder and water jacket walls.
K, Cooling-water space.
L, Cylinder exhaust port.
M, Inner sleeve exhaust port.
N, Outer sleeve exhaust port.
O, Exhaust passageway.
P, Piston.
Q, Piston pin.
R, Outer sleeve connecting-rod pin.
S, Inner sleeve connecting-rod pin.
T, Outer sleeve crank pin.
U, Center of crank shaft operating sleeves.
V, Inner sleeve crank pin.
W, Center of main crank pin.
X, Center of main crank shaft.

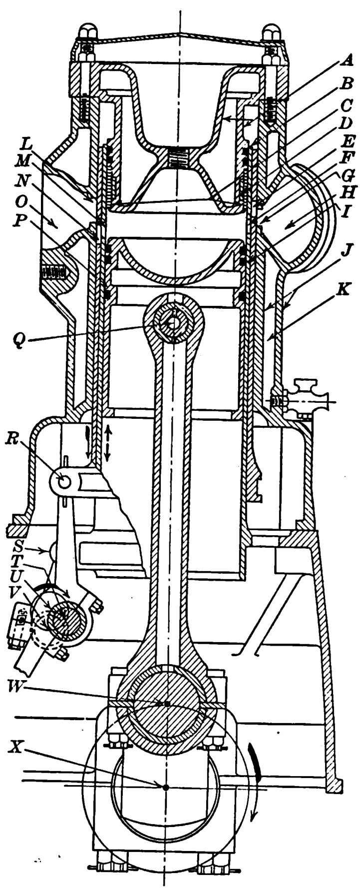


FIG. 370.—SECTION OF THE LYONS-KNIGHT SLEEVE-VALVE ENGINE

622. A practical sealing diagram for the Knight type of valve is drawn in Fig. 369, where it is shown that when the main crank $X\,W$ of Fig. 370 is on the explosion stroke and has turned 40°, the portway is sealed by the inner sleeve exhaust port overlapping the ring in the cylinder head by the amount $A\,C$. This value is equal to and is taken from $A\,C$ in Fig. 368. Should any gases leak through this seal it would still have to pass between the inner and outer sleeves the distance $A\,B$, which is also taken from $A\,B$ in Fig. 368. There is also a double seal, which is larger, that prevents leakage into the intake passageway when the crank is at 40°, as shown by the length of the ordinate above B that lies in the cross-section.

623. When the crank is at 140° and the engine exhausting, there is a triple seal protecting the intake passageway: first, by the amount $G\,F$ that the inner sleeve is above the cylinder head ring; second, by the amount $D\,E$ that the two sleeves overlap each other; and third, by the amount $E\,E_1$ that the outer sleeve intake port is above the cylinder wall intake port. The amounts of sealing for the intake and exhaust portways when the crank is 40° from the top on the compression stroke are also shown in Fig. 369, the former at $K\,J + I\,H$ and the latter at $L\,M + L_1\,K_1$. These values are determined as in the preceding cases.

A Crescent-Shaped Valve Working Between Piston and Cylinder

624. A crescent-shaped valve fitting in a corresponding recess in the cylinder wall is illustrated in Figs. 371 and 372. It slides up and down between the cylinder wall and piston, but does not encircle the piston as does the sleeve valve. Two crescent-shaped valves, one for admission and one for exhaust, are used in the Fischer motor which is shown in perspective in Fig. 371, where A is the crank case, B the bearing, C the main crank, D the connecting-rod end, E the box cam, F the cam pin follower, $F\,H\,G$ the bell crank connecting the cam and valve, I the crescent-shaped admission valve, J the water-cooling space between the two walls of the cylinder casting, K the engine piston, L the admission port in the crescent-shaped valve, M the intake passageway, N the spark plug, O the exhaust port in the exhaust valve, P the exhaust passageway, Q the crescent-shaped exhaust valve, R the cylinder walls, S the handhole cover, $T\,U\,V$ exhaust bell crank, W exhaust cam. The manner in which these valves are set in the engine is shown in regular projection in

cross-section in Fig. 372, where piston, valves, and cylinder walls are shown in practical proportions and are lettered as in Fig. 371.

Layout of Box Cams for Crescent Slide Valves

625. In the design and operation of these crescent-shaped valves one of the first considerations is to lay out the admission and exhaust cams so as to time properly the functioning of the ports *L* and *O* in the valves *I* and *Q*. This is done for the admission valve by laying out the groove in the box cam *E* so that it will cause the arm *F H* of the bell crank *F H G* to swing through such an angle and at such a time as will cause the pin at *G* and the valve *I* to move up and down at such speeds as will give the best opening and closing of the port; and then to hold the arm *F H* and the valve stationary during the explosion stroke and, if possible, to have the valve motion direction such that it is never opposite to that of the piston against which it slides. The exhaust cam is designed along the same general lines. The planning of the cams to secure these results is similar to that explained in paragraphs 579 to 581 inclusive. The cams used on this engine differ from those in common use, in that they give positive action to the valve during the entire time and do not depend on springs to return any part of the valve mechanism.

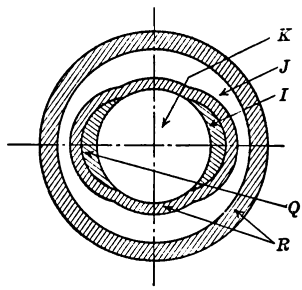


FIG. 372.—SECTION OF FISCHER VALVE AND CYLINDER

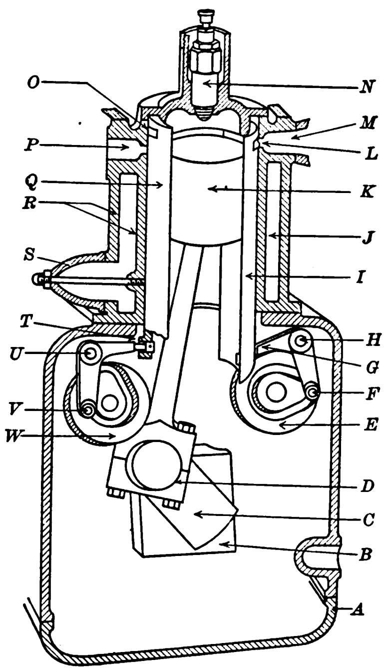


FIG. 371.—FISCHER ENGINE WITH CRESCENT-SHAPED VALVE SLIDING BETWEEN CYLINDER WALL AND PISTON

SPEEDWELL CYLINDRICAL VALVE

626. An example of a rotary full cylindrical valve as used on the Speedwell motor is shown in Figs. 373 and 374. The intake and exhaust valves with their

directions of rotation are indicated in the former Figure, and the valves and the piston are in their proper relative positions for the beginning of the suction stroke. The valves turn with uniform angular velocity with one-quarter the speed of the main shaft from which they are driven by suitable transmission and reduction gears. As the valves are set in the Figure it will be seen that there is a small amount of lead on both the admission and the exhaust closure. The full cycle of events may be readily followed by bearing in mind that the valve turns 45°, always in the same direction, for each stroke of the piston.

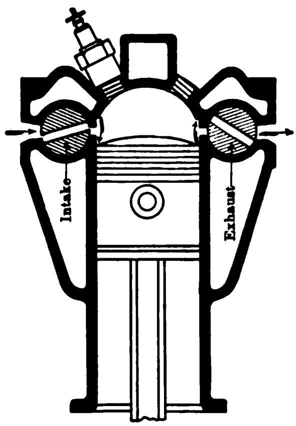


FIG. 373.—SPEEDWELL MOTOR WITH CYLINDRICAL VALVES

627. An illustration of the Speedwell cylindrical valve is shown in Fig. 374 where A and B, A_1 B_1, etc., are split eccentric packing rings which prevent leakage from one cylinder to another and past the ends of the valves. At C, C_1 and C_2 are the openings to the diagonal ports which are set 60° with each other, on six-cylinder engines, the valve shown being connected by an Oldham coupling at F with another identical part for operating a six-cylinder engine. The cylindrical surfaces shown at D, D_1 . . . are filled with rather fine longitudinal grooves which are designed to distribute the oil and to act as packing in preventing leakage around the body of the valve.

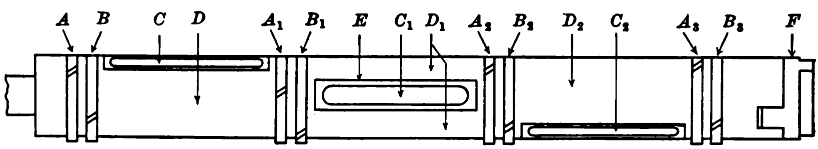


FIG. 374.—SPEEDWELL CYLINDRICAL VALVE

At the edges of the ports through the valves are smooth surfaces as shown at E, accurately machined for a close fit in the cylindrical chamber in which the valves rotate.

DARRACQ CYLINDRICAL VALVE

628. A cylindrical valve with a segmental opening as shown in Fig. 375 is used on the Darracq engine. The valve has a uniform

rotation at one-half the engine speed, admitting the fuel charge from the intake at I to the port at P during approximately the first quarter of the revolution, exhausting the burned gases at E during the fourth quarter-turn, and maintaining a closed port at P during a half-turn when compression and combustion are taking place. The

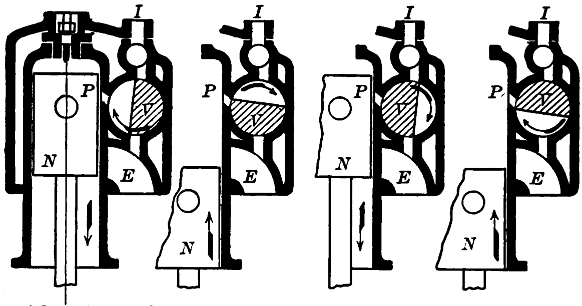


FIG. 375.—DARRACQ CYLINDRICAL VALVE

segmental pocket of the valve is open to the fuel supply channel during the second quarter-turn, but there is no suction to form the fuel mixture and consequently no fuel charge to be pocketed and lost during this period.

629. Cylindrical valves require protection from excessive heat and from pressure during rotation for best results, and this is admirably secured in the Darracq construction, although at the expense of other features which will be presently pointed out. It will be noted that the piston moves considerably above the port P on its up-stroke and closes the port for quite a period. This gives protection to the valve from the initial explosive heat and pressure. When the port is opened the valve is protected from the combustion heat and expansion pressure by the generous water pockets surrounding the valve. The passage of the piston above the port P, however, entraps some of the burned gases which might otherwise escape if the port P were above the piston. Also the piston must move down before the fuel charge can be drawn in, thus rendering a portion of the admission stroke unavailable for charging the cylinder. This form of valve has the mechanical advantage common to cylindrical valves in that it renders cams and springs unnecessary. It is simple in its idea, being nothing but a solid cylindrical bar running the length of the engine, with as many flat spots cut on it as there are cylinders. The machining and fitting, however, must be accurate, and must be maintained so as to prevent binding or leakage.

Cylindrical End Valve

630. A cylindrical end valve, as illustrated in Fig. 376, has been built and has been used in four-stroke engines in motor-boat work, giving quiet and smooth running over a period of a number of years. For an engine of four cylinders, four of these cylindrical-end or rotary disk valves are used. Each valve, *A*, has but a single port, *E*, and a stem, *D*, projecting up through the engine casting into a covered gear chest at the top of the engine. In this gear chest is a train of four helical spur gears, *B*, all in mesh successively so that each succeeding valve turns in an opposite direction. The valves are seated in the heads of the cylinders, and as they rotate at one-half the engine speed, their ports register alternately with the intake and exhaust passages in the cylinder casting. The last helical gear is driven by the one preceding it, and so on to the first gear, which is driven through a vertical shaft from the main engine shaft. Notwithstanding the quiet running of this engine, its manufacture has been discontinued, owing to its inefficiency after it has been running a while, or after it becomes thoroughly heated. When heated, the valves evidently warp and allow the gas to blow through, thus losing compression and power. On a test, one of these engines developed 22 horse-power at starting, but after twenty-five minutes only 15 horse-power was obtainable.

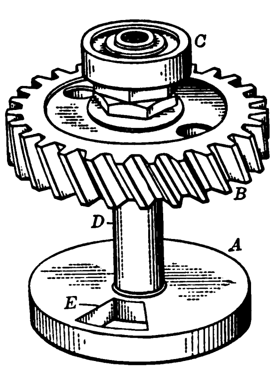


Fig. 376.—Cylindrical End Valve

Four-Stroke Engines without Cams, Push Rods, or Springs

631. Notwithstanding the lack of efficiency in the cylindrical-end valve just described, and the difficulty of preventing warping and leakage in rotating valves of all kinds, there is a continual effort being made in many quarters to perfect the working of such valves. The object is to do away entirely with all reciprocating parts on the engine excepting the main piston, and thus to avoid the use of cams, push-rods, springs, etc., and all exposed moving parts. This gives to the four-stroke engine a greater degree of simplicity of construction and secures for it some of the advantages of the two-stroke engine without materially sacrificing any of its own.

Cadillac V-Type Eight-Cylinder Engine

632. The eight-cylinder gasoline engine was first adopted in automobile work in America by the Cadillac Motor Car Co. in 1914. The eight cylinders are arranged in two rows of four each, the rows being in line with the arms of a letter "V" and the arms themselves set at 90°, as shown in section in Fig. 377, and pictorially in an outline sketch in Fig. 378. This type of engine has been in use for some time, and still is, in aeroplane engines, as will be described later.

633. The engine parts, with some of the characteristic detail operations, are as follows: *1*, Fig. 377, the main engine shaft; *1-2*, projec-

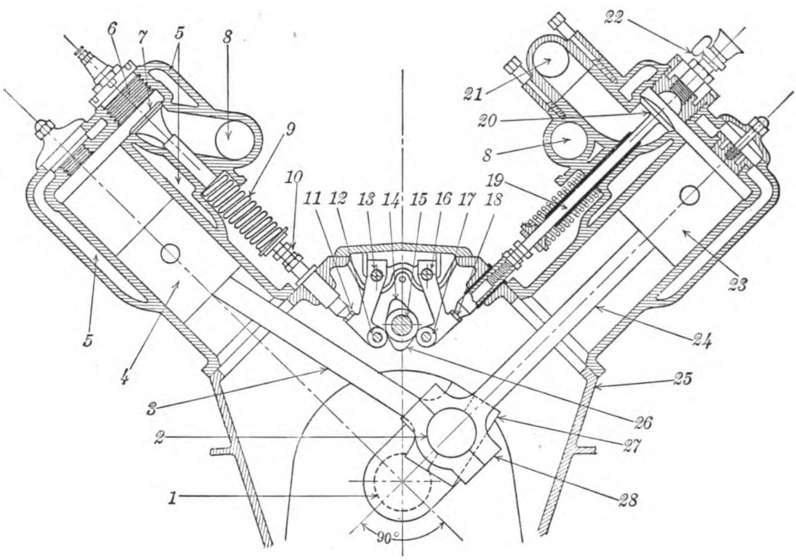


Fig. 377.—Cadillac 8-Cylinder Engine, Cross-Section Through Intake Valve on Left and Exhaust Valve on Right

tions of one set of cranks; *3*, the connecting rod for the first cylinder on the left, and *24*, the connecting rod for the first cylinder on the right; *4*, the piston; *5*, water-cooling spaces; *6*, valve cap into which spark plug is screwed; *7*, the admission valve with tulip-shaped head to facilitate intake of gas; *8, 8*, intake manifolds connected by a U-tube, and supplied from a single carburetor; *9*, valve-return spring; *10*, junction of valve stem and tappet; *11*, sliding surface between tappet and intake cam arm, *11-12-13*. This arm is one of the two arms which swing respectively about the pivots *13* and *13a* (Fig. 380) and permit a single cam, *14*, to operate the intake valves on two opposite cylinders; *12*, Fig. 377, roller on intake cam arm, which arm also takes practically all side strain from the tappet *11-10* for the reason that the junction surface at *11* is perpendicularly below the

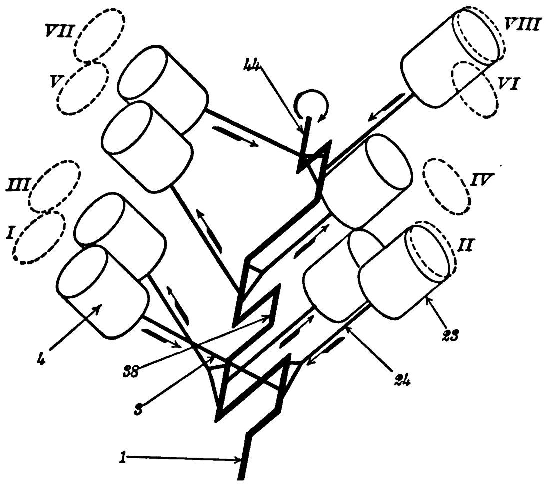


FIG. 378.—SHOWING CADILLAC PISTON POSITIONS WITH CYLINDER NO. VIII JUST FIRING

pivot *13* when the cam arm is half-way through its swing. The side rubbing of the two junction surfaces at *11* is, therefore, small. One of the two junction surfaces should be rounded slightly, however, to prevent the otherwise squared-edge from scraping, or a roller may be provided; *14*, intake cam operating two opposite cylinders; *26*, exhaust cam set just back of intake cam and operating same two opposite cylinders; *15*, cam shaft; *16-17-18*, exhaust cam arm; *19*, exhaust valve stem; *20*, exhaust valve; *21*, exhaust pipe for all four cylinders on the one side; *22*, priming valve; *25*, crank case; *27*, end of connecting rod *3*; *28*, end of connecting rod *24*. The crank shaft for this eight-cylinder engine is identical with one for a four-cylinder engine. Two connecting rods, however, are fastened to each crank pin in the former case, as indicated by the straight and the Y-ends of the crank ends of the connecting rods in Fig. 378, and by the actual construction in Fig. 379, where the Y-end of one rod is

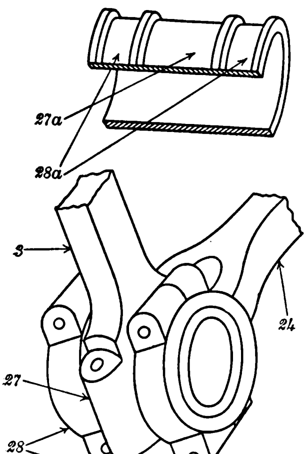


FIG. 379.—ONE CAM OPERATING INTAKE VALVE RODS FOR TWO DIFFERENT CYLINDERS

shown at *28* and the straight end of the other at *27*. The bushing *27a-28a* is held from turning in the Y-end of rod *24*, while rod *3* oscillates freely through an angle of about 32° on the outer and central surface of the bushing. The crank shaft is supported by bearings at three points, 1, 38, and 44, in Fig. 378.

Angle Between Cam-Arm Rollers

634. In order, first, that one cam, *14*, Fig. 380, may operate the intake valves on two cylinders, and second, that the angle, 135°, between the cam follower rollers *12* and *12a* may be determined, it is necessary not only to know the order of firing of the cylinders, but that that order should have some such arrangement as will permit the simplified cam construction here used. In this engine the order of firing is 1, 8, 5, 4, 7, 2, 3, 6, as given in Fig. 340 on page 288. The arrangement of the cylinders is shown by the Roman numbers in Fig. 378. Inasmuch as there is an explosion for each 45°-turn of the cam shaft in an eight-cylinder engine, and as the order of firing shows five 45°-intervals between the explosions in cylinders I and II, there must be a time interval in terms of a cam-shaft revolution of 225°. With the cam turning as shown in Fig. 380, it will be seen that

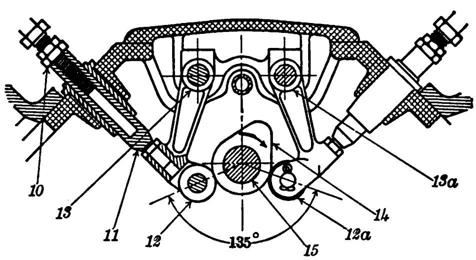


FIG. 380.—ONE CAM OPERATING TWO INTAKE VALVE RODS

it will start to move the cam arm *12a* which leads to cylinder No. II, just 225° after it has started to lift cam arm *12*, providing an angle of 135° is used as shown through the centers of the follower rollers. In the same way, a single exhaust cam is sufficient to operate the exhaust valves in opposite cylinders.

Auxiliary Drives and Shafts

635. The crank shaft *1*, *38*, *44*, Fig. 381, drives the cam shaft *15* through the silent chain *32*, and this in turn operates the generator shaft *37* through the chain *34*. A one-way clutch is indicated on the shaft, to the right of which is mounted a motor-generator at *37*. To

start the engine the gear *39* is first placed in mesh with the gear teeth cut on the rim of the flywheel *43*. Through suitable connections the motor-generator now furnishes the power to start the engine and as the engine reaches speed under its own power it drives the motor-generator, through the chains *32* and *34* and the one-way clutch, faster than it was running as a motor and turns it into a generator

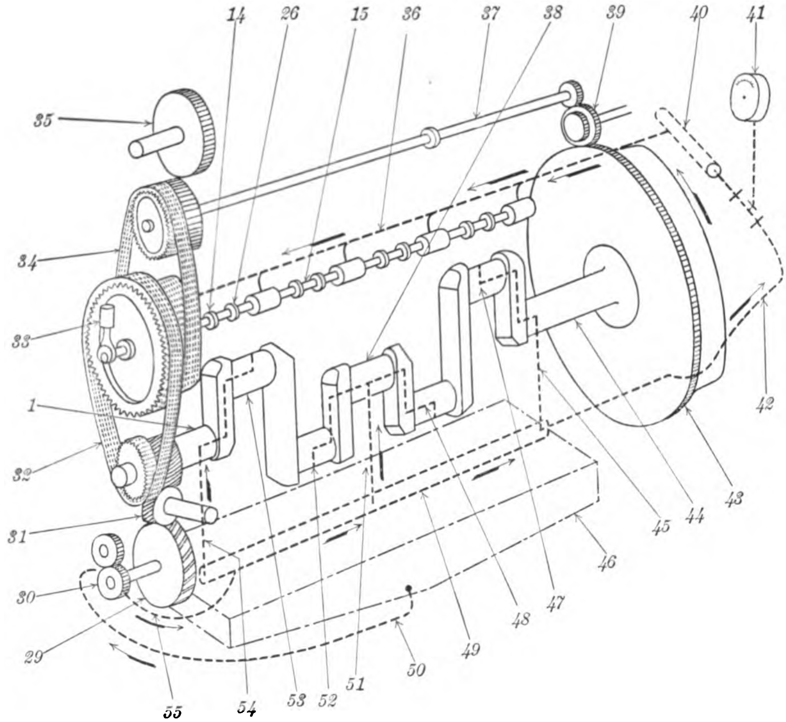


FIG. 381.—CADILLAC SHAFTING AND OILING SYSTEM

which charges a storage battery or supplies the lights, or both directly, all on what is known as the Delco system. When speed is reached, the spur wheel *39* is thrown out of gear with *43* by simply removing the foot pressure from the starting button. The wheel and spindle at *35* operate the tire pump. The eccentric pump at *33* provides air pressure for the fuel feed. A high-speed fan for drawing air through the radiator is placed on the end of the shaft *37* in front of the chain *34*. The cam at *14* operates the intake valves, and the one at *26* the exhaust valves on cylinders I and II.

A worm on the crank shaft *1* drives another worm wheel *31*, the shaft of which extends in both directions and drives two pumps which circulate the cooling water through the engine jackets, radiator, and pipes. The worm *31* drives, in turn, the wheel *29*, the shaft running from it, and, also the two ordinary spur wheels at *30* which are tightly enclosed and made to act as a pump in circulating the oil in the lubricating system.

Lubricating Systems

636. The lubricating system is as follows: A rectangular oil basin or reservoir is shown at *46*, Fig. 381. It is fitted into the bottom of the engine casing, and the large flat top surface is covered by a wire gauze to prevent foreign particles from dripping back into the basin with the return oil. The spur-gear pump *30* draws up the oil from the bottom of the basin through the pipe indicated by the dash line *50* and delivers it through the pipe *55* to the larger supply tube *49*. From this, small tubes *54*, *51*, and *45* lead to the crankshaft journals *1*, *38*, and *44*. Holes are drilled into the journals, the crank arms, and the crank pins so that oil may flow from the crankshaft bearings to the connecting-rod bearings at *53*, *52*, *48*, and *47*. As it flows out from all the bearings it is thrown about so as to lubricate the pistons, cylinders, and piston pins, after which it drops to the oil reservoir *46* and is used over again. In addition, some of the oil from the pump *30* passes on through the tube *49* to the adjustable pressure valve at *40*. Although this valve can be adjusted so as to give any pressure within certain limits for given speeds, the pressure will increase or decrease as the engine speed increases or decreases, it being necessary to have higher pressures at higher speeds. A pressure gauge is shown in the line at *41*. From the pressure-regulating valve *40* the oil overflows to a pipe *36* which carries the oil by gravity to small tubes extending into the cam-shaft bearings and to the silent chains *34* and *32*.

637. A common method of lubricating where holes are not drilled in the crank shaft and arms is the "splash" system. In this the bottom of the closed crank case acts as an oil well with the surface of the oil just high enough for the connecting-rod end to dip partially as it passes. At high speed the oil is thrown in all directions, reaching small wells or cups located at different points. From these, tubes carry the oil in a steady supply to some of the points and the splash reaches other points directly, thus lubricating the piston, cylinder, piston pin, crank-shaft bearings, and cam-shaft bearings, as well as the connecting-rod bearings.

Circulation of Cooling Water

638. The circulation of the cooling water in the new Cadillac eight-cylinder engine is maintained by a pump and is controlled by a thermostatic valve. There are three circuits for the cooling water, the two smaller ones through the pipes *61* and *66*, Fig. 382, being always open and in operation when the engine is running, and the third or principal one through the radiator *56* being open only after the engine has been running a short time and the water is well heated.

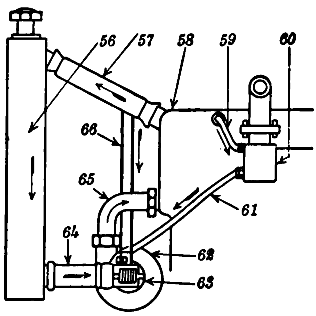


FIG. 382.—SHOWING CIRCULATION OF COOLING WATER

639. When the engine is started and the water is cool the automatic double-seat poppet valve *68*, Fig. 383, is closed, thus stopping circulation through the radiator. The cooling-water pump indicated at *62* then draws the cool water from the water-jacket casing *58* principally through the starting by-pass *66* and also through the carburetor by-pass *59–61*. Only a comparatively small part of the cooling water is now in circulation and it heats quickly. Its course is around and past the thermostat shown at *67* in Fig. 383, and as this becomes heated it expands and opens the double-seat poppet valve *68*, thus admitting the full supply of cooling water through the pipe *64* from the radiator.

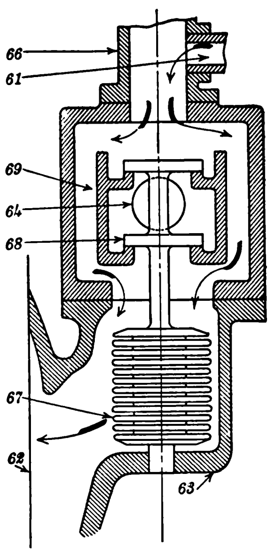


FIG. 383.—THERMOSTAT OPERATING CONTROL VALVE

640. The radiator consists of a great network of flat or round tubing through which the hot water from the engine cylinders passes in a great many small streams. The radiator is at the front of the car. A fan is provided just behind it to draw the air through more rapidly and assist the cooling process. The thermostat is made up of a series of sealed copper disks which contain a fluid which expands and contracts according to the temperature. The total movement of the disks is about

¼ inch. There is one pump and one thermostat for each set of four cylinders. The by-passing of the cooling water at starting enables the engine more quickly to reach its full power. To obtain full power from the fuel mixture it is necessary that the cylinder walls be heated to a certain degree. The thermostat also maintains a uniform temperature of cooling water in both sets of engines should there be any prevailing conditions to cause a difference, such, for example, as exposure of one side of car to cold winds.

Pump, Air, and Thermo-Syphon Cooling Systems

641. The cooling system just described depends upon a pump to force cooling water through the water jackets and radiator as explained.

Another cooling system makes use of air instead of water as the cooling agent. The air is caused to circulate freely around the cylinder walls under pressure of a fan, as in the Franklin air-cooled engine described in paragraph 596.

A third method of cooling known as the thermo-syphon system which uses water as the cooling agent is also in use. In this system there is no pump to force the cooling water. Briefly the water circulates through a circuit similar to that shown in Fig. 382 if we consider only the water jacket *58*, the pipe *57*, the radiator *56*, and a pipe back to the bottom of the water jacket, omitting all other parts. Then as the water is heated it rises to the top of the jacket and passes through the pipe *57* to the radiator, where it cools and falls to the bottom and flows back to the bottom of the water jacket by gravity. There it is again heated and repeats the circuit. It is essential that the radiator should be high enough to allow the cooled water to return to the cylinders by gravity. The circulating action is entirely automatic and the cooling effect is practically constant, with the temperature ranging from about the boiling point to some degrees below according to the cooling conditions. The thermo-syphon system requires more cooling water, which means that the water jackets, radiator, and pipes should be larger and that the pipe junctions, etc., should be smoother than where a pump is used. With an ordinary pump-cooling system the engine cylinders will become cooler as the engine speeds up, and will heat up more at low speed. The thermostatic valve in the Cadillac cooling system described in the previous paragraph regulates the amount of cooling water by the amount of the valve opening.

642. The eight-cylinder Cadillac has horse-power of 31.25, according to the Society of Automobile Engineers rating, with an actual

horse-power in many cases exceeding 70 at 2400 revolutions per minute in dynamometer tests. The cylinders are $3\frac{1}{8}$ inches bore, and the stroke is $5\frac{1}{8}$ inches. The eight-cylinder motor weighs 60 pounds less than the four-cylinder motor that it displaced. This is due to the fact that the cylinder bore is less and, therefore, the length of the cylinder casing and crank case is less. Also, the reciprocating parts are lighter in weight, due to the facts, first, that the number of power impulses per minute are increased in the eight-cylinder engine; and, second, that for a given horse-power the stress per impulse is always less as the number of impulses per minute increase. For the same sized cylinders the eight-cylinder engine of the V-type is about 30 per cent shorter than the regular six-cylinder engine where the cylinders are all in line. The connecting rods are $12\frac{1}{2}$ inches long between centers, are made of alloy steel, and are machined all over to reduce weight. The eight cams are integral with the cam shaft. All valves are $1\frac{9}{16}$ inches in diameter and have a lift of $\frac{11}{32}$ inch.

SECTION XI.—AEROPLANE ENGINES

Types of Construction

650. The gasoline engine for aeroplanes has been developed along four different lines of construction as regards arrangement of cylinders and valve gears, namely:

(a) The Vertical, or ordinary automobile type, with the multiple cylinders placed parallel and with the plane of their center lines passing through the axis of the shaft. Example, the Wright engine.

(b) The Diagonal or V-type, with two sets of multiple cylinders, the cylinders in each set being placed parallel. The planes of the axes of each set pass through the axis of the shaft and make an angle of 90° with each other. Example, the Curtiss engine.

(c) The Fixed Radial type, with the axes of the cylinder pointing toward a common center and all lying in a plane perpendicular to the engine shaft. The cylinders may be placed throughout the entire 360° or they may be arranged fan-shaped. Example of the former, the Salmson (English) engine.

(d) The Revolving Radial or Rotary type, with the axes of the cylinders pointing toward a common center and all revolving as one solid piece. The cylinder axes all lie in a plane perpendicular to the engine shaft. Examples, the Gnome and Gyro engines.

Wright Vertical Engine

651. The vertical type used by the Wright Company from the start is operated by a cam shaft as in multiple-cylinder automobile engines. This type of construction is diagrammatically illustrated for a four-cylinder engine at Y_1—Y_4 in Fig. 339. The vertical aeronautical engine is, however, much lighter in construction than the automobile type, weighing but 305 pounds complete including flywheel for a six-cylinder 60 horse-power engine, or about 5.1 pounds of engine material per horse-power. An example of this particular engine has 4⅜ inches bore and 4½ inches stroke, and runs at 1400 revolutions per minute at 60 horse-power. The vertical or upright type of aeroplane engine is also built by the B. F. Sturtevant Co., the Maximotor Co., and is largely used in Germany in aeronautical work. Horizontal opposed cylinders are also used.

Curtiss V-Type Engine

652. The V-type of engine as used in the Curtiss gasoline engine is illustrated by end and front views in Figs. 393 and 394. The eight cylinders are indicated by Roman numerals from I to VIII, the odd numbered ones being in front in Fig. 394 and also a small distance to the left of the even numbered cylinders as shown by the distance *M* that the projections of the center lines are apart and also by the distance *N* that the respective cams are apart, and further by the distance *O* that the connecting rods are apart. By arranging the eight cylinders in this way, they take up only about 20 per cent more longitudinal space than four upright cylinders of the same size. The connecting rods from each pair of cylinders are connected side by side on the same crank pin *50*, and thus require only four crank pins to be forged on the main shaft. With this form of construction the weight per horse-power, including both radiator and cooling water, has been slightly over five pounds. In the most recent design of V-type engine, where steel instead of iron is used in the cylinder construction, the weight per horse-power, for a 160 horse-power engine, has been brought down to 4.3 pounds per horse-power, including radiator and water. The weight of this V-type engine of 160 horse-power, without the cooling apparatus, is 570 pounds.

