May 20, 1947. C. L. HOLMES 2,420,779

OPPOSED PISTON ENGINE

Filed April 10, 1944 3 Sheets-Sheet 1

Fig. 1

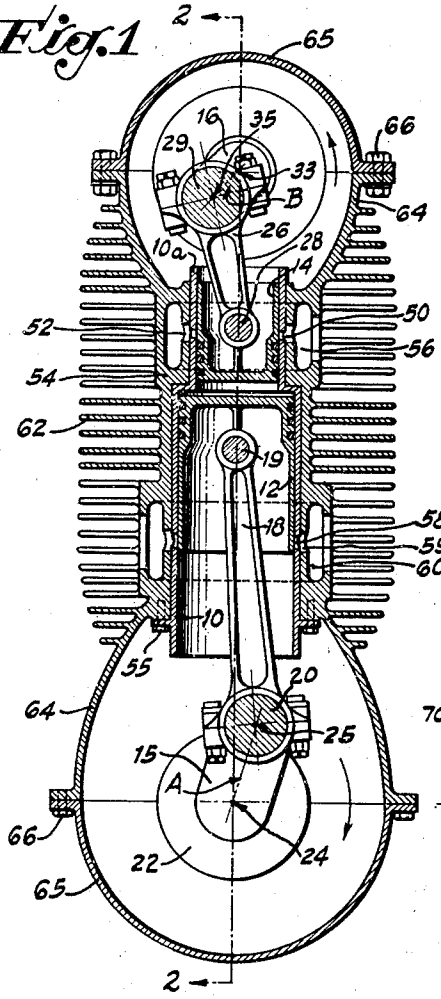


Fig. 2

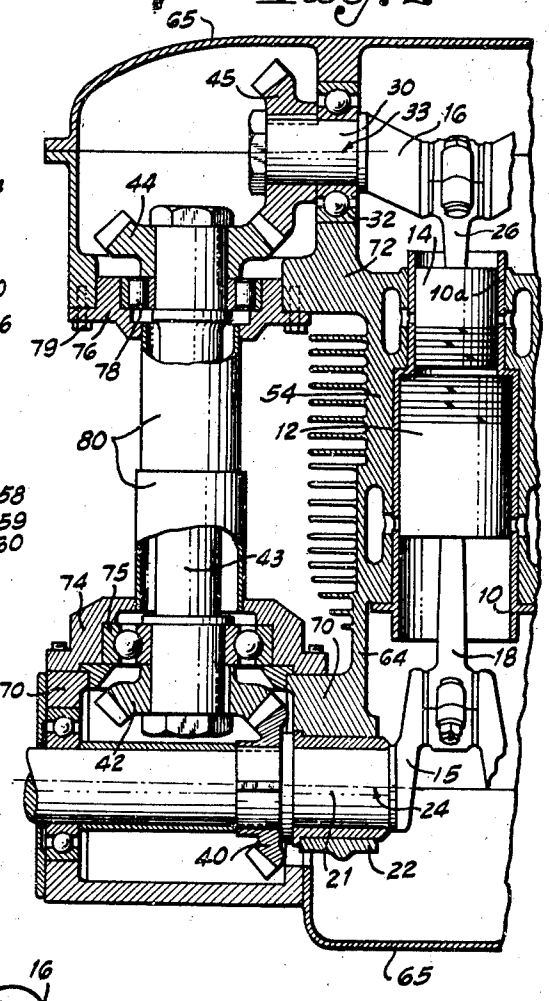


Fig. 3

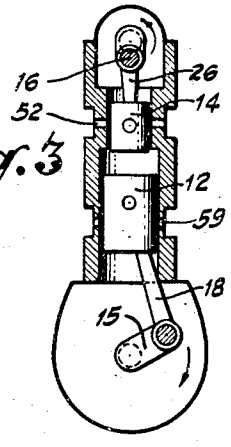
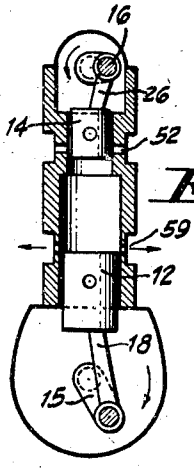


