June 16, 1931. H. W. HOUSTON 1,810,017

VARIABLE STROKE CAM ENGINE

Filed Nov. 20, 1928 4 Sheets-Sheet 1

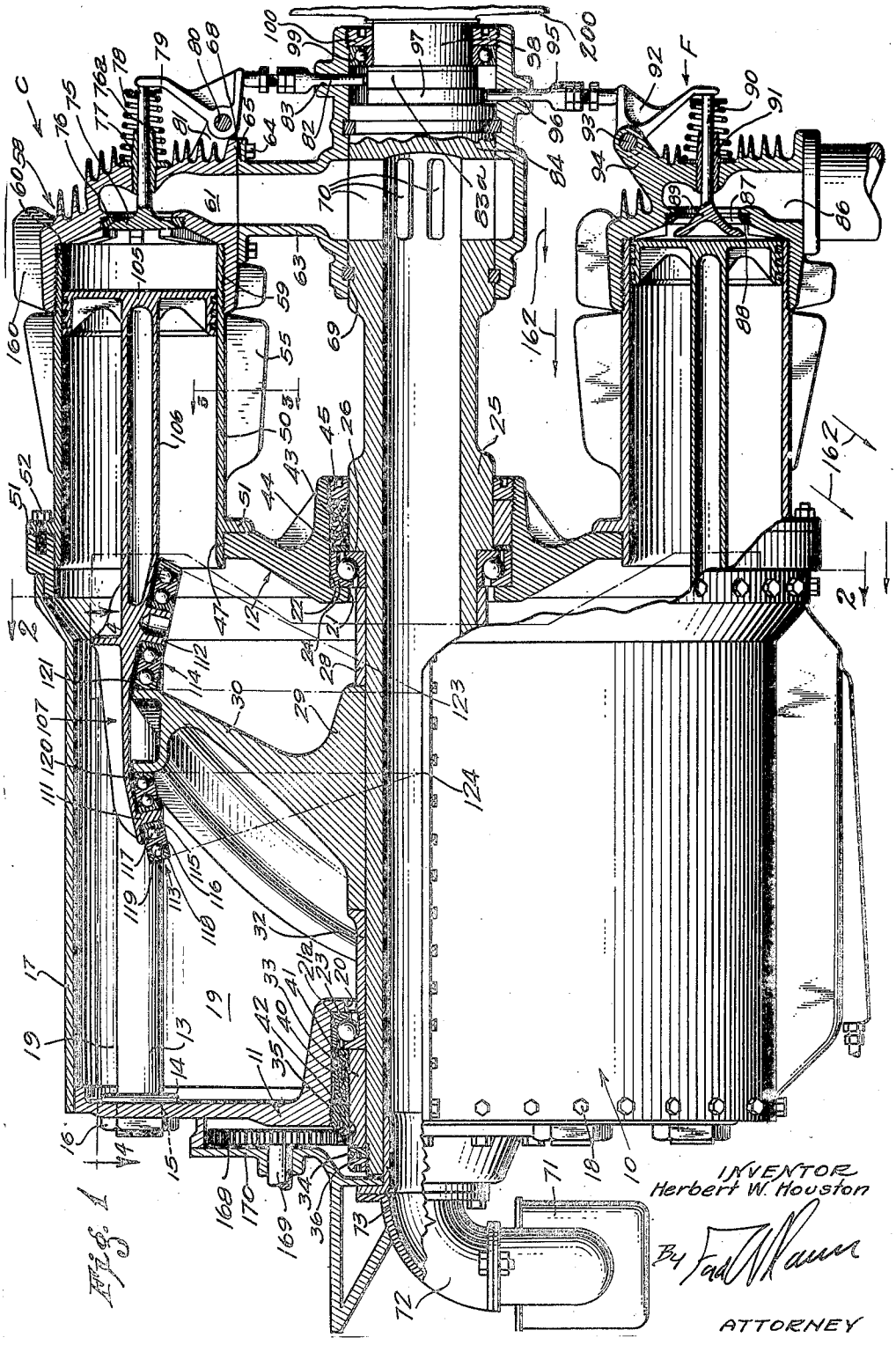


Fig. 1

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VARIABLE STROKE CAM ENGINE

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Fig. 2