653. The valves are the poppet type and are operated by the valve gear as follows: The timing gears *35* and *36* connect the main shaft *38* and cam shaft *40*. The cams are forged on the cam shaft and, in order to save space and weight, the intake and exhaust cam surfaces are forged together, *48 E*, for example, being the exhaust cam, and *48 I* being the double surface of the intake cam. This arrangement enables the designer to place the exhaust push rod *18* inside of the hollow intake push rod *16*. The form of the complete cam body for operating one of the cylinders is shown in the small illustration giving the section at $X-Y$ in Fig. 394. The solid outline shows the form of the intake cam and the dash lines the form of the exhaust cam. The cam is also shown in dash lines in Fig. 393. The cams for this engine, the "O. X." type, are formed and placed so as to time the valves as shown in the timing diagram, Fig. 395. For intake action the cam itself does not operate the valve directly, but is so formed that it permits the compression spring shown at *5 E*, Fig. 393, to push down the end *23 s* of the push-rod *17* as it passes the flat part *23 T* of the inlet cam, thus drawing down the intake rocker *21 E—27 E*, which in turn pushes down the valve

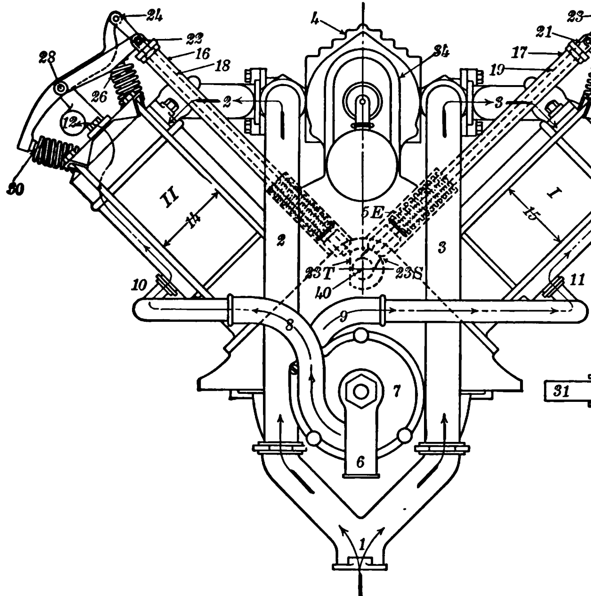


FIG. 393.—CURTIS 8-CYLINDER V-ENGINE, FRONT VIEW

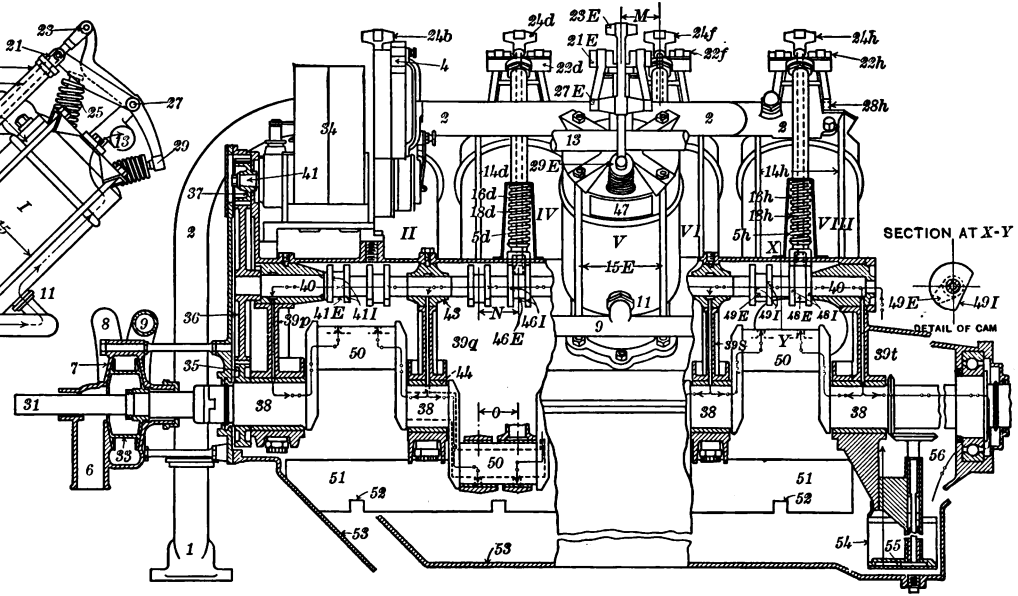


FIG. 394.—CURTIS 8-CYLINDER V-ENGINE, PARTIAL LONGITUDINAL SECTION

stem against the spring pressure *25*. The spring at *25* is weaker than the one at *5 E*, and its purpose is to reseat the valve, it being kept in mind that there is simply pressure contact between the intake rocker and the valve stem. The exhaust valve is opened as the lobe of the exhaust cam, shown by dotted lines at *49 E* in Fig. 394, and in projection at *23 S* in Fig. 393, passes under the exhaust push-rod *19* in Fig. 393. As this rod lifts it operates the exhaust rocker arm, pushing the end *29* down on the exhaust valve stem against the pressure of the spring shown in the Figure.

654. The intake rocker arm, although appearing as a simple lever at *21-27* in Fig. 393, is, in fact, a double arm as may be noted at *27 E*, with the exhaust rocker occupying the space between the two arms. A bridge connects the two arms and a spot face on this bridge presses against the valve stem *25* at the proper times. The end of the intake push rod has a fork-shaped end-piece, the two curved ends of which rest on the twin cam surfaces indicated at *48 I*. The end of the exhaust push-rod passes through the intake fork and has a rounded-end follower piece which bears against the exhaust cam *48 E*.

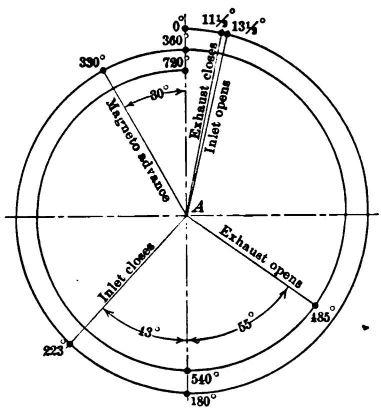


FIG. 395.—TIMING DIAGRAM FOR CURTIS "O.X." AEROPLANE ENGINE

655. Water is used for cooling the cylinders, it being drawn in from the circulating system through the supply pipe *6* by the rotary pump *33* and carried by piping, *8* and *9*, to the cylinders at *10* and *11*. After passing through the water jacket it comes out through the return pipes *12* and *13*. The lubricant or oiling system is indicated by arrow lines starting from the oil circulating pump at *55*. The channels for the oil are ordinary pipes, a hollow cam shaft *40*, channels in the engine frame *39 p* and *39 s* and bearings, a hollow crank shaft *38* with channels in the crank arms leading to hollow crank pins *50*, and finally the openings through the crank pins to the connecting-rod bearings from the ends of which the oil drips into a drip pan *51*. The connecting-rod ends dip in this pan as they pass in their lowest position, the oil being maintained at a constant level in the pan by the overflow tubes *52*, through which the oil passes to the oil reservoir *53* from which it is drawn by the pump *55* which is

operated by bevel gears *56* from the main shaft. The ignition is by means of a magneto *34* operated by a gear *37* from the large timing gear *36*. The exhaust port is simply a rectangular opening in the cylinder as shown at *47* on cylinder V, Fig. 394. Long bolts holding down the light weight cylinders to the engine frame are shown at *14* and *15*, there being four such bolts for each cylinder. The stepped block shown at *4*, Fig. 393, is an electric distributor board, the various wires being guided by each step.

Salmson Fixed Radial Engine

656. An aeronautical motor with nine cylinders placed radially in one plane and fixed in their positions is shown diagrammatically in Figs. 396 and 397. The crank ends of the nine connecting rods fit

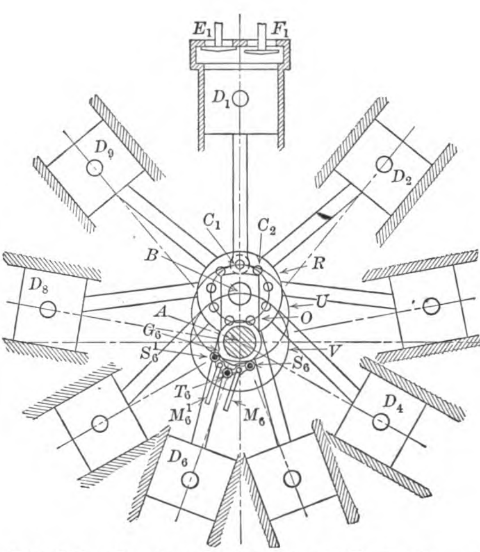


Fig. 396.—Salmson Fixed Radial Type of Aeronautical Engine, Diagrammatic End View

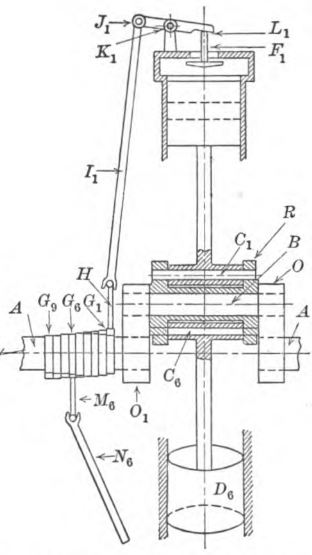


Fig. 397.—Salmson Engine, Diagrammatic Longitudinal Section

on nine pins, C_1, C_2 . . . of the spool R, and this spool turns freely on the main and ordinary crank pin, B.

657. The intake and exhaust valves, F_1 and E_1 respectively, for each cylinder are controlled by a single cam such as is represented at G_6 in Fig. 397. The cams G_1 . . . G_9 are rotated at one-half the engine speed. Each cam has two follower rollers, S_6 and S_6^1, Fig. 396, located at the ends of independent arms which swing on the fixed pivot T_6. The guide rod M_6 is attached to the arm T_6 S_6 and, through the push-rod N_6, it operates the admission valve of cylinder D_6 by

means of a rocker arm and poppet valve such as is shown at $J_1\ K_1$ $L_1\ F_1$, Fig. 397. Similarly, the roller S_6' operates the exhaust valve. With this arrangement, *i.e.*, using the same cam for both valves, the admission and exhaust periods must be the same.

Uniform Piston Motion in All Cylinders Due to Planetary Gear

658. With the simple arrangement of the spool and connecting rods shown in Figs. 396 and 397, it is possible that the spool will not have a uniform rotation on the crank pin, for the device is not a positive one mechanically. With such a large number of power cylinders, however, the spool would doubtless find a balanced position and maintain it with little or no oscillation. In order to make the valve gear action positive mechanically, a planetary gear is added in the Salmson aeronautical motor as shown diagrammatically in Figs. 398 and 399. The object of the planetary gear is to give the

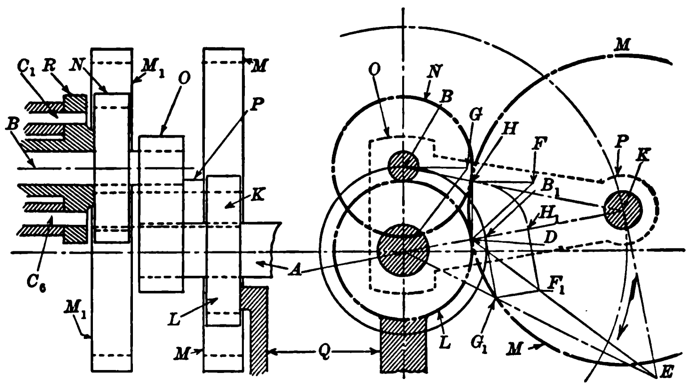


FIG. 398 FIG. 399

FIGS. 398 AND 399.—SHOWING THAT PLANETARY GEAR GIVES MOTION OF CIRCULAR TRANSLATION TO CONNECTING-ROD SPOOL

spool R, Fig. 396, a motion of circular translation, which would cause, for example, a vertical line on the spool to always move parallel to itself and thus compel C_1 to move in a circular path U equal in diameter to the circular path V of the main crank pin. The engine shaft A, Fig. 398, the crank O, the main crank pin B, the spool R, and the spool-pins $C_1 \ . \ . \ . \ C_6 \ . \ . \ .$ of Fig. 398 are all the same as in Fig. 396. A non-rotating spur wheel L is rigidly attached to the frame Q of the motor. It gears with the idler spur wheel M which is integral with a pin K. This pin turns freely in an arm P, which is forged onto the crank O. A spur wheel M_1, identical with M, is also integral with the pin K, and it gears with a fourth spur wheel N which is identical in size with L, and which is fastened to the spool R. With this arrangement the center of the spool-pin C_1 will always be vertically over the center of the crank pin B. To demonstrate this, it is only necessary to show that the resultant motions of two points, say B and H, Fig. 399, on the spur wheel N are parallel and equal. This is done graphically in Fig. 399, and explained in the following paragraph.

659. The crank O with its rigidly attached arm $O\,P$ revolves about the engine shaft A. Therefore, the pins B and K must have the same angular velocities, as represented by the equal angles $B\,A\,G$ and $K\,A\,E$, and the linear velocities will be represented by $B\,G$ and $K\,E$ respectively. Since the wheels M and L are in pure rolling contact on their pitch circles, represented by dash and dot lines, D is the instantaneous axis of the wheel M and it has an angular velocity about D represented by the angle $K\,D\,E$. The point H of wheel M also has the same angular velocity about D, and this is represented by the angle $H\,D\,F$. $H\,F$ is, therefore, the resultant linear motion of H on the wheel M and also of H on the wheel N, because M and N are engaging spur wheels and are in pure rolling action on their pitch circles. It only remains now to show that the motions $H\,F$ and $B\,G$ of the two points H and B on the wheel N are parallel and equal. They are parallel from the fact that the two pairs of wheels L and N and M and M_1 are equal respectively in size. That they are equal is shown by revolving $B\,G$ to $B_1\,G_1$ and $H\,F$ to $H_1\,F_1$, when the points G_1 and F_1 will fall on the lines $A\,E$ and $D\,E$ respectively. Then by similar triangles:

$$\frac{K\,E}{K\,A} = \frac{B_1\,G_1}{B_1\,A} \quad . . . (1), \qquad \text{also} \qquad \frac{K\,E}{K\,D} = \frac{H_1\,F_1}{H_1\,D} \quad . . . (2)$$

or, taking the equal values of $K\,E$ in (1) and (2)

$$\frac{B_1\,G_1 \times K\,A}{B_1\,A} = \frac{H_1\,F_1 \times KD}{H_1\,D} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (3)$$

By similar triangles, $K\,B\,A$ and $K\,H\,D$,

$$\frac{B\,A}{K\,A} = \frac{H\,D}{K\,D} \quad . . . \quad (4), \qquad \text{or} \qquad \frac{B_1\,A}{K\,A} = \frac{H_1\,D}{K\,D} \quad . \quad . \quad (5)$$

Substituting the value of $K\,A$ of (5) for $K\,A$ in (3),

$$\frac{B_1\,G_1 \times B_1\,A \times K\,D}{H_1\,D \times B_1\,A} = \frac{H_1\,F_1 \times K\,D}{H_1\,D} \quad . \; (6), \quad \text{or } B_1\,G_1 = H_1\,F_1 \; . \; (7)$$

$$\text{or, } B\,G = H\,F \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (8)$$

660. Although not relating to the mechanical motion of the valve gear it is of general interest to note that with this form of construc-

tion the nine-cylinder engine here represented weighs but 360 pounds and its rated horse-power is 130, thus giving a weight of less than 2.8 pounds per horse-power. The cylinder bore is approximately $4\frac{3}{4}$ inches and the stroke $5\frac{1}{2}$ inches. The speed is 1250 revolutions per minute. The casing of the motor is made of aluminum, the cylinders of steel turned from the solid, the water jackets of spun copper, the pistons of cast iron, and the connecting rod of steel. Further illustrations and description of this engine may be found in *The Engineer*, March 20, 1914.

Gnome Revolving Radial Engine

661. The Gnome engine, which is a French design, is of the revolving type, the cylinders being placed radially and revolving about the main shaft which is fixed. The crank pin is also fixed. The propeller is firmly attached to the circular casing to which the radial cylinders are rigidly connected. Any number of cylinders may be used, although 5, 7, or 9 are best adapted for engines in which the axes of the cylinders all lie in one plane. Where still higher powers are desired, 10, 14, and 18 cylinders may be used in two sets of cylinders side by side. The reason for selecting an odd number of cylinders depends on a regular order of firing of four-stroke multiple-cylinder engines and will be explained in a later paragraph. The revolving type of engine may be made exceedingly light in construction, its own revolving cylinders taking the place of a special flywheel, and giving efficient air cooling without providing water jackets and cooling water. The weight of a 50 horse-power engine, one having 7 cylinders with 120 m.m. stroke and 110 m.m. bore and rated at 1200 revolutions per minute, is 167 pounds, or $3\frac{1}{3}$ pounds per horse-power; while the weight of a 100 horse-power engine consisting essentially of two 50 horse-power engines is given as 220 pounds or 2.2 pounds per horse-power. The advantage of the revolving engine in the matter of weight is seen at once when compared, for example, with an eight-cylinder V water-cooled engine where the weight per horse-power is 4.3 pounds, or about double that of the revolving engine.

662. The general arrangement and action of the several parts of a seven-cylinder Gnome engine are shown diagrammatically in Figs. 400 and 401. A is a fixed shaft and B the fixed crank pin. H is a collar rigidly attached to the master connecting rod D. This collar turns freely on the fixed crank pin, and carries six small pins, such as shown at G_2 . . . G_4 . . . , Fig. 400, to which the remaining

six connecting rods are pivoted. Fuel mixture is admitted through the hollow shaft A to the crank case and up through a valve in the piston head E_1. This valve acts automatically and will be explained in a later paragraph in connection with an enlarged detail diagram.

The action of the engine is as follows: When the explosion takes place there is pressure in the cylinder and the piston is acted upon by three forces. At the phase E_2, Fig. 400, these forces are: the explosion pressure normal to the head of the piston, the thrust of the connecting

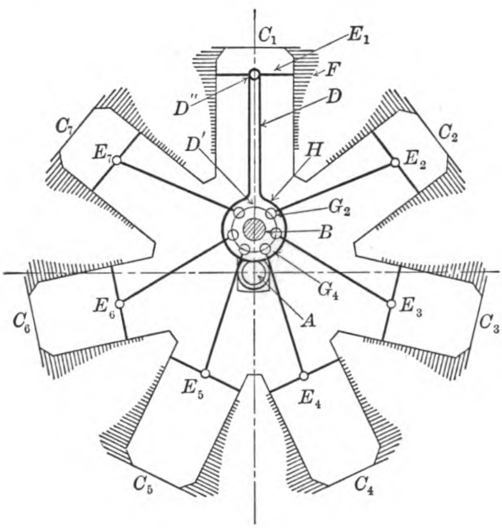


FIG. 400.—GNOME ENGINE, DIAGRAMMATIC END VIEW

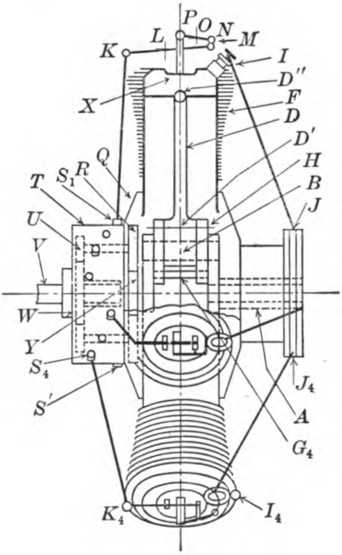


FIG. 401.—LONGITUDINAL SECTION AND SIDE VIEW

rod, and the side pressure of the cylinder wall. Since the cylinder is free to move, the reaction of this latter force results in an acceleration of the cylinder and is the direct cause of the rotation.

663. As the cylinder revolves it carries with it the crank chamber casing Q, Fig. 401, the cam casing T, and the propeller shaft V. The crank casing also carries a pair of free turning shafts to which are rigidly attached the toothed gear wheels R and U. These are also shown, on a larger scale, in Fig. 405. The former rolls on the fixed wheel Y, which is attached to the fixed shaft, while U drives the wheel W at half the engine speed. The wheel W connects, by means of a hollow shaft, with a simple bar cam which, as it revolves, comes into contact successively with seven different internally-formed cam-followers, which operate the push rods S_1 K and these operate the balanced levers K L M and N O P, Fig. 401, the latter pushing

down and opening the exhaust valve X. The fuel mixture is fired by the spark plug I, which connects by the wire $I\,J$ with a timer disk. The ribs at F run completely around the cylinder, their purpose being to add strength to the cylinder wall, which is only about $\frac{1}{16}$ inch thick and also to add a larger radiating surface than the cylinder alone would provide in the dissipation of the heat. These ribs are constructed with sharp edges which are favorable to heat dissipation and are longest near the top of the cylinder where the pressures and temperatures are highest. These ribs are integral with the cylinder, all of which is machined from a solid bar of steel. The piston also, shown in Fig. 403, is turned from a solid steel bar.

664. The cylinders are fitted into the crank casing in a unique manner without the use of ordinary fastening bolts. The bottom of the cylinder is enlarged, as shown at *22*, Fig. 402, and an inverted L-shaped groove *23* is cut into it, the same as a groove would be cut in a piston for a piston ring. The cylinder is slipped in through a circular opening in the crank casing drum *24* so far beyond its normal position that an L-shaped snap ring *26* may be readily sprung in place from the inside. The cylinder is then drawn out to its normal position when the projecting snap ring bears against a shoulder *25* which has been machined on the crank casing. Taper pins *27* are then driven into fitted holes in the crank casing so as to hold the snap rings and the cylinders in place.

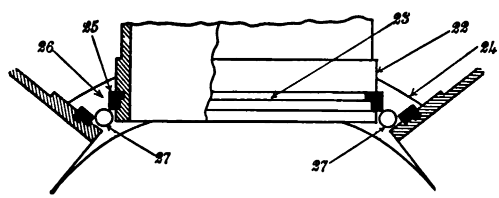


FIG. 402.—SHOWING METHOD OF ATTACHING GNOME CYLINDERS

665. A detail sketch of the automatic balanced inlet valve in the piston head E is shown in Fig. 403. This illustration and also others presented in connection with the engine show only the essential features of construction, omitting many of the refinements, such as ball bearings, liners, and other small parts necessary for attachment and adjustment. A valve casing *2* is fitted in the piston head E. This casing carries a cylindrical guide for the valve stem of the inlet valve *1* which seats on a faced surface of the casing. The cylindrical guide also carries pins at *4* for the balance weights *3*, the inner ends of the weights being tapered and fitting into suitable openings in the valve stem. The weights are in balance with the valve, and the valve is normally held to its seat under the action of a leaf spring *5*, which bears against small projecting pieces on the weights. As

the cylinders revolve at high speed, 1000 or more revolutions per minute, the valves would have to be held by heavy springs against the strong centrifugal force exerted by the valve were it not for these balance weights. Were such heavy springs used, difficulty would be experienced in contriving means to operate the valves at starting, when the action due to centrifugal force is negligible. As the piston moves down, a partial vacuum is formed in the cylinder above the

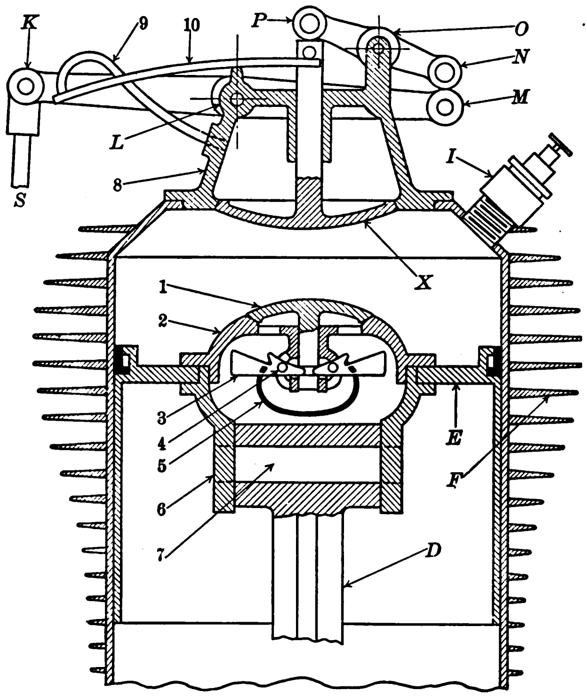


FIG. 403.—GNOME CYLINDER AND PISTON SECTION SHOWING AUTOMATIC INLET VALVE

head of the piston and the fuel mixture is drawn in through the valve, which is thus made to open automatically. The connecting rod *D* is attached to the lower part of the inlet valve casing which is fastened to the piston by a pin *7*.

666. The exhaust valve and the levers operating it are also balanced in order to nullify the effect of centrifugal force, as shown in Fig. 403. The exhaust valve *X* is held in place, when the engine is running, by its own centrifugal force, and when it is stationary by the leaf spring *10*, which is held at its fixed end by the bracket *9*. The cam-operated push rod *S K* swings the balanced rocker arm *K L M* about *L* as a fixed center and this rocker in turn rotates the

small balanced rocker $N\ O\ P$ about O as a fixed center and so moves down the valve stem of the exhaust valve against the pressure of the leaf spring (10). The contact between the end P of the rocker arm and the valve stem is simply that of pressure contact.

667. The form and arrangement of the cam-followers operating the exhaust valve are shown in Fig. 405. There are seven, one for each cylinder. These cam-followers are formed by internal projections *15* on a cylindrical ring *14*, the ring itself having radial arms *14* extending from opposite sides and running in cylindrical guides *16*

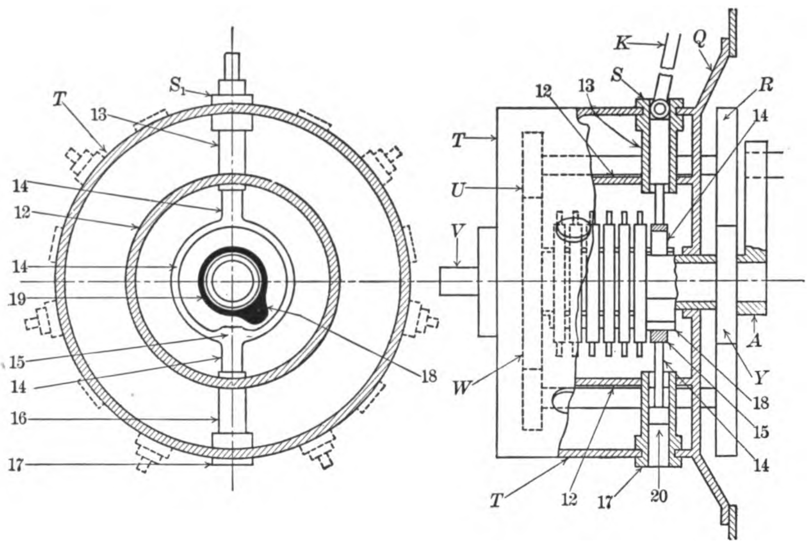


FIG. 404 FIG. 405

FIGS. 404 AND 405.—GNOME ENGINE, FORM AND ARRANGEMENT OF CAMS OPERATING EXHAUST VALVES

and *13*, which are firmly held in the cam-casing cylindrical shells T and *12*. The push rod $S\ K$ is attached to the end S of one of these guided arms. The cam rings revolve at the same speed as the engine cylinder while the cam itself, *18*, revolves at one-half the cylinder speed by means of the gear wheels $Y\ R\ U$ and W, as described in paragraph 663. The seven cam-followers with the guide rods and guide housings omitted, all in the same revolved positions, are indicated in Fig. 405.

668. The present illustrations show, diagrammatically, the details of construction on practically all the Gnome engines in the country at the present writing. A recent model, however, has a simplified form of construction at the following points: The gear wheels Y and R, Fig. 405, are placed to the left of the cams and next

to the gear wheels *W* and *U*. The cam projection *18* is machined on the periphery of a disk of about the same diameter as the ring *14*. The follower rod below *S*, which is guided by the tube *13*, has a roller on the lower end which rolls on the external surface of the new cam disk described in the preceding sentence. The guide *13* is somewhat longer in the later design, and is sufficient to hold the follower steady without running a projecting arm to the opposite side of the cam case, as shown at *17* and *20*. With the cam-follower working externally, the push rod *K* now does its work on the push or up-stroke and this also simplifies the construction of the valve gear at the end of the cylinder, where the second rocker arm *N O P*,

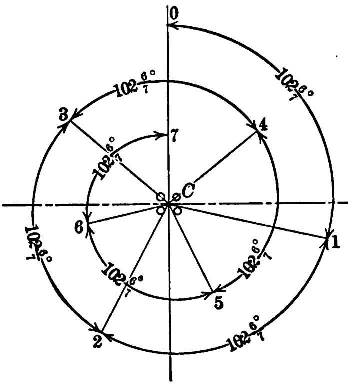


FIG. 406.—FIRING INTERVALS FOR 7-CYLINDER GNOME ENGINE

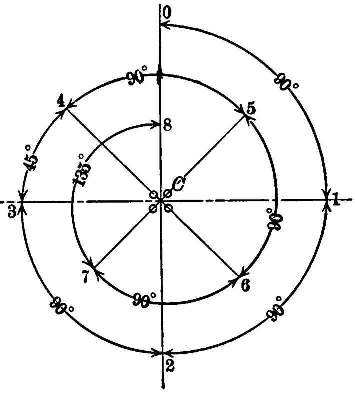


FIG. 407.—FIRING INTERVALS FOR 8-CYLINDER REVOLVING ENGINE

Fig. 403, is omitted. The length of the first rocker arm *K L M* is now made equal to the distance from *K* to the center of the valve stem and the supporting point *L* is moved further to the left so as to be at the center of the arm.

669. In revolving four-stroke engines an odd number of radial cylinders is necessary in order to secure evenly distributed power impulses. It must be kept in mind that in the four-stroke or "four-cycle" internal combustion engine the piston makes four full strokes in the cylinder from one explosion to the next. In one full turn of 360° of a Gnome or other revolving engine the piston makes two full strokes; therefore, it requires two full turns of the cylinder for each explosion. If all the cylinders fired successively as they passed a point at or near the top dead center there would be power in the engine during one full turn and no power at all during the next full turn. Therefore, every other cylinder must fire as it passes the upper dead center, and as a result there will be power applied at reg-

ular intervals throughout the two turns of the engine if an odd number of cylinders are used. If seven are used, each of the intervals between power impulses will be $102\frac{6}{7}°$, as illustrated in Fig. 406. If eight cylinders are used, and if every other cylinder fires as it passes the upper dead center, one of them would be called upon to fire after it had turned 360° from its previous firing position. This it cannot do in a four-stroke engine, because the piston, in this type, has only just completed its exhaust stroke after 360° and is

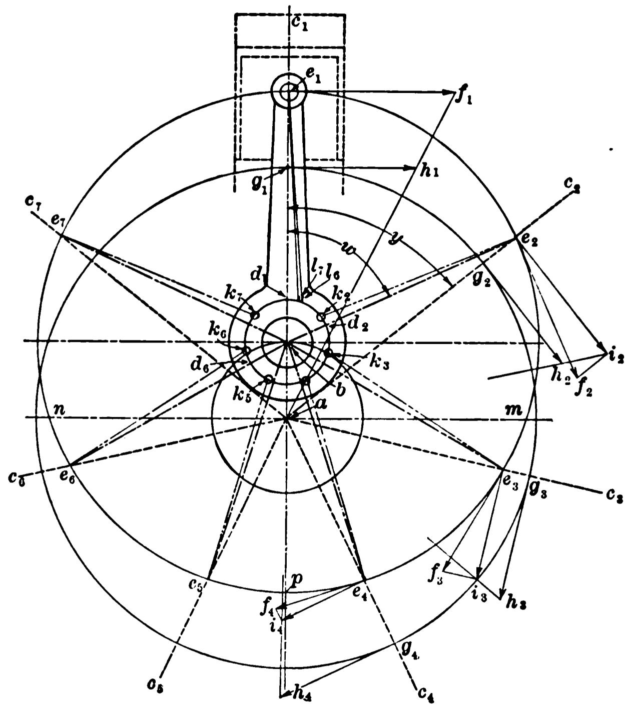


FIG. 408.—SHOWING EFFECT OF VARYING LENGTHS OF CONNECTING RODS FOR REVOLVING ENGINES OF THE GNOME TYPE

just beginning to draw in its charge. Therefore, if eight, or any even number of cylinders are used in a single circle, there may be a certain number of equally distributed impulses, but two of them must be irregular as shown in Fig. 407. The order of firing for a seven-cylinder revolving engine is as follows: 1, 3, 5, 7, 2, 4, 6.

Variation of Angular and Linear Motions in Revolving Engines

670. In building the Gnome engine the length of the master connecting rod measured from D' to D'' is made less than the pivoted

connecting rods from G_2 to E_2, etc., Fig. 400. The reason for this will appear from an inspection of Fig. 408. Let $a\,c_1$, $a\,c_2$, etc., be the positions of the axes of the seven cylinders, the master cylinder being at top dead center $a\,c_1$. The gudgeon or piston pin carrying the upper end of the master connecting rod is then at e_1. When the engine cylinders turn through 1/7 revolution, the center line of the master cylinder No. 1 will be at $a\,c_2$ and the master connecting rod will be at $b\,e_2$, the angular travel of the rod having been $65\frac{5}{28}°$, as represented at w, while that of the cylinder has been $51\frac{3}{7}°$, as shown at y. If the collar pins k_2—k_7 holding the crank ends of the six connecting rods are evenly spaced the pin at k_7 will then be at l_7, or $13\frac{3}{4}°$ beyond the top dead center when the cylinder axis $a\,c_7$ comes into top dead-center position. ($65\frac{5}{28} - 51\frac{3}{7} = 13\frac{3}{4}$.) The arc $k_7\,l_7$ equals the arc $d_1\,d_2$. If the piston in cylinder No. 7 is to be at the same point relatively to its cylinder as the piston in the master cylinder was at top dead center, and it must be to secure the same degree of compression of the mixture, then e_7 must be at e_1 as it crosses dead center and the connecting-rod length of cylinder No. 7 must be $l_7\,e_1$, or, it must be longer than the master rod by the amount that $l_7\,e_1$ exceeds $d_1\,e_1$. For a collar-pin radius ($b\,d_1$) of 1.3″, and a master connecting rod length ($d_1\,e_1$) of 6.53″, which are approximately the sizes of a seven-cylinder Gnome engine, the length of the connecting rod, $k_7\,e_7$, for cylinder No. 7 is found by computation to be 6.64″, or .11″ longer than the master rod.

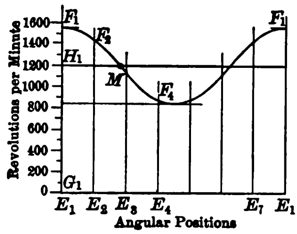


FIG. 409.—SHOWING VARIABLE ANGULAR VELOCITY OF MASTER CONNECTING ROD ON FIXED CRANK PIN

671. When the master cylinder has turned through 2/7 of a revolution (103° approx.), the collar will have turned through the angle $e_1\,b\,e_3$, or 120° approx., and the collar pin k_6 will have moved 120° to a point l_6, or 17° beyond d_1, when the piston e_6 is on top dead center. In order to have the piston e_6 at the same radial positions as e_1 and e_7 were when passing dead center, it will then be necessary to make the connecting rod for cylinder No. 6 equal to $l_6\,e_1$, which is even longer than the rod for cylinder No. 7.