Fig. 4



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Fig. 5

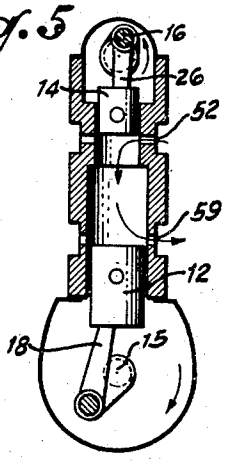


Fig. 7

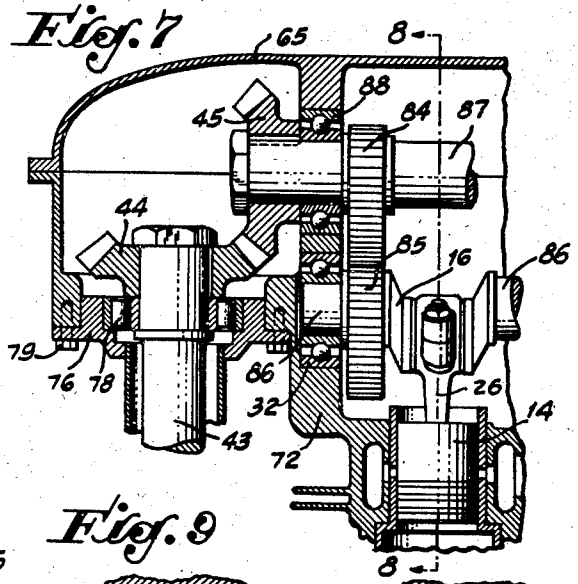


Fig. 8

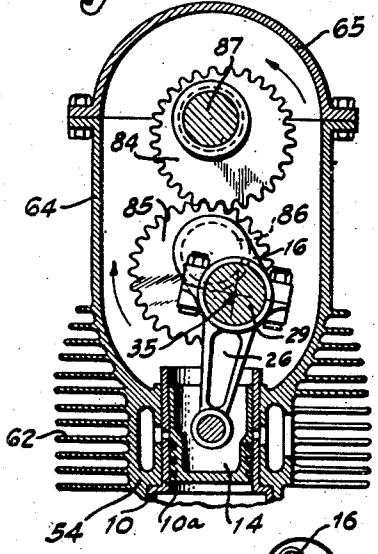
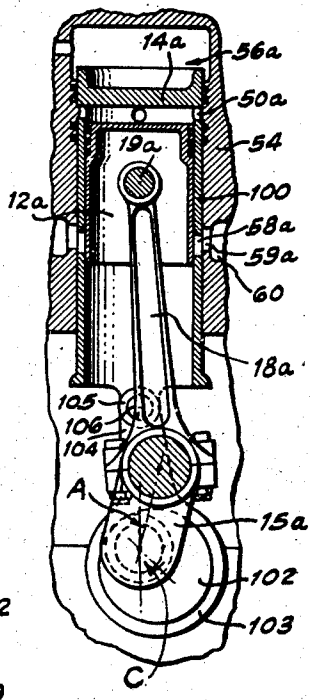


Fig. 9



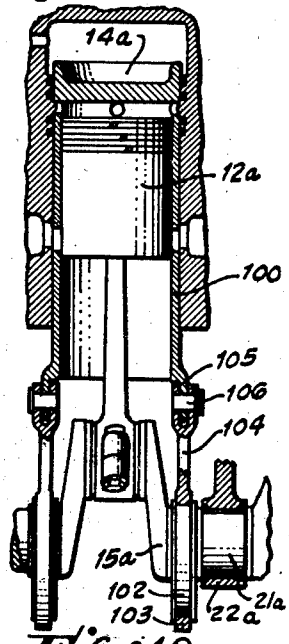
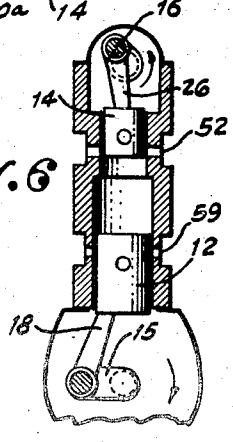


Fig. 10

Fig. 6



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Fig. 11

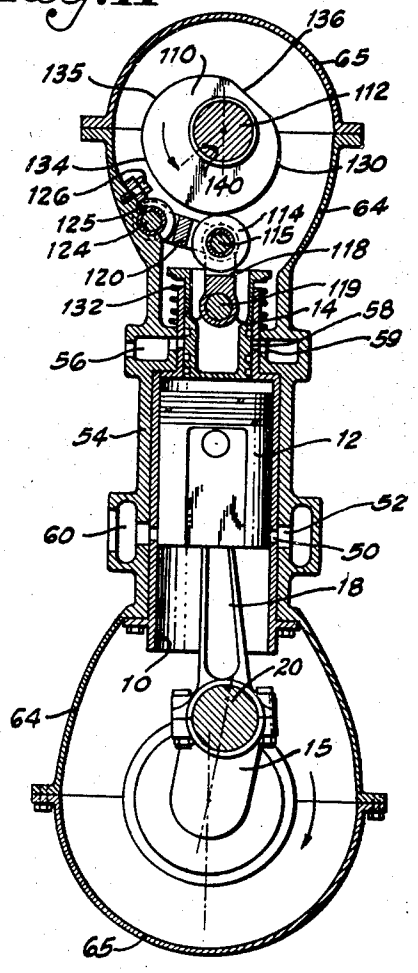
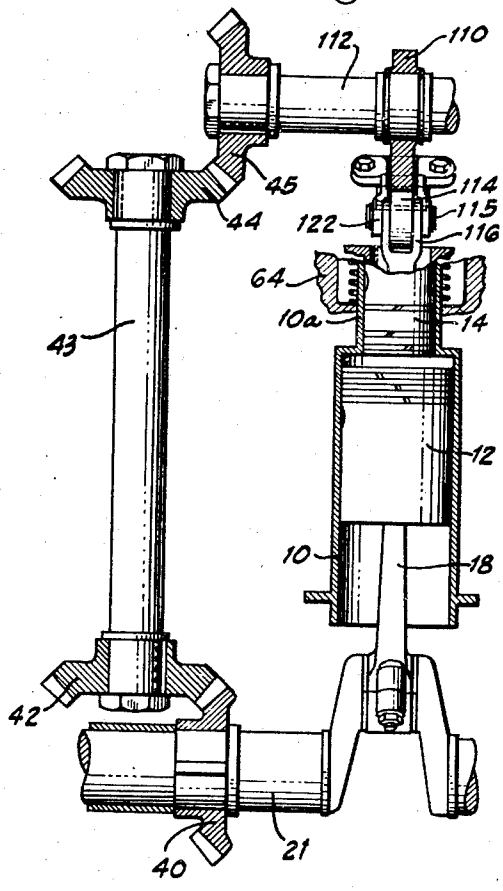


Fig. 12



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Patented May 20, 1947

2,420,779

UNITED STATES PATENT OFFICE

2,420,779

OPPOSED PISTON ENGINE

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Application April 10, 1944, Serial No. 530,246

31 Claims. (Cl. 123—51)

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This invention relates to two-cycle internal combustion engines of the compression ignition type and especially to opposed piston engines where one of the pistons trails the other.

One object of the invention is to increase power output, and generally to improve efficiency of internal combustion engines.

Heretofore, in employing opposed piston internal combustion engines, where a trailing piston has been used in connection with a leading working piston, the angular relationships of crank arms driving the two pistons has been such as to fail in production of maximum power output. For example, in some instances much of the power has been lost against the trailing piston, or the working piston has led the trailing piston at the time of maximum compression so far that much of the power has been lost.

Another object of this invention, therefore, is to provide for proper balancing of the leverages of the working piston crank arm and the trailing piston crank arm at the time of maximum compression, and to provide an angular relationship between the crank arms of the two pistons such that the trailing piston subsequent to maximum compression will follow the working piston through a substantial portion of its power stroke, whereby to cause a maintenance of effective power as close to the maximum as possible. To make the engine more efficient, the trailing piston is made smaller in diameter than the working piston so that the force of each explosion is much greater on the working piston than on the trailing piston at the position of maximum compression and through the power stroke.

