M. W. HALL.

INTERNAL COMBUSTION ENGINE.

APPLICATION FILED AUG. 11, 1917.

1,284,190.

Patented Nov. 5, 1918.

3 SHEETS—SHEET 1.

Fig. 1.

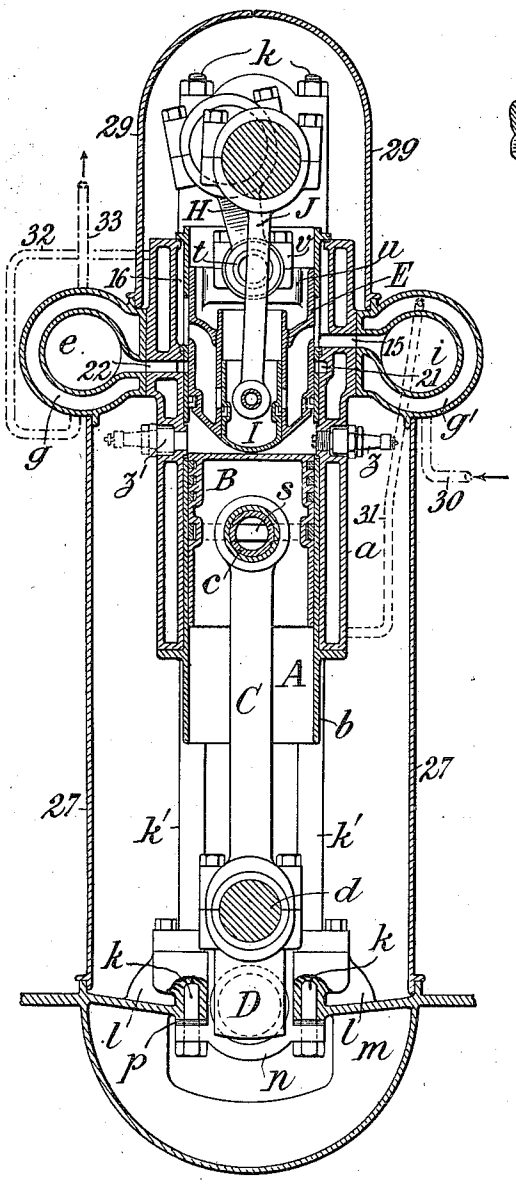
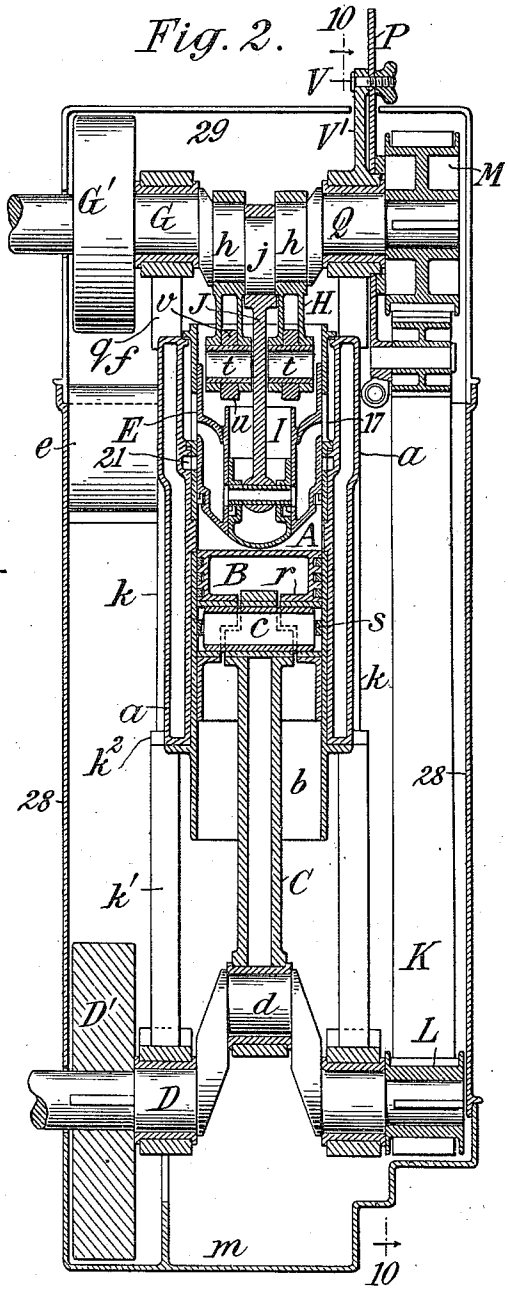


Fig. 2.