672. It may have been noted that while the revolving cylinders are considered because of their flywheel effect to have a constant angular velocity, the rotating collar on the fixed crank has a constantly varying velocity, the limits of which will be shown in a succeeding paragraph. This varying angular velocity of the collar together

with the evenly spaced connecting-rod pins K_2 . . K_7 makes it difficult to follow the relative connecting-rod positions during one full turn, and it is necessary to so follow them in order to lay out the design so as to secure equal compression in all cylinders as they pass the top dead center. The following table may aid in following the action of the several parts:

When the master cylinder has turned 1/7 rev., cylinder No. 7 is at top dead center, the collar has turned 65° and the collar pin k_7 is 13¾° to the right of d_1.
When the master cylinder has turned 2/7 rev., cylinder No. 6 is at top dead center, the collar has turned 120° and the collar pin k_6 is 17° to the right of d_1.
When the master cylinder has turned 3/7 rev., cylinder No. 5 is at top dead center, the collar has turned 162° and the collar pin k_5 is 8° to the right of d_1.
When the master cylinder has turned 4/7 rev., cylinder No. 4 is at top dead center, the collar has turned 198° and the collar pin k_4 is 8° to the left of d_1.
When the master cylinder has turned 5/7 rev., cylinder No. 3 is at top dead center, the collar has turned 240° and the collar pin k_3 is 17° to the left of d_1.
When the master cylinder has turned 6/7 rev., cylinder No. 2 is at top dead center, the collar has turned 295° and the collar pin k_2 is 13¾° to the left of d_1.
When the master cylinder has turned 7/7 rev., cylinder No. 1 is at top dead center, the collar has turned 360° and the master connecting rod passes through d_1.

673. Summing up the matter of connecting-rod lengths for revolving engines having auxiliary connecting rods evenly spaced on the collar, it is found that for the dimensions given in paragraph 670 and for uniform compression as each cylinder passes the top dead center:

1st, that the connecting rod for cylinder No. 7 immediately following the master cylinder must be longer than the master connecting rod;

2nd, that cylinder No. 2 immediately preceding the master cylinder should have a connecting rod the same length as No. 7;

3rd, that cylinders Nos. 6 and 3 should have the same length of rod and that this length should be greater than for cylinders 7 and 2.

4th, that cylinders Nos. 5 and 4 should have equal rods which are longer than the master rod, but shorter than the rods for any of the other cylinders.

If the six auxiliary rods are all made the same length, slightly longer than the master rod, and evenly spaced, the several cylinders will carry varying degrees of compression of the fuel mixture as they pass the top dead center. In all cases the several pistons will have slightly varying stroke lengths and slightly varying motions relatively to the cylinders on each stroke.

If the six rods are unevenly spaced on the collar, as at $d_1, d_2 \ldots d_6$, and if the rods are all made the same length and equal to the master rod there will be equal compression in all the cylinders at top dead center, but there will be material differences in the lengths of the strokes. All pistons will travel to equally distant points on their respective cylinders on the out-stroke, but will reach different radial points on the in-stroke. With varying strokes and with varying piston velocities relative to the cylinder walls, the exhaust and other events of the stroke will be somewhat affected.

674. The angular velocity of the collar which turns on the fixed crank pin, also which forms the enlarged end of the master connecting rods, and finally which carries the pivoted ends of the six connecting rods, is variable and ranges between 838 and 1562 revolutions per minute when the cylinders are revolving at their rated speed of 1200 revolutions per minute. This wide variation is shown by the diagram in Fig. 409. The straight horizontal line at H_1 represents the constant angular velocity of the engine cylinders at 1200 revolutions per minute, while the curved line F_1 F_4 F_1 represents the varying angular velocity or revolutions per minute at any phase of the main crank-pin collar. In connection with these variations of angular velocity it must be remembered that it is the connecting rod and the piston that are being thus retarded and accelerated each cycle from 1562 to 838 revolutions, and that the energy developed and required by this variation must go into and come from the uniformly rotating cylinders. This angular acceleration and retardation in each of the revolving cylinders are analogous to the linear acceleration and retardation of the piston and connecting rod in the ordinary reciprocating engine, and if there were only one cylinder there would be a side thrust in the bearing the same as in a single-acting reciprocating engine. With a number of cylinders set radially, as in the several types of revolving engines, this angular acceleration and retardation are more or less balanced, and when it is entirely balanced the forces involved will spend themselves in producing opposite bending moments in the respective cylinders.

675. The angular velocities mentioned above may be found as follows: a e_1, Fig. 408, is the distance from the center of the main or fixed engine shaft to a point on the master cylinder opposite the piston pin of the master connecting rod. Any length of line such as $e_1 f_1$ is taken at e_1 to represent the tangential linear velocity of the cylinder and of the connecting-rod pin, both of which must be the same at that instant. Linear velocity at unit distance is angular velocity, and, therefore, if any radial length is taken as a unit, and the linear velocity at that radial length is found, it becomes a measure of angular velocity as well as linear velocity. Since it is desired

only to compare angular velocities, or revolutions per minute in this instance, any unit for radial length may be taken, and the most convenient unit here is distance $b\,e_1$. It may then be noted that the same angular velocity of the cylinder which produced a linear velocity of $e_1\,f_1$ at the radial distance $a\,e_1$ would produce a linear velocity of $g_1\,h_1$ at the unit radial distance $a\,g_1$ where $a\,g_1$ equals $b\,e_1$. $g_1\,h_1$ now becomes a measure of the angular velocity or revolutions per minute of the cylinder, and its length is taken to represent 1200 revolutions per minute, and is laid off at $G_1\,H_1$ in the diagram Fig. 409. $e_1\,f_1$, Fig. 408, is the linear velocity of the master connecting pin at the same unit radius and, therefore, its length represents the rate at which it is turning about the crank pin and is laid off at $E_1\,F_1$ in Fig. 409, where it is seen to be about 1560 revolutions per minute, or, to be more exact, 1562 revolutions per minute, as found by a separate computation based on the relations of similar triangles and the known lengths of several of the sides.

676. To find the angular velocity of the collar at any other point as when the cylinder casing has turned 1/7 revolution, lay off the line $g_1\,h_1$, Fig. 408, representing the constant angular velocity of the cylinder at $g_2\,h_2$, find the tangential linear velocity $e_2\,i_2$ at the connecting-rod pin, and then the resultant linear velocity $e_2\,f_2$ of the connecting-rod pin itself. $e_2\,f_2$ is perpendicular to $b\,e_2$. $e_2\,f_2$, being at unit radius, is also a measure of the revolutions per minute and is laid off at $E_2\,F_2$ in Fig. 409.

The angular velocity of the master connecting rod around the crank pin will be equal to the angular velocity of the engine cylinders about the main shaft when the master cylinder center line $a\,c_1$ is in the horizontal position $a\,m$. This may be demonstrated directly by making a velocity construction at m similar to that made at e_2 and comparing the similar triangles that will be obtained. That the two angular velocities are equal at this phase is indicated by the curve $F_1\,F_4\,F_1$, Fig. 409, crossing the horizontal line through H_1 at M just before it reaches the ordinate through E_3. This corresponds to the amount that m, Fig. 408, is ahead of e_3.

677. Finally, it may also be of interest to note that the mechanism of the Gnome engine is identical with the essential elements of the widely known Whitworth motion, invented by Sir Joseph Whitworth in England, about 1850, and used on many types of shaping machines. In the Whitworth motion the shaft b is the driver turning the arm $b\,e_1$ with uniform angular velocity, and driving the arm $a\,c_1$ and the shaft a with variable angular velocity between the same relative limits as shown in Fig. 409. The ratio of the times of advance

to return for the cutting tool in the Whitworth motion would be as the circular arc $n\ e_1\ m$ is to the arc $m\ p\ n$, or as 43 to 29 for the proportions here given for the Gnome engine. The elementary revolving parts of the Whitworth motion are transferred into a Gnome type of engine by allowing the two arms $b\ e_2$ and $a\ c_2$ to turn freely on their fixed shafts and suitably enclosing the end of the former within the latter and then introducing an explosive charge at the pivoted and sliding junction e_2 of the two arms.

Gyro Motor

678. The Gyro motor is of the revolving type, the radially placed cylinders revolving about a fixed shaft, and the connecting-rod ends revolving about a fixed crank-pin center. In these general respects the Gyro is similar to the Gnome engine. The details of construction and operation, however, differ considerably. The Gyro engine is an American product and is manufactured by The Gyro Motor Co., Washington, D. C. It is a four-stroke engine. A former type of the Gyro, now discontinued, admitted the gas through a positively controlled valve in the piston head. The present type, put out in 1914, admits the fuel through ports in the cylinder wall, as explained in the following paragraphs.

679. The construction and operation of the Gyro motor are as follows: *1*, Fig. 410, is a gas-tight crank case which carries the cylinders and revolves with them; *5* and *15* are the two ends of the fixed shaft; *14*, the hollow part of the shaft through which a rich fuel gas mixture is admitted to the crank case; *6* and *13*, the fixed center cranks; *8*, the fixed crank pin; *12*, *12*, the cupped end or collar of the master connecting rod *24*, and *7-7*, the mating cupped half which is bolted to the end of the rod. This collar is split in this way to facilitate assembling. *11* is a bushing which fills the collar except for the openings for the six auxiliary connecting rods, and which provides a bearing surface for the master connecting rod on the fixed crank pin. *9*, *10*, *4* is one of the six auxiliary connecting rods.

680. As the piston *25* moves down relatively to the cylinder on its intake stroke the exhaust valve *27* is kept open for a greater part of the stroke by means of the disk cam *2* and the connecting mechanism. When the top of the engine piston *25* reaches the opening *33*, the fuel control piston *36* has moved up so that the port *35* in this piston is opposite *33* and the rich gas from the crank case is drawn in through the duct *38-37*. This duct is shown as entering the crank case just above *7* for simplicity of illustration, whereas on the engine itself there

is not sufficient room for it in this position and it is made to cross between the cylinders and enter the crank case just above *12*. The rich gas from the crank case mixes with the fresh air and both are compressed on the up-stroke. Ignition is from a spark plug at *26*. Auxil-

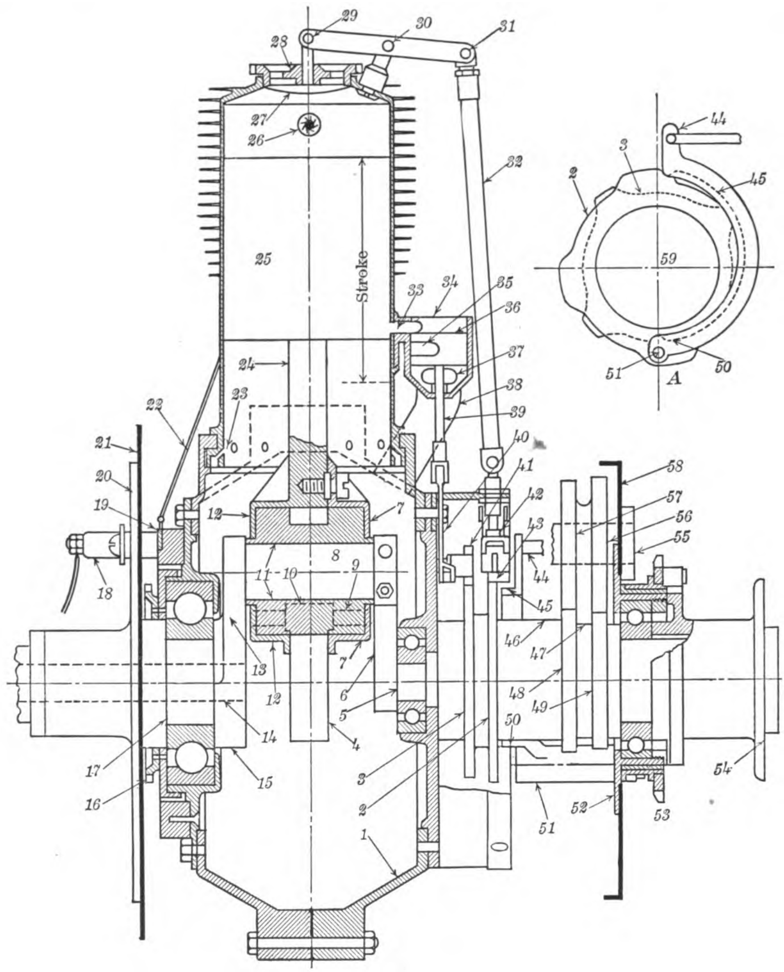


FIG. 410.—GYRO MOTOR.—DIAGRAMMATIC SECTION, AND DETAIL OF LOW COMPRESSION STARTING DEVICE

iary exhaust begins through the port *33* when the top of the engine piston uncovers it on the "down" or explosion stroke. At this phase the piston valve *36* is moved down by the cam *3* so that the port *33* is open to the atmosphere only. On the in-stroke, after the port *33* is opened, the exhaust valve *27* is opened against light pressure by the cam *2*, and the remaining burned gases are expelled as the piston moves out.

681. The cam disks at *2* and *3* contain four lobes each, and are illustrated in a front-view detail at *A* in Fig. 410. Both cam disks and the toothed gear wheel *48* are fastened to a sleeve *46* which turns freely on an inner sleeve *47*. The gear wheel *49* is connected, by the inner sleeve, with the crank case and consequently turns at the same speed. The offset differential gear wheels *56* and *57* turn about a pin *55* which is fixed in the stationary framework of the engine. The numbers of teeth on these four wheels are as follows: wheel *57* has 28 teeth; wheel *56*, 30 teeth; wheel *48*, 32 teeth; and wheel *49*, 30 teeth. When wheel *49* makes one turn, wheel *48* will make $\frac{30}{30} \times \frac{28}{32}$ turns, or $\frac{7}{8}$ of a turn in the same direction. It follows then, when the crank case has made two revolutions, that the cam disks have made $1\frac{3}{4}$ revolutions, and that the cam-follower roller at *41* and the one at *43* have each rolled over one-quarter of the periphery of its own cam. Therefore if four lobes are equally placed on the cam periphery as shown at *A*, Fig. 410, each exhaust valve on each cylinder and each auxiliary intake valve will be opened each second revolution by successive lobes on the cam. The cam-follower roller *41* is carried on one arm of a bent rocker which is pivoted to the crank case while the end of the other arm is connected to the link *40* which operates the auxiliary intake valve *36*. The roller *43* is carried on one end of a simple lever arm which is pivoted to an extension of the crank case. This lever arm is introduced to take the side pressure of the cam from the radial follower *42*. This follower takes its motion from the lever arm at a point directly over the roller *43* and transmits its motion through the connecting link *32* and the rocker *31-29* to the exhaust valve *27*.

682. For easy starting of the motor a separate "low-compression cam" is provided as shown at *44*, *45*, *50*, *51*. When starting, rod *44* is pulled to the right, thus rotating the semi-circular cam about the pivot *51* and making the cam surface shown by the dash line eccentric with respect to the center of rotation of the roller *43*. Therefore the roller will be lifted and the exhaust valve *27* opened on each "up" stroke of the piston, independently of the lobes on the cam *2*. The amount of compression relief that is desired is regulated by the distance through which the rod *44* is moved.

683. Other features of the Gyro engine are that an odd number of cylinders are always used in each ring of cylinders for the reason explained in paragraph 669, and the six auxiliary connecting rods are equally spaced on the collar *12-7*. Each of the six connecting rods is of the same length but they are longer than the master rod. Oil

holes for the purpose of lubricating the cylinder walls and piston pin are shown at *23*. *22* is the ignition wire to the spark plug *26; 19* is a spark distributor; *18*, a contact brush; *16*, a gear wheel for operating the oil pump and magneto, etc.; *20*, a rear mounting plate; *21*, a rear mounting frame; *58*, a front mounting frame; *52*, a front mounting flange or plate; *53*, a ratchet wheel for starting from seat; and *54*, the propeller flange.

Valve Operated by Oscillating Motion of Connecting Rod About the Piston Pin

684. The old type of Gyro motor, now discontinued in favor of the form shown in Fig. 410, is of special interest on account of the method of operating the fuel intake valve in the piston head. The motion for controlling this valve comes, in part, from the turning or oscillating action of the connecting rod on its wrist pin. This same peculiarity of action is illustrated with a somewhat different detail of application in one other case in this work in connection with one of the types of uniflow steam engine in Fig. 274, Vol. 1.

685. The center line of the fixed shaft of the revolving motor is shown at *1*, Fig. 411, the outside of the fixed hollow crank shaft at *2*, the fuel mixture intake at *3* through the hollow shaft into the closed crank case, the fixed crank at *4*, and the fixed crank pin at *6*. The sleeve-end *5* of the connecting rod *29* turns freely on the crank pin *6* and carries six pins evenly spaced. To these are attached the "auxiliary" connecting rods whose center lines are represented by *9*, *31*, etc. All of the above parts are illustrated diagrammatically without any attempt to show detail construction or proportions.

686. As the cylinders turn, counter-clockwise for example, the angle between the center lines of the cylinders and of the connecting rods varies, this angle being equal to a for the phase shown with the connecting rod at *9*. Considering now the mechanism at the other end of the rod *15*, it will be noted that *18* is the piston, *19* the intake valve, and *27* a counterweight which is pivoted at *25* to the piston frame and which balances the weight of the valve against centrifugal force. These engines run at 1000 to 1200 revolutions per minute. Another centrifugal weight *17* is pivoted at *16* to a lug on the connecting-rod end, and at the phase of the sectioned cylinder this second weight bears against the first at the point *26* to the right of the pivot and its centrifugal effect is added to that of the weight *27* in keeping the valve firmly to its seat. The spring at *28* serves to hold up the weight *17* at all times against the weight *27*, and to prevent dropping when at rest.

687. When one of the other six cylinders is in the position indicated by the center line *8* the connecting rod will make an angle a with it, as already explained, and approximately this same angle will also be traversed at a_1 by the pin *16* while the cylinder revolves from

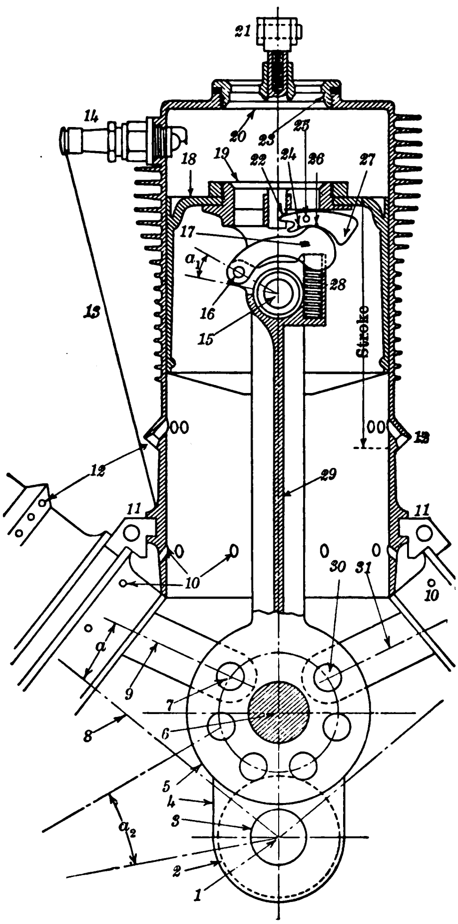


Fig. 411.—Old Type Gyro Motor, Showing Valve Operated by Oscillation of Connecting Rod

the vertical position to position *8*. The weight *17* will now be in contact with the weight *27* at the point *24* to the left of the pivot *25*, and the centrifugal effect of *17* will be acting to keep the valve open for the admission of the fuel gas. On the compression stroke the angle a_1 exists on the other side of the line *15-16*, and the weight *17* acts with the compression in keeping the valve seated. On the explosion stroke the excess explosive force keeps the valve closed during the period that the weight *17* tends to open it. When the normal to the surfaces of contact of the two weights passes through the pivot *25*, as it will do at some intermediate position between *24* and *26*, the centrifugal force due to the weight *17* exerts no control on the valve. When the cylinder has turned through about two-sevenths of a revolution the angle between the "auxiliary" connecting rod and the cylinder center line will be larger, as shown at a_2.

688. The exhaust valve *20* is operated in the same way as the exhaust valve in Fig. 410. A variable compression cam is also used in this motor and it controls the amount of compression at starting as at any other time, as desired, in the same way as explained in paragraph 682. The openings at *12* are auxiliary exhaust ports. There are no valves in these openings, so that the crank case, which contains a rich gas mixture, is open to the atmosphere at the phase shown by the sectioned cylinder. None of the gas escapes, however, for there is a suction intake purposely designed to add air to the enriched gas to form a proper mixture. The holes shown at *10* open into the crank case and they are for lubricating the cylinders. All of the small holes in the cylinder, it will be observed, are drilled at an angle such that the centrifugal effect on the oil particles will carry them in readily through the inner set at *10*, and will not permit them to be dashed or sprayed out at the outer set at *12*. The finger *22* on the balance weight *27* fits into a groove in the stem of the exhaust valve *19*.

SECTION XII.—THE GAS ENGINE

699. The history of internal combustion engines starts with the gas engine proper as distinguished from the gasoline and oil engines. The gas engine itself appears to be an outgrowth of an attempt to use gunpowder, an engine for which was devised by Christian Huyghens, a Dutch mathematician and astronomer, in 1680. The Abbé Hautefeuille of France is credited with having proposed a gunpowder engine two years earlier, although there appears to be no record of construction or experiment. A gas engine was patented by Robert Street in England in 1794. Samuel Brown took out patents in 1823 and 1826, and his was probably the first to do actual work. Other patents were taken out by W. L. Wright in 1833, and by William Barnett in 1838, those of the latter being of special importance because of the flame-ignition device and because of the first practical use of compression. Many other patents followed, the most important of which were A. V. Newton's in 1885, in which ignition was secured through contact with red-hot metal, and Barsanti and Matteucci's in 1857, in which a free piston was first used. The first gas engine to be actually constructed in quantity was built by M. Lenoir in Paris in 1860. It was afterward built in both France and England.

Lenoir, the First Commercially Successful Gas Engine

700. The Lenoir engine was not different in principle from previously attempted engines, but its proportions and its details of construction were better. The engine followed closely the lines of the steam engine. This may be seen in Figs. 420 and 421, which are from illustrations in "The Gas Engine," by Dugald Clerk. These illustrations were made from a Lenoir engine which for many years supplied the power for the repair-department in the Patent Office Museum at South Kensington, London. Mr. Clerk states that he found many of these engines running regularly after 20 years of service. The engine here illustrated developed one-half horse-power, the bore of the cylinder being 5½ inches and the piston stroke 8½ inches. No reference is found to show that any of these early engines developed more than one horse-power.

701. A governor controlled the supply of gas, which entered at *B* for the head end of the cylinder and at *C* for the crank end, Fig. 420.

The gas did not pass directly to the cylinder, but followed through a transverse port in the valve D to the side A, where it met the entering air from J, and both passed through a perforated or mixing plate back through the lower part of the valve into the main cylinder port. The fuel mixture followed the piston for about one-half stroke, when it was ignited by an electric spark at L. On the return stroke of the piston the burned gas was exhausted and the engine was again ready to take in a new charge. From this it will be seen that there was a power impulse for each stroke of the piston, and that in this sense the engine was double-acting, the same as the steam engine. In the gas

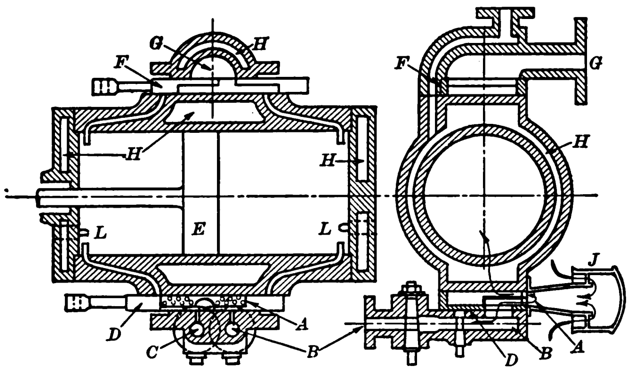


FIG. 420 FIG. 421

FIGS. 420 AND 421.—THE LENOIR, THE FIRST COMMERCIALLY SUCCESSFUL GAS ENGINE, 1860

engine, however, there was power for only half the stroke. There was no compression of the fuel charge, it being simply drawn in from atmospheric pressure and ignited just after the valve D had cut off the supply at about half-stroke. The intake valve D and the exhaust valve F were operated from separate eccentrics. The exhaust valve F was held open for a full stroke. A number of the drawings of the Lenoir engine that may be found show only a water-jacketed space around the main cylinder. It is believed that such illustrations were from specifications that had been set forth at an early date, rather than from actual working drawings of the successful engine, or from sketches of the engine itself. In the engine here described the cylinder heads and exhaust-valve casing are also water-jacketed, as indicated at H, H.

Otto-Langen Free-Piston Gas Engine

702. The next marked advance was the Otto-Langen free-piston gas engine, first built and exhibited at the Exposition in Paris in 1867. It was exhibited in the United States, and one of the engines is still preserved in the historical exhibit at Stevens Institute of Technology. A diagrammatic outline of the engine is shown in Fig. 422. The valve *B* controls admission, ignition, and exhaust. It is operated from an eccentric indirectly from the main shaft *K*. The charge of gas enters at *O*, the air at *N*, and both pass through a channel *P* on the under side of the valve to the port *D* and into the cylinder. During the first few inches of rise of the piston *E*, gas and air, without previous compression, are admitted under the valve *B* as just described. This valve then shuts off the gas and allows a flame at *A* to ignite the charge as follows: When the valve is in its lowest position a mixture of gas flows from *R* into the passageway *Q* of the valve, where it is ignited by the gas flame at *A* in the valve cover. As the valve slides up, the flame in *Q* flashes into the fresh mixture in the port *D* and explodes the charge. The piston shoots up with high velocity, finally creating a partial vacuum beneath it which, together with gravity, brings the piston to rest near the top of the cylinder, which is entirely open.

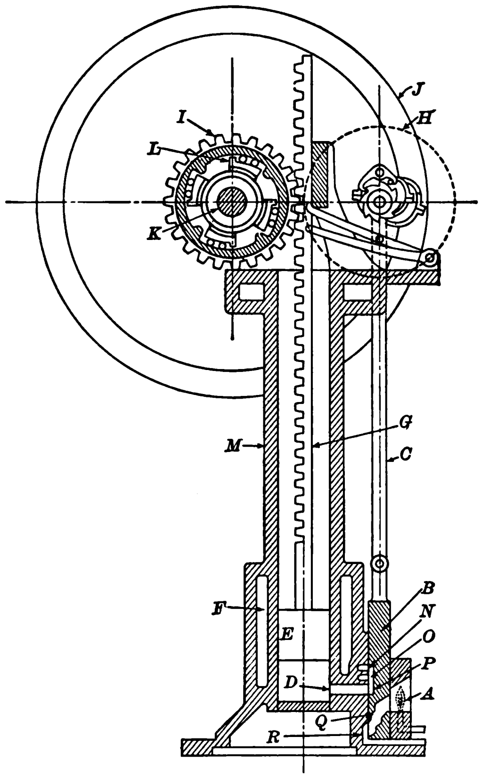


Fig. 422.—Otto-Langen Free-Piston Gas Engine

703. The piston rod *G* carries a toothed rack, heavily braced at the back, which gears with a toothed wheel *I*. This wheel is not

keyed to the shaft K, and so is turned freely on it while the piston is moving up. While the piston is moving down, the rack turns the gear wheel I in the opposite direction. This causes small rollers to wedge between the spiral surface on the inner side of the rim of wheel I and the spiral wedges which are shown at L and which grip a disk that is keyed to the shaft K. Thus the shaft K is driven on the down-stroke of the piston. On the up-stroke the direction of turning of the wheel I is again reversed, thus allowing the rollers to unlock automatically from their wedge grip, and permitting the shaft to turn in its constant running direction under the influence of the flywheel J. The downward motion of the piston, which is the working stroke, is caused by atmospheric pressure on the piston due to the vacuum underneath and by the action of gravity on the piston. During the downward stroke exhaust takes place through the opening in the valve B. The top or outside of this port is then just above the flame box. In larger sizes of engines the speed of the engine was regulated by adjusting the exhaust opening. Tests conducted by Dr. Robert H. Thurston at the Vienna Exhibition in 1873 showed that one of these engines having a piston diameter of 8.75 inches and a maximum stroke of 41.3 inches developed a maximum brake horse-power of 0.896 at 85.7 revolutions per minute. The flywheel and shaft made about three revolutions for each power stroke. The lifting of the piston through the few inches necessary to draw in the charge was accomplished by an ingenious arrangement of a pawl and wheel and levers and links connected with the governor, so that if the engine exceeded a desired speed the piston would not be lifted the few inches necessary to draw in the charge and there would be no explosion until the engine had slowed down. A heavy flywheel was used. The engine was extremely noisy and set up heavy vibrations when placed on a floor span, but it was economical in fuel.

Brayton Constant-Pressure Gas Engine

704. A new form of engine which had a marked but short period of success, and of which a large number were built, was patented by George B. Brayton, an American, in 1872. It was the only engine that had ever been used for any definite period in which the combustion took place at constant pressure for a considerable part of the stroke. Illustrated diagrammatically in Fig. 423, O is the fuel compression cylinder, the piston P being driven from the point R on a rocking or walking beam FQS. Air and gas were drawn into the cylinder O through an automatic valve, and on the up-stroke of the

piston the mixture was forced through a valve and the pipe L into a reservoir tank G and into the engine cylinder C through the intake valve H. This valve was operated by a lever which received its motion from a cone cam, which in turn was moved along a shaft by a governor, thus regulating the closing of the valve H and the fuel charge. A small jet of gas ran into the power cylinder at J just below a wire screen indicated at I, and a needle flame from it was kept burning constantly. As the main charge of carbureted air passed this flame it was ignited, and continued to burn as it entered the cylinder and until the valve at H was closed. The wire gauze was interposed be-

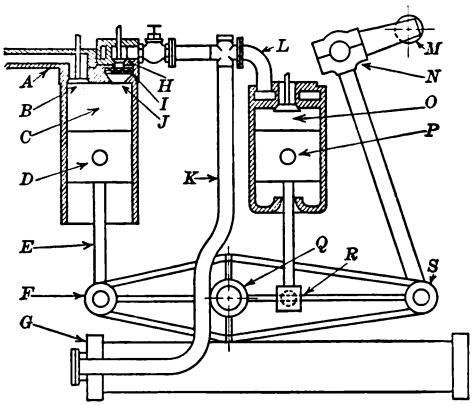


Fig. 423.—Brayton Constant-Pressure Gas Engine

tween the flame and the supply valve in an endeavor to keep the flame from striking back into the fuel receiver. The exhaust or burned gas passed out through the valve B and the exhaust pipe A.

705. The Brayton engine was not successful when running on gas owing to the failure of the wire screen through puncturing to keep the flame from flashing back and causing explosion in the receiver. The engine was then slightly changed so as to operate on light petroleum, by placing a pad at I, on which the liquid fuel was deposited by a separate pump. The pump O then compressed air only which passed through the saturated pad, thus forming a combustible mixture, which was ignited at J by means of a constantly burning flame, as it entered the cylinder. The governor shaft, which controlled the movement of the valve H and which operated the pump for supplying gasoline to the felt pad and to the burner, ran horizontally across the top of the engine and received its motion from the shaft M through a bevel-wheel connection.

THE OTTO, THE FIRST "FOUR-CYCLE" GAS ENGINE

706. In 1876, Nicholas A. Otto, of Deutz, Germany, brought out the "Silent Otto Engine," which was far superior in power and in economy to any internal combustion engine then in use. It was the first engine to embody distinctly the principles of the present-day "four-cycle" or four-stroke engine. These engines are still spoken of as operating on the Otto cycle. It was referred to as the "silent Otto," no doubt to distinguish it from the Otto-Langen engine, described in the previous paragraphs, and invented by Otto in conjunction with Langen about twelve years earlier.

707. The valve gear is operated as follows: A horizontal lay shaft *G*, Fig. 424, is driven from the main engine shaft by bevel wheels. A crank *I* is fastened to the end of the lay shaft and to it is connected a link *J* which runs across the end of the engine and moves the inlet valve *H* back and forth as the shaft *G* revolves. Air is admitted from *F* to the port *K* in the valve, and an instant later gas is admitted to the same port *K* from *M*, and the mixture passes through the inlet opening *N* to the cylinder on the suction stroke. On the compression stroke the valve *H* covers the opening *N* until time for ignition, which takes place from a confined flame in the opening *O*. This flame continues to burn for the short period while *O* is traveling from *P* to *N*. The ignited charge drives the piston for the full length of the third stroke of the cycle, and exhaust takes place through *Q* on the fourth or return stroke. The piston is shown at the head end of its stroke with a large clearance space at *D*.

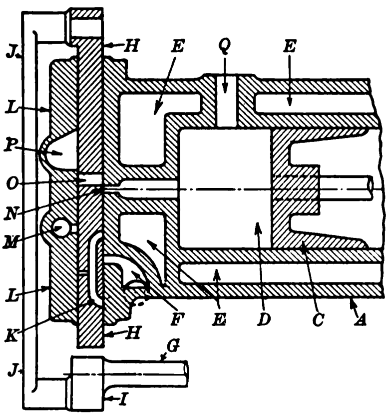


FIG. 424.—OTTO GAS ENGINE CYLINDER AND VALVE, HORIZONTAL SECTION

708. The method of producing a flame in the confined space is, briefly, to supply a jet of gas up through the opening *P* (see also Fig. 425), in the valve cover *L* and keep it constantly burning. A chimney is shown at *S* just above the burner at *P*. Air and gas are also supplied to the opening *O* in the valve through specially formed small openings and channels not shown in the illustration, this supply continuing while *O* is passing the steady flame at *P*. Just before the

channel O, filled with burning gas, reaches the main cylinder opening at N, it receives a fresh supply of the mixture that is now almost fully compressed in the engine, the supply coming through a very small opening not shown in the illustration. This supply from the engine not only reinforces the confined flame, but also raises the pressure of the small burning mixture in O and keeps it from being snuffed out when O reaches N.

709. All the valve gear used on the Otto engine takes its initial motion from the lay shaft shown at G, Fig. 424, and at T in Figs. 425–6, as described above. It has been shown how the main flat inlet valve receives its sliding motion back and forth. The exhaust valve

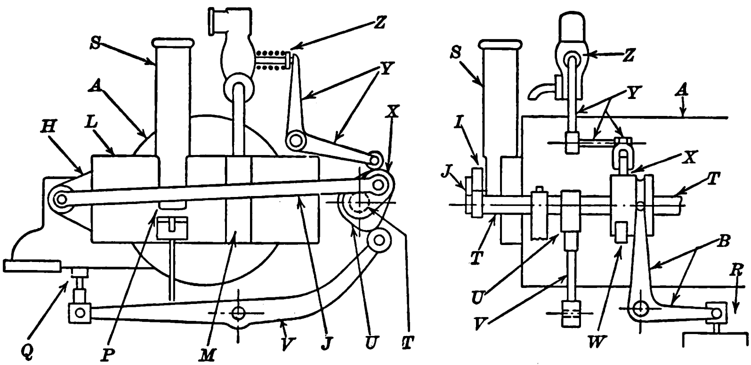


FIG. 425 FIG. 426

FIGS. 425 AND 426.—OTTO GAS ENGINE VALVE GEAR, DIAGRAMMATIC END AND SIDE VIEWS

is a poppet or lift valve and its location is indicated at Q in Figs. 424 and 425. It receives its motion through the rocker arm V from the cam U which is fixed to the lay shaft T.