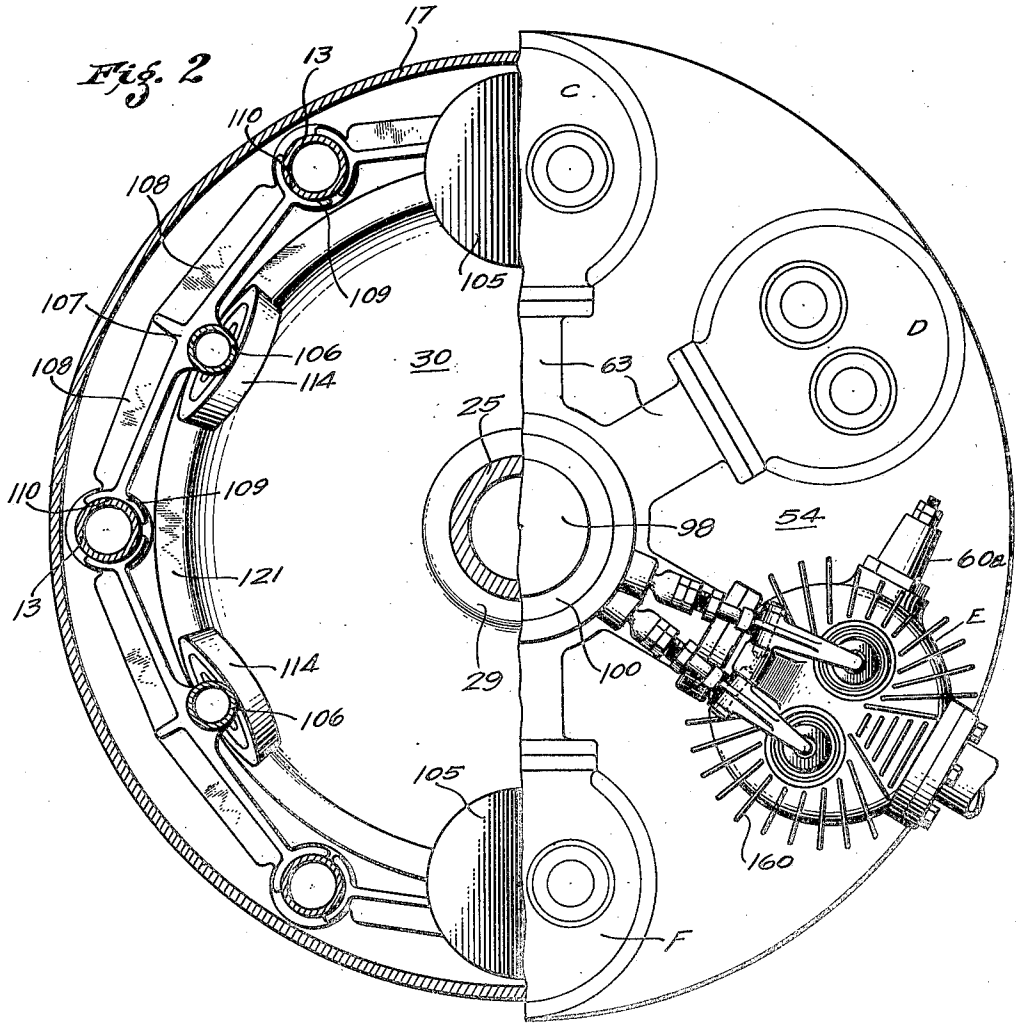
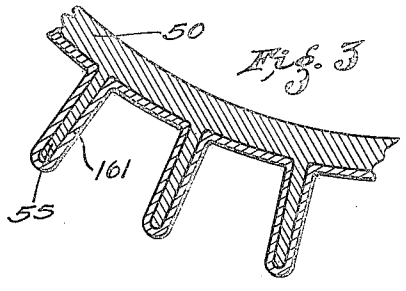


Fig. 3



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VARIABLE STROKE CAM ENGINE

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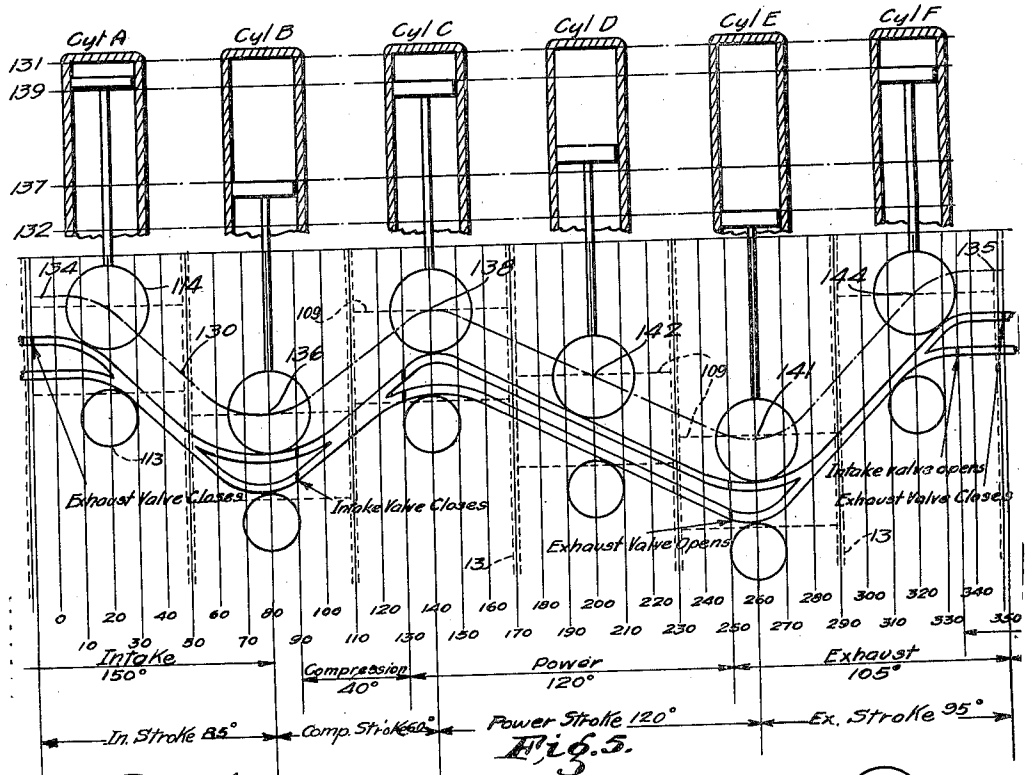


Fig. 5.