Another feature resides in using a shorter crank arm on the trailing piston in order to produce the stated function of controlling the leverage. By employing these characteristics in the trailing piston, the compression produced by the working piston as it approaches and reaches top dead center is increased by the smaller piston which at that time moves at substantially its maximum rate of speed, and then continues on to an optimum position of maximum compression. This position of maximum compression, or the beginning of the work stroke, is an optimum position of the working piston past top dead center wherein the leverages of the cranks are equal and the piston velocities are equal.

A further object of my invention is to provide an engine in which the pistons are set to obtain a compression ratio in accordance with the flash point of the individual fuel, and in which the

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combustible mixture is raised in temperature by compression between the opposed pistons to the flash point of the fuel whereby automatic ignition takes place. This does away with the necessity of providing spark plugs and the usual ignition system now used on internal combustion engines.

Another important object and feature of the invention is to so relate the angularity of the crank arms of the two pistons when in their positions of maximum compression as to obtain the most work from the power stroke, both from the standpoint of an optimum condition and from the standpoint of a practical range of operating conditions. The optimum condition appears to be that existing when the crank arm of the working piston is approximately 15° past its top dead center when the pistons are in their positions of maximum compression, the working piston leading the trailing piston approximately 60°, whereby the trailing piston is about 45° short of dead center at its position of maximum compression. A practical working range appears to exist when the crank arm of the working piston is anywhere between about 10° and about 25° past top dead center when the pistons are in their positions of maximum compression, the working piston leading the trailing piston within a range between about 40° and about 70°, the greater lead being employed especially as the position of the working piston past dead center increases toward 25° at its position of maximum compression.

Another inventive feature resides in an engine block having opposed pistons wherein a crankshaft of a working piston drives a crankshaft of a trailing piston through a drive shaft and gear train, the drive shaft bearings therefor and connected gears of the gear train being readily detachable and removable from a housing carried by the block.

In the accompanying drawings, wherein certain embodiments are disclosed for purposes of illustration only:

Fig. 1 is a longitudinal section taken axially through an engine constructed and arranged according to a preferred form of the invention;

Fig. 2 is principally a longitudinal section, parts being shown in elevation, taken approximately from the line 2—2 of Fig. 1, or at right angles to Fig. 1, and showing one means for actuating the pistons;

Figs. 3, 4, 5, and 6 are longitudinal sections corresponding to Fig. 1 at four successive posi-

tions following that of Fig. 1, and representing a complete cycle of operation.

Fig. 7 is a fragmentary section corresponding with the upper portion of Fig. 2 and showing a modified drive for the trailing piston;

Fig. 8 is a longitudinal sectional detail taken from the line 8—8 of Fig. 7 and corresponding with the upper portion of Fig. 1;

Figs. 9 and 10 are longitudinal sections at right angles to each other through a modified form of opposed piston engine embodying some of the present improvements;

Fig. 11 is a longitudinal section through another modified form; and

Fig. 12 is a similar view taken at right angles to that of Fig. 11.

According to the preferred form of the invention illustrated in Figs. 1 to 6, an engine cylinder 10 houses a reciprocating working piston 12, and a cylinder extension 10a houses a trailing piston 14 of smaller diameter than that of the piston 12. The larger piston 12 is connected with a longer crank arm 15 which gives it a greater stroke or throw than is given the smaller piston 14 which is driven by a shorter crank arm 16.

The working piston 12, which is the power piston, drives its crank arm 15 by means of a conventional connecting rod 18, one end of which is mounted on a wrist pin 19 carried by the piston 12, and the other end of which is mounted in a conventional manner upon a connecting rod bearing 20 carried by the crank arm 15 on a crankshaft 21 which is journaled in suitable bearings 22 to rotate about an axis 24. Thus, rotation of the center 25 of the connecting rod bearing 20 about the axis 24 of the crankshaft 21 measures the movement of the working piston 12 and its throw. Similarly, the trailing piston 14 is actuated by means of a connecting rod 26, one end of which is carried on a wrist pin 28 and the other end of which is mounted on a connecting rod bearing 29 carried by the crank arm 16 which in turn is carried by a crankshaft 30 working in bearings 32 about an axis 33. Thus, the travel of the center 35 of the connecting rod bearing 29 about the axis 33 of the crankshaft 30 measures the extent of the stroke or throw of the trailing piston 14.

The trailing piston 14 is driven from the working piston 12 in unison therewith, that is, by a 1:1 gear ratio, and with a relative angular set of the crank arms 15 and 16 to yield the required results as hereinafter described.

In the driving mechanism illustrated in Fig. 2, a bevel gear 40 secured on the crankshaft 21 meshes with a bevel gear 42 secured on a transmission shaft 43 carrying at its upper end a bevel gear 44 which meshes with a companion bevel gear 45 secured on the crankshaft 30. The lengths of the time cycles of the working piston and the trailing piston 14 are identical.

Since it is an object of the invention that the effective leverage of the two crank arms shall be equal at the time of maximum compression, and since it is a further object that the trailing piston 14 shall follow the working piston 12 from their positions of maximum compression for as great a portion of the power stroke as possible, and since it is a further object to place the crank arm of the working piston 12 at an optimum angle beyond dead center, it is necessary so to dispose the angular set of the crank arms 15 and 16 with relation to each other at the position of the pistons under maximum compression, as illustrated in Fig. 1, as to accomplish the indicated results. I have discovered that the desired effect can be obtained by setting the crank arm 15 of the working piston 12 at a point within a short angular range past top dead center, which shall be the position of maximum compression and which represents the best point for the commencement of the power stroke of the working piston 12. Thus, the trailing piston is so set in angular relation to the working piston that the working piston on the power stroke is traveling at an accelerated speed while the trailing piston is moving at a much slower rate and stops at dead center to act as a relatively fixed wall to the exploded gases. At the same time it is necessary to set the crank arm 16 of the trailing piston 14 so that it will have a maximum rate of travel during the interval when the working piston 12 is near top dead center and will continue to compress a fuel charge trapped between the pistons after the working piston 12 has practically ceased to travel.