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3 SHEETS—SHEET 2.

Fig. 3.

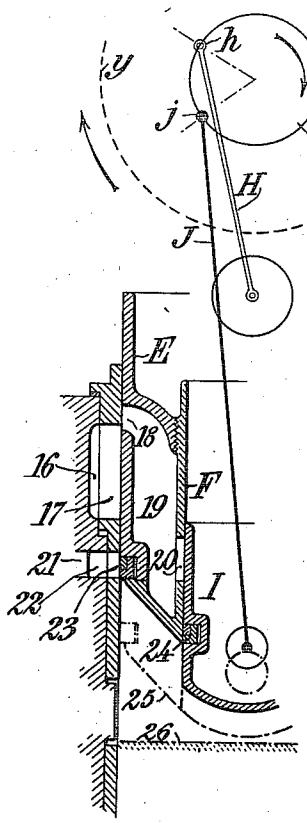


Fig. 4.

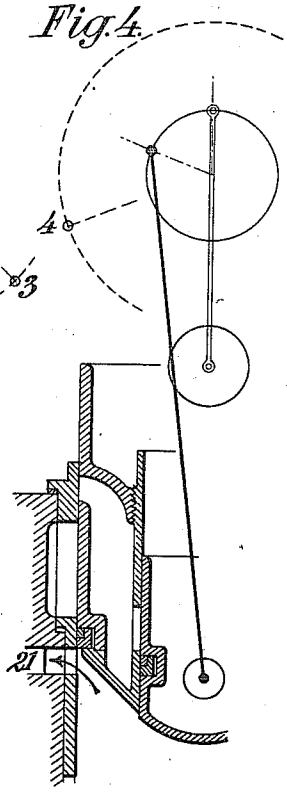


Fig. 5.

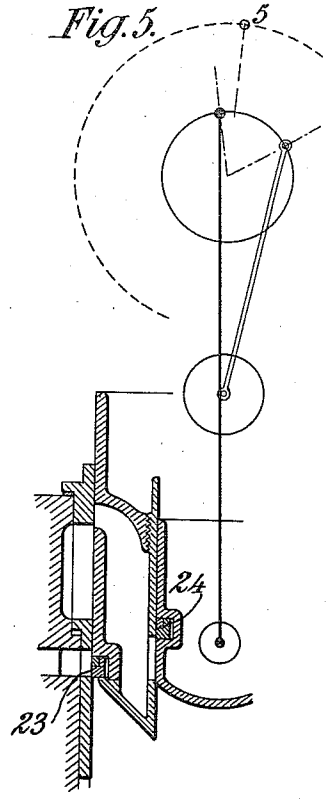


Fig. 10.

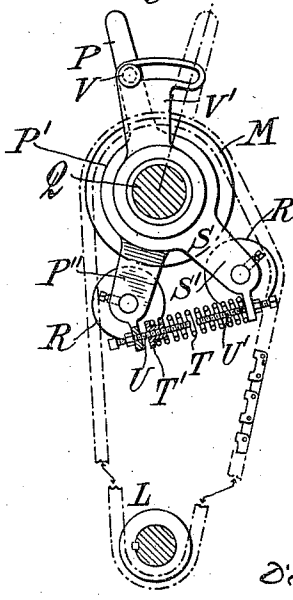


Fig. 11.

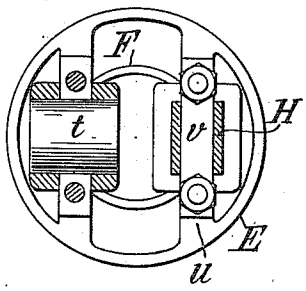
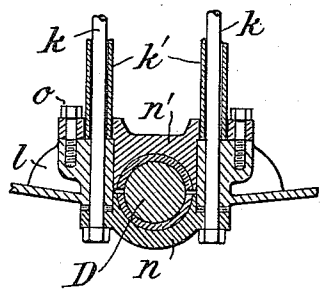


Fig. 12.



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3 SHEETS—SHEET 3.

Fig. 6.

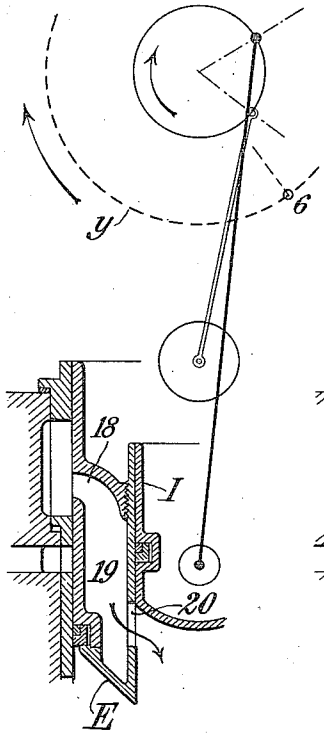


Fig. 7.