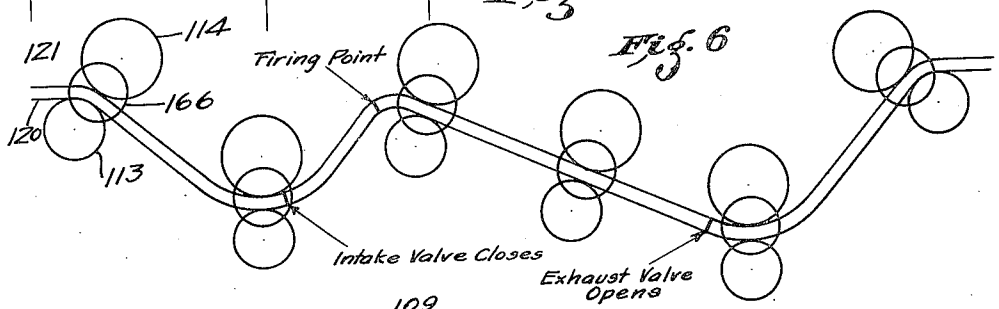


Fig. 6

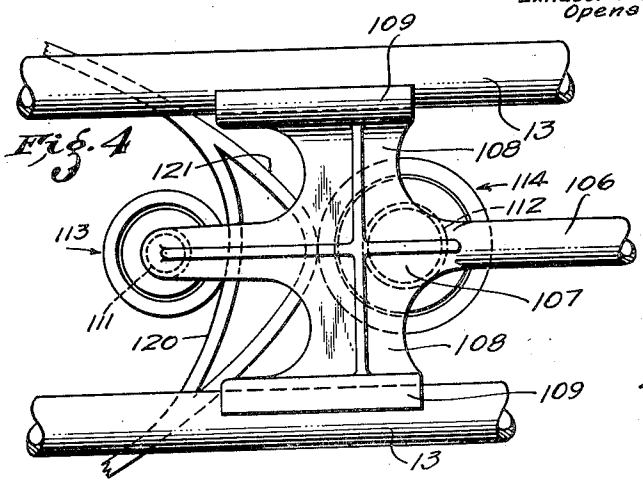


Fig. 4

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VARIABLE STROKE CAM ENGINE

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Fig. 7

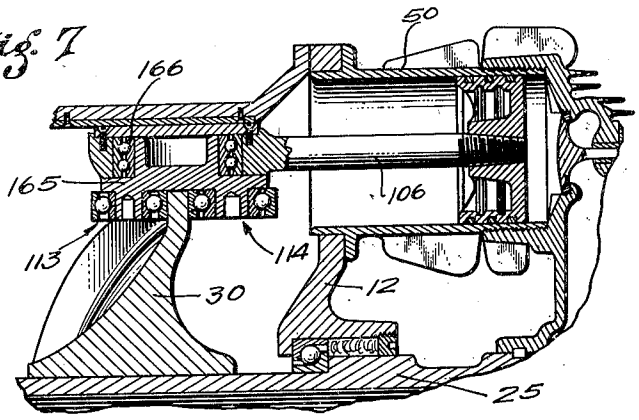


Fig. 8

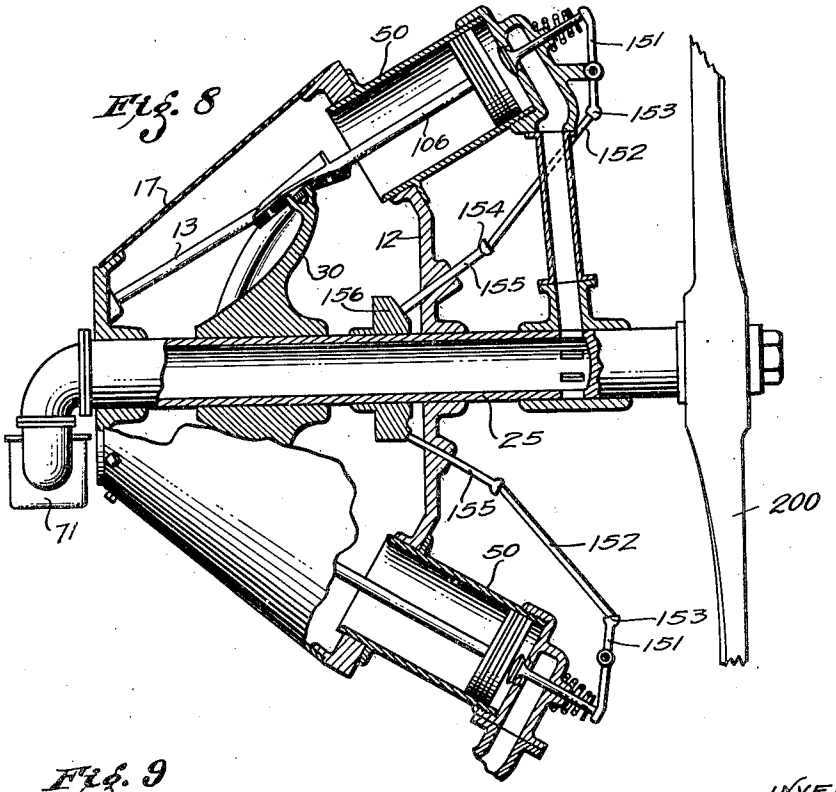
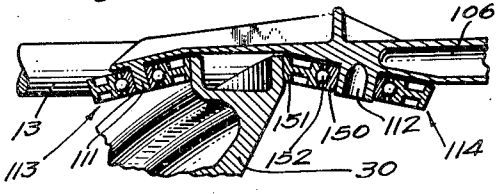