I have found that the angle indicated at A between the center line of crank arm 15 and the axis of the cylinder, which passes through the axis 24 of the crankshaft 21, has an optimum value of approximately 15° when the pistons 12 and 14 are in their positions of maximum compression. In other words, the crank arm 15 has a position 15° past top dead center when the parts are in positions of maximum compression. However, a satisfactory operating range lies between about 10° and about 25° past top dead center, and in any case somewhat less than 45° past top dead center when the pistons are at maximum compression. With the piston 14 trailing, the total optimum lead of the crank arm 15 of the working piston 12 over the trailing crank arm 16 of the trailing piston 14 is found to be approximately 60°. Thus, with crank arms 15 and 16 of satisfactory length and the crank arm 15 being in a position 15° past dead center when the pistons are in their positions of maximum compression, the crank arm 16 of the piston 14 will be trailing about 45° behind the axis of the cylinder, which corresponds with the section line 2—2 of Fig. 1, this angularity of the crank arm 16 being represented by the angle B formed between the axis of the cylinder and the center line of the crank arm 16 passing through the center point 35. While the optimum lead of the crank arm 15 over the crank arm 16 is approximately 60°, a satisfactory range appears to be between about 40° and about 70°, and in any event somewhat less than 90°.

In addition to the function of the trailing piston 14 to produce compression of the fuel charge between the pistons 12 and 14, it also serves the purpose of opening and closing a series of fuel inlet ports 50 annularly arranged around the cylinder. The ports 50 communicate with a corresponding series of ports 52 in the engine block 54 into which the cylinder 10 is fitted as a liner and in which it may be held as by means of screws 55. The ports 50 and 52 communicate with an annular fuel supply passage 56 in the engine block 54 to which passage 56 a suitable fuel mixture is applied under pressure by any means, not shown, such as a centrifugal pump. The piston 12 in turn operates to control an annular series of ports 58 and 59 in the cylinder 10 and engine block 54 respectively which lead to an annular exhaust passage 60 vented in any desired manner.

As shown, the engine block 54 may be provided with a plurality of radial fins 62 for dissipation of heat to the air. The opposite ends of the engine block 54 may be in the form of crankcases 64 to which suitable covers 65 are bolted as indicated at 66. For the purpose of carrying the gear train 40, 42, 44, and 45 and the transmission shaft 43, one crankcase 64 is provided with a laterally disposed housing 70 for the gears 40 and 42, and the other crankcase 64 is provided with a laterally disposed housing 72 for the gears 44 and 45, these housings 70 and 72, together with the crankcases 64 and their covers 65 carrying the bearings 22 and 32 for the crankshafts 21 and 30. In order that the transmission shaft 43 and the bevel gears 42 and 44 mounted thereon may be removed, a cap 74 which carries bearing parts 75 for the end of the shaft 43 adjacent the gear 42, is detachably mounted on the housing 70. Similarly, a cap 76 is detachably secured to the housing 72 and carries bearings 78 adjacent the bevel gear 44 for positioning the corresponding end of the shaft 43. As indicated, the bearing elements 78 are roller bearings which facilitate removal of the parts. Thus, when the caps 74 and 76 are disconnected by removing screws 79, the cap 76 may be dropped down so that the bevel gear 44 may be raised and moved sidewise until the bevel gear 42 is lifted out of the housing 70, whereupon the bevel gear 44 and the shaft 43 together with a telescopic housing 80 for the shaft 43, may be dropped down and removed.

In Figs. 7 and 8 there is shown a modified drive for the trailing piston 14 which is adapted to speed up the travel of the trailing piston as it approaches its dead center. This is accomplished by employing two eccentric gears 84 and 85. The gear 85 is eccentrically mounted on a stub crankshaft 86 which carries the crank arm 16 for the trailing piston 14. The eccentric gear 84 is carried upon an operating shaft 87 journaled in suitable bearings 88 in the housing 72 and the cover 65, the bevel gear 45 being secured to one end of the shaft 87. Otherwise, in this modification the parts are arranged as in the form of Figs. 1 and 2. It will be apparent that as the eccentric gear 84 is rotated through the gear train including the bevel gear 45, it will impart additional speed of rotation to the stub crankshaft 86 when the trailing piston 14 is approaching its dead center position, this relation of the parts being shown in Fig. 8.

In Figs. 9 and 10 another modification is shown wherein both pistons are driven from the same crankshaft. Here the working piston 12*a* actuates the connecting rod 18*a* and imparts rotation to the crank arm 15*a* on the crankshaft 21*a*, as in the form of Figs. 1 and 2. However, a trailing sleeve piston 14*a*, corresponding with piston 14, instead of being actuated by means of a connecting rod and crankshaft similar to that shown in Figs. 1 and 2, has an integral sleeve 100 which operates in a cylinder formed in the engine block 54, the sleeve 100 serving also as a liner in which the working piston 12*a* reciprocates. The sleeve 100 is driven from the crankshaft 21*a* through the medium of an eccentric 102 serving as a crank and working in a circular ring 103 provided with an actuating arm 104 pivoted to an extension 105 on the end of the sleeve 100 as indicated at 106, a second oppositely disposed extension 105 being also provided for connection of a similar drive.

The sleeve 100 is provided with exhaust ports 59*a* adapted to register when in the exhaust position with exhaust ports 59*a* in the engine block 54, which exhaust ports communicate with the annular exhaust passage 60. The extreme inner end of the piston sleeve 100 just within the piston head 14*a* is provided with inlet ports 50*a* adapted to communicate with a fuel supply chamber 56*a* corresponding in general with the annular fuel supply passage 56 of the form of Figs. 1 and 2, fuel under pressure being supplied to the passage 56*a* by a centrifugal pump or otherwise, not shown. With the arrangement shown, the trailing piston 14*a* operates with respect to the working piston 12*a* in the same manner as the trailing piston 14 of Figs. 1 and 2 operates with respect to the working piston 12, the sleeve 100 acting somewhat as a sleeve valve and being ported to provide for intake and exhaust at proper stages of the cycle, as will be apparent to those skilled in the art.