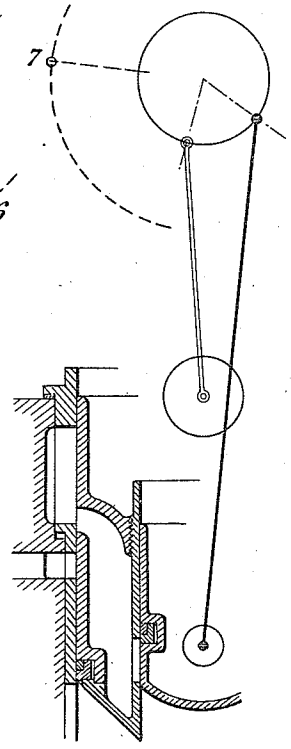


Fig. 8.

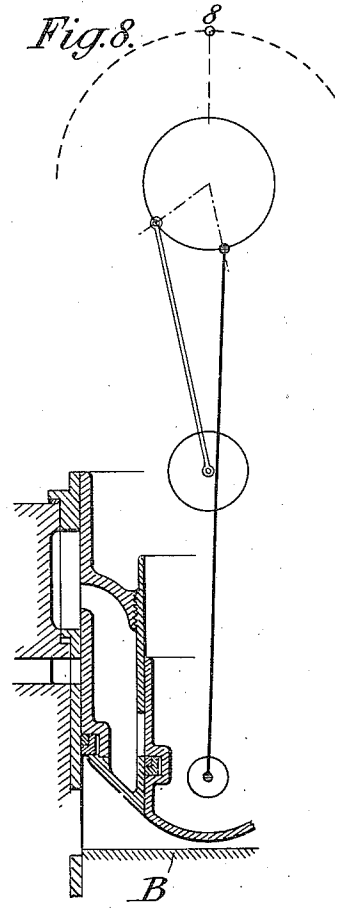
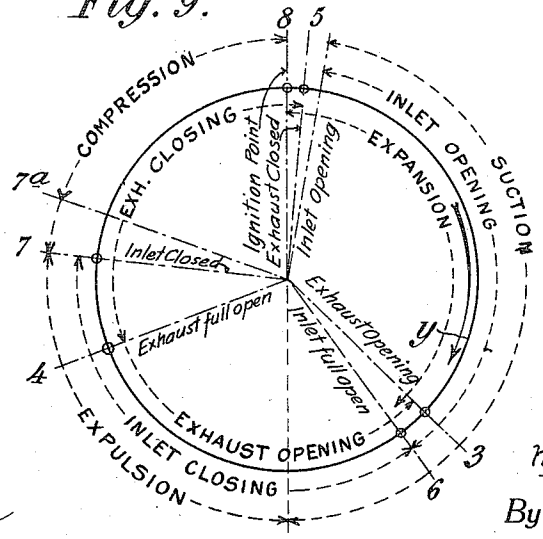


Fig. 9.



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UNITED STATES PATENT OFFICE.

MILAN W. HALL, OF NEW YORK, N. Y., ASSIGNOR, BY MESNE ASSIGNMENTS, TO PISTON-VALVE MOTOR COMPANY, A CORPORATION OF DELAWARE.

INTERNAL-COMBUSTION ENGINE.

1,284,190. Specification of Letters Patent. Patented Nov. 5, 1918.

Application filed August 11, 1917. Serial No. 185,680.

To all whom it may concern:

Be it known that I, MILAN W. HALL, a citizen of the United States of America, residing in the borough of Manhattan, city, county, and State of New York, have invented certain new and useful Improvements in Internal-Combustion Engines, of which the following is a specification.

This invention relates to internal combustion engines for combustible or explosive gas or vapor, of the four-cycle type.

The objects of the invention are the production of a motor with a positively actuated and reliable valve-mechanism, adapted for high speed, and for the attaining of high efficiency.

The invention is characterized by the construction of the inlet and exhaust valves as pistons working the one within the other and receiving the pressure of the expanding gases, under such conditions that a sensible proportion of the total power of the motor is developed against these piston valves. The valves are connected to a valve crank shaft, which is suitably geared to the main crank shaft so as to turn at half the speed of the latter, as is typical of four-cycle motors. Thus the valves, instead of being moved by power taken from the main crank shaft, are impelled forcibly by the explosions and become power-generating members of the engine. In a suitable embodiment of the invention as much as twenty per cent. of the total power of the motor is developed by the expansion of the burning gases against these piston valves.