Fig. 9



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Patented June 16, 1931 1,810,017

UNITED STATES PATENT OFFICE

HERBERT W. HOUSTON, OF LOS ANGELES, CALIFORNIA

VARIABLE STROKE CAM-ENGINE

Application filed November 20, 1928. Serial No. 320,579.

My invention relates to reciprocating engines, and more particularly to a variable stroke cam-engine which finds special utility in the field of air transportation.

Two types of airplane engines are at present in wide use, one containng a plurality of cylinders arranged in one or more rows, these cylinders mutually cooperating to rotate a crank-shaft on an axis parallel to the rows of cylinders, and the other type containing radially extending cylinders, these cylinders usually being made stationary in a radial engine, but are sometimes rotated in which event the engine is termed a rotary engine. Of late the radial, air-cooled engines, have become very popular, but these engines are open to the objection that they have a relatively high wind resistance.

It is an object of this invention to provide an internal-combustion engine of the four-cycle type in which the cylinders are parallel to and spaced around the main shaft or at an angle relative to the shaft, the pistons in these cylinders being operated by a circular cam-plate rotating with the shaft.

This cam-plate is preferably formed so as to move each of the pistons through a complete cycle of events each time the shaft turns through 360° of rotation. This permits the accomplishment of another object of this invention, that of providing an engine of low speed relative to the horse-power developed.

My invention is not, however, limited to the four-cycle engine, nor is it limited to an internal-combustion engine, although I have chosen to describe it in this capacity. The basic design may be utilized in a compressor or other pumping device without departing from the spirit of this invention.

It is another object of this invention to provide an engine having a low frontal area, few working parts, and which is light in weight.

Another object of this invention is to provide an engine in which the piston rods move parallel to the axis of the cylinder at all times, thus eliminating the necessity of a piston pin, and allowing the pistons to be shortened to a length only sufficient to carry the piston rings. Due to the fact that the main shaft makes but one revolution for each complete cycle, it becomes possible to operate the valves directly from this shaft, and it is an object of this invention to provide a main shaft carrying a drive cam and also cams utilized for operating the intake and exhaust valves.

Another object of this invention is to provide a main shaft which is hollow and which communicates with the interior of the cylinders through suitable ports, so that the space inside this shaft may be utilized for supplying a combustible mixture to the cylinders.

A further object of this invention is to provide an internal-combustion engine adapted to be secured directly to a propeller, the thrust of the propeller being counter to the pressure exerted on the piston during the power stroke, thereby eliminating large thrust bearings which have been heretofore necessary in radial engines.

It is a further object of this invention to provide a cam engine in which the piston does not have the same velocity and displacement or subsequent strokes of a cycle.

A further object of this invention is to provide a new cycle permitting a relatively lengthy expansion stroke and a relatively short compression stroke, thus effecting an increase in volumetric and thermal efficiency, a smoother flow of power, a more silent exhaust, and an engine which runs much cooler than ordinary internal-combustion engines operating on previous cycles, together with other important advantages to be hereinafter brought out.

A further object of this invention is to provide a novel method of cooling the cylinders by forming fins thereon and coating these fins with a material of high heat conductivity.

Still a further object of this invention is to provide a novel bearing for reducing bearing speeds.

It has been found that with a relatively fast compression stroke involving only a small portion of the 360° cycle, and a relatively slow expansion or power stroke distributed over an extensive portion of the 360° cycle, there is produced a marked increase in the efficiency, particularly in the

horse-power output at slow speeds. Therefore it is another object to provide an engine in which the compression stroke of the piston is faster than the power stroke thereof.

Various other objects and advantages will be more fully apparent from the following description of the accompanying drawings, which form a part of this disclosure and which illustrate a preferred embodiment of the invention.

Referring to the drawings in which I have illustrated several embodiments of my invention,—

Fig. 1 is a diagrammatic view, partially in section, illustrating a pair of oppositely disposed cylinders of the preferred embodiment of my invention.

Fig. 2 is a diagrammatic end view, partially in section, taken in the direction of the arrow 2 of Fig. 1.

Fig. 3 is a sectional view illustrating my method of cooling cylinders.

Fig. 4 is a view taken along the line 4—4 of Fig. 1, this view illustrating the cross-head and bearings of my invention.