As shown in Fig. 9, the angle of the crank arm 15*a* with respect to the axis of the cylinder is the same as that shown in Fig. 1, and establishes the angle A as about 10° at the position of maximum compression shown. The piston 12*a* is shown as having a lead of 60° over the piston 14*a*, the crank arm 15*a* of the working piston 12*a* being set at 10° past top dead center and at its position of maximum compression, and the effective angle of the eccentric 102 as represented by the angle C being 50° short of its dead center.

For operation of the construction of Figs. 1 to 6, inclusive, the working piston 12 and the trailing piston 14 are so set that, in their positions of maximum compression, the angularity of the crank arm 15 of the working piston 12 with respect to the crank arm 16 of the trailing piston 14 represents an optimum condition. This optimum condition is illustrated in Fig. 1, where the center line of the crank arm 15 is set at 15° past dead center as represented by the angle A and has a lead of 60° over the crank arm 16, whereby the crank arm 16 is in a position 45° short of dead center as represented by the angle B existing between the center line of the crank arm 16 and the axis of the cylinder. Here the trailing crank arm 16 is shorter than the crank arm 15, and the diameter of the trailing piston 14 is smaller than the diameter of the working piston 12.

Also, the trailing crank arm 16 and the working crank arm 15 are so set that the leverages are equal at their positions of maximum compression. These conditions are represented by the following formula, the angles being the optimum angular positions of the cranks at the moment of maximum compression:

Sine angle A×length of crank arm 15
=sine angle B×length of crank arm 16

In other words, the sine of the included angle generated by the working crank arm 15 as it moves from top dead center to an optimum angular position, times the length of the working crank arm 15, is equal to the sine of the included angle generated by the trailing crank arm 16 as it moves to dead center from an optimum angular relation to the working crank, times the length of the trailing crank arm 16, as shown in Fig. 1.

The centers 25 and 35 of the connecting rod bearings 20 and 29 are, therefore, equidistant from the axis of the cylinder and the pistons. Where the gear train 40, 42, 44, 45 rotates the crank arms 15 and 16 in opposite directions, as indicated by the arrows in Fig. 1, the centers 25 and 35 are necessarily on opposite sides of the

axis of the cylinder when the pistons 12 and 14 are in their positions of maximum compression. Where a drive is employed as in Figs. 7 and 8 so that the crank arms 15 and 16 are rotated in the same direction, then the centers 25 and 35 will lie on a line parallel with the axis of the cylinder when the pistons 12 and 14 are in their positions of maximum compression.

Therefore, in order for the leverages to be equal at the moment of maximum compression, it is necessary for the trailing crank arm 16 to be shorter than the crank arm 15, thereby maintaining maximum effective power through the power stroke. The trailing crank arm 16 follows the working crank arm 15 through the power stroke for 45° and yields higher power efficiency than the present conventional type engine.

To make the engine more efficient, the trailing piston 14 is smaller in diameter, thus using less effort to bring the pistons to the position of maximum compression. Since the working piston 12 reaches top dead center 15° before maximum compression, maximum compression is accomplished by the trailing piston 14. At the moment of impulse, or at the beginning of the power stroke, the force on the pistons is directly proportional to the area of the pistons, and the thrust therefore is much greater on the larger working piston. By this construction, which is shown in Fig. 1, a large portion of the surge of power at the moment of impulse is made immediately effective on the power crank shaft when at about 15° past top dead center. This feature makes this engine far superior to the present conventional engine because the conventional engine's position of maximum compression is at top dead center.

A further result of the smaller trailing piston and the lesser effort needed on the compression stroke is a higher compression engine which makes it possible to ignite the fuel by compressing the charge to the ignition point.

Although the leverages of the crank arms 15 and 16 are the same when they are in their positions of maximum compression, the force of the explosion exerted on the piston 12 is greater than on the trailing piston 14 because the diameter of the trailing piston 14 is smaller than that of the working piston 12, thereby reducing the area of its piston head, and, hence, the amount of power which it exerts on its cranks when under the influence of compression and power application. In addition to employing the angular sets at the optimum conditions as above indicated, the angular set of the crank arm 15 of the working piston 12 may be varied between about 10° and about 25° past dead center at the position of maximum compression, the total lead of the working piston 12 being between about 40° and about 70°. In general, the total lead will be larger as the angular set of the crank arm 15 beyond top dead center at the position of maximum compression is increased toward 25°.

When operating the engine shown in Figs. 1 to 6, with the respective crank arm relations described, compression ignition will take place when the pistons are approximately in the positions shown in Fig. 1 or at a time slightly prior to their reaching such positions. The position of Fig. 1 indicates the commencement of the power stroke, the crank arms 15 and 16 moving in the direction of the arrows. Fig. 3 illustrates diagrammatically the position of the pistons 12 and 14 when they have traveled 45° from the position of Fig. 1, and Fig. 4 indicates the position of the pistons 12 and 14 after they have traveled through the principal portion of the power stroke, the position of Fig. 4 representing 90° of travel from the position of Fig. 3. At the position of Fig. 4 the exhaust ports 59 are uncovered. In traveling from the position of Fig. 1 to the position of Fig. 3, the trailing piston 14 will have followed the working piston 12 for the purpose of maintaining as far as possible the high compression accompanying the early portion of the power stroke. Thereafter, the trailing piston 14 begins to recede as the working piston or power piston 12 continues on its work stroke.

Following complete uncovering of the exhaust ports 59 by the piston 12, as indicated in Fig. 4, these ports remain completely open for an appreciable interval and then remain partially uncovered for a further interval. During the latter portion of this period the trailing piston 14 recedes to partially uncover the fuel inlet ports 52, with the result that the fuel which is introduced under pressure serves to scavenge the residual exhaust gases, both the inlet ports and the exhaust ports being open for approximately 30° of rotation. This condition is indicated in Fig. 5. By the time that the scavenging operation has reached a desired stage, the working piston 12 will have moved on its compression stroke to close the exhaust ports 59 as indicated in Fig. 6, at which stage the crank arm 15 will have traveled approximately 45° from the position of Fig. 5, and the trailing piston 14 will have closed the intake ports 52 to initiate compression.