The invention affords a means of operating under high compression, in such manner as to attain the greatest expansion, and also of avoiding excessive cooling loss, resulting in important economy of fuel. It also provides means for regulating the compression at will, so that it may be brought to the closest practical approach to the point of pre-ignition.

The preferred embodiment of the invention is shown in the accompanying drawings, wherein,—

Figure 1 is a vertical transverse section;

Fig. 2 is a vertical longitudinal mid-section;

Figs. 3 to 8 inclusive, are fragmentary diagrammatic views, being half sections through the valves and valve ports, and illustrating successive positions in the cycle of the engine;

Fig. 9 is a diagram showing the crank path of the main crank shaft, and the points of opening and closing of the respective valves;

Fig. 10 is a fragmentary section taken approximately in the planes of the line 10—10 in Fig. 2, showing means for regulating the compression;

Fig. 11 is a plan of the exhaust valve, partially in section, showing the construction of the pitman connections therewith;

Fig. 12 is a section through the crank shaft bearing.

The drawings illustrate a single cylinder engine, but it is to be understood that the invention may be applied to engines having any of the customary number of cylinders, as, for example, two, three, four, six, eight, twelve, etc.

In the drawings, A is the engine cylinder, shown as made up of a jacketed casting *a* and lining *b;* B is the piston or plunger, connected by the usual pitman C with the crank *d* on the crank shaft D. The pitman connects with the plunger by means of a wrist pin *c* which for lightness is shown as being tubular. The cylinder is extended upwardly considerably above the top stroke of the plunger, and within this extended portion, which is jacketed and has a lining *f*, slides a plunger-shaped piston valve E which constitutes the exhaust valve. This valve is formed annularly, having an inner cylindrical portion F within which moves a plunger-shaped piston valve I constituting the inlet valve. The exhaust valve E and inlet valve I are connected to cranks or eccentrics on the valve shaft G by means of pitmen H and J respectively. To balance the construction there are two pitmen H engaging twin eccentrics *h h*, and between them one pitman J engaging a single eccentric *j*. These eccentrics are shown as having equal throw, although this is not essential. In the diagrams Figs. 3–8, only the eccentric centers are shown, the circle *x* indicating the path of these centers. Of course, cranks may be used instead of eccentrics.

The inlet conduit *i* (or in a multi-cylinder engine the inlet manifold) is extended hori-

zontally along one side of the engine, and the exhaust conduit *e* (or manifold) is extended along the opposite side. These inlet and exhaust conduits are provided with water jackets *g g'*. From the inlet conduit *i* there leads an inlet passage 15 extending through the cylinder jacket *a* and communicating with an annular cavity 16 between this jacket and the lining *f*, and thence with a prolonged port opening 17 in this lining, whence the inflowing mixture passes by a port 18 in the outer side of the exhaust valve E into an annular chamber 19 in said valve, and thence by an inlet port 20, when this port is uncovered by the inlet valve I, (see Fig. 6) into the cylinder.

The exhaust conduit *e* communicates with the cylinder through an exhaust port 21 which is best extended annularly around the cylinder through its lining *f* and co-inciding groove in the jacketed casting, and through a communicating exhaust passage 22.

The valve pistons E and I are best provided with packing rings 23 and 24 respectively, to make tight joints with the cylindrical linings through which are formed the ports 20 and 21 respectively.

It will be understood that the valves E and I are essentially reciprocally-acting slide-valves which for convenience and symmetry are developed annularly into cylindrical form. The reciprocal operation of these valves with respect to the plunger may now be followed, it being understood that the valve shaft G is in any manner connected to the crank shaft D so as to revolve at half the speed of the latter, as is typical with four-cycle motors.

The diagrams Figs. 3 to 8 inclusive, show the positions of the valves at those positions of the power piston or plunger which correspond to the crank positions shown in Fig. 9. In Fig. 9 the crank positions corresponding to the valve positions of the preceding diagrams are indicated by the figure numbers thereof. In Fig. 9 the circle *y* indicates the crank path; and in Figs. 3–8 the dotted circle *y* indicates a portion of this crank path (but of smaller radius) superposed around the circle *x* to show on each figure the angular position of the crank which corresponds to the valve positions shown.