710. The speed of the engine is controlled by a governor whose location is indicated at R. If, for example, the speed of the engine increases, the point R of the governor bell-crank B is moved up and the upper arm will slide the cam drum W along the shaft T until the cam projection on the drum does not, as it revolves, come under the roller X of the offset lever Y Y Y. In this case the lever will receive no motion, and none will be imparted to the gas-admission valve at Z. Therefore there will be no gas at M, Fig. 424, when the suction stroke takes place and only air will be drawn in. This will be compressed and the flame will be delivered at N just the same, but there will, of course, be no explosion and the air will be discharged on the fourth stroke. If the engine slows down sufficiently during the two revolutions, or during subsequent revolutions, the governor will swing

the bell-crank *B* and so move the cam drum along until the projection *W* rotates under the roller *X*. When it does so the gas valve at *Z* will again admit gas to the engine through the openings at *M* in the valve cover and at *K* in the valve itself.

The Clerk, the First "Two-Cycle" Gas Engine

711. The first successful "two-cycle" or two-stroke internal combustion engine was invented by Dugald Clerk, of England, in 1880. Previous attempts had been made during a number of years by Mr. Clerk and others to produce an engine that would work with a compressed charge in the engine cylinder and also give a power stroke for each revolution of the engine shaft. Notwithstanding the present-day simplicity of the two-stroke engine, its development for commercial use proved a slower and more difficult problem than did the four-stroke engine.

712. The Clerk engine operates as follows: An auxiliary cylinder with a piston *O*, Fig. 427, driven from the main shaft and 90° ahead of the main crank, was used to draw in the air and gas, and to deliver the mixture to the engine cylinder *C*. The auxiliary cylinder and piston were known as the "displacer." The mixture was timed to enter the cylinder *C* just after the piston *B* had uncovered the exhaust ports *A*, *A*, in order to drive out the burned gases. The quantity of the mixture, the cylinder proportions, and the piston positions and speeds were such, however, that the piston *B* had advanced far enough on its return stroke to cover the ports *A* before any of the fresh mixture could escape through the exhaust. The engine piston on its return stroke compressed the charge, and it was ignited at *Q* by a small flame burning in a cavity in the slide valve *F*. The piston was thus driven down on its power stroke, near the end of which the exhaust ports were uncovered, and the cycle repeated.

713. Some of the structural features and details of the engine were ingenious, and the principles involved are to be found in different grades of successful engines manufactured at the present time. It is to be noted first that the admission was at the head end of the cylinder, which added to the efficiency and to the ability to regulate speed, although it lacked in simplicity of construction. When the piston *O* of the "displacer" moved away from the head end it drew in the gas through a somewhat tortuous passage starting at *K*, through the port *I* in the slide valve, into the opening *J*, all in Fig. 427, then to the annular space *U*, Fig. 428, and through a series of fine holes which opened on the valve seat of the valve *T*. This valve was automatically lifted by the suction of *O*, and so the gas passed through the

fine holes, and, as it did so, it met with a jet of pure air coming up through the tube *V*. The air and the gas became mixed and were drawn through the pipe *L* over the baffle-plat *M* into the cylinder *N*. As the piston *O* returned it forced the mixture back through the pipe *L*, and thus automatically lifted the valve *S* and admitted the charge to the conical compression chamber *P* and to the cylinder. The charging cylinder, or "displacer," was not a compressor in the

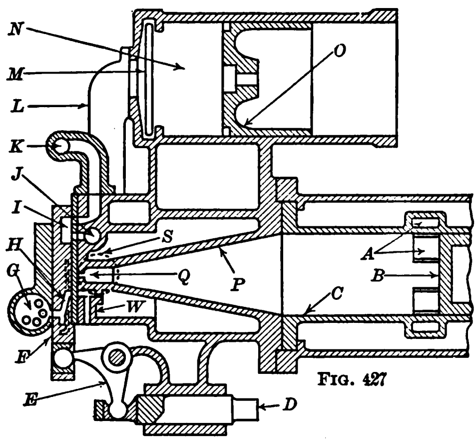


FIG. 427

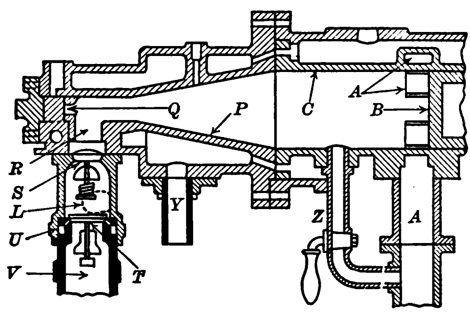


FIG. 428

FIGS. 427 AND 428.—CLERK GAS ENGINE. HORIZONTAL AND VERTICAL SECTIONS. THE FIRST COMMERCIAL "TWO-CYCLE" GAS ENGINE

working sense, such compression as it gave being simply to operate the check valve and to drive the charge into the cylinder.

714. The ignition of the charge was accomplished by means of a confined flame in a cavity in the valve *F*. A combustible mixture was supplied to the opening *H* in the valve *F* by means of small holes not shown except as indicated by the dash lines in the cross-sectioning. This mixture came from the fresh charge in the engine cylinder and was kept from back-firing by a grating or gauze where it entered the

valve cavity. A flame was kept constantly burning at G and this ignited the gas in the cavity, the products of combustion passing out through a small opening at W. As the valve moved up, the confined flame at H was released as the upper end of the opening H came opposite Q and the charge was then ignited. There were other details of construction which helped to regulate the action of the flame, but they were too small to show in the present illustration.

Further description of this and other valves that have been described thus far in this section may be found by consulting some of the older works on gas engines, notably "The Gas and Oil Engine," by Dugald Clerk, and "Gas and Petroleum Engines," by William Robinson.

Methods of Controlling the Speed of the Gas Engine

715. Gas and oil engines are usually made in large sizes for producing larger units of power than gasoline engines, and it becomes necessary to provide means for automatically maintaining a practically normal speed for varying load. All automatic regulation of internal combustion engine speed comes from a governor which may be of the pendulum, centrifugal, or inertia type, or it may be based on the compression and flow of gases or fluids. Each of these types is illustrated in the following paragraphs.

716. The following general methods of controlling the speed of internal combustion engines may be used, and each method may be accomplished in several different ways. Each way calls for a special type of valve gear or valve, or both, to accomplish the desired results, but the details of the construction of the gears in any one type may vary greatly. The great number of gear types, only a few of which can be mentioned in this work, are probably due to the fact that later manufacturers have found it necessary to devise original methods of regulation due to the patent protection afforded to the earlier builders.

1. Hit-and-Miss Method, by keeping
   - (a), The main admission valve closed on the suction stroke,
   - (b), The fuel valve closed where a separate fuel valve is used,
   - (c), The exhaust valve open during suction, thus breaking a vacuum and preventing the admission valve from lifting if it is automatic.

2. Quantity Method, by
   - (a), Cutting down the rate of fuel supply during the entire suction stroke, called Throttling Method,

(b), Cutting off altogether the full rate of supply after the piston has traveled a part of the suction stroke, called Cut-off Method.

(c), Drawing in the fuel at full rate for full suction stroke and then forcing part of it back before the fuel valve is closed.

In the quantity method the ratio of gas to air remains constant, thus giving a uniform explosive mixture.

3. Quality Method, by
   (a), Changing the ratio of gas to air, thus making the mixture lean or rich,
   (b), Retaining some of the exhaust gas, thus reducing the quality of the gas.

   In the quality method the quantity remains constant, thus giving constant initial compression.

4. Combination Method, by
   (a), Changing the quantity and quality at the same time.
   (b), Using either quantity or quality at high loads and Hit-and-Miss at low loads.
   (c), Using either quantity or quality and ignition timing methods.

5. Ignition Timing Method, by
   (a), Causing the spark or flame to ignite the charge at different positions of the piston.

Hit-and-Miss Types of Construction

717. A valve gear for the hit-and-miss method in which the admission valve is closed on the suction stroke is shown in Fig. 429. With the gear in the position shown, the sharp edge at the end of the arm K will engage the "V" at the end L of the stem of the admission or mixture valve, and the valve will be lifted each revolution of the cam shaft F. If the governor weights move out, the bell-crank B will rotate a small distance about its fixed center and will draw the arm K to the right so that when it is lifted by the cam F it will miss the end L of the valve stem and no fuel will be admitted that stroke. Lack of fuel on one or more cycles will cause the engine to slow down until the governor

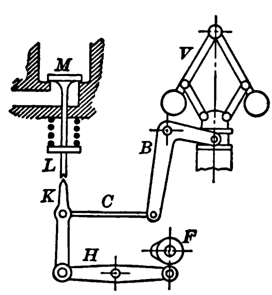


Fig. 429.—Hit-and-Miss with Flyball Governor

weights at V have moved in far enough to push K over until it again engages with L.

718. A pendulum governor used in connection with the hit-and-miss method is shown in Fig. 430. With the engine running at normal speed the pendulum weight V, backed by the compression spring N, does not lag behind enough to keep the "pick" or sharp-edged rod K from engaging with L and lifting the inlet valve M. The rod K is fastened to a stub projection on the governor arm. The compression spring N seats also on a stub of the governor arm, and on a projection from the arm H. When the engine speed increases, the pendulum weight lags and turns the "pick" K down so that it misses the stem of the valve M.

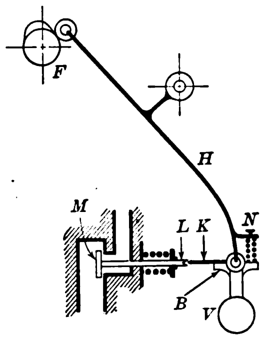


FIG. 430. — HIT-AND-MISS WITH PENDULUM GOVERNOR

719. An air-compressor regulator for the hit-and-miss method is shown in Fig. 431. The rod R slides back and forth and carries with it the cylinder T. A stationary piston is held inside the cylinder by the stud S, which is fastened in the engine frame. A regulator needle valve at Q is adjusted to let the compressed air out of the cylinder on its left-hand stroke fast enough, at normal speed, so that it will not accumulate enough pressure to move the piston at O against the compression spring at that place. Above normal speed, pressure does accumulate and drives out the piston O, to which is attached a "pick" K. With K out at the position shown by the dash lines it will not engage with L and the admission valve M will not be opened.

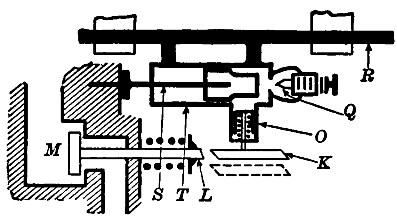


FIG. 431.—HIT-AND-MISS WITH COMPRESSED-AIR REGULATOR

720. A hit-and-miss valve gear for the case where two valves are used, one for gas and one for the gas-and-air mixture, is shown in Fig. 432. The gas valve is at G, and it remains closed when the governor weights fly out and draw the rod K to the right so that it misses L on its up-stroke. In this case air only enters the cylinder at M, and this serves to scavenge more thoroughly the engine and to give a richer mixture and a more powerful stroke when the governor again places K under L.

721. An example of hit-and-miss governing by keeping the exhaust

valve open and thus breaking the vacuum on the suction stroke is shown in two views in Fig. 433. This case is for an automatic intake valve which depends on a vacuum to open. The governor weights V, at normal speed, hold the "stop" C in the position shown in the lower view. As the governor weights fly out, the "stop" C moves down between a faced edge D of the engine frame and the end E of the rocker H. The exhaust valve X will be then moved from its seat and will be kept open a certain amount all of the time. The push rod F, which receives a back-and-forth motion from a cam, does not now reach the rocker, or if it does it only serves to open the exhaust valve still further. When the engine resumes its normal speed the "stop" C is withdrawn and the rod F regularly opens the exhaust valve.

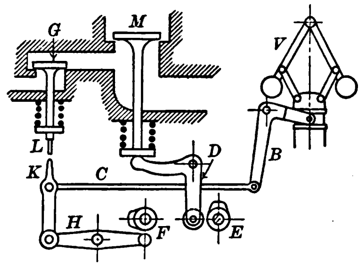


FIG. 432.—HIT-AND-MISS WITH SEPARATE GAS AND INTAKE VALVES

722. There are numerous forms of construction for the hit-and-miss method of governing in addition to those illustrated above, one of the most prominent of all forms being that of the shifting cylindrical cam as illustrated in connection with the Otto engine in Fig. 426. A slightly different form of this same construction consists in moving the roller follower along by means of the governor, instead of the cam cylinder, so that the roller and cam projection will not meet at high speeds.

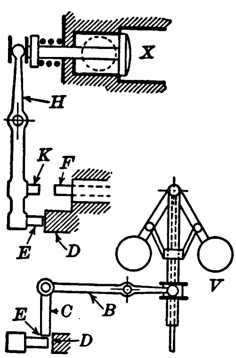


FIG. 433.—HIT-AND-MISS KEEPING EXHAUST VALVE OPEN

The Quantity Method

723. The quantity method of speed control for gas engines is embodied in many different forms of construction, nearly all of them giving constant periods of admission but cutting down the volume by reducing the opening through which the mixture passes on its way to the cylinder. The ratio of gas to air, however, is designed to remain the same at all loads. The principal disadvantage of this form of governing is that a reduced volume in the cylinder gives a reduced compression and reduces the economy or efficiency of the engine. Its advantage is that it maintains a practically uniform combustible mixture. The various types of construc-

tion differ not only in the valve-gear arrangement, but in the forms of the valves as well.

724. The use of a butterfly valve for throttling the mixture is shown in Fig. 434. Gas is admitted through an inside tube at G and air through an outer tube at A. The mixture then passes by the throttle valve T as it is drawn into the cylinder through the intake valve M by the suction of the piston. The valve M is opened and closed at the same times at all loads by means of a cam, not shown, which acts through the push rod E and the rocker H. As the load decreases and the engine tends to speed up, the governor weights at V fly out and raise the governor end of the rocker C, which rotates the butterfly valve clockwise, and thus reduces the area through which the mixture must pass.

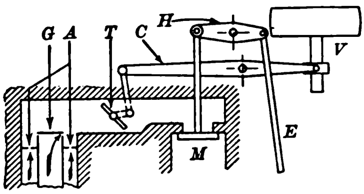


FIG. 434. — QUANTITY GOVERNING, THROTTLING FOR FULL STROKE WITH BUTTERFLY VALVE

725. Separate disk valves for throttling the gas and the air, together with a different type of valve gear, are illustrated in Fig. 435. The main admission valve M is operated from a cam not shown. The gas valve is represented at G and the air valve at A. The valve gear operates as follows: The cam F swings the hanging arm H and this in turn pushes the distance-block, or die, K. The die K is suspended by rod C from the governor lever B, and its position, or height, is determined by the governor. The die K transmits motion to the upright arm D, and this in turn gives the final motion to the rod carrying the gas- and the air-valve disks. The position shown is for full load, and the valves will receive the maximum opening. As the load falls and the governor speed increases, the end B of the governor lever rises and carries with it the distance-block K, which now receives a smaller motion from the arm H and transmits also a smaller motion to the arm D and to the valves.

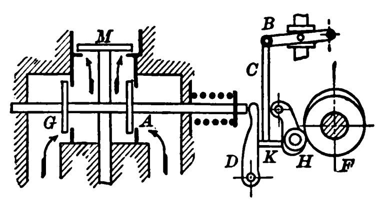


FIG. 435.—QUANTITY GOVERNING, THROTTLING FOR FULL STROKE WITH DISK VALVES

726. A valve gear quite different in appearance from the one shown in Fig. 435, but identical in mechanical principle, is shown in Fig. 436. The distance-block K in the former Figure and the roller K in the latter are both moved by the governor and both serve to

change the driving radius of the arm H and the follower radius of the arm D. The latter will have a smaller angle of swing, as K is moved toward the center of rotation of the driving arm H. In Fig. 436 a combined sleeve and disk valve is used. The sleeves at G and A and also the disk valve at M are all fixed to the one valve stem. The part of the sleeve at G has a slightly larger lap at the top than the sleeve at A has. This is to permit the pure air to scavenge the cylinder before the gas-and-air mixture enters.

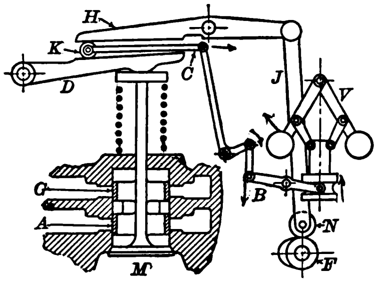


FIG. 436.—QUANTITY GOVERNING WITH SLEEVE AND DISK VALVES

727. A prominent form of valve gear that has been largely used is illustrated in Fig. 437. The rocker H has no permanent point of support and is, in fact, a floating lever, the fulcrum of which is changed by the governor to give varying degrees of motion to the disk valves at N and M. The left-hand end of the lever H is pinned to the valve stem L. The gear is shown in the position for a light load because the fulcrum arm K is close to the valve stem. The right-hand end of H receives a constant motion from the cam F at all loads. If the governor weights and links in the governor case V are arranged so as to turn the governor lever B clockwise as heavier loads come on the engine, the bell-crank K will turn counter-clockwise and the fulcrum arm K will move to the right and cause the valve stem and valves to receive a greater motion and so admit more of the fuel mixture.

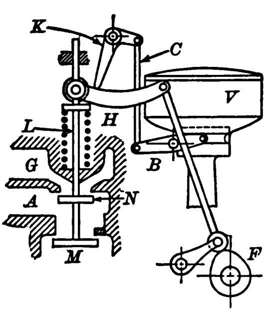


FIG. 437.—QUANTITY GOVERNING WITH FLOATING LEVER

728. A quantity governing valve gear which would admit the mixture at a full rate of flow, but which would cut off the admission after the piston had moved in a fraction of its suction stroke instead of the full stroke, is shown in end and side views in Fig. 438. This is called the quantity cut-off gear. The gas and air enter at G and A and pass into the engine cylinder through the main valve M, which has a constant lift each time it is opened. The period of opening, however, is determined by the governor. In the position shown the governor acts through the governor rod B and holds the cam drum F

in its extreme right-hand position. The cam projection E_2 on the drum subtends an angle w_2, and as this projection passes under the roller D the valve M is lifted. Since the angle w_2 is about 60° in the position shown and since the cam shaft Q rotates at one-half the speed of the engine shaft, the valve M will be lifted while the engine crank is traveling 120°, or about two-thirds of the stroke. The position shown corresponds to a light load. For full load the governor would cause the cam drum F to slide to the left along the lay shaft Q, to which it is connected by a feather or sliding key, and would bring the projection E under the roller D. This projection takes up 90°, and, therefore, there will be admission to the cylinder for the full 180° crank turn, or for full suction stroke. The lift of the valve is the

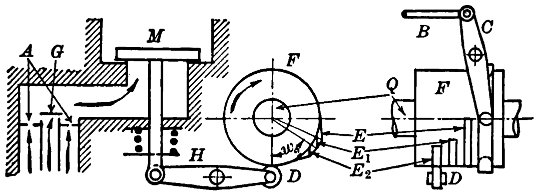


FIG. 438.—QUANTITY GOVERNING, CUT-OFF METHOD

same in each case because the cams E_2, E_1, and E are all of the same height.

729. Quantity valve gears with variable cut-off are rarely met with, and in cases where they have been used the gear has been such as to change the time of the opening of the valve as well as the closing. This would be illustrated by considering that all the cam projections E, E_1, and E_2 in Fig. 438 were the same size, and each one advanced a small amount ahead of the preceding one.

730. Quantity valve gears which admit the fuel mixture for the full suction stroke and then hold the admission valve open so that part of the mixture may be driven back on the compression stroke before the valve closes may be illustrated also in Fig. 438 by considering that the cam projection E, which includes 90°, is the smallest of several cam projections on the cam drum, and that the others include angles greater than 90°. This would mean that as these other cam projections were placed under the roller D by the governor the valve M would remain open while the crank traveled through more than 180°, or while the piston moved down on its suction stroke and back part of its distance on the return stroke.

The Quality Method

731. The quality method of governing is generally accomplished by changing the amount of gas that goes to make up the fuel mixture. When the amount of the gas in the mixture is small it is said to be a "lean" mixture, and its ignition becomes uncertain, and when large it is termed a "rich" mixture. Whether lean or rich, it is characteristic of the quality method that the intake valve should have full lift or opening, that the cylinder should be filled with the fuel mixture each stroke, and that full compression should be attained at the end of each compression stroke.

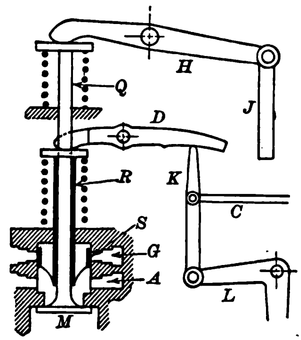


FIG. 439.—QUALITY GOVERNING

732. A gear that will give quality governing is represented in Fig. 439, where M is an intake poppet valve, attached to the stem Q, S is a sleeve valve attached to the hollow valve stem R, A is an air admission port, and G is a gas port. The rocker H receives a constant swing from a cam and always opens the valve M the same amount. The bell-crank L also receives a constant swing from a cam, but the motion it gives to the sleeve valve S varies with the position of the upright arm K. The position of K is controlled by the governor through the rod C, and when K is moved to the left the gas port will be opened wider and a richer mixture will enter the cylinder.

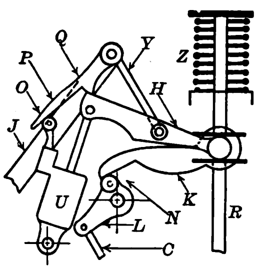


FIG. 440.—QUALITY GOVERNING WITH TRIP GEAR

733. Quality governing is frequently accomplished with trip gears, as illustrated in Fig. 440. The valve stem R operates the gas valve, not shown, which has a variable period of opening, as will be described, while the intake or mixture valve, also not shown, has a constant period of opening. The gas valve and its stem R receive motion through the lever H, which is designed to roll on the curved lever K. This curved lever is hinged at the right-hand end to the frame of the engine, and at the left end it rests on a roller N supported by an arm of the bell-crank L. The rod C comes from the governor, which thus determines the position of the roller and the elevation of the left end

of K. The rod J has a constant up-and-down motion and is guided at the top by the link Y. At the top is also pivoted a latch P, which bears on the end of the lever H. As the rod J is drawn down it carries with it the lever H until the arm of the latch is thrown over by the roller O far enough to trip at Q. The spring Z then returns the valve to its seat, the seating impact being cushioned by the dashpot U.

Quantity and Quality Methods Combined

734. A combination of the quantity and quality methods of governing is illustrated in Fig. 441. The intake valve is at M and the fuel gas valve at G. Both are operated from one cam at F. The opening of the intake valve M is regulated by the position of the arm K, which in the position shown has its greatest lift due to the fact that it is a maximum distance from the pivot of the curved arm L on which it rests. As the engine speeds up, due to lighter load, the governor V, through the set of links and levers at B, E, C, rotates K to the right, and the valve M is given a smaller motion, thus causing the mixture to be throttled. A horizontal arm N is fastened to the rod J. Between this arm and the rod O there is some clearance. With K in the position shown, N would be moved up a maximum amount and the gas valve at G would receive its full motion, thus admitting a full percentage of gas. With K rotated counter-clockwise, G would receive less motion and consequently smaller percentage of gas would be admitted. The proportions of the valves and their motions are such that this gear gives a richer gas at low loads than a quality governor alone, and a higher compression than the quantity governor alone.

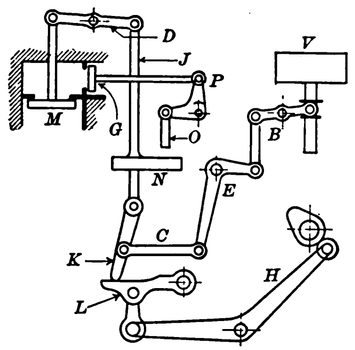


FIG. 441. — COMBINATION QUANTITY AND QUALITY VALVE GEAR

735. A special gear giving variable quantity with high compression when the lean mixtures are reached is shown in Fig. 442. The intake valve M is operated from the cam F and has constant lift and constant opening. The gas valve at G has constant lift and variable opening, and the same is true of the air valve A. The latter is operated from the stepped cam E. The gas valve G is operated directly by the air valve coming in contact with the gas-valve stem. There is a small clearance so that the air valve opens slightly ahead of the gas

valve. The roller operating against the cam E is moved on its pin in the direction of its axis by a governor. At full load it is in front of the pair of steps indicated at X X_1, and at light load it is in front of the pair of steps indicated at Z Z_1. Considering that the cams are turning counter-clockwise, the rocker K will be rotated its full distance at once and the valves A and G will be opened a maximum amount. As the cam revolves, the valve G will close when the roller moves down the first step at X, and an instant later the air valve will close when the roller moves down the second step X_1. At light load the valves will be lifted their maximum amounts at the same time as before, but the gas valve will close sooner because the step Z will come under the roller sooner. The air valve will close later, however, than it did for full load because the second step down at Z_1 comes under the roller later. This gives a lean mixture, but it will be highly compressed.

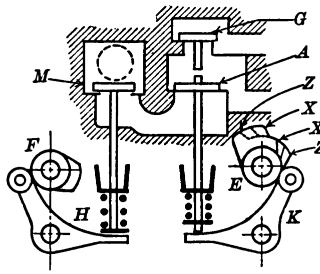


FIG. 442.—LEAN MIXTURES WITH HIGH COMPRESSION

Ignition Timing Method

736. The best time to ignite fuel mixture in an engine running under normal load at normal speed depends on the richness and compression of the mixture and the proportions of the engine. This brief statement, however, includes a wide and varied number of combinations which can not be satisfactorily arranged for in advance, and the matter of timing for best results must be left to adjustments after the engine is placed under its regular working conditions. Even then the accidental variations of fuel and mixture render the timing by governor action an unsatisfactory method of speed regulation, although it has been tried on the basis of advancing the time of ignition as the richness of the fuel mixture is decreased. Where hand regulation is practical the timing method may be used. It will be evident that if an engine is running at normal load and speed under constant conditions with the ignition at or near the top of the stroke, it will develop less power if the ignition is made to come later in the stroke when the fuel mixture is expanded and when the ignited charge will have less time to act on the piston.

Special Forms of Speed Control in Two-Stroke Gas Engines

737. Most of the illustrations of methods thus far given have been for appl:cations to four-stroke gas engines although the principles may be applied, in part at least, to the two-stroke type.

Reid Two-Stroke Gas Engine

738. An unusual type of small two-stroke gas engine, but a successful one, is manufactured by the Joseph Reid Gas Engine Company at Oil City, Pa. It operates on natural gas, admits fuel at the head end of the cylinder, and has hot-tube ignition. It is built in sizes up to 40 horse-power, and over eight thousand have been manufactured.

739. Its construction and operation are as follows: As the auxiliary or charging piston *O*, Fig. 444, moves away from the head end of its cylinder it creâtes a vacuum, automatically lifts the check valve *F*, Fig. 443, and draws in a charge through the passageway *G*, the air coming through the regulating valve *D*, and gas through the regulating valve *E*. On the return stroke of the charging piston the compression automatically closes the check valve *F* and the compressed

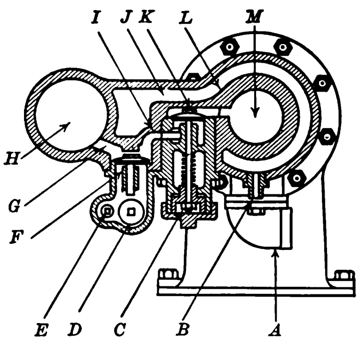


Fig. 443

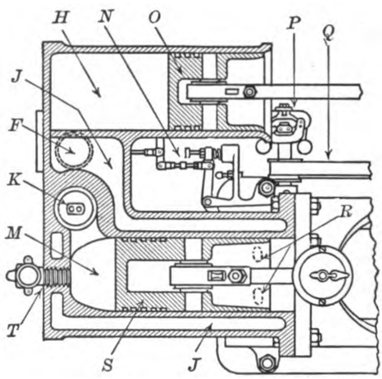


Fig. 444

Figs. 443 and 444.—Reid Two-Stroke Gas Engine, Hot-Tube Ignition, and Head-End Admission

charge passes on through the port *I*, automatically lifting another check valve *K* against a light spring pressure, until it reaches the engine cylinder *M*.

740. The fresh charge reaches the head end of the cylinder just after the piston has opened up the exhaust port on the crank end,

and thus secures an effective scavenging of the burned gas. When the charging piston reaches the head end of its stroke the check valve K is closed by the spring. Forcible seating of the check valve is prevented by means of a plunger C on the end of the valve stem which fits in a closed chamber so as to form an air cushion. This check valve is also held firmly on its seat by the compression of the fresh charge as the engine piston moves toward the head end.

741. The ignition of the charge is obtained by means of the hot tube shown in Fig. 445. The thread T screws into the cylinder head at T, Fig. 444. A portion of the compressed charge is forced through the opening U, Fig. 445, up into the long hole drilled in the nickel tube W. This tube is kept at red-hot heat through a jet coming from a Bunsen gas burner at V. The openings at Z and Z_1 furnish air for the flame which burns in the igniter chimney Y. This chimney is lined with asbestos, as indicated at X, and is open at the top.

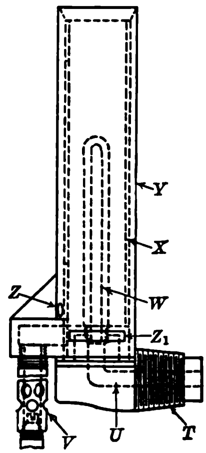


FIG. 445.—HOT-TUBE IGNITER, REID GAS ENGINE

742. The speed of the engine for light or heavy loads is regulated by a centrifugal governor P, Fig. 444, which is connected through the linkage shown in Fig. 446 to the cylindrical valves at E and D, Figs. 443 and 446. These valves move so as to reduce the port openings *2* and *8* to both air and gas when the load decreases and so to reduce the entering charge. The gear operating these valves is so adjusted that the quantity of air to gas at all loads

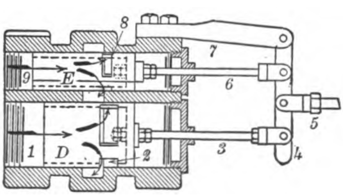


FIG. 446.—CONTROL VALVES

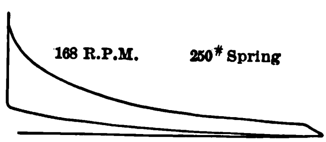


FIG. 447.—INDICATOR CARD FROM REID GAS ENGINE

is in the same proportion, thereby simply cutting down or increasing the quantity of the mixture while the quality remains the same. The rod *5* leads to the governor. An indicator card from the engine cylinder is shown in Fig. 447.

743. The exhaust pipe is shown at A, Fig. 443, the cooling-water outlet at B, the water-jacket space at J, the mechanism from the governor to the regulating valves at N, Fig. 444, and the governor belt at Q. The exhaust ports are shown at R. When the engine is used in locations where there is danger from gas in the surrounding air a safety igniter, which is in effect a hot-tube igniter such as described above but covered with a gauze, is attached in place of the regular one.

Koerting Two-Stroke Double-Acting Engine

744. The speed-control mechanism of the Koerting two-stroke double-acting gas engine is illustrated in Fig. 448. The main cylinder is at L, the admission valves at K and K_1, the exhaust ports at M, and the piston at N. The piston is not quite at the end of the stroke. Its length for a double-acting engine, it will be observed, must be twice the length of the crank minus the width of the exhaust port.

745. A pump at B supplies fuel gas through the ducts G and G_1 which lead directly to the admission valves K and K_1. A pump at R supplies air at low pressure through the ducts A and A_1, also directly

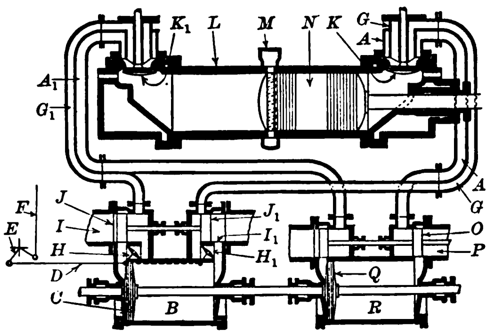


FIG. 448.—KOERTING GAS-ENGINE SPEED CONTROL

to the admission valves. Both pumps are of the same size and both are on one rod, which is operated from a crank on the engine shaft. The pump crank is set about 110° ahead of the engine crank. In some Koerting designs the pump cylinders B and R are of different diameters in order that the piston areas may be in the same ratio as the gas and air in the explosive mixtures, but in the later designs the cylinders and pistons have been made of the same diameter. The regulation of the proportions of air and gas, in this latter case, has been affected by regulating the amount of gas that the gas pump is allowed to drive into the cylinder.

746. The valves J and O are operated from an eccentric on the main shaft. Starting at the phase shown in Fig. 448, O is moved to the right, when air is forced through the duct A up to the head of the admission valve K. This valve, being still closed, the air presses on for a short distance into the gas duct G. The distance the air penetrates into the gas duct depends on how widely open the throttle valve H_1 has been during the suction stroke of the piston C. This throttle valve is under the control of the governor through the rod D, bell-crank E, and rod F. Presently the admission valve K opens and the control valve J_1 closes. For a very short period air is then driven into the engine cylinder from both the air and gas ducts A and G. This air serves to expel the burned gases and to form an air stratum against the piston head. The fuel gas supply from the pump B soon reaches the admission valve K, and it is mixed with the air from the cylinder R as it enters the cylinder to form the combustible charge.

Structural Details of Gas-Engine Valves and Valve Gears

Allis-Chalmers Intake Valve and Valve Gear

747. A structural detail of the Allis-Chalmers double-acting four-stroke gas engine is illustrated in Fig. 449. Rolling cam arms operate the valves A and C, the former being for admission of the fuel mixture to the main cylinder port Y and the latter for admission of the pure fuel gas from the chamber W to the chamber Z, where it is mixed with the pure air from X.