As the working piston 12 is approaching its top dead center position and passing thereover, the trailing piston 14 is traveling at its maximum rate and, therefore, continues to compress the injected fuel even though compression movement of the piston 12 has ceased. Under these conditions any appropriate liquid fuel will be ignited by reason of the ability of the engine upon proper setting to produce compression ignition for any such fuel. The angular relation of the two pistons is so set that the power piston completes most of its accelerating motion on the power stroke while the trailing piston continues its slower velocity over dead center, acting as a comparatively fixed wall to the exploded gases. Then on the compression stroke the opposite is the case; as the power piston is approaching its top dead center position and passing thereover, the trailing piston is traveling at its faster rate and, therefore overtakes the power piston at the optimum position of maximum compression after dead center.

In the operation of the form of Figs. 7 and 8, the same results are obtained, with the further advantage that the eccentric gears 84 and 85 cause the trailing piston 14 to approach the power piston 12 more quickly and produce maximum compression more quickly than is accomplished with the engine shown in Fig. 1.

With the form of Figs. 9 and 10, the relation of the leverages so that the leverage of the working crank arm 15*a* shall not exceed that of the eccentric 102 is attained solely by means of the total lead of the working crank arm 15*a* over the eccentric 102, the crank arm 15*a* of the working piston 12*a* having an angularity of about 10° past top dead center at the maximum compression position. Here the crank arm 15*a* leads the eccentric 102, which is effective as a crank arm, in the same manner that the crank arm 15 leads crank arm 16 of the other forms, a lead of about 60° being illustrated. The eccentric 102 causes

trailing piston 14a to move with relation to the working piston 12a in exactly the same manner as the trailing piston 14 of the other forms moves with relation to the working piston 12.

Each of the forms herein disclosed is adaptable also for use as compressed air and steam engines.

From the foregoing it will be apparent that a two-cycle compression ignition internal combustion engine employing opposed pistons is provided having high efficiency, the working stroke, which follows immediately upon attainment of maximum compression, commencing at a most efficient advance of the working piston past its top dead center.

It will also be apparent from the use of the eccentric 102 in the form of Figs. 9 and 10, that cams, eccentrics, and similar means for imparting reciprocating motion by conversion of rotary motion and vice versa are equivalents of the crankshafts and crank arms of the other forms. Therefore such devices are intended to be included in the terms "cranks" and "crank arms" of the claims. Also, other forms of drive between the upper and lower shafts than the bevel gears illustrated may be employed, such as spur gears, chains, and the like.

The optimum positions of the pistons at the point of maximum compression may be measured also in terms of proportions of their full strokes from their innermost positions. Thus, when the working piston 12 at the position of maximum compression is 15° past its top dead center and the trailing piston 14 is 45° short of its dead center position, the piston 12 has moved past dead center about 2% of its full working stroke and the piston 14 is about 15% to 18% short of the limit of its compression stroke. This is a ratio of about 1:8 measured in terms of the proportion of working stroke traveled and the proportion of compression stroke yet to be traveled at the maximum compression positions.

In the form of Figs. 11 and 12, the same movement of the trailing piston 14 is accomplished by means of a cam 110 and the same ratio of proportions of stroke traveled at maximum compression is maintained thereby as with the cranks and the eccentric of the other forms. Here the same pistons 12 and 14 are employed as in the form of Figs. 1 to 6, as are the same cylinder 10, engine block 54, and associated parts, and the same crank arm 15 connecting the rod 18 and the crankshaft 21 for the piston 12. Similarly, the same arrangements of ports 50, 52, 58, and 59 and fuel inlet and exhaust passages 56 and 60 are employed. The same gear train 40, 42, 44, 45, and gear shaft 43 drive a shaft 112 which carries the cam 110. Cam 110 bears upon a roller 114 mounted upon a pin 115 carried between bifurcations 116 on the upper end of a connecting rod 118 mounted on a wrist pin 119 in the trailing piston 14. The upper end of the connecting rod 118 and the roller 114 are positioned by means of a link 120 which has fingers 122 pivoted on the outer ends of the pin 115, the opposite end of the link 120 being pivotally mounted at 124 on a bearing retained by a bracket 125 and screws 126 on the wall of the adjacent crankcase 64.

The cam 110 is shown as having a propelling section 130 which steadily moves the piston 14 on its compression stroke through the medium of the roller 114 and the connecting rod 118 against the pressure of a spring 132 sufficiently heavy to hold the roller 114 properly against the cam 110 at all times.

The cam section 130, as shown, advances the piston 14 during about 105° of rotation of the shaft 112, and is followed by a constant or holding portion 134 which holds the piston 14 at the inner limit of its stroke for about 90° of rotation of the shaft 112. The portion 134 is followed by a cam section 135 which permits the piston to move on its outward stroke during about 105° of rotation of the shaft 112. This is followed with a flat section 136 representing about 60° of shaft rotation, which permits further outward movement of the piston 14 at an increased rate to retain the intake ports 58 wide open for an adequate interval of time. The section 136 is followed by the section 130 which causes closing of the intake ports 58 and produces the compression stroke.

The cam 110 is so set that maximum compression is obtained when the working piston 12 is at an optimum position, as when its crank 15 is about 10° past dead center or has moved about 2% of its power stroke, and the cam section 134 holds the trailing piston 14 at the inner limit of its stroke from the time when the piston 14 of the form of Figs. 1 to 6 would normally still have about 15% to 18% of its compression stroke to travel, this interval being for about 90° of rotation of the shaft 112, as above indicated.

In this modification the cam section 130 is formed on the arc of a circle drawn about a center indicated at 140 which is on a line through the axis of the shaft 112 and set at an angle of about 45° short of the axis of the cylinder 10 when the trailing piston 14 has been forced down by the cam section 130 to its position of maximum compression, this occurring when the working piston 12 is at its optimum position for maximum compression, e. g., about 10° past top dead center.