Let us start on the assumption that the explosive mixture has been drawn in and compressed, and is at the point of ignition. This is the condition shown in Fig. 8, and in Fig. 3 the dotted lines 25, 26 show respectively the lower position of the valves and the upper position of the plunger at the point of ignition. During the power stroke the plunger B descends to the crank position 3 in Fig. 9, at which point the exhaust valve begins to open, this being the position shown in Fig. 3. During this effective portion of the power stroke the respective valves are moved up from the positions indicated by the dotted line 25 to those indicated by the full lines in Fig. 3. In so moving they are impelled by the pressure of the burning gases and communicate power through their pitmen H and J to the crank or eccentric centers, whereby a certain fraction of the power generated by the engine is communicated to the valve shaft.

While the crank shaft is turning from positions 3 to 4 (Fig. 9) the valves are moving from the positions shown in Fig. 3 to those shown in Fig. 4. During this movement the exhaust valve is opening and the exhaust or scavenging operation is inaugurated. This exhaust is continued during the remaining ascent of the plunger, during which time the exhaust is closing, this closure being completed at approximately the top stroke of the plunger or preferably shortly thereafter, as indicated by crank position 5 in Fig. 9, at which point the valves are in the positions shown in Fig. 5, the exhaust having just closed and the inlet being about to open. The opening of the inlet which thereupon ensues, is effected mainly by the downward movement of the exhaust valve, which carries the port 20 down beneath the bottom of the inlet valve I; as this movement continues the valve I moves down, although more slowly, until the full open position of the inlet shown in Fig. 6 is attained, at which time the main crank is in position 6, Fig. 9. During the descent of the plunger from position 5, through position 6, to its bottom stroke, the explosive mixture is being drawn in from the inlet conduit through ports 15—20. These ports remain partially open during a portion of the up stroke of the plunger and while the inlet valve is closing; that is to say, until the valves reach the position shown in Fig. 7, at which time the crank has reached the point 7 in Fig. 9. During this portion of the up-stroke of the plunger its effect is to expel a portion of the mixture previously drawn in by forcing it back through the inlet ports into the inlet conduit *i*. The effect of this partial expulsion of the charge is both to insure a more exactly measured charge of mixture at all speeds, and to delay the period of compression. While some compression commences toward the closing of the inlet valve, that is to say, in advance of the position 7, the major compression takes place during the ascent of the plunger from point 7 to the top of its stroke indicated by crank position 8 in Fig. 9, which is the position of ignition, being that shown in Fig. 8. Thereupon the compressed charge is ignited and the operation just described is repeated.

A study of the valve movements during this cycle indicates that during the intake or suction stroke both valves are descending, so

that the suction is due wholly to the descent of the plunger. During the period of expulsion of excess mixture the valve E is nearly stationary and the valve I is descending, so that the latter valve contributes to the expulsion. During the period of compression (positions 7 to 8) the valve E is rising and the valve I is descending. Toward the close of the compression the packing rings 24 are passed beneath the inlet ports 20 (see Fig. 8) so as to make a tight joint and prevent leakage of the compressed mixture back through the inlet ports; and this condition continues during the entire power stroke, *i. e.*, from the positions of Fig. 8 to those of Fig. 3.

The displacement of the valve piston during the effective portion of the power stroke may be seen by comparing the dotted line 25 with the full lines indicating the bottoms of the valves in Fig. 3. During this displacement the valves become power pistons which are being forced up by the pressure, and power is thus delivered to the valve shaft. The power thus transmitted through the valve pistons is in proportion to their displacement as compared with the coincident displacement of the plunger. As the valve shaft and crank shaft are geared positively together, the power thus imparted to the valve shaft may be transmitted through such gearing to the crank shaft; or conversely, the power from the crank shaft may be transmitted to the valve shaft. In a multi-cylinder engine having sufficient units, the power impulses to the valve shaft may become continuous, or even overlap, so that a steady flow of power is imparted to the valve shaft. Power may be taken from the engine by means of either or both these shafts, for which purpose both shafts are shown in Fig. 2 as projecting out through the inclosing casing of the engine.

The feature last referred to is of advantage for certain uses where it is desirable to avail of two different speeds generated by the engine. Thus, for example, in aviation work, where it has been found that the most efficient propeller speed is about 1,200 revolutions per minute, the propeller may be driven from the valve shaft at that speed while the bulk of the power is developed at the main crank shaft at a speed of 2,400 R. P. M., whereby the weight of the motor per horse power is greatly diminished as compared with an engine the crank shaft of which turns at the preferable speed for the propeller.