Fig. 5 is a crank diagram development of the preferred embodiment of my invention, illustrating the positions of the different pistons.

Fig. 6 is a somewhat similar crank diagram of an alternate form of my invention.

Fig. 7 is a sectional view illustrating the bearing structure utilized in this alternative form of my invention.

Fig. 8 is a sectional view of a third form of my invention.

Fig. 9 is a sectional view illustrating an alternative form of bearing which may be utilized in my invention.

Referring particularly to Fig. 1, the preferred embodiment of my invention is illustrated by the numeral 10, and comprises primary and secondary supporting structures 11 and 12 held in spaced relationship by a plurality of cross-head guides 13, these guides being preferably hollow, as best illustrated in Fig. 2, so as to eliminate any excessive weight. The outer surface of these cross-head guides is preferably ground smooth between shoulder portions 14 formed at each end, these portions being adapted to respectively engage the primary and secondary supporting structures 11 and 12. The ends of the cross-head guides extend through openings 15 in the primary and secondary supporting structures 11 and 12, these ends having nuts 16 threaded thereon which firmly clamp the supporting structures relative to the cross-head guides 13.

A cover-plate 17 extends around the cross-head guides 13 and is held in engagement with the outer periphery of the primary and secondary supporting structures by means of bolts 18, this cover-plate preventing any dust or foreign matter from reaching the mechanism in a chamber 19 between the primary and secondary supporting structures, and also preventing any escape of oil from this chamber. Bores 20 and 21 are respectively formed through the primary and secondary supporting structures, these bores being in axial alignment with the axis around which the cross-head guides 13 are positioned.

Counterbores 21*a* and 22 are adapted to receive the outer races of bearings 23 and 24, the inner races of these bearings being adapted to engage the outer periphery of a hollow main shaft 25 so that this shaft is journalled therein. The inner race of the bearing 24 engages a shoulder 26 of the shaft 25, this bearing being of the thrust type so as to resist any thrust of the main shaft in a leftward direction. Surrounding this shaft and also engaging the inner race of the bearing 24 is a sleeve 28, this sleeve engaging a hub 29 of a drive cam 30 which is suitably secured in fixed relationship relative to the shaft 25 by any suitable means. The other end of the hub 29 is engaged by a sleeve 32 surrounding the shaft, this sleeve also engaging the inner race of the bearing 23, as clearly illustrated in Fig. 1. A clamping sleeve 33 engages the opposite side of the inner race of the bearing 23, and is clamped against this inner race by a nut 34 through a gear 35 which is mounted on the main shaft 25. The nut 34 thus clamps the sleeves 28, 32 and 33, the hub 29, and the inner races of the bearings 23 and 24. A lock-nut 36 keeps the nut 34 from loosening. It should thus be seen that any axial thrust on the main shaft 25 in a rightward direction is taken by the bearing 23.

The outer diameter of the clamping sleeve 33 is less than the inner diameter of the counterbore 21*a*, the intermediate space being filled by packing 40 compressed against a washer 41 by a packing nut 42. A similar packing 43 is positioned in the counterbore 22 and is compressed against a washer 44 by a packing nut 45.

The packings 40 and 43 thus insure that no leakage of oil will be permitted around the shaft 25 nor will any dust or foreign fluid enter the chamber 19 inside the cover-plate 17.

The secondary supporting structure 12 is provided with a plurality of openings 47 grouped around the axis of the main shaft 25, one of these openings being formed between each pair of cross-head guides 13. Into each of these openings is extended the walls of a cylinder 50, this cylinder having a flange 51 which is suitably bolted to the secondary supporting structure 12, as by studs 52. The cylinders 50 are not interconnected, so that there is an air circulation space 54 therebetween.

I have shown six cylinders 50 in the particular engine illustrated in the drawings, these cylinders being lettered A–F inclusive for the

purpose of distinction. For the purpose of illustration I have indicated only the cylinders C and F in Fig. 1, while in Fig. 2, the cylinder E is the only one shown in full lines the position of the other cylinders being suitably indicated, it being understood that all of the cylinders are of a construction similar to that illustrated in Figs. 1 and 2. Outward extending fins 55 are formed on each cylinder 50 for cooling purposes, these fins being either cast integrally with the cylinder, or tacked, brazed, or welded to the cylinder walls.

Suitably secured to the forward end of each of the cylinders 50 is a cylinder head 58. I have illustrated each head 58 as being held in fixed relationship relative to the cylinder by means of intermeshed threads 59 formed on the cylinder and in a cavity 60 of the cylinder head, but this particular form of attachment is not essential, and I am not limited thereto. Suitable spark plugs 60a are secured to the head and ignite a combustible mixture in the compression chamber of each cylinder.

Each of the heads 58 has an intake port 61 therein, the ports of each of the cylinders being in open communication with the interior of a head-spider 63 which is suitably secured to each head by means of bolts 64 extending through flanges 65 of the head-spider and being threadedly received by the cylinder head 58. The cylinder heads 58 cooperate with the head-spider 63 to form a cylinder head-spider structure 68 which is adapted to support the forward ends of the cylinders 50 and to surround the main shaft 25, this shaft passing through an opening 69 of the head-spider 63.