This arrangement thus gives the maximum impulse to the piston 12 after it has passed dead center, as in the other forms, but in this form the piston 14 is not to be advanced farther after maximum compression, and its position at maximum compression is to be held during much of the power stroke of the piston 12, for example, during 90° of revolution of the shaft 112 with the cam 110 shown. Therefore, the motion of the piston 14 on its inward stroke is checked and this is accomplished by forming the cam section 134 on the arc of a circle about the axis of the shaft 112 as a center.

Thus, this form produces, in all the essential particulars, the same effect as a crank arm whose center line passes through the axis of the shaft 112 and through the center point 140 when the pistons 12 and 14 are in their maximum compression positions and where such a crank arm trails the crank arm 15 by about 55°, there being the difference, however, that further inward movement of the trailing piston after maximum compression is eliminated.

While I have shown various embodiments which my invention may assume, it is to be understood that these are employed solely for purposes of illustration since they will suggest many modifications to those skilled in the art. Therefore I intend to cover all variations that fall within the scope of the appended claims, as above indicated.

I claim as my invention:

1. In combination in a two-cycle internal combustion engine: a cylinder; a pair of opposed

pistons operating on a 1:1 ratio in said cylinder, one of said pistons constituting a working piston and the other constituting a trailing piston; a relatively long crank operatively connected with said working piston; and a relatively short crank connected with said trailing piston for operating the latter, the long crank being set relative to the short crank so that the working piston leads the trailing piston, and so that the leverages of the two cranks when they are at their positions of maximum compression are substantially equal.

2. A combination according to claim 1 wherein the trailing piston has a smaller diameter than that of the working piston.

3. A combination according to claim 1 wherein the crank of the working piston has a lead of less than 90° over that of the trailing piston.

4. A combination according to claim 1 wherein the crank of the working piston has a lead over that of the trailing piston between about 40° and about 70°.

5. A combination according to claim 1 wherein the crank of the working piston has a lead of about 60° over that of the trailing piston.

6. A combination according to claim 1 wherein the position of the crank of the working piston at maximum compression is about 15° past dead center.

7. A combination according to claim 1 wherein the position of the crank of the working piston at maximum compression is past dead center a distance less than 45°.

8. A combination according to claim 1 wherein the position of the crank of the working piston at maximum compression is between about 10° and about 25° past dead center.

9. A combination according to claim 1 wherein the lead of the crank of the working piston over that of the trailing piston is between about 40° and about 70°, and means are provided to accelerate the travel of the trailing piston as it approaches top dead center.

10. A combination according to claim 1 wherein the position of the crank of the working piston at maximum compression is between about 10° and about 25° past dead center and the lead of the crank of the working piston over that of the trailing piston is about 60°.

11. A combination according to claim 1 wherein the trailing piston has a smaller diameter than the working piston, the lead of the crank of the working piston over that of the trailing piston is about 60°, and the position of the crank of the working piston at maximum compression is about 15° past dead center.

12. In combination in a two-cycle internal combustion engine: a cylinder; a pair of opposed pistons operating in said cylinder and having a common axis, one of said pistons constituting a working piston and the other constituting a trailing piston; a relatively long crank arm operatively connected with said working piston and having its axis of rotation extending at right angles to the common axis; and a relatively short crank arm connected with said trailing piston and having its axis of rotation extending at right angles to the common axis, the two crank arms being connected to operate in unison, the long crank arm being set relative to the short crank arm so that the working piston leads the trailing piston and so that the two crank arms make angles with the common axis so that, when the pistons are at maximum compression, the sine of the angle that the long crank arm makes with the common axis, times the throw of the long crank arm, is equal to the sine of the angle that the short crank arm makes with the common axis, times the throw of the short crank arm.

13. A combination according to claim 12 wherein the trailing piston has a diameter smaller than that of the working piston.

14. A combination according to claim 12 wherein the centers of connection of the crank arms with their respective pistons at their positions of maximum compression are on a line approximately parallel with the axis of the cylinder, and the trailing piston has a diameter smaller than that of the working piston.

15. A combination according to claim 12 wherein the crank arm of the working piston has a lead over that of the trailing piston of between about 40° and about 70°.

16. A combination according to claim 12 wherein the position of the crank arm of the working piston at maximum compression is between about 10° and about 25° past dead center.

17. A combination according to claim 12 wherein the diameter of the trailing piston is smaller than that of the working piston and the crank arm of the working piston has a lead over that of the trailing piston of less than 90°.

18. A combination according to claim 12 wherein the trailing piston has a diameter smaller than that of the working piston and the crank arm of the working piston has a lead over that of the trailing piston of between about 40° and about 70°, and the position of the crank of the working piston at maximum compression is between about 10° and about 25° past dead center.

19. A combination according to claim 12 wherein the centers of connection of the crank arms to the pistons at their positions of maximum compression are approximately equidistant from the axis of the cylinder, and the position of the crank arm of the working piston at maximum compression is approximately 15° past dead center, the trailing piston having a diameter less than that of the working piston.

20. In combination in a two-cycle compression ignition internal combustion engine: a working piston; a trailing piston arranged in opposed relation to the working piston and having a diameter smaller than the diameter of the working piston; a crank arm operatively connected with said working piston; and a second crank arm operatively connected with said trailing piston, said crank arms being operatively connected to work in unison, the crank arm of the working piston being set somewhat past top dead center when the pistons are in their positions of maximum compression and so leading the crank arm of the trailing piston when said pistons are in such positions that the leverage of the crank arm of the trailing piston does not appreciably exceed the leverage of the crank arm of the working piston.

21. In combination in a two-cycle compression ignition internal combustion engine: an engine cylinder; a working piston reciprocatingly mounted in one end of the cylinder; an opposed trailing piston reciprocatingly mounted in the other end of the cylinder; a first crank arm operatively connected with said working piston; and a second crank arm connected with the trailing piston to operate in unison with the first crank arm, the second crank arm being shorter than the first crank arm, the first crank arm being set between about 10° and about 25° beyond its

top dead center when the pistons are in their positions of maximum compression, said first crank arm leading said second crank arm between about 40° and about 70°.