With an engine of the proportions shown in the drawings, the division of power generated between the crank shaft and valve shaft is approximately as 4 to 1; or, in other words, the valve shaft generates about twenty per cent. of the power. These proportions may, of course, be greatly varied.

This invention thus provides a new type of internal combustion engine in which, instead of applying tappet valves or slide valves which are driven solely by power taken from the crank shaft, and which themselves have no part in the effective generation of power,—the new construction renders the valves and valve shaft an appreciable part of the power-generating unit. Thus, frictional losses are avoided and the efficiency of the engine is increased. The engine embodies also a degree of simplicity heretofore unattained in four-cycle motors. The construction of gas-impelled valve-mechanism thus provided by this invention enables much higher speed to be maintained than is possible with motors of conventional design; in fact, its speed possibilities are limited only by perfection of design, material, workmanship, lubrication, dynamic balancing, and other minor details.

Another feature in which this invention has important advantages is in the securing of an unusual degree of fuel economy. To this end it avoids in an important degree the deficiencies of ordinary internal combustion engines as respects compression, expansion, and cooling loss. Compression as a rule being maintained in existing engines at a point only about half of that which is possible, it results that an expansion of only three to four volumes can be attained. Excessive cooling loss being a concomitant of low compression and limited expansion ratio, contributes to a low economy.

In the improved engine these defects have been practically eliminated. To this end the motor is designed to operate normally at the highest possible compression permissible with a pre-mixed charge of fuel and air, means being provided to maintain the compression at a point just below that of pre-ignition. The timing of the inlet valve whereby it remains open after the suction stroke and enables a portion of the indrawn mixture to be re-expelled, results in the measuring of a correct volume of the mixture, so that after compression and at the moment of explosion, the compressed mixture bears the correct ratio to the volume of the expansion stroke of the motor (considering both the main plunger and the valve pistons), whereby a much greater ratio of expansion is available than with engines as heretofore constructed.

The reduction of cooling loss is accomplished principally by the diminution in the volume of the required combustion space within the cylinder, together with such concentration of this space that the least possible external wall surface is presented to the cooling medium. The engine has no fixed cylinder head with its cooling jacket. Instead, the valve pistons E I occupy the place of a cylinder head and these pistons

are cooled solely by the circulation of mixture through the chamber 19 in the outer piston. This has the double effect of cooling the valve pistons and of heating the incoming mixture.

The combined economies thus effected by operating the motor normally at maximum compression, and utilizing the greatest available expansion and reducing the loss in cooling, result in a reduction of fuel consumed of from fifty to sixty per cent.

Maximum compression is practically attainable only by compressing as closely as possible to the point of self-ignition of the charge, and this can be attained in practice only by providing means for controlling (either at will or automatically) the compression so as to keep it just below such point. To accomplish this some means for varying the extent of compression during the running of the engine is requisite. As the compression is determined by the portion of the compression stroke of the plunger intervening between the closure of the inlet valve and the completion of the plunger stroke at the igniting point, a convenient means of varying the compression is available by changing the valve timing so as to be able to lead or lag the inlet valve closure relatively to the crank shaft while the engine is running, so that such timing as will best afford the desired compression may be availed of under any existing running conditions. This result may be accomplished in various ways, but most conveniently by varying the running relation of the valve shaft G to the crank shaft D so as to give the valve shaft more or less lead or lag, as the case may be, relatively to the crank shaft. A convenient means of accomplishing this result which has several practical advantages over other mechanical expedients that might readily be resorted to, will now be described.

The connection between the shafts D and G, shown in the drawings, is by means of a so-called silent chain K running on sprocket wheels L and M having teeth in the proportion of 1 to 2 and fixed respectively on the shafts D and G. Such a chain as ordinarily applied would hold the shafts in an invariably rotative relation. But with the supplemental means best shown in Fig. 10, this relation may be varied so as to bring about the desired retardation or lag of the valve shaft relatively to the crank shaft. For this purpose the chain is given a slack corresponding to the extent of rotative displacement desired (which may be as much as 15° for the valve shaft, corresponding to 30° for the crank shaft), and a device in the nature of a double belt tightener is provided for taking up this slack on either one side or the other of the chain. This device in its illustrated form is applied as a lever P having an annular portion P′ which turns around a support provided for it on one of the bearings Q for the valve shaft G. This lever has an arm P″ which carries the belt-tightening loose pulley R. On another parallel ring or annular hub S which encircles the same bearing Q, is formed an arm S′ carrying a belt-tightening pulley R′. The arms P″ and S′ are forced apart by a spring T which is compressed between projecting lugs formed on the arms so as to provide for a slight yielding of the pulleys R R′ sufficient to take up the slack effectively, the spring having sufficient stress to keep the chain tight during the normally maximum pull or transmission of energy between the shafts. Two screws U U′ are provided, which are adjustable with respect to the lever arms, and the ends of which may strike together and form an absolute stop in the case of an abnormal pull on the chain. For adjusting the stress of the spring one end thereof rests against a nut T′ which may be screwed in or out on one of the screws U U′ in order thereby to more or less compress the spring. The lever P may be set in any angular position by means of a set nut or clamp V engaging an arc-shaped slot in a fixed bracket V′ so as to hold the lever P in any adjustment.