As previously mentioned, the main shaft 25 is hollow, and the interior of this shaft is in open communication with the intake ports 61 through a plurality of openings 70 extending radially through the main shaft in a position indicated in Fig. 1. A combustible mixture is supplied to the intake ports 61 through the main shaft 25 from a carburetor 71 which communicates with the interior of the main shaft through an elbow 72, there being a sleeve 73 extending into the main shaft to prevent any leakage.

The combustible mixture is thus distributed to each of the cylinders from a central point, thus insuring that the same quality of gas is supplied to each of these cylinders. This arrangement eliminates separate manifolds extending to each cylinder, and with the arrangement illustrated it is possible to make use of the inertia of the combustible gas mixture passing through the main shaft to secure a supercharging effect.

The admission of combustible gas to each of the cylinders is controlled by an intake valve 75 which is illustrated as being of the poppet type and engageable with a renewable valve seat 76 screwed into the cavity 60 of each cylinder. This intake valve has a stem 76a slidable in a stem guide 77, and is held in closed position by a spring 78. The valve 75 is actuated by a rocker-arm 79 pivoted on a pin 80 held in straps 81 extending from the cylinder head 58. One face of the rocker-arm 79 is engaged by a tappet 82 of adjustable length, this tappet extending through a tappet guide 83 of the head-spider 63 and extending into the opening 69 to engage an intake cam 83a formed on a solid portion 84 of the main shaft 25. A similar intake valve and operating mechanism is provided for each cylinder. Thus, as the main shaft is turned, the intake valves are opened and closed at the correct instant due to the cooperation between the tappets 82 and the intake cam 83a.

Similarly, each of the cylinder heads 58 is provided with an exhaust port 86 which communicates with the interior of the cylinder through an opening 87 in which is secured a renewable valve seat 88. This seat is resiliently engaged by an exhaust valve 89 having a stem 90 journalled in a guide 91 and being actuated by a rocker-arm 92 pivoted on a shaft 93 supported in straps 94, this valve mechanism being similar to the intake valve mechanism just described. An adjustable tappet 95 engages the rocker-arm 92 and extends through a guide 96 formed in the head-spider 63, the end of this tappet engaging an exhaust cam 97. Thus, as the main shaft rotates, the exhaust cam actuates the exhaust valves 89 of each of the cylinders 50, opening and closing these valves at the correct instants and in the desired sequence.

That part of the solid portion 84 of the main shaft 25 which extends forward from the cams 83a and 97 has been termed a drive shaft 98, and it is to this shaft that the propeller is directly connected. The shaft 98 is journalled in a suitable bearing 99 held in the opening 69 of the head spider 63 and protected from foreign substances by a nut 100.

Positioned in each of the cylinders 50, and slidable therein in a direction parallel to the axis of the main shaft 25 is a piston 105 having a piston rod 106 suitably secured thereto or formed integrally therewith. The end of this piston rod terminates in a cross-head 107 which is shaped as best illustrated in Figs. 1, 2, and 4.

This cross-head is provided with extensions 108, on the free ends of which are formed shoes 109 having arcuate surfaces 110 which bear against the smooth periphery of the cross-head guides 13 in a manner best shown in Fig. 2. The axes of all of the cylinders 50 are parallel to the cross-head guides 13 so that the cross-heads 107 sliding along the cross-head guides 13 may move the pistons parallel to the walls of the cylinder 50. In other words, the piston rod 106 is moved along the axis of the cylinder 50. This arrange-

ment does away with any piston pins and eliminates the necessity of a long skirt on the piston to maintain this piston in axial alignment with the cylinder.

Extending downward from the cross-head 107 are pin structures 111 and 112 on which ball or roller bearings 113 and 114 are mounted. These bearings are similarly formed, though of different size, the bearing 113, for instance, having an inner race 115 tightly engaging the pin structure 111, an intermediate race 116 separated from the inner race and journalled thereon by bearings 117, and an outer race 118 separated from, and journalled relative to the intermediate race 116 by balls 119.

The outer periphery of the race 119 engages a cam surface 120 of the drive cam 30. Similarly, the bearing 114 is formed of inner, intermediate, and outer bearing races separated by two series of balls, the periphery of the outer race engaging a cam surface 121 formed on the drive cam 30.

It is preferable to position the bearings 113 and 114 in non-parallel planes, as best illustrated in Fig. 1. This ordinarily necessitates tapering the outer periphery of the outermost race at each bearing, as illustrated in this figure, the cam surfaces 120 and 121 being shaped to correspond to the shape of the periphery of the outermost races. I prefer to form the periphery of these outermost races in a frusto-conical shape so that a continuation of this periphery would define a cone with the apex at the center of rotation of the main shaft 25, the apex of the imaginary cone formed by the bearing 114 being indicated by the numeral 123, while the apex of the imaginary cone formed by the bearing 113 is indicated by the numeral 124. These apexes are separated as indicated in Fig. 1. By this arrangement there is no force component tending to move the cross-head radially away from the drive cam 30, inasmuch as the cam surfaces 120 and 121 at the point of contact with the bearings 113 and 114 are always parallel and extending radially from the axis of rotation of the main shaft 25. It should be understood that the outermost diameter of the drive cam 30 is the same throughout the length of this cam, so that as the cam rotates with the drive shaft 25, the cross-head 107 and piston 105 are moved along the axis of the cylinder 50.