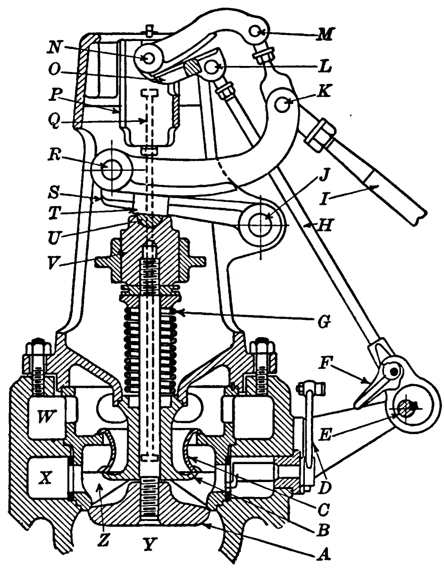


FIG. 449.—INTAKE VALVES AND VALVE GEAR, ALLIS-CHALMERS GAS ENGINE

748. The eccentric rod I receives its motion from an eccentric on a side or lay shaft, which in turn is geared to the main shaft and which runs at one-half the speed of the main shaft in four-stroke engines and at the same speed in two-stroke engines. The present illustration is for a four-stroke action, the exhaust valve being opposite the valve A. As the rod I moves up and down it oscillates the curved arm KR about

R and the arm has pure rolling or partial rolling and sliding contact on the arm $J S$ which swings about J. On the lower part of arm $J S$ is a flat-faced surface T, which bears against the split cylinder U, which acts as a rotary-sliding joint between the arm $J S$ and the valve-guide head V. The eccentric rod I has a constant stroke and transmits to the valve A a regular motion admitting the fuel mixture in uniform quantities and at regular periods. The regulation of the explosive charge is accomplished by the double-seat poppet valve C, which admits a variable quantity of fuel gas from W to Z, according to the position of the governor which controls it. The governor rocks the shaft E and the eccentric which is keyed to it, and the governor motion is thus transmitted through the eccentric rod H to the supporting arm $N O L$, which is pivoted to the engine frame at N.

749. The pivot M of the curved arm $M N$ receives its motion up and down from the lay shaft through the eccentric rod I and the link $K M$, while the point N is attached to the gas valve crosshead P. It will be noted that there are two distinct pivot actions at N, one fixed in the engine frame for the arm $N O L$ and the other fixed in the crosshead P for the arm $N M$. As M is drawn down, $M N$ rolls on $O L$, which is stationary for any one position of the governor, and raises the point N and the gas valve C. The gas valve C and the crosshead P are connected through two rods, one of which is represented by the dash lines Q. From the arrangement of the gear it will be noted that as the eccentric rod I moves down, it will cause the valve A to move down and C to move up, thus opening both valves. The gear is arranged so that the gas inlet valve opens a little later and closes a little earlier than the main inlet valve. As the engine speeds up the governor tilts the cam rest $O L$ by dropping the end L, and the lift of the gas valve C is decreased, the valve opening however being still later and the closing still earlier than the main inlet valve. Both valves have their maximum opening at the same time in all cases, and this is made to agree, in timing, as nearly as possible with the maximum velocity of the piston.

750. The exhaust valve is operated from the same lay shaft as the admission valves, but in the case of the exhaust valve the gear is simpler, inasmuch as the exhaust opening is always the same and at the same time, and, therefore, only one valve and one pair of cam arms are necessary. The handle at F is a regulator handle which turns an eccentric, against which the end of the regulator eccentric rod H rests. When F is turned 180° the rod H is dropped in its socket a distance equal to twice the eccentricity and the lower gas valve lever or cam rest $O L$ is swung so low that the upper valve lever $M N$ does

not come in contact with it at all, and consequently the gas valve *C* will not be raised, and the fuel will be cut off.

751. The arm at *D* adjusts the mixture by rotating a sleeve shown in solid between the air chamber *X* and the mixture space *Z*. This sleeve has ports similar to those in the valve cage, and as the sleeve is rotated about its vertical axis the ports may be made to register, giving full opening, or they may be decreased in area to any desired size. The arm *D* being once adjusted remains so for a given quality of fuel.

752. The cam or rolling arms, it will be noted, are designed to have a strong leverage and slow motion at the instants the valves are raised from their seats. The slow motion at opening insures quiet operation and long wear, and, in addition, it corresponds to the slow motion of the piston at this phase. The strong initial leverage, particularly in the case of the exhaust valve, insures the opening of the valve with comparatively light-weight links and arms.

Westinghouse Intake Valve and Gear

753. The Westinghouse gas engine is operated at the intake by a poppet valve and a multiported regulating sleeve, the immediate mechanism controlling both being illustrated in Fig. 450. The poppet valve at *1* is opened as follows: Rod *16* pulls down a rolling cam having a fixed pivot at *11*. This in turn presses down a follower cam pivoted at *14* and pinned at *10* to the poppet valve stem. The poppet valve is closed by the compression spring at the top of the valve stem. The regulating sleeve *2* has a set of ports *18*, which are staggered around the circumference of the sleeve, the upper sets of ports admitting air from the air chamber *17* and the lower set admitting gas from the chamber *19* as the ports pass the corresponding openings in the bushing *3*. The sleeve is moved down and up at the same times that the poppet valve is, the downward motion of the sleeve, however, coming from a pin *13* through the floating lever *15–6* to the trunnion joint *9* which connects with a tubular projection of the sleeve. For any one constant speed of running, the fulcrum point *6* of the floating lever remains stationary and the regulating sleeve moves down and up over the same range of travel each cycle, thus permitting the ports in the sleeve and bushing to open up the desired amount.

754. If the load should be thrown off, the engine would speed up, the governor would rotate the rock shaft *4* a small amount and raise the point *6* to a new position and hold it there so long as the increased speed continued. With *6* raised and *15* still having the same up-and-down motion the point *9* and the sleeve will move up and

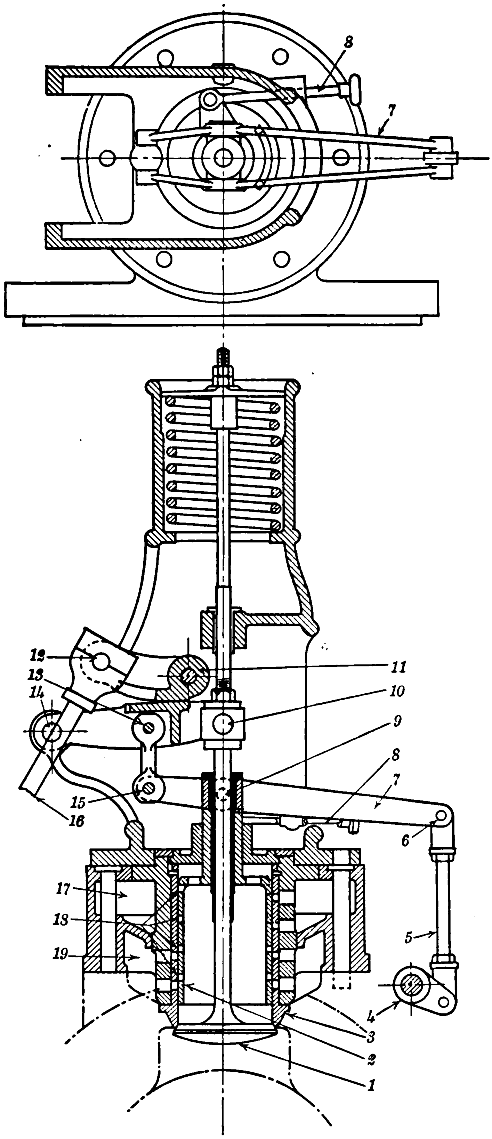


FIG. 450.—INTAKE VALVE, REGULATING SLEEVE AND GEAR, WESTINGHOUSE GAS ENGINE

down the same amount as before, but the range of motion will be slightly higher and the parts in the sleeve and bushing will not be so widely opened.

755. At maximum rated speed the regulating sleeve has zero lap and the valve gear is designed so that the ports in the sleeve and bushing will begin to open just as the poppet valve starts to open, and also so that the ports will close just as the poppet valve closes. For lighter loads the ports have greater lap, begin to open after the poppet valve opens, and they close before the poppet valve does.

756. The regulating sleeve is designed so that it may be rotated by an adjusting screw *8*, thus varying the circumferential opening of the gas and air ports so that the ratio of the mixture for different qualities and kinds of gas may be fixed without stopping the engine or disturbing the other parts of the valve gear. The ports are so laid out that as one set is decreased in area the other set is correspondingly increased, thus maintaining a constant port area for all circumferential positions of the sleeve. In all positions of the sleeve the air ports will lie immedi-

ately above the center of the gas ports for all load changes, and the air in its descent will be thoroughly mixed with the entering gas, forming a good burning mixture.

Westinghouse Water-Cooled Exhaust Valve

757. The exhaust valve of the Westinghouse gas engine illustrates the method of cooling large valves by causing water to flow through them. In Fig. 451 the cooling water enters the large hollow va ve stem at *28*, flows down to *35*, cooling the heated surface at that point, and is returned up through the tube inserted in the hollow valve stem to the outlet represented by the dotted circle at *27*.

758. The exhaust valve is operated by the rod *39* through the rolling cam, which is pivoted at *25*, and the oscillating follower cam *26*, which is pivoted at a point just beyond *38*. The follower cam has a semi-cylindrical bearing piece *21* seated in it so as to allow the bearing surfaces of contact between *21* and *20*, the capped end of the valve stem, to be always horizontal. This cylindrical piece will oscillate slightly on its cylindrical surface, and it will also slide a little on its plane surface against the cap *20*. It should be noted that the rod *39* is always in tension, as is also the similar rod *16* of the intake valve gear, and this is always a desirable feature in designing, as it may often contribute to simpler constructions, and it will always give smoother action, especially after long periods of wear.

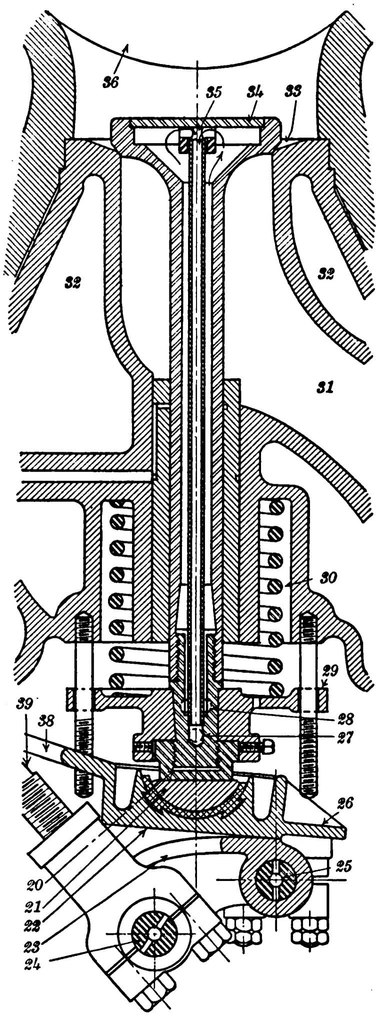


Fig. 451.—Water-Cooled Exhaust Valve, Westinghouse Gas Engine

SECTION XIII.—OIL ENGINES

769. Engines using kerosene, fuel oil, crude oil, or any of the distillates of petroleum are known as oil engines. Owing to the original difficulty of vaporizing these heavier oils, the establishment and development of the oil engine came later than that of the engines using the lighter liquids such as naphtha and gasoline. In oil engines the fuel oil must be broken up into an oil mist or vapor by passing through a nozzle or vaporizer of some specially designed form, after which it is mixed with air in proper proportions to form a combustible mixture.

Oil Engines Grouped

770. Oil engines may be grouped as follows:

1. Those using a high-grade fuel oil such as kerosene and operating with a special heated carburetor and electric spark. These engines are, in effect, modified forms of the gasoline engine and take the full fuel mixture into the cylinder and compress it.

2. Those in which the fuel or crude oil is sprayed into a separate red-hot vaporizing chamber during the time that the pure air is being compressed in the cylinder and in the vaporizer. These engines may be termed low air-compression oil engines, the compression ranging from a little less than 300 pounds per square inch to much lower pressures.

3. Those in which the fuel oil passes through an atomizer directly into the engine cylinder at about the time that the pure air is compressed in the cylinder. The compression in these engines ranges about 500 pounds per square inch and the heat due to compression alone ignites the oil-vapor mixture. These engines are known as Diesel oil engines.

Meitz & Weiss Oil Engine

771. One of the earliest American oil engines and one of the best known of the oil engines at the present time is the Meitz & Weiss engine. It was first built in 1897 and for many years was known as a kerosene engine, but its design at the present time enables it to operate on kerosene, distillate, fuel oil, or crude oil and alcohol.

772. Their horizontal type is shown in Fig. 462. It is a two-stroke engine and its method of operation is as follows: Air is drawn

through the openings $C_1 C_2$ in the engine foundation through the port D into the crank case, where it is compressed as the piston H moves in toward the crank on its forward stroke. This compressed air then passes through the cored passageway R to the port N. Just to one side of N steam is introduced from the steam dome O, as will be explained presently, and these two elements enter the cylinder when the piston uncovers the ports at N, and the first blast serves to scavenge the previous charge through the exhaust port G, which opens a trifle earlier. A short time after the exhaust port closes, a charge of the liquid fuel is injected at M, directly against the baffle plate I,

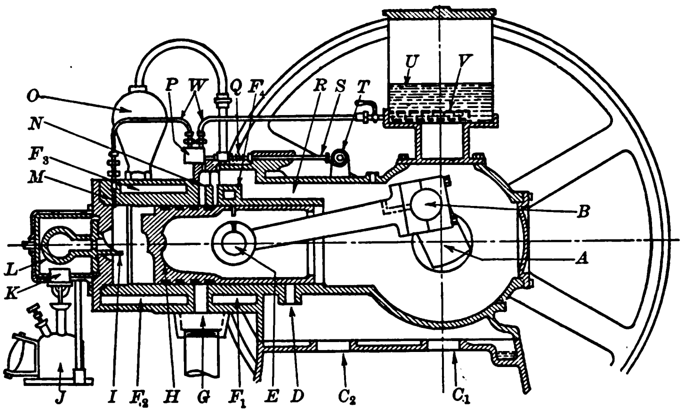


FIG. 462.—MEITZ & WEISS OIL ENGINE

some of the mixture entering the red-hot ignition bulb L. Upon completion of the return stroke the charge is compressed, and this compression, together with the heat of the ignition bulb, serves automatically to fire the mixture without the use of any electric spark or flame or other device.

773. When starting, the ignition bulb is heated by an auxiliary kerosene blue burner lamp shown at J, and, after starting, the bulb remains heated as a result of the continuous firing of the fuel charges, and the starting lamp is extinguished.

774. The admission of steam into the air and oil mixture is effected by drawing off into the steam dome O the steam that is generated in the water jacket $F_1 F_2 F_3 F_4$, and injecting it at N with the air supply. The steam, by its partial dissociation, liberates oxygen, which aids in a more complete combustion of the fuel. The water jacket in this engine acts as a boiler, the water in the jacket being maintained at a uniform level by a specially constructed auto-

matic supply tank attached to the engine. The lower section $F_1 F_2$ of the water jacket is filled with water, while the upper section $F_3 F_4$ is filled with steam.

775. The regulation of the engine speed is effected by changing the quantity of kerosene or other oil that may be used, by automatically changing the length of stroke of the plunger in the fuel pump *P*, which takes the liquid fuel from the tank *U* through the filter *V* and delivers it at *M*. The pump plunger is connected to the rod *S*, which is pushed forward by the rocker arm *T*, and backward by the spring *Q*. The rod *S* does not quite reach to the rocker arm, there being a small gap, as shown, for purposes of adjustment. The rocker arm *T* connects with the governor, which causes it to swing through a larger or smaller arc, according to the speed of the engine. The rocker arm *T* is also shown in Fig. 463, illustrating the

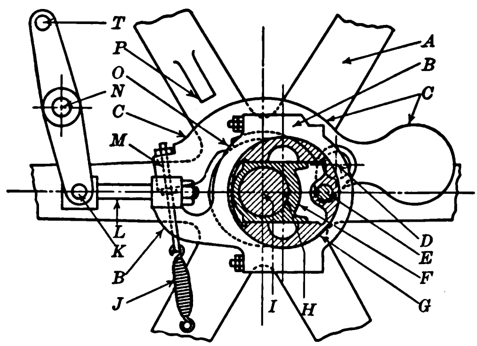


FIG. 463.—MEITZ & WEISS GOVERNOR

governor mechanism. The governor weight *C* is pivoted to the flywheel at *D*. This weight has attached to it an arm *D E*. It has also a spot-faced surface at *O*, which acts as a stop against the wheel hub, and a faced bearing surface at *M* for the nut which holds the governor-spring rod. The eccentric strap *B* carries the link *L*, which operates the rocker arm *K N T*. The eccentric sheave *G* slides back and forth on the slide block *F*, which in turn is keyed to the main engine shaft *I*. The governor is shown in position for the engine at rest, and the eccentric radius is a maximum and is equal to *I H*. As the engine speeds up, the governor weight rotates about *D* and moves the pin *E* and the eccentric sheave across the slide, and the distance *H I* is decreased, also the swing of the rocker-arm pin *K* is decreased. The lug *P* on the flywheel arm acts as an outer stop to the governor weight *C*, although the mechanism is so proportioned and adjusted that the eccentric motion becomes so small before *C* reaches *P* that the end T of the rocker arm does not swing full across the gap between T and S, Fig. 462, and consequently the pump plunger is not moved and no fuel oil is fed to the engine.

776. One of the operating mechanisms for the Meitz & Weiss oil engine is illustrated in Fig. 464. The governor shaft K, shown also in a detail side view at K_1, has pinned to it, by means of the pin L, a section of a spherical body J, which takes the place of an ordinary eccentric sheave. The spherical surface of this body is surrounded by an eccentric strap W, which is connected by a short rod to the push-ball end I. The spherical body has attached to it two arms, Y Y, which rotate with it about the shaft K_1. These arms are shown in their outer position with the governor springs U U in full tension, and the center of the eccentric sheave corresponds with the center of the shaft at T, and, therefore, the ball-end I has no motion. In the normal position the arms Y Y will be in toward the shaft, and the center T of the eccentric sheave will be revolved down about L_1 to, say, T_2. The ball-end at I will then be moved up and down a distance equal to 2 T T_2, and this motion will be transmitted to the fuel oil pump rod H and to the plunger S. If the engine speed now increases, the distance T T_2 is reduced and the plunger S will pump less oil. The fuel oil is drawn from the supply tank through the pipe A and is delivered to the engine through the pipe at D. An air cock is shown at C, its object being to relieve the supply-pipe system of air, which is essential in securing a uniform oil supply.

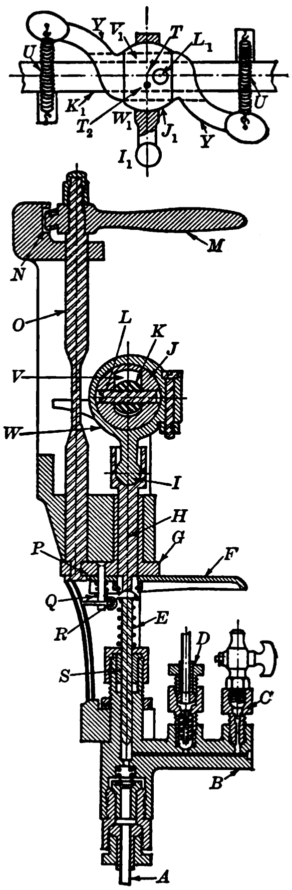


FIG. 464.—SPEED REGULATOR, MEITZ & WEISS OIL ENGINE

777. A starting handle is shown at F. This handle fulcrums on the edge shown at P, and as it is pressed down it bears on the plunger head R. The plunger is returned by the spring E when the pressure on the handle F is released. In this way as much oil as desired may be obtained when the engine is being started. The spring at Q is

merely to hold the handle in place against the end of the plunger-guide sleeve *G*.

778. A means of regulating the engine speed for varying conditions is provided at the handle *M*, which has a slightly inclined sector at *N*. This sector works in a narrow groove in the engine framework, and as the handle is turned it is lowered, and so also are the vertical rod *O*, the plunger-guide sleeve and sleeve arm at *G*, the handle *F*, and the plunger *S*. This allows the plunger guide *H* to drop, and, in the present illustration, the push-ball *I* would not come into contact with *H* at all. If the eccentric sheave at *J* is moved down, *I* will move up and down each cycle, but a part only of its motion will be imparted to the plunger and so less fuel oil will be pumped. In this way the stroke of the plunger *S* may be changed by the handle *M*, while the eccentric sheave maintains the same eccentricity.

De La Vergne Oil Engine

779. One of the earliest builders of gas and oil engines in this country was the De La Vergne Machine Company of New York

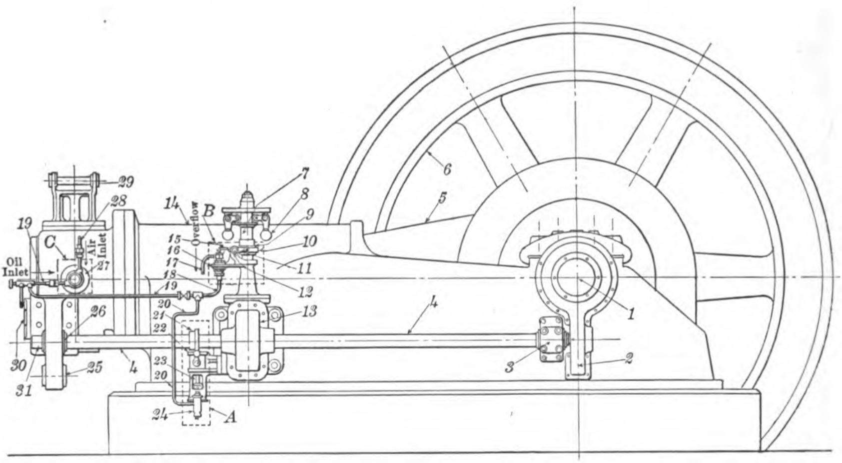


FIG. 465.—DE LA VERGNE OIL ENGINE, TYPE F. H.

City. They manufactured the Hornsby-Akroyd oil, and the Koerting gas engines under European patents, but modified for American conditions, and many of these engines are in successful operation. They have lately given up the building of the Koerting gas engine and are putting out the De La Vergne oil engine, which was designed entirely in their own works. The action of this engine is explained in the following paragraphs:

A general view of the De La Vergne oil engine, "F. H. Type," is shown in Fig. 465. It is a four-stroke engine and operates on cheap fuel or crude oils. When the fuel oil is too thick to run freely a special heating device utilizing the waste heat of the "cooling" water is installed to warm the heavy oil.

780. In Fig. 465 the main engine shaft is at *1*, the flywheel at *6*, and the engine cylinder at *14*. The engine piston with a conical head is shown at *45* in Fig. 467. In Fig. 465 three important groups of the valve-gear mechanism are blocked out in dash lines and indicated by the letters *A*, *B*, and *C*, and these are illustrated in detail in Figs. 468, 470, and 471 respectively.

781. The remaining parts indicated in Fig. 465 are: the worm gear casing at *2*, the lay-shaft bearing at *3*, and the lay shaft *4*, which

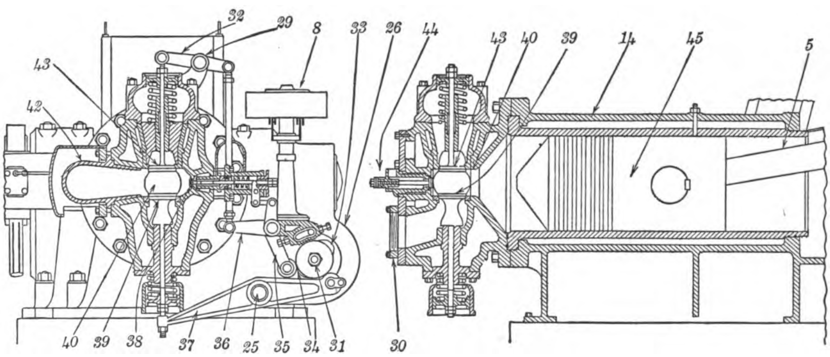


FIG. 466 FIG. 467

FIGS. 466 AND 467.—DETAIL SECTIONS OF DE LA VERGNE OIL ENGINE

is driven at half the speed of the main shaft. As the governor weights *8* fly out they press down the sleeve *7*, which carries the governor-arm ring *10*. The mechanism and action at this point are taken up in connection with Figs. 469 and 470, and described in paragraphs 785–787. A lay-shaft bearing is indicated at *13*. The pump-plunger cam *21* is attached to the lay shaft *4* and it drives the plunger *23*, which pumps the fuel oil through the action of automatic valves in the casing at *24*. The parts here indicated from *21* to *24* are shown in detail in Fig. 468. From delivery pipe *20*, the fuel oil takes two courses, one through the pipe *18* to the overflow valve at *16*, where the amount of escaping fuel oil is regulated by the governor, and the other through the pipe *19* to the spray valve at *27*, which admits the oil to the cylinder. The details of construction at the overflow valve and at the spray valve are shown in Figs. 469, 470, and 471 respectively. A supporting bracket for the intake valve rocker is shown at *29* in Figs. 465 and 466, also a supporting bracket *26–25* for the lay shaft *4* and for the exhaust valve rocker *37*. A two-stage air compressor is used on this particular engine to supply air at about 700 pounds per square inch pressure, which is needed to inject the fuel oil into the engine. In some other types of the De La Vergne oil engine it has been proposed to do away with this air compressor and to design the plunger pump at *A*, Fig. 465, and other affected parts, so that the pump itself will develop the pressure necessary to deliver the oil to the vaporizer.

782. The action of the De La Vergne oil engine, "F. H. type," is as follows: Atmospheric air is admitted by the valve *43*, Figs. 466 and 467, on the suction stroke, and fuel oil is injected through the spraying valve at *38*, Fig. 466, with an air pressure behind it of about 700 pounds per square inch at or near the end of the compression stroke. The fuel oil comes out of the spraying nozzle *38* as a fine mist, the heavier particles of the oil being projected to the far end of the red-hot vaporizer *42*, where they are ignited. Upon starting the engine, this vaporizer must be brought to a red heat by a blast lamp which is discontinued after a few minutes of running. Then the red heat of the vaporizer is maintained by the heat of combustion due to regular running. The necessary heat for the ignition of the fuel mixtures of oil-mist and air is given by a combination of the red heat of the vaporizer and the heat due to the compression of the compressed air, which compression is about 280 pounds per square inch in this engine. A storage tank of compressed air at 100 pounds pressure is provided for starting the engine, and this starting air is admitted by the valve shown at *44*. Fuel injection, under regular running, takes place from 8° to 12° before head-end dead center to 18° to 22° after; the exhaust valve *39* opens about 40° ahead of crank-end dead center and closes about 40° after head-end dead center; and the air valve opens 35° before head-end dead center and closes 25° after crank-end dead center.

783. The mechanism for pumping the fuel oil is illustrated in detail in Fig. 468. The cam *21* on the lay shaft *4* gives a constant stroke to the plunger *23*, which, on its upstroke, draws in the fuel oil through the pipe indicated at *56*. This oil passes by the check valve *55*, which is raised automatically, and on the down stroke of the plunger it is forced up past a second automatic check valve at *54* and into the delivery pipe indicated at *20*. Both check valves are assisted in closing by light compression springs. The valve chamber of the plunger pump is closed at the bottom by a pipe plug *57* and at the top by a safety valve *53*, which works against a heavy compres-

sion spring in case the delivery pipe line becomes clogged. This spring is adjusted to 2,000 pounds.

784. Although the stroke of the plunger is constant for any one operating condition, it may be varied at will by the adjusting screw in the bell crank *49*. The end of the screw *51* bears against the pump housing, and if it is screwed in, the bell crank *49* will rotate slightly counter-clockwise about *48* as a center and press down the cylindrical head *47* of the plunger. This head carries the roller follower *46* further away from the lay-shaft center and gives a certain amount of play between the roller and the inner radius of the cam. Consequently the cam will not strike the roller until later, and the plunger stroke will be decreased. In some designs the play mentioned above is taken up in the interior of the plunger head, so that the striking action will come on suitable fiber surfaces and the roller thus be left free to be in continuous contact with the cam. This form of construction is shown in Fig. 473. A hand lever is provided at *50*, so that the plunger may be operated by hand so as to supply more oil at starting than the cam is designed to give. A heavy compression spring at *52* serves to drive the plunger *23* through its return or suction stroke.

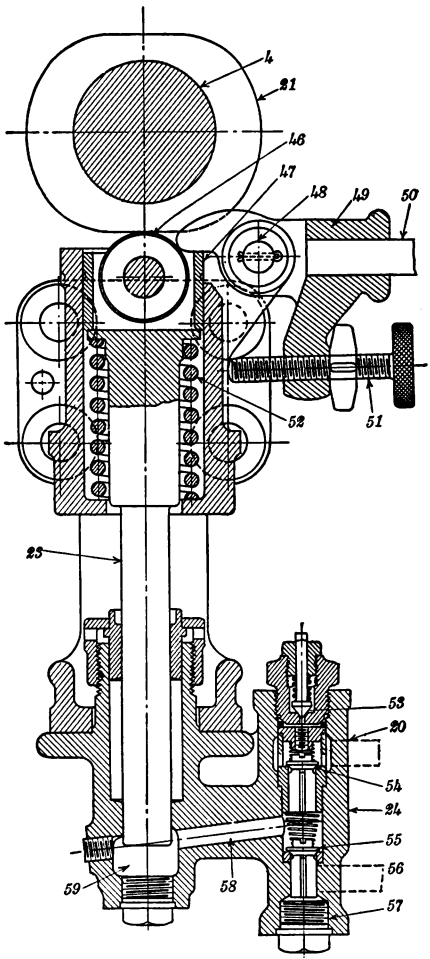


FIG. 468.—DE LA VERGNE OIL PUMP

785. The mechanism regulating the amount of fuel oil that is permitted to go to the engine is shown in Figs. 469 and 470. Fig. 469 is a section on the center line X X of Fig. 470. After the fuel

oil leaves the plunger pump at A, Fig. 465, it passes through the pipe *20*, and a portion of it continues on through the pipe *19* to the engine while the other portion goes through the pipe *18* and overflows from the pipe *17*, from which it is returned to the fuel-oil supply tank to be used over again. The amount that overflows is regulated by the governor, as explained in the following paragraph.

786. If the load is reduced, for example, the governor weights fly out and press down the governor ring *10*, Fig. 470, and also the right-hand end of the rocker lever *11*. This lifts the end *71* of the second rocker *15*, which has a fixed pivot at *70*, and depresses the end *69*. At this point is pivoted one end of the floating lever *68*, the

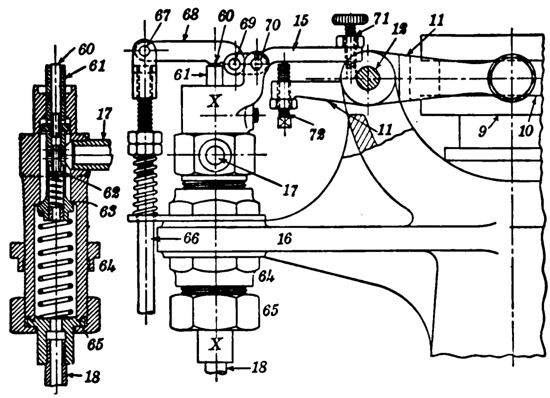


FIG. 469 FIG. 470

FIGS. 469 AND 470.—SHOWING CONTROL OF FUEL OIL OVERFLOW

other end of which, *67*, has a slight but constant vibratory motion, which comes through the rod *66* from a cam on the lay shaft. The floating lever presses, at *60*, on a central valve stem attached to the disk valve *62*, which is kept in a constant vibratory motion by the mechanism just referred to. This vibratory motion is designed to keep the small annular passageway around the disk valve from gumming and sticking with the thick oils and any foreign matter. It also assists the governor in quickly responding to fluctuations in speed. Repeating the assumption made at the beginning of the paragraph, it may now be noted that if the load decreases and the pivot *69* is lowered, the floating lever *68* will open the disk valve *62* further, and more fuel oil will be allowed to pass and to overflow through the discharge pipe *17*.

787. If there should be a sudden or large decrease in load the point *69* would be dropped so low that the seat *60* of the floating

lever would not only press down the central valve stem *60*, but it would also press down the annular stem *61* and open the larger disk valve *63*, to which it is attached. Thus a very free passage would be provided for the pumped fuel oil and only a very little would flow toward the engine. The distance between the top of the stem *60* and the top of *61* is only 1/32 inch. A further adjustment for taking care of quick action by the governor is found in the screw *72*, which, it will be observed, will act more rapidly on the rocker *15* than will the contact through the screw *71* if it is adjusted to do so. For regular running, however, the screw *71* is the active one, and *72* is set to come into play only as required by any unusual conditions. In some of the engines of this type, the builders omit the floating

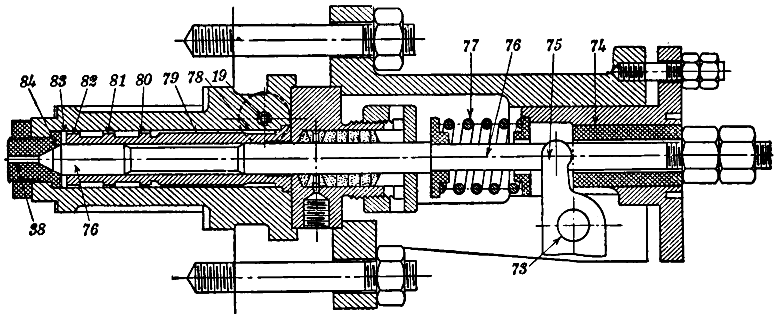


FIG. 471.—FUEL-OIL SPRAY VALVE

lever *68* and the vibratory action and allow the end *69* of the rocker *15* to press directly on the valve stem *60*. This, of course, holds the disk valve *62* a fixed distance from its seat for any one speed of running.

788. The spray valve and the method of injecting the fuel oil into the cylinder by means of air compressed to about 700 pounds per square inch is shown in detail in Fig. 471. This figure, it will be remembered, is a horizontal section of that part of Fig. 465 which is blocked out at *C*. At the latter place are shown the injecting-air pipe *28* and the fuel-oil pipe *19*. This latter pipe is indicated at *19* in Fig. 471, and the air pipe is at right angles to it and in the same plane. Both empty into the small annular space *78*. This space communicates through a series of open annular spaces and fine broken channels as at *79*, *80*, *81*, and *82* to the space *83* at the final cone seat *84* of the valve. This valve is lifted a very small amount from its seat at the end of each compression stroke through the double lever *75*, which is pivoted at *73* and which is actuated through suitable links and levers from the cam shown at *34* in Fig. 466.

789. In the type of engine in which fuel oil is forced into the engine vaporizer directly from the plunger pump without the aid of compressed air, as referred to in paragraph 781, a much simpler form of spraying nozzle is used. Such a nozzle is shown in Fig. 472, where A is the wall of the engine cylinder. Fuel oil from the plunger pump

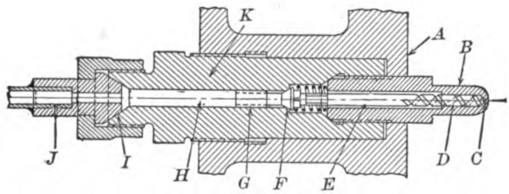


FIG. 472.—A SIMPLE FORM OF OIL-SPRAYING NOZZLE

enters at J, passes along the opening H, through longitudinal slots in the cylindrical guide G for the automatic check valve F, around the spray-nozzle pin E, through the spiral grooves D in the pin, and finally through the very fine orifice or nozzle at C which is drilled in the nozzle piece B.