22. In combination in a two-cycle internal combustion engine: a cylinder; a reciprocating closed end sleeve in the cylinder and constituting a trailing piston; a working piston operating in said sleeve and having a common axis with the sleeve; a relatively long crank arm operatively connected with said working piston and having its axis of rotation extending at right angles to the common axis; and a relatively short crank arm connected with said closed end sleeve and having its axis of rotation extended at right angles to the common axis, the long crank arm being set relative to the short crank arm so that the working piston leads the trailing sleeve piston and such that the two crank arms make angles with the common axis so that when the pistons are at maximum compression, the sine of the angle that the long crank arm makes with the common axis, times the length of the long crank arm, is equal to the sine of the angle that the short crank arm makes with the common axis, times the length of the short crank arm.

23. In combination in a two-cycle internal combustion engine: a cylinder; a working piston reciprocably mounted in said cylinder; a trailing piston reciprocably mounted in said cylinder in opposed relation to said working piston; means adapted to impart a relatively long stroke to said working piston; and means adapted to impart a relatively short stroke to said trailing piston, said two means and said pistons being so set relative to one another that they are at their positions of maximum compression when the working piston has moved on its outward stroke to a position corresponding with a position at approximately 15° past dead center of a crank having said long stroke, and said trailing piston has moved on its inward stroke to a position corresponding with a position at approximately 45° short of dead center of a crank arm having said short stroke.

24. In combination in a two-cycle internal combustion engine: a cylinder; a working piston reciprocably mounted in said cylinder; a trailing piston reciprocably mounted in said cylinder in opposed relation to said working piston; means adapted to impart a relatively long stroke to said working piston; and means adapted to impart a relatively short stroke to said trailing piston, said two means and said pistons being so set relative to one another that they are at their positions of maximum compression when the working piston has moved on its outward stroke about 2% of its travel on said outward stroke, and said trailing piston has moved on its inward stroke to a position about 15% to 18% short of its normal inward limit on said inward stroke.

25. In combination in a two-cycle internal combustion engine: a cylinder; a pair of opposed pistons operating in said cylinder, one of said pistons constituting a working piston and the other constituting a trailing piston; a relatively long crank operatively connected with said working piston; a relatively short crank connected with said trailing piston for operating the latter, the long crank being set relative to the short crank so that the working piston leads the trailing piston, and so that the leverages of the two cranks when they are at their positions of maximum compression are substantially equal; and means to accelerate the travel of the trailing piston as it approaches top dead center.

26. In combination in a two-cycle internal combustion engine: cylinder means; a pair of opposed pistons operating in said cylinder means, one of said pistons constituting a working piston and the other constituting a trailing piston; a relatively long crank arm operatively connected with said working piston; and a relatively short crank arm connected with said trailing piston, the two crank arms being connected to operate in unison, the long crank arm being set relative to the short crank arm so that the working piston leads the trailing piston, so that the two crank arms make angles with their respective pistons such that, when the pistons are at maximum compression, the sine of the angle that the long crank arm makes with the axis of its piston, times the throw of the long crank arm, is equal to the sine of the angle that the short crank arm makes with the axis of its piston, times the throw of the short crank arm.

27. In combination in an internal combustion engine: cylinder means; a pair of opposed pistons operating in said cylinder means, one of said pistons constituting a working piston and the other constituting a trailing piston; a relatively long crank operatively connected with said working piston; and a relatively short crank connected with said trailing piston for operating the latter, the long crank being set relative to the short crank so that the working piston leads the trailing piston, and such that the leverages of the two cranks are substantially equal when they are in their positions of maximum compression and the trailing piston after maximum compression will follow the working piston for a substantial portion of the power stroke.

28. In combination in a two-cycle internal combustion engine: cylinder means; a working piston reciprocably mounted in said cylinder means; a trailing piston reciprocably mounted in said cylinder means in opposed relation to said working piston; means adapted to impart a relatively long stroke to said working piston; and means adapted to impart a relatively short stroke to said trailing piston, said two means and said pistons being so set relative to one another that they are at their positions of maximum compression when the working piston has moved on its outward stroke to a position substantially beyond dead center of a crank providing said long stroke, and said trailing piston has moved to the limit of its inward stroke, the means for imparting said short stroke to said trailing piston including means for maintaining said trailing piston at said limit of its inward stroke for a substantial portion of the outward stroke of said working piston.

29. A combination according to claim 28 wherein the means for imparting stroke to said trailing piston is a cam provided with a holding portion adapted to retain the trail piston at the inner limit of its stroke.

30. In combination in a two-cycle internal combustion engine: cylinder means; a pair of opposed pistons operating in said cylinder means, one of said pistons constituting a working piston and the other constituting a smaller trailing piston; a relatively long crank arm operatively connected with said working piston; and a relatively short crank arm connected with said trailing piston, said two crank arms and said pistons being so set relative to one another that when they are at the position of maximum com-

pression the leverages of the two crank arms and the velocity of the pistons are equal, and subsequent to such position the velocity of the trailing piston is much slower than that of the working piston, the trailing piston then coming to rest at dead center and thus serving as a comparatively stationary wall to the exploded gases for a substantial portion of the power stroke.

31. A method of operating an internal combustion engine having a cylinder and a pair of opposed pistons in said cylinder and having operating cranks, one piston constituting a working piston and the other piston being a trailing piston, said method comprising the steps of: moving said pistons so that the working piston leads the trailing piston; causing said working piston to assume a position past top dead center when said pistons are at positions of maximum compression and causing said pistons to have substantially equal leverages in said positions; imparting substantially equal velocities to said pistons when at said positions of maximum compression; moving the trailing piston toward its top dead center immediately following maximum compression, and simultaneously increasing

the velocity of the working piston; and bringing the trailing piston to rest at its top dead center while moving the working piston at substantially its greatest velocity.

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