The shape of the drive cam 30 is an important feature of this invention, a development of the cam surfaces on this drive cam being illustrated in Fig. 2. Before proceeding, however, to describe this cam in detail, it is advisable to point out wherein the results obtained by the use of the cam of my invention differ from those previously realized by the use of a crank-shaft.

All internal-combustion engines heretofore designed have utilized a piston which moved forward and backward in a cylinder over the same path of travel for each successive stroke in the cycle. In a four-cycle engine, the first stroke is an intake stroke which draws the supply of combustible mixture into the cylinder, this mixture being compressed on the compression stroke and ignited. The pressure in the compression chamber at this instant becomes very large and power is derived from the piston when moved through the subsequent power stroke. Subsequently the exhaust valve opens and on the exhaust stroke of the piston the expended gases are forced from the cylinder. This completes one cycle of a four cycle engine. Heretofore the piston has travelled over the same path of travel in forming these strokes, and the speed of travel of the piston at any particular point in its path of travel is always the same irrespective of whether or not the piston is on a power, compression, exhaust, or intake stroke. Furthermore, the maximum movement of the piston is invariably completed in 180° of rotation of the crank-shaft to which the piston is connected. There is thus no possibility of increasing any of the strokes beyond 180° unless, of course, the timing of the valves may be varied so as to open or close before the piston reaches dead center, as is usually done, to vary the duration of the different functions of the cycle, but this does not vary the length of the piston stroke.

For instance, one type of radial motor which is in great demand at the present is designed so that the power stroke extends over 150° of crank-shaft revolution. The actual length of the power stroke is only a trifle greater than 77% of the total stroke of the piston.

I have designed the cam of my invention so that the power stroke extends over a substantially greater portion of the total stroke of the piston. Furthermore, the path of travel of the piston is not the same on successive strokes of a given cycle.

Referring now to Fig. 5, I have plotted a development of the cam of my invention whereby these results are obtained. Horizontal distances are proportional to the rotation of the main shaft 25, while vertical distances are proportional to the displacement of the piston relative to the cylinder. The outer peripheries of the bearings 113 and 114 have been indicated, it being clear that the center line of these bearings is always parallel to the axis of the cylinder 50. The cam surfaces 120 and 121 are illustrated as respectively contacting the bearings 113 and 114.

The position of the pistons in each of the cylinders A to F inclusive at a given instant are indicated in Fig. 5. It should be clear that the displacement of the piston is equal to the displacement of the axis of the bearing 114. Thus, for the sake of clearness, I have drawn a line 130 indicating the locus

of the center of the bearing 114, this line also indicating the locus of the piston relative to the cylinder. A horizontal line 131 defines the maximum advanced position of the piston relative to the cylinder, while the line 132 defines the maximum retracted position of the piston relative to the cylinder. The cylinders A to F inclusive are diagrammatically illustrated in Fig. 5 and the pistons in these cylinders are illustrated in their substantially correct positions at a given instant of time when cylinder C is at the end of the compression stroke. At this instant the piston of cylinder B is at the end of its intake stroke, while the piston of cylinder E is at the end of its power stroke. Similarly, the successive positions of any one piston for a complete revolution of the main shaft is indicated by the position of the cylinders A to F inclusive.

In laying out a development of the cam surfaces 120 and 121, the line 130 is first drawn so that the desired cycle is obtained. The outline of these cam surfaces is then determined by drawing the successive positions of the peripheries of the bearings 113 and 114 and drawing the lines defining the cam surfaces 120 and 121 tangent to these peripheries. The result is a profile cam such as illustrated in Fig. 5, this cam being so designed that the cam surfaces thereof are always in engagement with the peripheries of both bearings 113 and 114.

In this embodiment, the distance between the cam surfaces 120 and 121 is not a constant, due to the fact that the center lines of the bearings 113 and 114 are not perpendicular to the cam surfaces at all times.

Before following through a complete cycle, it should be clear that in Fig. 5 no piston is illustrated in its extreme advanced position, in which position the top face of the piston would lie along the line 131. This condition is reached when the center of the bearing 114 lies adjacent a point 134 or 135. The position of the piston when the center of the bearing 114 is adjacent the point 134 is, however, illustrated in the lower portion of Fig. 1. Referring to this figure, it should be clear that at this time the piston has only a very small clearance with the top of the cylinder head.

It will be noted that at the bottom of Fig. 5 there is illustrated the piston stroke or displacement analysis of the cycle, and immediately above this there is illustrated the functional analysis of the cycle as determined by the timing of the respective valves and of the ignition.