790. Some of the earlier designs of the De La Vergne oil engine embodied a mechanical means of varying the supply of fuel oil to the engine without making use of the overflow method described in paragraphs 785–7. Briefly this means consisted in giving a variable stroke to the pump plunger by means of a shifting wedge which was under the control of the governor. This interesting device is illustrated diagrammatically in Fig. 473. A is the lay shaft of Fig. 465 and $B\,D\,C$ the plunger cam. With the cam turning as shown by the arrow, the follower roller and the cylindrical head S to which it is attached will be driven in the distance $V\,Z$ before the flat surface T of the head will traverse the clearance space $T\,U$ and come in contact with the fiber layer surface U of the plunger head. The cam, continuing to turn, will then drive the plunger the distance $Z\,D$, and this will be proportional to the volume of oil taken into the pumping chamber at M and delivered through the automatic check valve at N to the vaporizer of the engine. If now the load on the engine is decreased, the governor

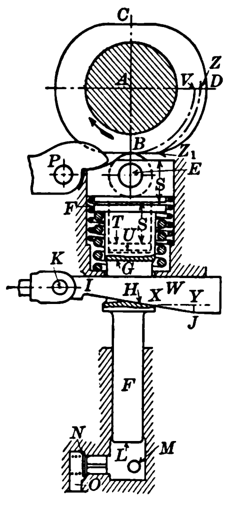


FIG. 473. — VARIABLE STROKE PLUNGER PUMP

will move the wedge W to the left, and this will increase the clearance space at T U and cause the cam-follower head S to strike the fiber surface U later, and thus move the plunger through a still smaller distance. This will, of course, offer less displacement for the oil that is taken in and will deliver less oil to the engine vaporizer. In some designs the plunger F and the roller-follower head S are all in one piece, but this permits the cam to take the blow at Z_1, where it first comes into contact with the roller E B.

791. A cam diagram which is useful in practical work is shown in Fig. 474. This diagram shows clearly the overlapping action of

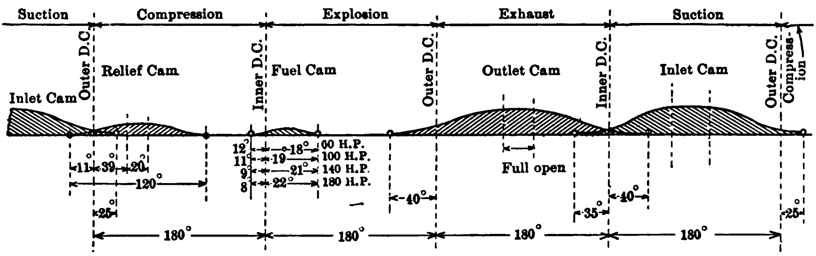


FIG. 474.—CAM DIAGRAM FOR AN OIL ENGINE

the several cams. It also shows that the inlet and exhaust cams remain constant for varying sizes of the engine, but that the angle of action of the fuel cam changes.

Bolinder Oil Engine

792. A reversing marine oil engine, sometimes referred to erroneously as a semi-Diesel engine, was invented in Sweden before the advent of the Diesel engine. The Bolinder engine is of the two-stroke type and works on low air compression and uses a hot-bulb ignition. It is quite similar in construction to the two-stroke gasoline engine, the principal differences in operation being (1) the compression of pure air only in the crank case; (2) the injection of pure liquid fuel oil near the end of the compression stroke, and (3) the hot-bulb ignition. This engine has thus far been built in sizes giving from 5 to 640 horse-power and is in use on a large number of vessels in foreign countries. It is now being introduced in this country.

793. The Bolinder engine cylinder is illustrated in section in Fig. 475. Pure air is admitted through the automatic air valves at *4* and *5*, into the crank case on the up stroke of the piston *7*. When the piston is on the next down stroke this air is compressed to about 10 pounds per square inch. Near the end of the down stroke the

piston uncovers the port *9* and the crank-case compressed air enters the engine cylinder. It is then compressed to about 125 pounds per square inch on the up stroke of the piston. As the piston nears the top of its stroke fuel oil is injected through the nozzle *11*, by means of a fuel pump, into the hot bulb *12*, where it is mixed with the air and ignited. When the piston is near the end of the explosion or down stroke, the burned gases are exhausted through the port *8*. This port is opened a trifle before the air-intake port is opened, thus allowing the pressure of the exhaust gases to be reduced so as not to blow back into the crank case. The automatic air-intake valves at *4* and *5* in the crank case are simply thin brass plate spring disks as indicated at *5* in the detail sketch at *A* in Fig. 475. On the up stroke of the piston these disks curl up against a saucer-shaped guard as indicated at *13* and allow air to enter at the openings shown in the crank-case wall *14*. On the down stroke of the piston the compressed air in the crank case forces the valve disks flat on the crank-case walls, thus closing the ports in *14*. There are six ports to each valve, and each valve disk is split well toward the center into six segments.

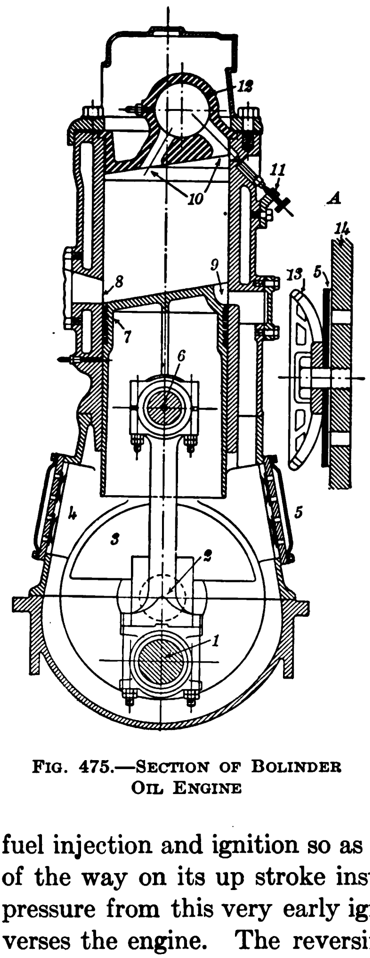


FIG. 475.—SECTION OF BOLINDER OIL ENGINE

794. The reversal of the engine is accomplished in a simple manner by changing the timing of fuel injection and ignition so as to occur when the piston is only part of the way on its up stroke instead of when it is near the top. The pressure from this very early ignition drives the piston back and reverses the engine. The reversing gear is designed so as to permit the engine to slow down considerably before the reverse charge is fired, thus reducing to a minimum the effects of sudden stopping and starting on the crank shaft and bearings. The shaft and bearings are made specially heavy.

795. The mechanical gear for accomplishing the reversal is shown in Fig. 476, where the gear is in position for running, either forward or astern. When running either way the eccentric shown by dash lines *16*, keyed on the engine shaft *1*, gives an oscillating motion to the bell-crank *24*, which turns on the fixed center *25*. A free arm *26*, which is attached to the end of the bell-crank, is pressed down by a spring *27*, so that a guide block on it slides on a smooth guide surface, which is part of a second bell-crank or "angle piece" *21*. This angle piece is held in the position shown when the engine is under way, either ahead or astern. Consequently the tappet *28* at the end of the arm *26* presses the stem *29* of the fuel-pump plunger and feeds the fuel oil to the engine cylinder through the discharge pipe *31*, always just as the piston nears the top of its stroke.

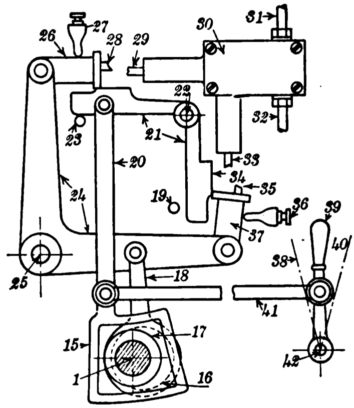


Fig. 476.—Reversing Gear of Bolinder Oil Engine

796. To change from ahead to astern, the lever *39* is thrown to *40*, thus drawing over the rod *20* and its rigidly attached box link *15*, so that the inner surface of the left side of the box link presses against the periphery of the disk *17*, which is keyed to the engine shaft. The friction of the revolving disk against the surface of the link lifts the link and its arm *20*. This swings the horizontal arm of the angle piece *21* up from its stop pin *23* and lifts the arm *26* and tappet *28* so that the tappet no longer engages with the stem *29* of the fuel-pump plunger. At the same time, the vertical arm of the angle piece *21* moves to the left against the stop pin *19*, thus permitting the spring at *36* to push the arm *37* and tappet *35*, so that the latter now engages with the stem of another plunger, *33*, in the fuel pump, so that fuel oil is injected into the cylinder when the piston is only part way up on its air-compression stroke. By this time the engine has slowed down somewhat. Ignition then takes place and the piston is forced back and the engine reversed. To change from reverse to ahead, the lever is moved to *38*, and when reversal is effected the lever is moved back to *39*, so that the disk *17* may run free from the box link *15*.

797. When starting the engine, it is necessary to heat the bulb with a blow lamp and to inject the fuel oil by means of a hand pump. On smaller sizes, the engine starts by giving the flywheel a quick

half turn in the opposite direction from which it is to run. In larger sizes, above 30 horse-power, and in all two- or four-cylinder engines compressed air is used to turn the engine. The pressure required for starting is obtained from the exhaust gases of the engine itself while-running and is stored in a receiver cylinder. The speed of the engine is regulated by a hand lever controlling the fuel-oil admission. When running at half speed or slower, the air supply from the crank chamber is reduced by means of a damper in the passageway leading to the cylinder. This reduced air supply permits the cylinder to remain sufficiently heated to enable the engine to run without relighting the blow lamp. If the engine is to be run above its normal power, fresh water must be injected into the engine cylinder to prevent overheating and loss of power.

SECTION XIV.—DIESEL OIL ENGINES

806. The Diesel oil engine was patented by Rudolf Diesel, of Munich, Germany, in 1892, when he first began experimenting with small-sized engines. The application of the principles used in the operation of the engine and the form of construction were so radical and its economy was so low that it attracted and compelled world-wide interest. Simultaneous attempts were made in many countries to adapt it to larger powers and to simplify its construction, until at the present time it is built in large sizes and is widely used.

807. The Diesel engine is constructed on both the four-stroke and two-stroke principles. The former type, with its ignition every second revolution, offers an inherent advantage in permitting of longer cooling periods for the dissipation of the intense heat peculiar to this type of engine, and which acts on the piston head and cylinder walls. Notwithstanding this, many large two-stroke engines, with circulating cooling water in the piston, are installed in marine service. Taking both the two-stroke and four-stroke types, it is estimated that about 500 vessels are equipped at the present time with Diesel engines. Diesel engines aggregating many thousand horse-power, mostly of the four-stroke type, are installed for land or stationary service.

808. The Diesel engine is sometimes referred to as a slow-combustion engine in that the ignition of the fuel takes place as it enters the cylinder, and at approximately constant pressure for an appreciable period of time. This is in contradistinction to the gasoline and gas engines, which are explosive in their ignition characteristics.

809. The Diesel oil engine operates by—

(*a*) Admitting into the cylinder a charge of pure air at or near atmospheric pressure and temperature.

(*b*) Compressing the air in the cylinder to between 450 and 500 pounds per square inch. The simple act of compressing this pure air raises its temperature to between 1,000° F. and 1,100° F.

(*c*) Injecting a small quantity of crude oil by means of air which has previously been compressed to about 800 pounds per square inch and cooled. This mixture of the air and oil forms a fine spray in passing into the cylinder through the specially formed admission valve, and, upon meeting with the hot compressed air in the cylinder, burns at approximately uniform pressure, forming a gas which expands and which is released at about 35 pounds pressure per square inch at about 700° F. for normal conditions.

(*d*) Exhausting the burned gases. when pure air is again admitted and the process repeated.

810. The Diesel engine differs fundamentally from gasoline and gas engines in that these compress the fuel mixture instead of pure air, and the temperatures of ignition of the gas-and-air mixture always limit the compression pressure in the gasoline cylinder to much lower values than are attained in the Diesel engine or even in ordinary oil engines. Most of the ordinary oil engines compress only the air, fuel oil being injected at or near the end of the stroke. The compression is much lower, however, than in the Diesel engine, and a special hot-metal surface must be relied on to ignite the charge.

811. The Diesel engine employs the poppet type of valve and the cam form of valve gear. The Diesel engine is regulated by the amount of fuel oil that is supplied to the spraying nozzle or fuel admission valve, and this amount is under the control of the governor.

Diagrammatic Arrangement of Diesel Engine with Air-Pump, Water-Cooling, and Fuel Connections

812. A Diesel oil engine with its various essential connections is shown in diagrammatic form in Fig. 486. The diagram here shown and the text in condensed form are reproduced from an article prepared by Professor W. E. Dalby for the Institution of Naval Engineers, London, as reported in *The Engineer*, London, April 24, 1914. The engine cylinder is shown at *2*, the piston at *3*, and the main shaft at *1*. The fuel-oil valve is shown at *5* and the fuel-oil pump, which supplies oil to the valve, and the fuel oil supply tanks are indicated in the diagram. The fuel oil reaches the small opening surrounding the fuel valve as pure oil without any entrained air or foreign matter, and it is there met by cooled compressed air at 715 pounds per square inch, in this case. This pressure produces a blast the instant the fuel valve is lifted by the valve gear, and carries with it a definite amount of oil into the engine cylinder. The amount is regulated by a governor-operated plunger pump shown in Fig. 488.

813. The fuel valve is a steel spindle with extremely fine winding and inclined channels milled in the head, as indicated in Fig. 487, and as the air blast drives in the oil it converts it into a fine mist. The energy of so converting the fuel is shown by a reduction in pressure from 715 pounds per square inch for the injection air to 529

pounds at the highest point of compression as shown by the indicator card, Fig. 489. The opening at *5*, Fig. 486, through which the fuel spray enters the cylinder, is about one-twentieth of an inch in diameter.

814. The compressed air for injecting the fuel is supplied by the Diesel engine itself, while running, from the main shaft *1*, through a

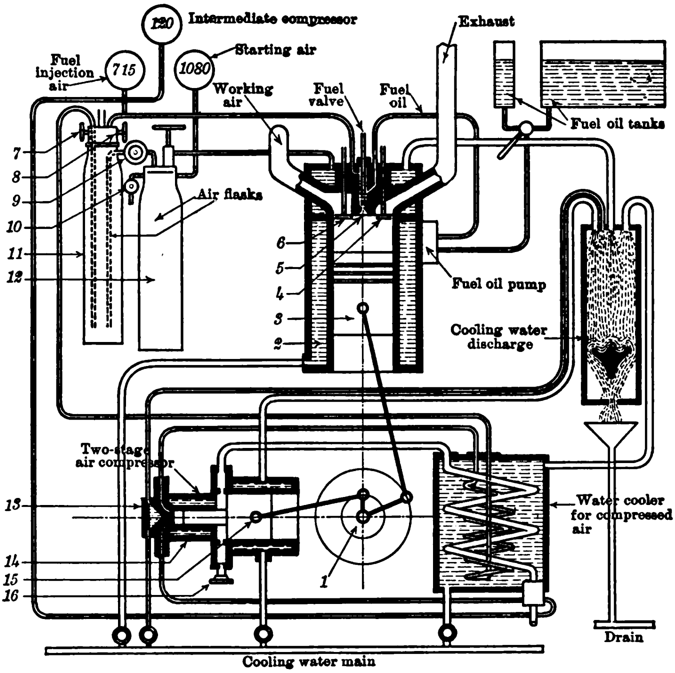


FIG. 486.—DIAGRAM OF FOUR-STROKE DIESEL OIL ENGINE SHOWING COMPRESSED-AIR AND WATER-COOLING SYSTEMS

two-stage air compressor as marked in Fig. 486. In larger engine sizes, three-stage air compressors are used. The method of cooling the compressed air between the two stages of compression and after its final compression may also be traced in the diagram. The intermediate pressure between the two stages of compression is seen to be 120 pounds per square inch. The pressure of 715 pounds for the fuel-injection air is regulated by the valve *7*. The starting-air storage flask *12* is also filled by opening the valve *9*. The working air is at atmospheric pressure, or under a pressure of a few pounds in two-stroke engines, in which case it is separately compressed in additional auxiliary compressors.

815. Attention is specially called to the fact that in Diesel oil

engines air under three different conditions is used for three different purposes: First, the "scavenging" or working air, which is taken into the cylinder at or near atmospheric pressure and is compressed in the engine cylinder to 450 to 500 pounds per square inch; second, the injection air, which forces the fuel oil into the cylinder and which is separately compressed to 700 pounds or more and cooled; third, the starting air, which is used only to start the engine, but which must be separately compressed to 800 pounds or more and stored.

816. The valves are always placed in the cylinder head, and there are at least five in each four-stroke engine—namely, the fuel valve *5*, Fig. 486, the working-air valve *6*, the exhaust valve *4*, and the starting-air valve, which is not shown, but which lies in front of the fuel valve, and a relief valve in back of the fuel valve. In larger sizes of engines two working-air valves and two exhaust valves are sometimes used in four-stroke engines, and four scavenging valves have been used in two-stroke engines. Each valve is moved in one direction, either up or down as the case may be, by having its valve stem lifted or pushed down by a rocker arm, which in turn is moved by a cam. Each valve is moved in its return direction usually by a compression spring.

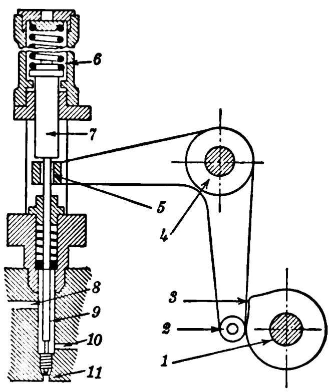


FIG. 487.—FUEL-OIL VALVE, BELL CRANK AND CAM

817. The cam, rocker arm, and valve stem for moving the fuel valve in the present case are shown in Fig. 487. The cam projection *3*, it will be noted, is very small, because the time for fuel admission is small and the lift of the valve is only 3/32 of an inch. The compression return spring is at *6*, the injection air inlet at *8*, the fuel-oil inlet at *10*, and the milled valve head and opening into the cylinder at *11*. The cam shaft *1* is driven by gears from the main shaft at one-half engine speed in a four-stroke engine. Rocker arms and cams for the remaining valves are shown in their proper order and arrangement in connection with other forms of engines described in later paragraphs.

Fuel-Oil Pump with Governor Gear

818. A method of automatically regulating the charge of fuel oil that is delivered to the fuel valve by the plunger pump is explained

in detail in the two following paragraphs. Briefly, the method consists in using a plunger with constant stroke and displacement and allowing it to draw in each time a uniform quantity through a suction valve that lifts automatically, but which is held open for longer or shorter periods by the governor mechanism, so that all or any part of the charge drawn in by the plunger may be returned through the open suction valve. That which is not so returned passes up through the automatic pump delivery valves to the fuel valve.

819. The mechanical detail of a mechanism for regulating the fuel-oil charge is shown more or less diagrammatically by section and

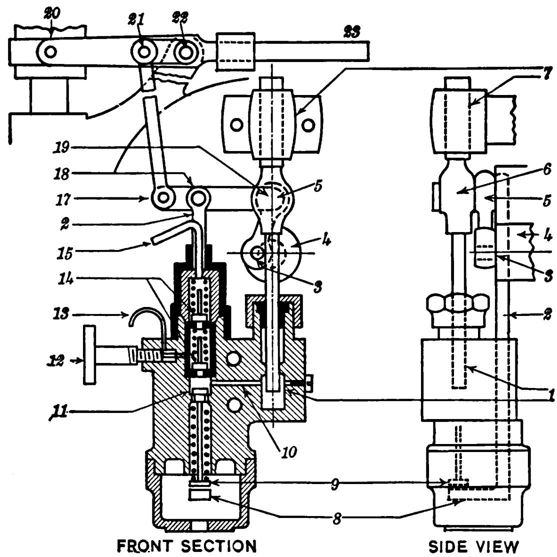


Fig. 488.—Front Section Fuel Pump and Governor Valve Gear

side views in Fig. 488. The pump plunger *1* is operated through the connecting rod *5–3* from a pin *3* in the end of the cam shaft *4*. The suction valve *11* lifts automatically as the plunger *1* lifts and closes automatically as the plunger descends, providing the arm *8* does not interfere with the end *9* of the suction valve stem. If the suction valve is permitted to close automatically as the plunger starts on its down stroke, all of the fuel-oil charge will be passed through the delivery valves *14* to the fuel-oil pipe *15*, which leads to the fuel valve *5* in Fig. 486. A valve at *12*, Fig. 488, is used to test the fuel-oil flow. When it is opened, oil will squirt out regularly at the pipe *13* if the gear is working properly.

820. The method of changing the quantity of fuel oil supplied is as follows: As the engine speed increases above normal the governor sleeve *20*, Fig. 488, moves up, rotating the governor weight arm *20–23* about a fixed pivot *22*, to which a return crank *22–21* is rigidly attached. The connecting rod *21–17* then swings the points *17* and *18* of the floating lever up about the movable fulcrum *19*, which is pivoted to the plunger crosshead and moves up and down with it. So long as the governor holds the sleeve *20* and the pivot *17* in any one position, the suspension rod *2* and its attached arm *8* will have a constant up-and-down motion, which will be to the plunger motion in the ratio of the distances *18–17* to *19–17*. The limiting planes of this motion will vary with the location of the point *17*, and if, as supposed above, the governor raises *17*, the arm *8* will rise higher and remain in contact with the suction valve-stem head *9* longer

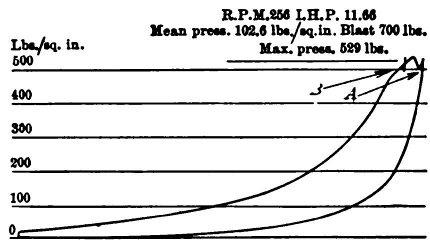


FIG. 489.—DIESEL OIL ENGINE INDICATOR CARD

and keep the suction valve open longer, thus permitting more of displaced oil to return through the suction valve and less to go through the delivery valves *14* to the engine. The method of determining the limiting motions of the several members of the valve gear mechanism is shown in further detail in a later paragraph describing another form of gear construction.

Indicator Card for Diesel Engine

821. An indicator card for a Diesel engine is illustrated in Fig. 489. The cam was set to close the fuel valve at 4.3 per cent of the stroke, which is indicated by the short vertical reference line on the card. The character of the ignition line *AB* at the top of the card should be specially noted. In theoretical cards this is frequently shown as a horizontal line, and builders endeavor to construct the valves and valve gear, and to proportion the engine and control the fuel, so that there will be ignition at constant pressure. But, in general, the actual cards obtained show the characteristics indicated

in Fig. 489. A full set of cards in the paper referred to in paragraph 812 shows that the burning of the mixture takes place at slightly rising pressure at full load, and at decreasing pressures at light loads.

Diagrammatic Arrangement of Valves, Levers, and Cams

822. A full set of valves, valve rockers, and cams for a Diesel engine of the four-stroke type is illustrated diagrammatically in Fig. 490. The engine cylinder is represented at V, the piston at P, there being very little clearance, the pure air inlet valve at A, the oil or fuel valve at O, the exhaust valve at E, the starting valve at S, and a relief or safety valve at R.

823. The Diesel engine is started by opening the valve S and admitting a charge of air that has been previously compressed, and

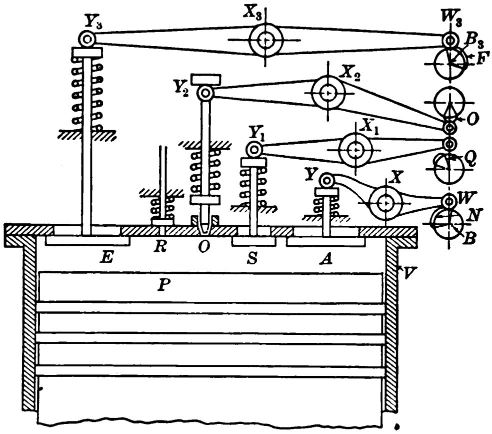


FIG. 490.—DIAGRAMMATIC ARRANGEMENT OF VALVES, ROCKERS, AND CAMS FOR DIESEL ENGINE CYLINDER

allowing the engine to run as an air engine for a few revolutions when the fuel is turned on and the starting air turned off. In order to run the Diesel engine, it is only necessary to supply a cam valve gear that will operate the four valves S, A, O, and E at their proper times.

Timing Diagram and Cam Layout for Diesel Engine

824. The first step in laying out a valve gear so that the valves will open and close at their proper times is, obviously, the drawing of a time diagram as shown in Fig. 491, where SR is the diameter of the crank-pin circle and CF, CK, etc., are the crank positions where

the several events in the cycle are desired to take place, the direction of turning being as shown by the arrow. The numerals at the extremities of the radial lines show the order of events, it being kept in mind that in this four-stroke type of construction the crank pin makes two complete revolutions while one heat cycle is being completed.

825. Since each of the valves A, O, and E, Fig. 490, can open only once in two revolutions of the engine, the cams which operate the

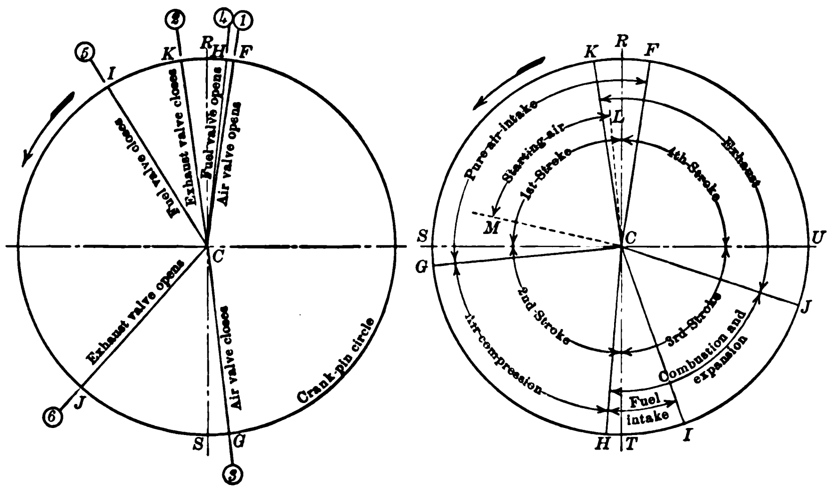


FIG. 491.—CRANK SHAFT TIMING DIAGRAM. FIG. 492.—CAM SHAFT TIMING DIAGRAM.
TIMING DIAGRAMS FOR DIESEL FOUR-STROKE ENGINE

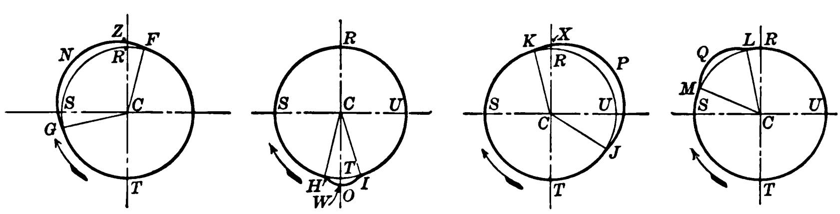


FIG. 493 FIG. 494 FIG. 495 FIG. 496
FIGS. 493–496.—CAMS WITH CAM ANGLE TAKEN FROM FIG. 492

valves will have to be fixed to a cam shaft which revolves at one-half the speed of the main shaft. This is accomplished by connecting the main and cam shafts by toothed gearing having a velocity ratio of 2 to 1. Then during each stroke of the piston, each of the cams will turn through 90°, and if another time diagram is constructed, as in Fig. 492, each of the cardinal points R, S, T, and U will represent the end of the stroke, and this new diagram will help in the cam layouts. For example, the air cam starts to open at F just before the crank reaches the top dead-center position; it remains open for a little more than one full stroke and closes at G just after the crank reaches the bottom dead-center position represented by S. Then taking the angle $F\ C\ G$ from Fig. 492, the limits of the cam projection are established at F and G in Fig. 493. The cam curve $F\ N\ G$ with its proper rise to give the desired valve motion is then drawn in.

826. The precise location of the keyways on the cam and on the cam shaft are absolutely essential to the timing of the valve and the proper running of the engine, and these should be laid out and the reduction gear wheels so assembled that when the piston is at top dead center, the line $C\ R$ on the cam will be vertical; and then the air valve will be open the small amount indicated by $R\ Z$ in Fig. 493. Similarly the fuel cam should be mounted on the cam shaft so that the line $C\ T$, Fig. 494, is vertical when the piston is again at top dead center; then the fuel valve will be open by an amount represented by $T\ W$. The exhaust cam, Fig. 495, is to be treated the same way.

827. Some of these cams, as will be explained in a later paragraph, are mounted on the same shaft at B, Fig. 490, but because of the difficulty of illustration, they are shown at $B \ldots B_3$, where the respective cams and also levers may be more clearly represented. Practically the levers lie one behind the other. It is customary in most designs to have the air and exhaust valves open inwardly while the fuel valve opens outwardly. Also in some larger sizes of engines there are two or more air admission valves, but they are all operated by multi-forked or combination levers from the same, or from duplicate cams.

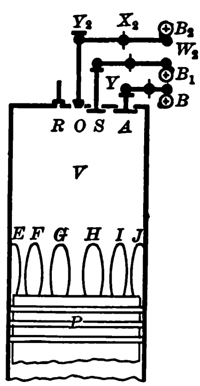


Fig. 497.—Diagram of Two-Stroke Diesel Oil Engine

Diagrammatic Arrangement of Ports, Valves, Rockers, and Cams for Two-Stroke Diesel Engine

828. In the two-stroke engine, which is shown diagrammatically in Fig. 497, there is no exhaust valve. Exhaust ports are shown instead at the bottom of the cylinder at E, F, etc. These run clear around the cylinder. Still further, the pure-air valve at A may be and is also omitted in some forms of construction, and in this event one-half the openings as at H, I, J are air ports, while the other half are the exhaust ports. No separate valves are used in connection with these ports, their opening and closing being accomplished entirely by the piston P.

829. Briefly the operation of the two-stroke engine is as follows, considering that the valve A is used for working air and that the ports E . . . J are for exhaust only. Pure air, compressed to about four pounds per square inch, is admitted by the valve A when the piston P is near the bottom of its stroke, thus driving the burned gases before it through the exhaust ports E, F, ... After the piston passes the top of the ports on the up stroke, this pure air is compressed to between 450 and 500 pounds per square inch, with a temperature of about 1,000° F. and the fuel is then injected and ignited just before the piston reaches the top of its stroke. Expansion takes place on the downward stroke until the exhaust openings E, F, . . . are again uncovered by the piston, and the heat cycle is repeated.

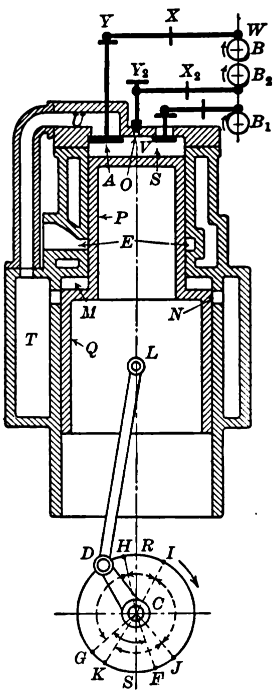


FIG. 498.—TWO-STROKE DIESEL OIL ENGINE WITH CRANK TIMING DIAGRAM

When the air valve A is omitted the cycle of events is the same, except that the air is admitted by the piston as it uncovers the openings E-G while exhaust is taking place at H-J. This method, although requiring fewer parts in construction, does not give as good scavenging as when the valve A is used.

830. The two-stroke engine requires a modification of the timing diagram shown in Fig. 492, the modification consisting in taking only the crank-pin positions R and T as dead-center positions. This new timing diagram may be shown directly on the crank circle as in Fig. 498, where the angle $H\ C\ I$ is the fuel-admission angle, $I\ C\ J$ the expansion angle, $J\ C\ K$ the exhaust angle, $F\ C\ G$ the pure-air admission angle, and $G\ C\ H$ the air-compression angle. Cams may now be constructed to operate the valve lever arms during these angles, as already explained in connection with Figs. 492 to 496, and when constructed they will be placed on a shaft at B_1 in Fig. 498. In this Figure the cams at B, B_2, B_1 are shown one above the other for clearness of illustration, but in practice some of them are on the same shaft, or are distributed on two parallel shafts, according to the detail design of different manufacturers.

831. A method for compressing the pure scavenger air, which is also the working air that is finally compressed to about 500 pounds

per square inch in the engine cylinder, the scavenge air being compressed to only 4 or 5 pounds per square inch, is shown in Fig. 498. The piston at *Q*, which is a part of the main or working piston *P*, is called the scavenge piston. As it descends it draws into the cylinder *M* a charge of air at atmospheric pressure, the port at *N* being opened and the port at *T* closed. On the up stroke the port at *N* is closed and the port *T* opened so as to exhaust the compressed air at 4 or 5 pounds pressure into the spaces at *T* and *U*, which serve as a receiver until the air is drawn off into the working cylinder *V*, when the valve *A* is opened. Another method, and one that is generally used for compressing the scavenge air, is to provide special and totally independent scavenge pumps, the piston rods of which are worked by long rocker arms from the main crosshead, or by eccentric rods or other mechanical means.

832. The compressed air at about 800 pounds per square inch which is used to inject the fuel is compressed in a separate multistage compressor which may be operated directly from the main shaft, as explained in paragraph 814, or through other mechanical connections, or independently. This highly compressed air is then cooled so that it will not ignite the oil which it injects into the cylinder.

Fuel Valve or Atomizer Detail for Diesel Engine

833. The oil enters the main cylinder in a highly pulverized condition mixed with the compressed air by means of a pulverizer, a form of which is shown in Fig. 499. Fuel oil from the fuel pump is admitted as shown, and is forced down through channel *5* and onto the perforated disks *4*, which are fastened to the valve-stem guide *7*. These disks are perforated with fine holes, which are staggered in the succeeding plates. As the injection air enters through the opening *6* it forces the oil from the disks and through the openings with a high velocity, effectively disintegrating the liquid into a fine spray. This spray also passes through fine channels, *3*, cut on the surface of the fine cone guide, shown in solid black, on its way to the cylinder through the nozzle *1*. The cone valve is shown at *2* at the end of the valve stem *8*, which is pushed down by the fuel rocker arm.