Let us assume that the relative positions shown in full lines in Fig. 10 are such as to cause the opening and closing of the valves to the varying positions indicated on the diagram Fig. 9, and in corresponding positions shown in the diagrams Figs. 3 to 8 inclusive; and that these positions correspond to the maximum compression which is practicable, that is to say, just below the point of pre-ignition. If then the operator finds that pre-ignition is occurring, he has, in order to avoid this, to make such adjustment as will retard the valve shaft. This is accomplished by releasing the lever P and moving it over more or less toward the right or dotted line position. This has the effect of letting off chain on the right and taking up chain on the left, which, if done while the crank shaft is held stationary, would turn the valve shaft backward any number of degrees up to the maximum (say 15°). The operator should, in normal running so retard or lag the valve shaft to an extent only sufficient to avoid liability to pre-ignition. Having found this point, he should clamp the lever P fast in its new position. Such retardation of the valve shaft has the essential effect of retarding the point in the crank shaft at which the inlet valve closes, so that instead of closing in the position 7 in Fig. 9, it may close in a higher position, as, for example, that shown at 7^a in Fig. 9. This prolongs the

period of free expulsion of the mixture and retards the beginning of the compression and consequently cuts down the portion of the stroke during which compression can occur, so that compression is reduced.

The described construction also is capable of imparting to the engine the capacity for self-regulation of the compression at starting. Assuming that a self-starter or a crank is applied to turn the shaft D, the pull therefrom straightens the chain on the right-hand side by the yielding of the spring T, so that while the lever P and pulley R remain stationary, the lever arm S′ and the pulley R′ yield to the extent of the gap between the screws U U′. This action causes the valve shaft to lag to a corresponding extent, thus closing the inlet at a later point in the compression stroke and reducing the compression, as is desirable in starting. It is mainly for the purpose of regulating this starting operation that the adjustable screws U U′ and spring adjustment T′ are provided.

Some desirable details of construction remain to be described.

For rigidly connecting together the parts of the engine, tie rods *k k* are provided having nuts at their opposite ends holding in position the bearings which support respectively the crank shaft and valve shaft. The lower base of the engine is shown as a casting *m* which forms the crank case and has transverse brackets *l* approaching each other for supporting the crank shaft bearings. These brackets *l* are perforated vertically, forming sockets for the passage of the tie rods *k*. Beneath the brackets are received projecting ears of a bearing block *n* constituting the lower half of the crank shaft bearing. The upper half *n′* (Fig. 12) of this bearing is fastened down by screws *o* to the tops of the brackets *l*, and constitutes the rigid member of the bearing. The lower member *n* is adjustable for wear by removing shims *p* and substituting thinner ones. The greater part of the wear is against the lower members *n*. The tie rods *k* pass up through tubular struts *k′* and ears *k²* formed on the cylinder casting *a*, and are continued upward above the cylinder casting through tubular bearing supports *q* preferably constructed as a separate member and integral with the lower members of the bearings for the shaft G.

The power pitman C is shown as connected with the plunger B by a preferably tubular wrist pin *c*, the pitman head being widened on the lower side to give the maximum contact surface for transmission of power and narrowed on the upper side to give less contact surface for pulling down the plunger during the suction stroke; and the plunger having a wrist pin housing *r* which is reciprocally widened at the lower side and narrowed at the upper side to give it the maximum bearing surface against the pin during the power stroke. The pin is held in position longitudinally so that its ends cannot come into contact with the cylinder lining, by means of a ring *s* which is sprung around the plunger, entering a groove therein, and which crosses the opposite ends of the pin. This ring *s* might constitute one of the packing rings of the plunger.

The pitmen H engage short wrist pins *t t*, preferably tubular, which are clamped at their middles with cross bars or bridges *u* cast as part of the valve piston E, being held rigidly down upon these bridges by clamping caps *v* similar to bearing caps. The pitmen H are forked and engage the protruding ends of the pins *t*. In Fig. 11 the valve piston E is shown in plan with the pitmen H in section, that on the left just beneath the cap *v* which is removed, and that on the right just above the cap.

In Fig. 1 a spark plug *z* is shown in place. On the opposite side of the cylinder a socket *z′* is shown which may receive another spark plug, as shown in dotted lines, or may be closed by a solid plug.

For a single-cylinder engine it is desirable to provide both the crank shaft D and valve shaft G with fly wheels shown at D′ and G′, respectively, but for a multi-cylinder engine one or both of these may be omitted.

It is desirable to case in the engine and for this purpose a casing is shown consisting of side plates 27, 27 (Fig. 1), end plates 28, 28 (Fig. 2), and top plates 29, 29 (Figs. 1 and 2), these latter meeting at the center and forming hoods housing the ends. The inlet and exhaust manifolds are shown as projecting outside of the casing.

It is preferable to introduce the cooling water around the intake manifold, then to circulate it through the jacket *g′*, then to conduct it to the bottom of the jacket in the cylinder casting *a* and from the top of this jacket to circulate it through the exhaust jacket *g*. This flow is shown diagrammatically in Fig. 1 in dotted lines, 30 being the inlet to *g′*, 31 the duct leading from the top of the latter to the bottom of the cylinder jacket, 32 a duct from the top of said jacket to the bottom of *g*, and 33 the outlet to the radiator.

The purpose of introducing the coolest water around the intake manifold is to cool the mixture before its admission through the inlet ports and into the annular chamber 19, as this cooling of the intake mixture is relied upon to cool the valves. This is especially desirable because the reëxpulsion from the inlet valve of some of the mixture during the return stroke of the plunger drives out some residue of the heated spent gases

mixed with the expelled combustible mixture, and this latter is also heated in circulating into and out of the cylinder and through the valves, so that it becomes desirable to cool the gases by the circulation of cool water through the intake jacket.

It is to be understood that the invention is not limited to the details of construction shown and that the general arrangement, proportions and engineering design may be considerably modified without departing from the invention.

What I claim is:—

1. In an engine of the described type having a main cylinder and piston valves within said cylinder affording the sole control for the inlet and exhaust, and a valve shaft operatively connected to said valves, whereby the piston valves are adapted to recede during the power stroke and thereby communicate power to the valve shaft, said valves coöperating to control the inlet, and one of said valves acting alone to control the exhaust.

2. In an engine of the described type having a main cylinder and piston valves within said cylinder affording sole control for the inlet and exhaust, and a valve shaft to which said valves are connected, a piston and crank shaft to which said valve shaft is connected said piston valves and shaft being related to cause the piston valves to recede during the power stroke, and thereby to communicate power to the valve shaft, said valves coöperating to control the inlet, and one of said valves acting alone to control the exhaust.

3. In an engine of the described type having piston valves affording sole control for the inlet and exhaust, valve actuating cranks to which said valves are connected, a piston and power shaft, and a connection between said power shafts and valve actuating cranks whereby the piston valves have continuous uninterrupted movement coinciding with the crank throw, and said valves coöperating to control the inlet, and one of said valves acting alone to control the exhaust.

4. In an engine of the character described, the combination of a cylinder having inlet and exhaust ports, piston valves within the cylinder which afford the sole control for the inlet and exhaust, a valve shaft operatively connected to said valves, a piston and crank shaft to which said valve shaft is operatively connected, the said valves being relatively movable and constructed and arranged whereby they coöperate to control the inlet, and whereby one of said valves acts alone to control the exhaust.

5. In an engine of the character described, the combination of a cylinder having inlet and exhaust ports, valve mechanism within the cylinder affording the sole control for the inlet and exhaust, said valve mechanism including relatively movable concentrically arranged cylindrical pistons, an actuating shaft for said pistons, and said pistons being constructed and arranged whereby in the action of said shaft they coöperate to control the inlet, and whereby one of said pistons acts alone to control the exhaust.

6. In an engine of the character described, a cylinder having separate inlet and exhaust ports, the combination of piston valve members arranged one within the other in said cylinder and affording the sole control for the inlet and exhaust, a shaft for said members, a piston and crank shaft, said crank shaft having connection with the valve shaft whereby said valves are positively actuated in both directions by the crank shaft, and said valves being constructed and arranged whereby they coöperate to control the admission through the inlet, and whereby one of said valves acts alone to control the exhaust.

In witness whereof, I have hereunto signed my name.

MILAN W. HALL.