834. In all forms of construction it is essential to clear the pulverizer at each admission, so that exact amounts of oil may be injected each time. The speed regulation depends on admitting definite quantities of the mixture each time, and this quantity is controlled by the governor, which acts directly on the fuel pump and regulates the amount of oil delivered at each injection. One of the methods,

and a quite general one, so far as principle is concerned, for accomplishing this is explained in detail in the following paragraph.

Detail of Governor Gear Movements

835. The governor and fuel pump are shown in diagrammatic form in Fig. 500. The heavy lines show the mechanism in position for the engine running at full load. The action of the entire controlling gear is as follows: A plunger C obtains its motion, D_1 to D_2, from an eccentric or a crank mounted on the cam shaft or an intermediate shaft A. As the plunger rises, it draws in a supply of oil from

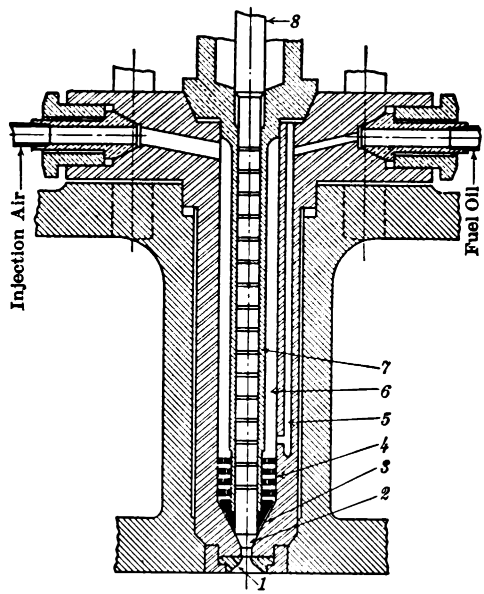


FIG. 499.—COMMON TYPE OF FUEL VALVE OR ATOMIZER

the oil well at I through the valve H. As the plunger moves down it will drive a quantity of oil equal to its displacement up through the check valve F through the passageway G to the fuel valve and into the engine, *providing the valve H is seated during the entire time that the plunger is moving down.* If the valve H is not seated during this entire period, some or all of the oil will be driven back into the oil well at I. It is just here that the governor control comes in, for it regulates the time that the valve H is to be closed while the plunger is descending.

836. The fuel pump and gear are so designed that when the engine is running at full load and rated speed the plunger is half-way

up on the suction stroke, or at D, Fig. 500, when the valve H opens, and it remains open until the plunger is again at the half-way point on its downward stroke. This relative motion is secured by pivoting the two ends of the floating lever $B\ O$ at B on the pump crosshead, and at O on the governor rod, the latter point remaining stationary for any one speed of running. M then moves to M_1 and back again while D moves to D_1 and back, and likewise L, J, and H all move to the sub *1* positions and back again in the same time. The object in not closing the valve H when the plunger is at the top of its stroke is to give time for any air that may be in the oil to be driven off through the open valve H before it is delivered to the engine. The nature of the contact at J between the lever $L\ K$ and the valve stem $H\ J$ is simply one of pressure in the up direction, the spring being relied upon to close the valve H. Thus as the point J of the lever moves to J_2 it will not be in contact with the valve stem at all.

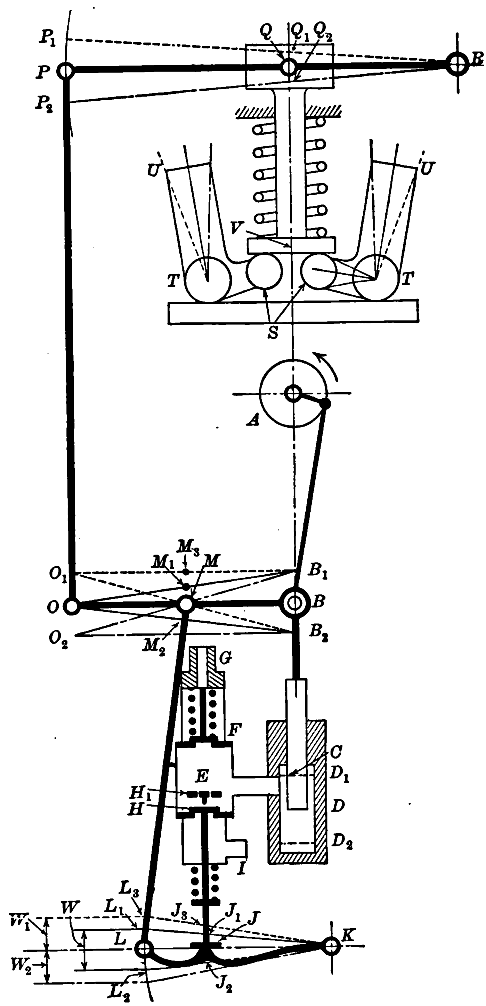


FIG. 500.—SHOWING GOVERNOR AND FUEL-PUMP MOTIONS IN DETAIL

837. At no load the suction valve is allowed to seat only when the plunger is at the bottom of its stroke, and this means that the valve H will be open practically all the time, and all the oil that the plunger draws in through H will be driven back to the oil well through the same valve, and none of it will be delivered to the engine. The manner in which this is accomplished may be seen by noting the

movements of the valve gear in Fig. 500 as follows: At no load the governor weights T U fly out to the dash-line positions, raising the point Q of the governor weight to Q_1 and swinging the governor lever R P about its fixed fulcrum to R P_1. This lifts the governor connection rod P O to P_1 O_1 and causes the floating lever to turn about a new temporary center at O_1 between the dash-line limiting positions O_1 B_1 and O_1 B_2 as the crosshead B moves up and down. The point M now moves between the limits of M and M_3, and similarly the points L and J now move between the limits L L_3 and J J_3. J therefore becomes the lowest point of the valve-stem lever stroke, and this occurs as B and C are at the bottom of their strokes. Therefore the valve H is seated only for an instant and no oil can get to the engine.

838. Summing up the motions of the lever L J K, Fig. 500, it will swing through the arc represented by W when the engine is running at full load, through the arc represented by W_1 when the governor weights are full out in the dash-line positions, and through the arc represented by W_2 when the governor weights are full in, in the dash-and-dot line positions. In this latter position the valve gear will not move the valve H at all, and it will be free to act as an automatic valve, and remain open during the entire upward stroke of the plunger, and will close at the beginning of the downward stroke, in which case the full displacement charge of the plunger will be delivered to the engine.

Diagrammatic Arrangement of Diesel Engine with Fuel Pump, Governor, and Air-Compressor Connections

839. A paper on the "Recent Developments in the Manufacture of the Diesel Engine," giving an excellent account of the status of the Diesel engine in this country, was presented by Mr. H. R. Setz before the St. Louis local meeting of the American Society of Mechanical Engineers and was printed in full in the journal of the society in the issue of December, 1914. From this valuable paper we are privileged to reproduce, in paragraphs 840 to 855 and in paragraph 892, some of the material relating to valves and valve gears. For our present purpose the illustrations are reproduced with a different system of referencing and with some of the structural details omitted. The text is also rewritten to accord with the general plan of this work.

840. A simplified diagram of a four-stroke engine showing the compressed air, fuel pump, and governor mechanisms is given in Fig. 501. The valve gear for controlling the fuel supply is indicated

by the reference numbers from *4* to *17*. The operation of this gear is explained in detail in connection with Fig. 500 in the preceding paragraphs. A vertical two-stage air compressor is indicated at *32–29*. The atmospheric intake valve at *33* opens on the down stroke of the low-stage piston and closes on the up stroke when the valve *30* leading to the inter-cooler *28* is opened. The air, partially compressed, is then drawn into the high-stage cylinder when the piston *29* moves down and is delivered to the receiver *24*, which is always open to the annular space in the fuel-valve cage *20, 19, 18*.

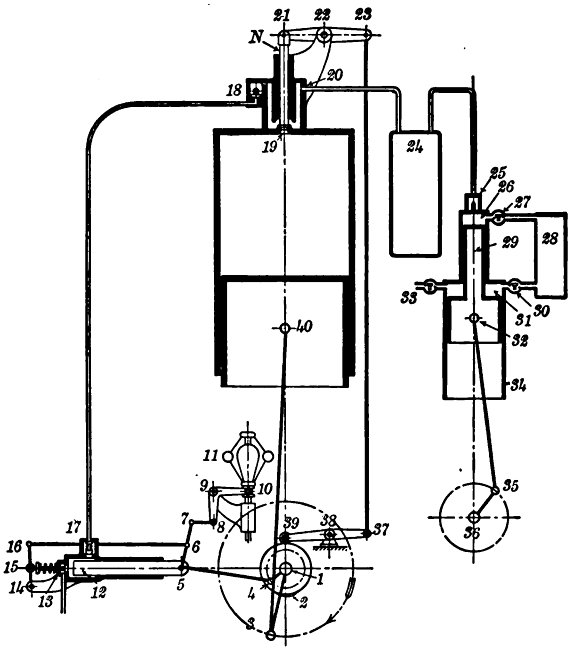


FIG. 501.—DIAGRAM OF FOUR-STROKE DIESEL OIL ENGINE SHOWING COMPRESSED AIR, FUEL PUMP, AND GOVERNOR MECHANISMS

It will therefore be seen that the fuel pump *12* must supply the oil against the injection air pressure which always exists in the fuel-valve cage, and that the injection air must be at a higher pressure than the compressed atmospheric air at the end of the compression stroke of the main piston, *40*. The clearance between the piston and the cylinder head in Diesel engines is extremely small.

841. In recent practical designs the cam shaft, carrying cam *2* and other cams, is placed at the top of the cylinder so that the rockers represented by *21–23*, Fig. 501, operate directly from their respective cams. This cam shaft is driven by a vertical shaft which

connects with the main shaft by bevel gear wheels at each end. This vertical shaft also operates the governor *11* and the fuel pump *12*, which are situated at the upper end of the shaft so as to give a more compact construction of the valve gear. In Fig. 501 the atmospheric-air admission valve, together with its stem, rocker arm, push rod and cam, is not separately shown, but may be considered as being in back of the fuel-admission valve *19*. Likewise the exhaust valve, and the air-starting valve, together with their control-gears, may be considered also in line with the fuel-valve gear from *19* to *39* as just described. The starting-air flask or receiver is charged from the fuel-injection receiver *24*, and it, together with its connections and valve, may be considered in line with *24–20*. The air-cooling and the water-cooling systems are also omitted in this illustration, but these as well as other omitted parts may be understood by referring to Fig. 486 and accompanying text.

Detail of Rocker-Arm Arrangements and Mountings

842. The usual arrangement of the cam shaft running alongside of the cylinder heads, so that the rocker arms to the valve stems may be operated directly without the intervention of rods, is shown in Fig. 503. This Figure, together with Figs. 502 and 504, shows also a common detail of valve-gear construction in Diesel engines in the mounting of two of the rocker arms on eccentric sheaves which are manipulated by hand in starting the engine. To start the engine, the handle *15* is placed in the position *17*. This handle, it will be seen from Fig. 504, is part of an eccentric sheave which supports the hub *29* of the rocker *11–18*, and also the hub *30* of the bell crank *9–24*. With the handle at *17* the eccentric *14* is down and the roller at *18* is dropped so as to come into contact with the starting cam *20*. The starting valve is shown at *2*. At the same time the roller *24* is swung to the left so that it does not come into contact with the fuel cam *22*. The engine then runs on the compressed starting air for a few turns. Then the handle is swung back to *15* and the roller *18* is lifted so that it does not engage the cam *20*, while the roller *24* is moved over so as to engage the fuel cam. With the handle at the position *16*, both rocker-arm rollers are out of range of the cam projections and the engine will not run at all.

843. A top view of a Diesel engine cylinder head and valve rocker arms is shown in Fig. 502. The exhaust valve is located at *32*, and is operated from the cam *21* through the rocker arm *27*, *28*, and *32*, which is mounted on an ordinary rock shaft *12*. The atmospheric-admission or injection air valve is located at *33* and is

operated from cam *19* by the rocker *26*, *31*, and *33*. The fuel-valve nozzle is shown at *3* in Fig. 503. It will be noted that the starting-air valve *2* opens inwardly. It does not open automatically, however, because the stem *6* of the valve is the same diameter as the disk *2*, and it is therefore balanced. In addition the spring at *7* serves to

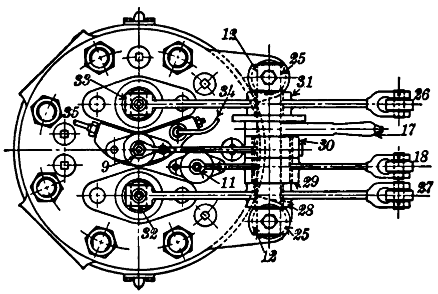


FIG. 502

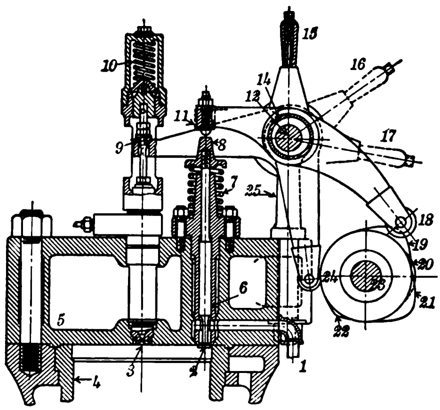


FIG. 503

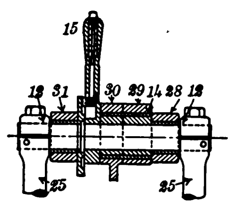


FIG. 504. — DETAIL SHOWING ECCENTRIC ACTION

FIGS. 502 AND 503.—PLAN AND ELEVATION SHOWING ARRANGEMENT OF VALVES AND CAM SHAFT IN FOUR-STROKE DIESEL ENGINE

keep it closed, except when its resistance is overcome by the rocker *11*, *29*, and *18*.

Pneumatic Control of Starting-Air Valve

844. In some forms of marine valve gears the air-starting valve is operated pneumatically, thus simplifying the construction and providing means for starting or stopping from different stations. In Fig. 505, *23* represents the cam shaft, and *19*, *22*, *21* represent the fuel-air admission, fuel oil, and the exhaust cams respectively, which operate the rockers *28*, *30*, and *31*, shown in Fig. 502. A quick-

opening valve at *44*, Fig. 505, permits air from the air-starting pipe *46* to pass through the branch pipe *41* to the rotary distributor valve *20*, which periodically admits starting air to the upper side of the piston *42* on the starting-air valve stem. This presses down the valve *2* and admits starting air to the main cylinder at *4*. The air

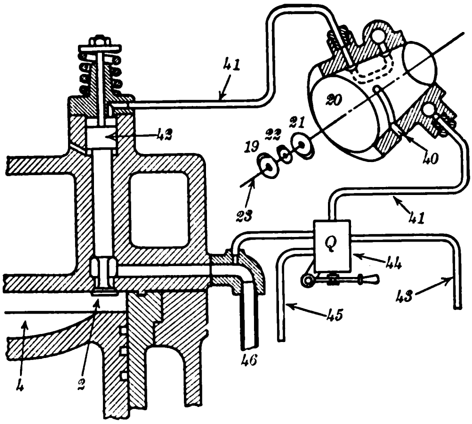


FIG. 505.—PNEUMATIC CONTROL OF STARTING-AIR VALVE

pressure here takes the place of the cam *20* and the rocker *29* in Figs. 502–504. As the distributor valve *20* rotates further it connects up the pipe *41*, through suitable channels, to the atmosphere at *40*, thus relieving the pressure on the valve *42* and permitting the valve to close by means of the spring at the end of the valve stem. This action continues to alternate so long as the valve *Q* is open. The valve *Q* is so arranged that the engine may be started either from the platform or engine-room floor.

Types of Fuel Valves

845. The injection of fuel into the cylinder of the Diesel oil engine is one of the most difficult problems in valve and valve-gear design, as the following considerations will show: About 1/25 to 1/35 of a second is available, on some types, to inject the fuel charge gradually; the fuel oil must be converted in this short time from a liquid into very minute particles, which have to be uniformly distributed in the injection air to form a proper mixture for sustained combustion; the fuel oil must be delivered against a pressure of over 700 pounds per square inch.

846. Several types of valves, or atomizers, for accomplishing this

are in use. One of the best known types has already been illustrated in Fig. 499, and described in paragraph 833.

847. Another type of atomizer is shown in Fig. 506, in which the fuel oil in liquid state is received from the fuel pump in channel *5*. The oil fills the annular space *3* and nearly fills the series of holes which slant upward at *4*. The space at *7* is filled with the highly compressed injection air. When the cone valve *2* is lifted by the fuel rocker, as illustrated at *9*, *30*, and *24*, Figs. 502–3, pressure on the oil is relieved at *4* and it flows up through the holes and mixes with the inrushing air.

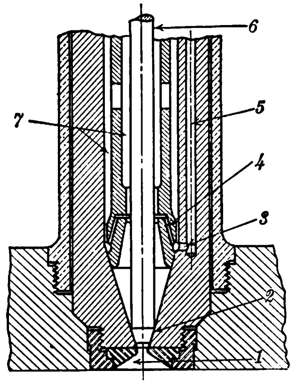


Fig. 506.—Fuel-Oil Injection Valve

848. A form of sprayer or fuel nozzle used where the fuel oil has a high flashpoint, making it necessary to add a small amount of low flashpoint oil so as to give a more rapid ignition, is shown in Fig. 507. Two fuel pumps are used, one for pumping the lighter oil into the annular space *5*, and one for pumping the heavy oil into space *6*. The lighter oil enters the channels *5* and passes directly to the head of the valve at *2*. When the valve is raised, the air through *7* carries before it the heavy oil through the atomizer disks *4* and the finer oil from *5*, the latter entering the cylinder first.

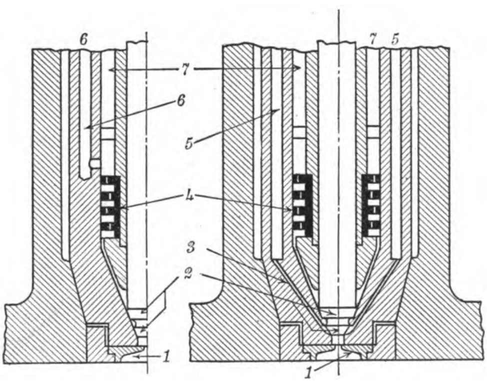


Fig. 507.—Sprayer for Separate Fuel Oils

Types of Fuel Pumps

849. The fuel pump delivers the oil to the fuel-valve cage as shown at *5* in Fig. 499. The amount of oil that is required at each admission is small, being 0.141 cubic inches for a 100 horse-power engine at 165 revolutions per minute. The delicacy of the regulation is shown, in this case, by the fact that the quantities that should be delivered at ¾, ½, ¼, and 0 load are 0.30, 0.28, 0.26, and 0.21 cubic inches, respectively. In another case, a 500 horse-power engine consumed 0.56 cubic inch per heat cycle at full load.

850. Several forms of fuel-pump gears are in use for accomplishing this. The design of these gears is based on one of the following three principles of operation:

1. Variable stroke of the pump plunger.
2. By-passing the fuel oil by means of a mechanically operated pump intake valve which permits some of the oil taken into the plunger cylinder to be pumped back to the source of supply before the intake valve is allowed to close.
3. Variation of clearance volume in the fuel-pump cylinder.

851. A gear built on the principle of varying the stroke of the pump plunger is shown in Fig. 473. and is described in the text accompanying that Figure.

Two eccentrics, one turning on the periphery of the other, as shown in the illustration of the Armington & Sims steam engine gear on page 154 of Volume I, are also used to give a variable stroke to the pump plunger.

These gears are under the control of the governor and are not suited for operating directly against the high pressures involved in the strictly Diesel engine, because of the load imposed on the governor, due to the packing and friction involved when high pressures are used. They have been used, however, on modified types of the Diesel engine where an open fuel-injection nozzle is used. In this form of engine the nozzle *C*, Fig. 508, in the cylinder head is always open to the cylinder, and the fuel oil is injected into the chamber *D* from the fuel-pump pipe *E* when the compression is low, that is, when the piston is at the bottom of its stroke as shown at *A*. The oil thus injected does not enter the cylinder through this small open nozzle, but remains deposited at the bottom of space *D*, and is violently injected into the cylinder when the high-pressure injection air is admitted from *H* by the lifting of the spindle valve *F*. This occurs, of course, when the piston is at the top of its stroke.

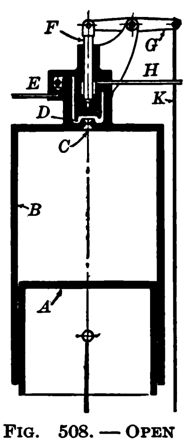


Fig. 508. — Open Fuel-Injection Nozzle

852. Another form of open fuel-injection nozzle with an auxiliary injection-air piston, which does away with the regular compressor for injection air, is shown in Fig. 509. The fuel oil is injected by the pump *7* into the fuel-oil space *21* as described in the preceding paragraph. The auxiliary piston *24* is operated by the cam *1*, which permits the piston *24* to remain down and keep the port *23* open so

that air compressed in the main cylinder *18* is also compressed in the space above the piston *24*. Just before the main piston reaches the top of its stroke the steep part of the cam *1* moves the rocker *3* and quickly raises the auxiliary piston *24* so that air under high pressure is injected at *21*, thus driving the deposited oil into the engine cylinder.

853. A variable stroke plunger pump of special construction, which is effective under governor regulation even under high delivery pressure, is shown in Fig. 510. In this pump two plungers are used, one being termed the measuring plunger, *13*, and

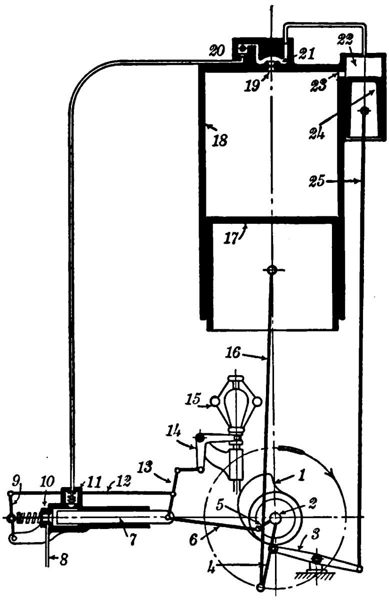


FIG. 509.—OPEN NOZZLE WITH AUXILIARY INJECTION-AIR PISTON

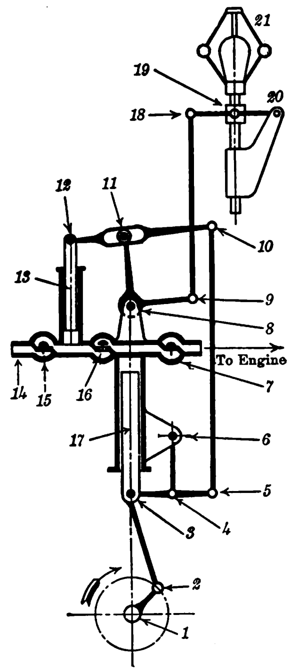


FIG. 510. — SPECIAL VARIABLE STROKE FEED PUMP AND VALVE GEAR WITH MEASURING PLUNGER

the other the delivery plunger, *17*. The fuel oil from the supply tank enters at *14*, being drawn in by the plunger *13*, which receives its motion from the shaft *1* through the cranks and levers, as shown. This plunger measures the oil to suit the required load, because the governor changes the position of the fulcrum *11*, making the pump stroke longer when the engine is below speed, and shorter when above speed. *9*, *8*, and *11* is a bell crank. *15* and *16* are automatic valves, *15* lifting, and *16* remaining closed on the up stroke of the plunger *13*. *15* remains closed, and *16* lifts on the down stroke of *13*. Thus the measured amount of oil is delivered to the force-pump plunger *17*, which delivers in turn to the atomizer case against the high-pressure injection air. In this way the governor does not have to operate against high resistance, as it would have to do, for example, if the plunger *13* were also the force pump as well and had to slide back and forth against a tight stuffing-box packing.

854. The by-pass type of fuel pump is illustrated in skeleton outline at *12* to *17*, Fig. 501, and also at *J, H, C, G* in Fig. 500. It is also shown in a diagrammatic form at *8* to *14* in Fig. 488. Each of these is described in detail in the text referring to these Figures, the general principle being, in each case, that the pump plunger draws in a charge of fuel oil from the supply tank through an automatic valve that is lifted with the rise of the plunger, and that this automatic valve is kept open by the governor during a greater or less portion of the down stroke of the pump plunger, thus allowing a portion of the oil to be pumped back into the supply tank. As soon as the governor allows this valve to close, the remaining charge is forced through another automatic spring-loaded valve to the atomizer valve. In this gear also it will be noted that the governor is not called upon to operate against any heavy resistance that comes from or is due to the high-pressure injection air.

855. The variable clearance type of fuel pump controls the fuel-oil charge by changing the clearance volume surrounding the plunger. This is shown in Fig. 511. The plunger *4* receives a constant stroke

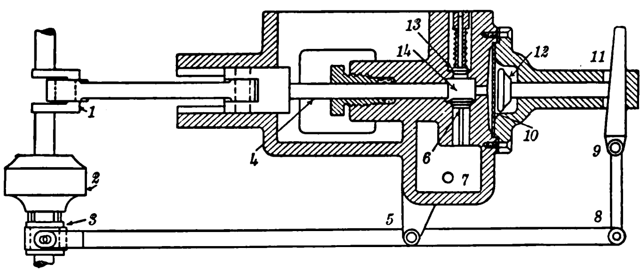


FIG. 511.—VARIABLE CLEARANCE VOLUME TYPE OF FEED PUMP

from the crank *1*. On each suction stroke a uniform quantity of fuel oil fills the pump chamber *14* from the supply chamber through the automatic valve *6*. If the engine is below speed, the revolving weights in the governor at *2* draw in the end *3* of the long rocker arm, which is supported at *5*, and so pull down the wedge *11*. *10* is a flexible diaphragm, and it is pressed back against the flat head *12* as the pump plunger *4* moves forward on its pumping stroke. If the wedge *11* has allowed the head to move back from its present position, the diaphragm will spring back to it and the plunger clearance volume will be increased. With this increased volume more oil will be stored in the plunger chamber at the end of the plunger stroke and less will have been delivered to the atomizer valve.

Practical Examples of Diesel Oil Engines

856. Having brought out diagrammatic illustrations of the Diesel engine, together with more or less diagrammatic representations of fuel valves, fuel pumps, and governor gears in the preceding paragraphs, several practical applications to well-known makes of Diesel engines will now be made. A four-stroke non-reversing engine is given first in the following paragraphs, after which the fundamental principles of reversing mechanism are illustrated. Practical reversing engines are then illustrated and described.

The New London Ship & Engine Company Diesel Four-Stroke Engine

857. The valve gear for a 120 horse-power four-stroke Diesel engine, developed by the above company and now manufactured by them at New London, Conn., under the name of "Nlseco," is illustrated in diagrammatic form in Fig. 512. This company also builds a two-stroke Diesel engine developed by the Machinenfabrik Augsburg-Nürnberg A. G., and generally referred to as the "M. A. N.," or Nürnberg engine.

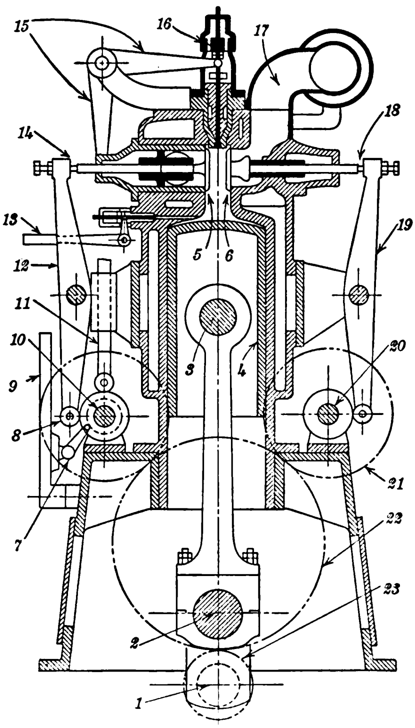


Fig. 512.—The New London Ship & Engine Co. Diesel Engine

858. The main shaft, connecting rod, and piston are represented at *1* to *4*. The air-inlet valve is shown at *5*, the exhaust valve at *6*, and the spray or oil-fuel valve stem at *16*. The air-starting valve is in front of the plane of the section, and therefore not shown in the illustration.

859. The motion for operating the several valves comes from a spur wheel *23* on the main engine shaft and is carried through the large transmission wheel *22* to the two wheels on the cam shafts *10* and *20* respectively. A cam on the shaft *10* moves the vertical rocker *12* at proper intervals and opens the air-inlet valve *5*. Another vertical rocker similar to and immediately back of *12* receives motion from another cam on the shaft *10* and transmits it through the bell crank *15* to the fuel valve stem *16*. A cam on the shaft *20* transmits motion through the vertical rocker *19*, which opens the exhaust valve *6*.

860. For starting the engine the lever *9* is drawn over, thereby giving endwise motion to a small countershaft at *7*. This small shaft carries an arm which transmits the longitudinal motion to two cams which are cut on one piece of metal which is free to slide lengthwise on the shaft *10*. The fuel cam is thereby thrown out of line with its vertical rocker, while the starting-air cam is thrown in line with its rocker. The working-air cam is fixed on the cam shaft, and thus the valve it operates is always in service, whether the engine is running on starting air or on fuel. When the engine gains headway the lever *9* is thrown back to its regular running position. In practice, four or more cylinders are placed in line and the starting lever *9* is made to control only one-half the number of cylinders for starting air. The remaining cylinders take working air and fuel in regular manner as soon as the starting air in the other cylinders turns the engine, and when sufficient speed is attained ignition takes place and the engine runs on half the cylinders under its own power. Then the lever *9* is returned to its running position and all of the cylinders are working.

Fuel-Pump Control

861. The fuel-oil supply is controlled by hand and by governor action in a 180 horse-power "Nlseco" Diesel oil engine as shown in Figs. 513 and 514. The governor shown in Fig. 513 is of special mechanical interest, although it is replaced in someof the later types by the usual form of flyball governor as indicated in Fig. 515. Returning to Figs. 513 and 514, a heavy flywheel is shown at *12*, carrying pins at *4* and *11*, from which are hung governor arms *3* and *13*, which swing out, when running, against the tension of the springs *2* and *14*. Each governor arm carries a sector on which are cut gear teeth as at *5*. As the weights swing out about the pins, the gear teeth engage with teeth on the sleeve *7*, which is then rotated through a small arc on the flywheel hub *10*. This sleeve has a 45° slot, in which fits a pin, *31*, that

is fixed in the flywheel hub. The slot is shown in a small detail view at *32* in Fig. 514. As the sleeve is rotated, as just explained, it is also advanced in the direction of the axis of the shaft *1*, because of the action of the 45° slot against the pin. The longitudinal motion

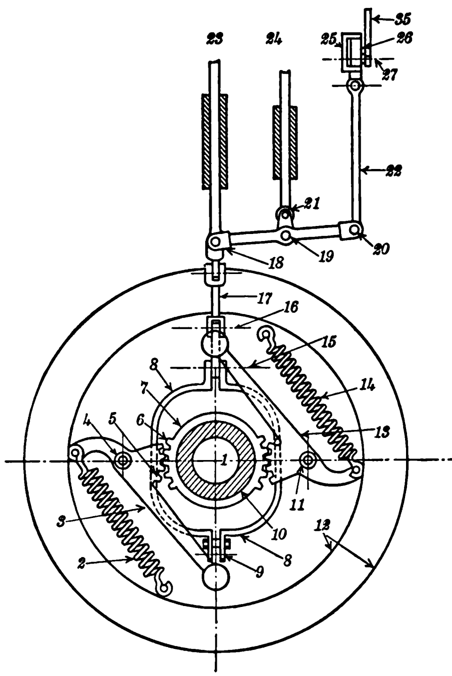


FIG. 513

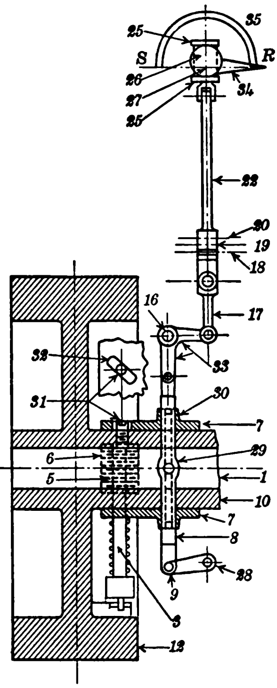


FIG. 514

FIGS. 513 AND 514.—GOVERNOR AND FUEL-PUMP GEAR, NEW LONDON SHIP & ENGINE CO. DIESEL ENGINE

of the sleeve gives a swinging motion to the yoke *8* about the point *9*. The sleeve and yoke are connected with an ordinary ring joint and trunnion pins *29*.

862. As the yoke swings to one side or the other of its vertical position it rotates the bell crank *33* on the fixed pin *16*, and this rotation moves the link *17* up or down and gives a definite position to the end *18* of the floating lever *18–20* for the corresponding position of the governor weights. The position of *18* will change for each fluctuation of speed in the engine, and so will the rod *24*, which regulates the discharge of the fuel pump to the engine cylinders, as explained in connection with Fig. 515. The position of the point *20* of the floating lever is regulated by hand by the handle *34*, which

rotates a circular disk *26* about an eccentric center *27*. The circular disk works in a box-frame *25*, to which is connected the rod *22*. The point *20* is shown in the illustration in its highest position, and so is the point *21* of the regulating rod, which now permits a maximum fuel supply to pass to the engine for a normal load. When the handle is turned 180° to the position *S*, the points *20* and *21* will be in one phase of their lowest range of working, and in this case no fuel oil at all is allowed to pass to the engine.

863. In the mechanical construction of the gear it may be pointed out that the teeth *6* on the sleeve *7* are longer than the teeth *5* on the governor arm. This is shown in Fig. 514, the object being, of course, to permit the sleeve to move endwise without sliding the teeth out of contact. The rod at *23* is merely a guide rod for the end *18* of the floating lever.

864. The amount of fuel oil delivered by the fuel pump depends on the time that the suction valve is held open. This is illustrated in Fig. 515. The pin *8* receives a constant back and forth motion,

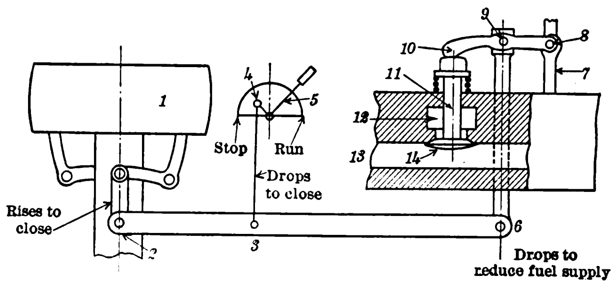


FIG. 515.—FUEL-PUMP CONTROL, "NLSECO" ENGINE

thus giving the pump plunger *7* a constant stroke. The end *8* of the lever *8–10* also has the same motion. The elevation of the supporting pivot *9* of the lever may be varied by the governor *1*, or by the hand lever *5*. When the point *9* is lowered, the suction valve *14* is held open longer and more fuel oil is returned to the source of supply at *12* and less goes to the engine through the channel *13*. When the point *9* is raised the suction valve is permitted to close sooner and more fuel oil is pumped to the engine. The fuel pump is arranged with one plunger for each cylinder.

Reversing of Oil Engine

865. The smaller sizes of regular oil engines and of Diesel engines are generally made to turn in one direction, and in such cases reversal in direction of turning of the follower shaft is accomplished

by means of a mechanical reversing gear outside of the engine itself. These mechanical gears usually depend on a planetary gear arrangement, friction device, or on reversible propeller blades. For larger engines such reversing mechanisms become too cumbersome and the effort to manipulate them too great. The engine itself must then be reversed, by causing the piston to stop at some intermediate point on its up stroke and forcing it to return without having reached its top dead-center position.

866. The valve gear necessary to permit reverse directions of rotation *when an engine is under way* is in itself comparatively simple in an oil or Diesel engine, for all that is necessary is to have two sets of cams, one "forward" and the other "reverse," properly set side by side on the same shaft and to slide the shaft along a small distance so that the several valve rockers will rest on their respective cams. But in the Diesel engine, particularly, the several mechanical movements that are necessary during the period that the reversal is being effected serve to complicate the gear. In the first place, when reversal is desired, the fuel oil must be cut off, and this requires mechanism that will lift the rocker from the cam. In the second place, mechanism must be provided to drop the starting-air rocker onto the starting-air cam, it having been raised above its cam during the regular running of the engine. In the third place, mechanism must be provided to operate the exhaust-valve rocker in four-stroke engines. Finally, there must be no interference between cams and rockers as the cam shaft is moved longitudinally.

867. Reversal in two-stroke oil engines may be effected by using one set of cams for the running parts if some compromise is made on the periods of admission of the scavenging air and fuel oil. This one set of cams need only be rotated through an angle of 30° to 40°. To accomplish this, it is necessary that the fuel angle TAF, Fig. 516, and the scavenge-air angle HAL should be so positioned that a diametral line such as DJ bisects both angles. In the illustration the fuel angle is taken as 45°, with a preadmission of $2\frac{1}{2}$°. The line DJ, which bisects this angle, therefore makes an angle of 20° with the vertical center line, and if the scavenge-air period is taken at 100° the preadmission of air must be 30°. If, now, reversal is to be effected, it is only necessary to turn the cam shaft carrying the fuel and scavenge-air cams through an angle of 40° in order to have a fuel preadmission of $2\frac{1}{2}$° and a preadmission of 30°, the same as before, but for reverse running. The exhaust, of course, need not be considered, for it is controlled by the piston, and the angle RAS, during which it takes place, is equally divided on both sides of the

vertical line for both forward and reverse running. With the one set of running cams here described, it is necessary, during the period that reversal is being effected, to provide mechanism for lifting the

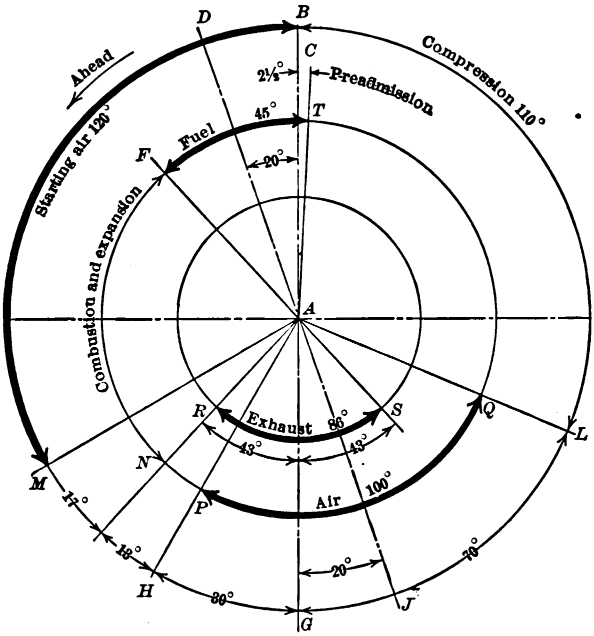


FIG. 516.—VALVE DIAGRAM FOR TWO-STROKE SINGLE-ACTING ENGINE

fuel and scavenge air-rockers off the cams so that they will be prevented from functioning during the periods necessary to slow down and get up speed with starting air in the reverse direction. Mechanism for controlling the starting-air lever is also required.

868. Two principal methods of mechanical movement for operating the main cam shaft have been pointed out in the two preceding paragraphs, one consisting in a sliding of the shaft in the direction of its axis and using two sets of cams, and the other consisting in a rotation of the shaft with one set of cams. The details and combinations used in practical applications where the additional requirements of lifting and dropping rocker arms at proper times must also be considered vary widely with different makers. The fundamental operations, however, are the same in all makes, and in the following paragraphs full detail directions for practical reversing will be given for two types of two-stroke engines and for one type of a four-stroke engine.

Krupp-Diesel Two-Stroke Reversing Engine

869. A simple method of control for a two-stroke Diesel marine engine is used in the type manufactured at the Krupp works in Germany. The engine and the essential features of control are

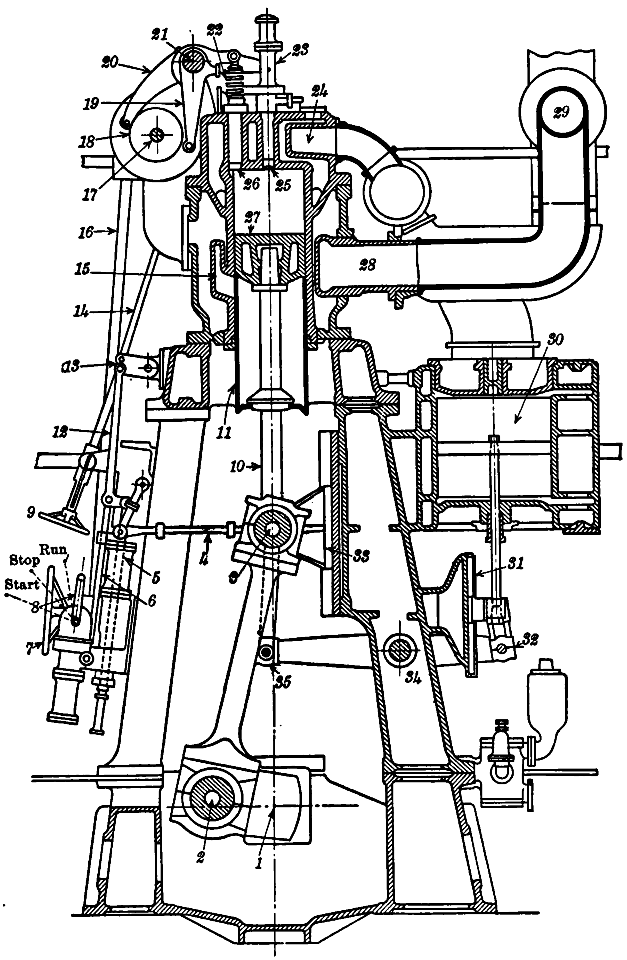


FIG. 517.—KRUPP-DIESEL TWO-STROKE MARINE ENGINE

illustrated in Fig. 517, which represents a cross-section of a cylinder developing about 190 horse-power. Six cylinders similar to this one are arranged in two independent groups of three each on one of the propeller shafts, while six other cylinders are similarly arranged on the other propeller shaft. These engines, developing 2,300 brake horse-power at 140 revolutions per minute, are installed on the oil-tank ship *Hagen.*

870. The main shaft is indicated at *1*, crank pin at *2*, connecting rod at *2–3*, hollow piston rod at *10*, carrying cooling water from the swivel-jointed pipe *4* to the piston *27*, crosshead and crosshead guide at *33*, and cylindrical piston guide at *11*. One of the exhaust-port openings is indicated at *15*, and the exhaust pipe leading to a small exhaust funnel at *28–29*. The fuel valve is shown at *25*, and the air-starting valve at *26*. Two scavenging-air valves are used on each of these cylinders, one being immediately in front and one in back of the fuel valve, and all three lying in the plane of the crank shaft. The scavenging air is supplied through the pipe *24* from the pump *30*. This pump is operated from the crosshead pin *3* through a short return arm *3–35*, a rocker *35–34–32*, and a short link to the auxiliary crosshead which operates on the guide *31*.

871. The operation of the valve gear for starting, stopping, and reversing is as follows:

For starting ahead, the handwheel *7* is turned to the marked starting position. The handwheel shaft rotates the vertical shaft *16* through a universal worm-gear joint, and this shaft drives the cam shaft *17* a few inches in the direction of its axis by means of a forked arm and collars. The cam shaft has fixed to it two independent full sets of cams, one for running ahead and one for astern. The end motion of the shaft *17* places the ahead cams under their proper rocker arms, *19* and *20*, etc.

All of the rocker arms are mounted on eccentric sheaves on shaft *21* and are so adjusted that when the operating lever *8* is in the "stop" position, all of the rocker-arm rollers are thrown out of reach of their respective cams. This permits the cam shaft *17* to move endwise, by turning the handwheel *7* as stated above, without interference.

872. When the operating lever *8* is moved to the "start" position, a pneumatic plunger rod *5* is operated, moving the intermediate rod *12*, which is guided at its lower end by the rod *6*, the double-arm rocker *13*, the second intermediate push rod *14*, finally giving a motion of rotation to the shaft *21*. This rotation causes the eccentrically mounted rockers to move in such a way that the starting-air rocker comes into contact with its cam while the fuel rocker is held free from its cam. The scavenge rocker is also held free from its cam in order that the roller may not interfere with the longitudinal shifting of the cam shaft. An example, in detail, of eccentrically

mounted rockers is shown in Figs. 502 to 504. The starting air is turned on by means of a small handwheel located just below the lever *8*, but not appearing in the illustration. When the engine is under way with the starting air as motive power, the lever *8* is moved up to the running position when the starting-air rocker is thrown out of contact with its cam and the scavenging-air and fuel rockers are thrown in.

873. Each set of three cylinders is operated by one lever as at *8*, while the single handwheel *7* operates for all six cylinders on the one propeller shaft. One set of three engines out of the twelve can thus be made to furnish all the power if slow speed is desired. In reversing, the lever *8* is put in the stop position and the handwheel *7* turned until it comes into "reverse" position. The operations described above for hand lever *8* are then repeated.

874. In case of derangement of the pneumatic plunger, the rocker-arm shaft *21* may be rotated into the starting, stopping, or running positions, as desired, by mechanical means alone through the handwheel *9*, there being one such handwheel for each set of three cylinders. In most engines of other manufacture, the motion here obtained regularly by purely mechanical means through the handwheel *7* is accomplished by pneumatic action. The effort required to operate this wheel is not an objection in engines of this size, and it tends to simplicity and reliability in operation.

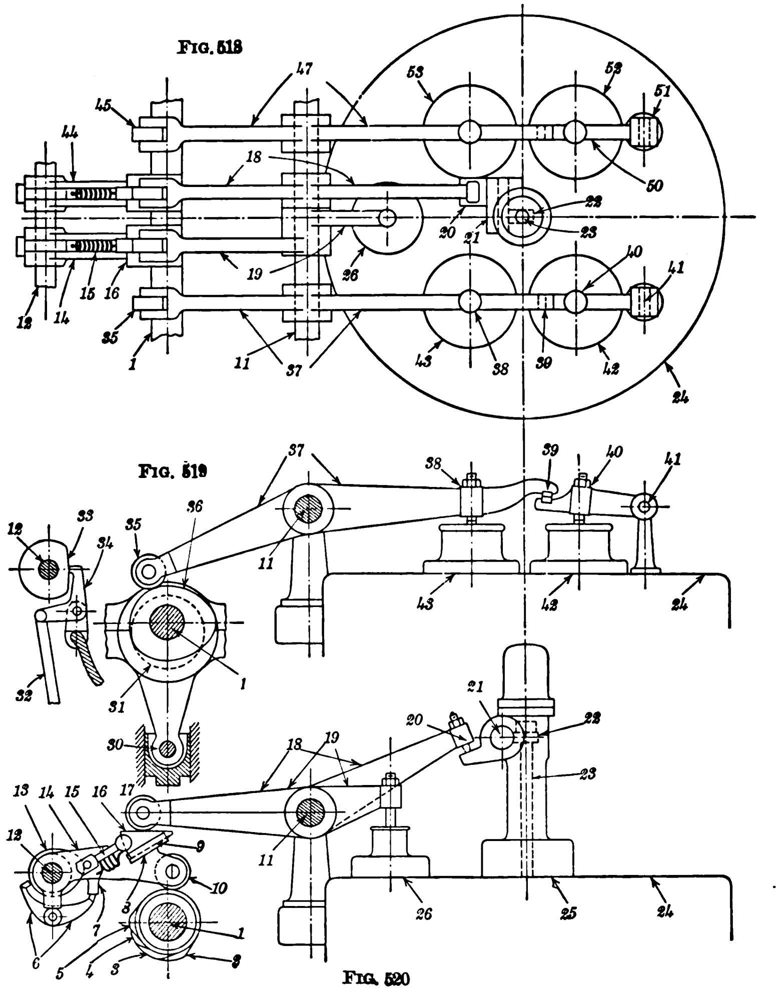
Carels-Diesel Two-Stroke Reversing Gear with Gradual Cut Out and Admission

875. The Carels-Diesel engine, developed at Ghent, Belgium, has a valve gear that is distinctive in several of its features, the principal one being that the starting air is gradually cut out and the fuel oil gradually introduced when starting the engine and when effecting reversals.

876. In Figs. 518–520, the mechanism is shown diagrammatically for one of the cylinders of a four-cylinder two-stroke marine engine of 1,000 horse-power for the four cylinders. A top view or plan is shown in Fig. 518, while a front view or elevation is shown in two sections in Figs. 519 and 520 in order to illustrate the several parts.

877. When the engine is running regularly, the cam shaft *1* is rotated by means of a vertical drive shaft from the main engine shaft, worm-gear connections being used. The fuel oil is admitted when the cam projection *5* passes under the roller *10*, thus lifting the interfuel lever *14* about the center *12*. This lever has an adjustable flat surface at *16*, which lifts the roller *17* of the main fuel rocker *18*,

thus lowering the point *20* and turning the rocker *21* counter-clockwise. This rocker has an arm represented by dash lines at *22*, which lifts the fuel-valve stem *23*. On the cam shaft *1* there are also attached two cams represented at *36* in Fig. 519, and projected at *35* and *45*



Figs. 518–520.—Carels-Diesel Valve Gear, Giving Gradual Cut-Out of Starting Air and Gradual Admission of Fuel

in Fig. 518. When these pass under the rollers on the rocker arms *37* and *47* respectively, the four scavenge-air valves indicated at *42*, *43*, *52*, and *53* are pressed down and opened.

878. Four valves are used in this engine to allow ample opening

for the quick admission of the large quantity of low-pressure air required for running. This air, which also gives a clean exhaust, is under pressure of 4½ pounds per square inch. The two scavenging valves at *43* and *53*, it will be noted, are opened directly by the rocker arms while the two valves at *42* and *52* are opened by auxiliary rocking arms shown at *40* and *50*. The contact blocks between the arms are indicated at *39*. The arms are so proportioned that the distance *41–40*, Fig. 519, is to *40–39* as *11–38* is to *38–39*, thus giving the same opening to each scavenge valve. Being a two-stroke engine, the exhaust is effected through ports uncovered by the piston near the bottom of the stroke, and so no exhaust valves are used in the cylinder head.

879. Starting or reversing is accomplished gradually and quickly as follows: Assuming that the engine is running and it is desired to reverse, the cam shaft *1* is turned approximately 30° by means of a compressed-air motor located at the operating stand, thus turning the two scavenge-air cams indicated at *36* the same amount and causing them to operate the scavenge rockers and the scavenge valves at the proper time for reverse running. The fact that a 30° turn will do this is explained in paragraph 867 in connection with Fig. 516. The maneuvering shaft *12*, Fig. 520, is then rotated, and suitably placed cams *13* on this shaft allow the wedge *16* to be drawn in on the sliding seat *9* by means of the spring *15*. This cuts out the contact between the surface *16* and the roller *17* by increasing the space between them, and consequently the fuel rocker *18* is not moved when the cam projection *5* comes under the roller *10*. A further rotation of the maneuvering shaft *12* causes the cams on it to turn the curved rocker *6* counter-clockwise, so as to swing the auxiliary levers *14* and *44*, Fig. 518, clear of all the cam projections on the shaft *1*. The maneuvering shaft *12* is next moved endwise, carrying the auxiliary or interstarting and interfuel levers *14* and *44* with it, so that these levers are in line with a new set of reverse cam projections on the shaft *1*. The reverse fuel cam is indicated at *4*, Fig. 520, and the reverse starting-air cam at *2*.

880. The auxiliary levers are always free to turn on the shaft *12*, but are required to move endwise with it, because of collars at both sides of the lever hubs. A further rotation of shaft *12* places the cams on it so that the interstarting lever *14* drops on its reverse cam *2*, and the high-pressure air is admitted against the advancing piston, thus slowing down the engine and soon causing it to turn in the opposite direction. A still further rotation of shaft *12* places one of the cams so that the interfuel lever *44* drops on the reverse cam *4*,

and the engine then picks up on its own power. In this final rotation of shaft *12* the cams on it cause the starting and fuel wedges, both represented at *16*, Fig. 520, to move out of and into contact respectively, so as gradually to cut out the starting air and admit the fuel oil.

881. At the time that the fuel oil is being gradually introduced the cam *33*, Fig. 519, also on the maneuvering shaft *12*, is being rotated so as to control the suction valves of the fuel pump through the bell crank *34* and the rod *32*, and so regulate the amount of fuel oil injected. The fuel-oil pump is connected to the crosshead *30* and is operated from the eccentric *31*, which is keyed to the cam shaft *1*. The description here given is applied to one cylinder for simplicity, and it illustrates fully the order of operations for controlling the engine. In practice usually only one-half of the total number of cylinders are equipped with the wedge gear. The other cams, however, are timed a little differently, so that the other cylinders will pick up after the reversing and starting cylinders with the wedge control have done their work. Although the gear appears complicated, it is simply controlled by two levers and a handwheel, and reversal of a 1,000 horse-power engine from full speed ahead to astern has been effected in six seconds. In a trial at maneuvering, 63 reversals were accomplished in 42 minutes, with more than half of the storage high-pressure starting air still unused.

Craig-Diesel Four-Stroke Reversing Engine

882. A Diesel oil engine of American design is shown in Figs. 521 to 524. It is manufactured by the James Craig Engine & Machine Works, Jersey City, N. J. The engine from which the Figures are arranged is a six-cylinder, four-stroke, air-starting, and reversing marine engine. The cylinders have a bore of 9½ inches and a stroke of 11 inches, and are individually cast. All valves are in the cylinder head which is water-cooled, as are the cylinder walls. The engine is rated to develop 180 horse-power at 400 revolutions per minute.

883. When the engine is running ahead, three of the four valves in the cylinder head are in regular use, namely, the air valve *26*, Fig. 523, the oil-injection valve *27*, and the exhaust valve *28*. These are operated by rockers similar to *45*, *42*, and *34*, respectively, Fig. 521. The rocker arms are driven by the ahead cams *15*, *16*, and *18*, respectively, on cam shaft *14*, which is driven at one-half the engine speed through gears in the gear box *10*. In this Craig engine an auxiliary exhaust is used for the first time in Diesel oil engines, and this is shown at *56* in Fig. 524. The auxiliary exhaust passage-

way *55* is just about to be opened as the piston is nearing the bottom of its stroke. This auxiliary exhaust valve is operated by a separate cam shaft *67* and cam *60* through the valve stem *58*. It is claimed that 80 per cent of the hot exhaust gases escape through this auxiliary port, leaving the remainder to expand and cool and pass out of the regular exhaust valve *28*, which is timed to open just before the auxiliary valve port *55* is closed by the piston *24* on its way up. The auxiliary exhaust valve *56* is opened before exhaust takes place, that is, before the piston uncovers *55* on its way down, and, therefore, neither exhaust valve has to be opened against working pressure.

884. When the engine is running "reverse" the same valves as described above are the only ones in use, the auxiliary exhaust valve being operated by the same cam turning in opposite direction, while the three valves in the cylinder head are operated by the reverse cams *15R*, *16R*, and *18R*, which are also rigidly attached to the cam shaft *14*, and which are placed in position under the respective valve-stem rollers *19*, *20*, and *22* by moving the cam shaft endwise through the lever handle *1*.

885. The engine is started by compressed air, previously compressed and stored, on all six cylinders, and after it has speeded up the starting air is thrown off the three forward cylinders and air and oil are admitted, whereupon the forward set starts running under its own power. Starting air is then cut off from the other three cylinders and air and fuel admitted. The entire engine is then running under its own power.

886. The mechanism necessary to carry out the starting operations mentioned in the preceding paragraph and the order of operation are as follows: Lever *1* is placed forward in notch *A*, as shown in Fig. 523, thus moving the cam shaft *14* endwise and placing the "forward" cams *15* to *18* in position under the valve-stem rollers *19* to *22*. The lever *2*, which controls the forward three cylinders, is then swung from its neutral position *N* to the starting notch at *S*, thus rotating an outer hollow control shaft *52* through a bevel-wheel connection at *5*. This motion automatically throws the fuel valve out of action and puts the starting-air valve into action, as will be explained in detail in paragraph 888. Lever *3* is next drawn over to notch *S*, accomplishing exactly the same results as lever *2*, but on the three rear cylinders through the inner control shaft *53*. Each of the control shafts, *52* and *53*, carries a cam which opens a valve in the starting-air supply pipe when the levers *2* and *3* are placed at *S*.

887. With the above movements of the valve gear, the engine will

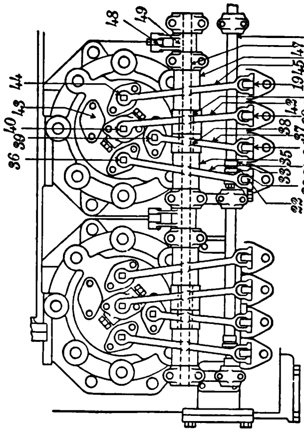


FIG. 521

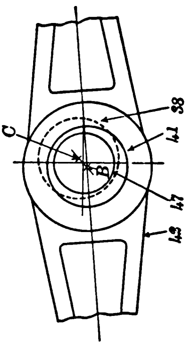


FIG. 522

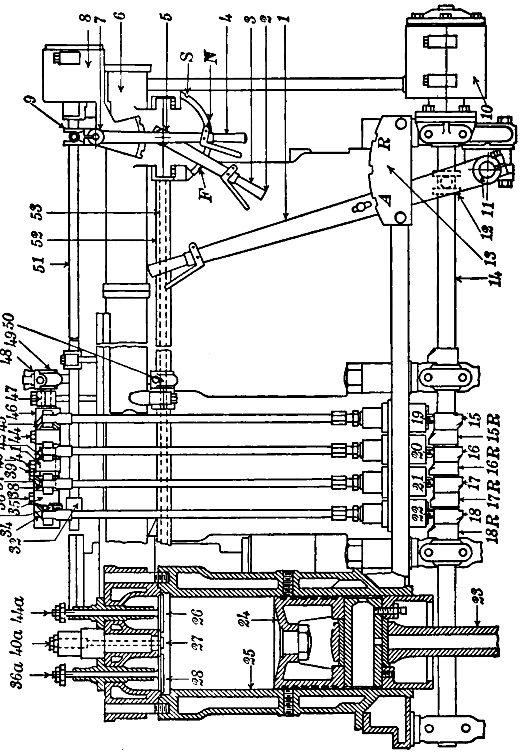


FIG. 523

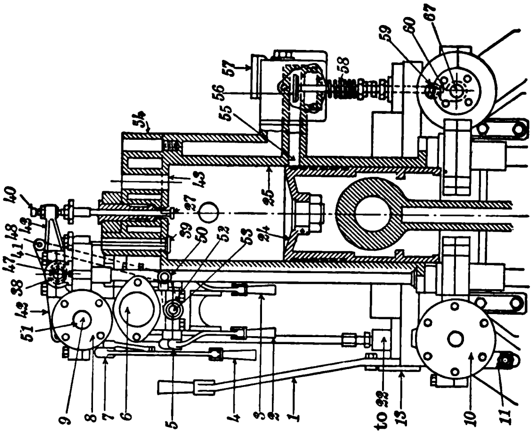


FIG. 524

FIGS. 521-524.—CRAIG-DIESEL FOUR-STROKE REVERSING ENGINE start up under the sole action of compressed air. After a few revolutions the lever *2* is thrown over to the fuel notch *F*, and the starting-air valve is thereby thrown out of action and the fuel valve into action, thus admitting fuel oil to the forward three cylinders. The oil is then ignited by the heat of the air compressed in the cylinders, and the forward three are running under their own power. Lever *3* is then thrown over to *F*, and similarly the three rear cylinders immediately begin to run under their own power. The engine is stopped by cutting off the fuel-oil supply. To reverse the engine, the main lever *1* is thrown over to notch *R* and the other levers, *2* and *3*, are then manipulated exactly as for running ahead.

888. The detail of the mechanism by which the fuel valve is thrown out of action and the starting-air valve into action, and vice versa, is as follows: When lever *2* is thrown over to notch *S* for starting it rotates the outer control shaft *52* through a bevel wheel at *5*. To this control shaft is attached an arm *52–50*, Fig. 524, which moves, in turn, a connecting link *50–48*, an arm *48–49*, Fig. 521, and a rock shaft *47*, which runs in front of the three forward cylinders. This rock shaft has fixed to it two eccentric sheaves for each cylinder, one on which the fuel rocker arm *42* turns, the hub *41* acting as the eccentric strap; and the other on which the starting-air rocker arm *37* turns. The atmospheric air admission and the exhaust rocker arms are mounted concentrically on this shaft *47*. The eccentrics, and shaft *47*, are shown on enlarged scale in Fig. 522. The two eccentric sheaves with centers at *B* and *C* are set at 180° relatively to each other, so that when one of them throws one valve out of action the other eccentric throws the other valve into action. The means by which each eccentric throws its respective valve out of action, thus allowing it to be kept closed by means of a spring, is to raise the hub of the rocker so that when the outer end, say *20*, Fig. 521, is raised and lowered in regular order by the cam *16*, the inner end *40* will move up and down, but will not move down far enough to touch the valve stem with which it comes in pressure contact only.

889. The pipe at *6*, Fig. 523, is for the admission of atmospheric or fuel air. The opening at *43*, Fig. 524, is for a pop-safety valve and by-pass for taking indicator cards. One fuel pump serves for the six cylinders and is driven from the back cam shaft *67*. From the pump the oil goes to a distributor where it is divided into six streams by calibrated needle valves. A relief cam shaft which is used to relieve the high compression when making valve adjustments, etc., is shown at *51*, and it has mounted on it two relief cams side by

side as indicated at *32*, one for forward and the other for reverse running. This relief cam shaft is shifted longitudinally by lever *4* so as to bring either the forward or reverse cam under the roller *33*, which roller is attached to the exhaust rocker arm *34*. Thus the outer end of the arm is lifted by the cam on each compression stroke. The lifting of the outer end of the rocker arm causes the inner end to move down, and opens the exhaust valve indicated at *36*, Fig. 521.

Comparison of Prime Movers

890. An interesting diagram showing superposed indicator cards for the three most recent types of reciprocating engines and also the relative piston displacements and clearance volumes is shown in Fig. 525. This diagram, which is referred to in paragraph 839, gives a scale of cylinder pressures so that the maximum and average

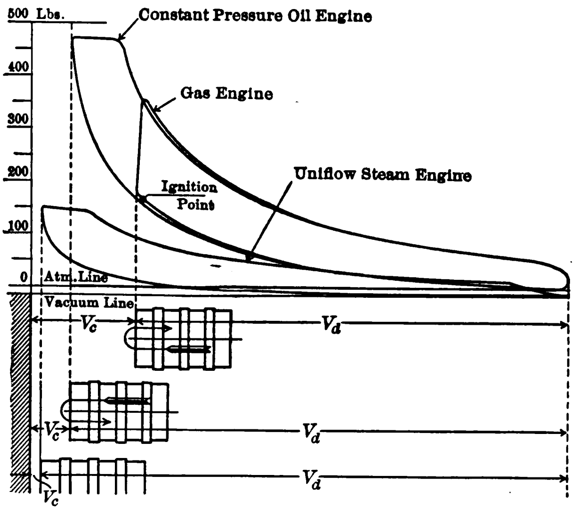


FIG. 525.—COMPARATIVE INDICATOR CARDS OF DIESEL, GAS, AND UNIFLOW ENGINES

pressure values for the high-compression oil engine (Diesel) for the producer gas engine and for the uniflow steam engine may be readily compared.

891. An approximate rating of the various prime movers in general use, based on thermal efficiency and on the cost of fuel per brake horse-power per hour is given in the following table, and diagrammatically in Figs. 526 and 527. In making up this table the test data for steam engines were taken from Gebhardt or Barrus,

the producer gas engine from Mathot, the gasoline engine from Dr. Riedler's reports, the Diesel oil engines from Wells & Tayler, and the low-compression oil engine from manufacturer's report. The other data were taken from practice or assumed. The steam in the single-cylinder multiple-valve engine was at 40° superheat, and in

	Type of Engine	Efficiency Based on Heat in Fuel	Fuel per Brake Horse-Power per Hour	Cost of Fuel per Brake Horse-Power per Hour
		Per Cent	Pounds	Cents
1.	High-Compression Oil (Diesel)	38.2	0.37	0.102
2.	Low-Compression Oil	28.3	0.50	0.137
3.	Producer Gas	23.8	0.765	0.103
4.	Locomobile (Superheat)	17.5	1.04	0.139
5.	Triple Expansion	16.6	1.10	0.148
6.	Single Cylinder (Uniflow)	15.6	1.165	0.156
7.	Compound	15.0	1.21	0.162
8.	Steam Turbine	14.0	1.30	0.174
9.	Gasoline Auto	22.0	0.645	1.10
10.	Single-Cylinder, Multiple-Valve	8.6	2.10	0.282
11.	Single-Cylinder, Single-Valve	6.1	2.98	0.400

the compound at 296°; the boiler efficiencies were taken at 70 per cent; coal at 0.134 cent per pound or $3 per long ton; gasoline at 1.7 cents per pound or 10 cents per gallon for 0.71 specific gravity; oil at 0.275 cent per pound or $1.07 per barrel of 50 gallons for 0.936 specific gravity; heating value of coal 14,000, and oil and gasoline each at 18,000 British thermal units per pound. The results are:

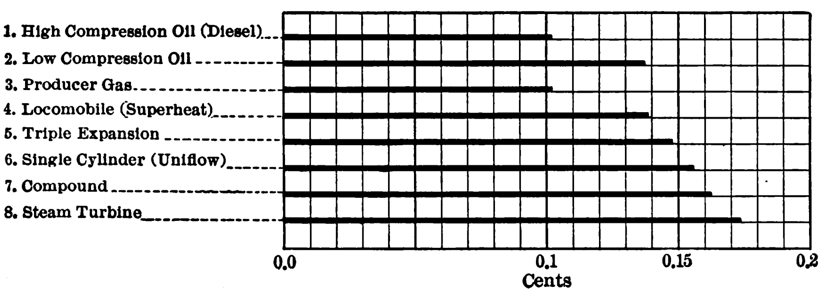


FIG. 526.—COMPARISON OF COSTS OF FUEL PER BRAKE HORSE-POWER PER HOUR FOR SEVERAL TYPES OF HIGH-POWERED STEAM AND INTERNAL-COMBUSTION ENGINES

892. It is not to be understood from the above table that an engine with high thermal efficiency and low cost of fuel is best for a

given installation, all things considered. On the contrary, the conditions may easily be such that an engine having low thermal efficiency and even high fuel cost may be the most practical for a given service. Some of these conditions will be referred to in paragraph 894. A full and intelligent comprehension of them involves the subjects of thermodynamics and power engineering to which numerous texts and reference books are devoted. Also in the above table and in Figs. 526 and 527 the engines with reference numbers from 1 to 8 are for the higher-powered prime movers. Those from 9 to 11 are for smaller powers. In all of these types of engines the efficiencies will be lower and the costs much higher in the small sizes.

893. In Fig. 527 the dash-line curve is based on the use of oil under the boiler instead of coal where steam engines are concerned.

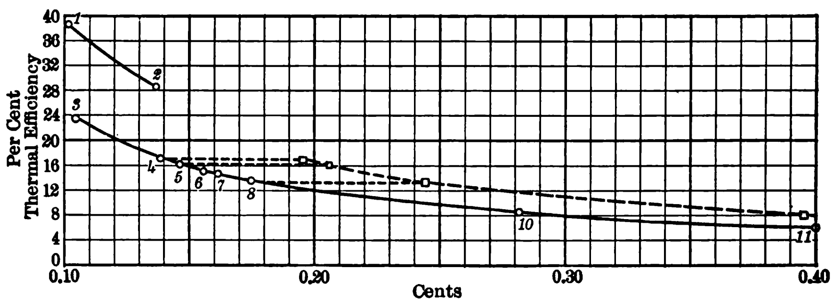


FIG. 527.—DIAGRAM SHOWING THERMAL EFFICIENCY AND COST OF FUEL PER BRAKE HORSE-POWER PER HOUR FOR SEVERAL TYPES OF PRIME MOVERS. DASH LINE IS FOR OIL INSTEAD OF STEAM UNDER BOILER

It is shown there, for example, that the cost for one brake horse-power per hour for a highly efficient triple-expansion engine is .148 cent when coal at $3 per long ton is used, and that it is .206 cent when oil at $1.07 a barrel of 50 gallons is used as boiler fuel. The efficiency of an oil-fired boiler is here taken as 8/7 of that of a coal-fired boiler. When the two curves coincide, oil and coal for boiler fuel will cost the same per brake horse-power per hour. This occurs, under the conditions assumed in this comparison, approximately when 4 barrels of oil cost the same as one ton (2,240 pounds) of coal.

894. The initial investment for the complete power plant, the costs of maintenance, labor, and fuel in the particular locality where the installation is to be made, the nature of the service, the floor space required, etc., must all be considered in selecting a type of engine. Taking into account these and other items controlled by

local conditions it will be found that there are useful fields in which each of the prime movers may excel in ultimate economy. With this in mind the student may better appreciate the special features of valves and valve gears, if any, that are likely to be most used in future power development.

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