In the preferable functional cycle the intake valve opens at the 335° line of the diagram and remains open to the 80° line while a charge is being drawn into the cylinder by the advance of the piston from the line 131 to the line 137, thus the intake function extends through 115° of the main shaft cycle. The compression function extends through 40° of the main shaft cycle from the 80° line at which the intake valve closes, to the 130° line at which point the charge is ignited, and the power function extends through 120° of the main shaft cycle from the point of ignition at the 130° line, to the 250° line, at which point the exhaust valve is opened. The exhaust function, of course, takes place between the opening and closing of the exhaust valve and extends from the 250° line to the 355° line where the exhaust valve closes, thus the exhaust function extends through 105° of the main shaft cycle, there being an overlap of 10° with the intake function, during which overlap both the intake and the exhaust valves are open due to the fact that the intake valve opens at the 335° line or 20° prior to the closing of the exhaust valve at the 355° line.

In this cycle the power function takes place over 120° or one-third of the 360° shaft cycle, thus insuring not only a longer expansion than has heretofore been obtained, but also carrying the expansion through a greater portion of the cycle than has heretofore been possible. This extended power function is in a large measure made possible by reason of the use of only 40° of the cycle for the compression function, since it is not desirable to unduly shorten the time of the intake or exhaust functions.

Now considering the diagram for the purpose of analyzing the various piston strokes, it will be apparent that when the bearing 114 is adjacent the point 134, the piston is in the maximum retracted position with its top at the line 131. As the center of the bearing 114 follows the curve 130 to the point 136 on the 80° line, the piston is advanced to the line 137, as illustrated by the piston in the cylinder B of Fig. 5; thus the intake stroke of the piston has been completed with 85° rotation of the main shaft. The center of the bearing 114 then follows the curve 130 to the point 138 on the 140° line to retract the piston to the line 139, as illustrated by the piston in the cylinder C of Fig. 5; thus the compression stroke of the piston will be completed with an additional 60° of rotation of the main shaft, it being apparent that this retracted position of the piston is short of the maximum retracted position indicated by the line 131 to provide a compression space above the piston of a volume determined by the permissible compression ratio of the particular fuel used. During the succeeding power or expansion stroke of the piston the center of the bearing 114 follows the curve 130 from the point 138 on the 140° line to the point 141 on the 260° line, passing through the point 142, the piston advancing to the maximum advanced position at line 132, as illustrated by the piston in the cylin-

der E of Fig. 5, this maximum advanced position being a material distance beyond the line 137 which indicates the advanced position of the piston on the intake stroke. The piston thus is advanced from the retracted position 139 to the maximum advanced position 132 to complete the power or expansion stroke with an additional 120° rotation of the main shaft. From the point 141 the center of the bearing 114 follows the curve 130 to the point 135 on the 355° line to fully retract the piston to the line 131, thus completing the exhaust stroke with an additional 95° rotation of the main shaft in a completion of the stroke cycle of the engine.

Thus we have a cycle, in which the piston is advanced from the line 131 to the line 137 during an intake stroke involving 85° of rotation of the engine shaft; in which the piston is then retracted from the line 137 to the line 139 during a compression stroke involving 60° of rotation of the engine shaft; in which the piston is then advanced from the line 139 to the line 132 during a power stroke involving 120° of rotation of the engine shaft; and in which said piston is next retracted from the line 132 to the line 131 during an exhaust stroke involving 95° of engine shaft rotation.

As previously pointed out, it is of great importance that the cycle include a quick compression stroke relative to the power stroke, and since the speed of each stroke depends upon the relationship between the length of the stroke and the lineal travel of the cam during such stroke, the mean average speed of each stroke for each degree of cam travel may be determined by dividing the length of the stroke by the number of degrees of the cam cycle involved in said stroke. For example, if the power stroke is 3 inches over 120° of cam travel, the mean average speed of the piston during the power stroke would be .025 inches per cycle degree of the cam, and if the compression stroke is 2 inches over 60° of cam travel, the mean average speed of the piston during the compression stroke would be .033 inches per cycle degree of the cam, thus the compression stroke of the piston would be quicker than the power stroke. This is substantially the ratio of the compression and power strokes shown in the diagram. However, if the power stroke were increased to 4 inches while retaining the 120° cam travel, we would again have .033 as the mean average piston speed, and the piston speed on both the power and the compression strokes would be the same since the ratio between the length of each stroke and the degree of cam travel involved in each stroke would be the same. Therefore it becomes necessary in order to secure a quick compression stroke relative to the power stroke that the ratio between the length of the compression stroke and the degree of cam travel involved in the compression stroke be less than the ratio between the length of the power stroke and the degree of cam travel involved in said power stroke.

In the cycle illustrated in the diagram, the exhaust stroke is the fastest in the cycle, the speed of the intake stroke is about 82% as fast as the exhaust stroke, the speed of the compression stroke is about 91% as fast as the exhaust stroke, while the speed of the power stroke is only about 66% of the speed of the exhaust stroke, the speed of the compression stroke thus being approximately 27% faster or quicker than that of the power stroke. While these relative speeds of piston travel during the various strokes have been found to be satisfactory in certain instances, it is to be understood that they may be varied to meet various conditions, as long as the cycle includes the principle of a compression stroke which is faster or quicker than the power or expansion stroke of the cycle.

The advantage of this cycle should be obvious. The small clearance space between the piston and the cylinder head at the end of the exhaust stroke and at the beginning of the intake stroke insures that all of the spent gases are forced out of the cylinder, and the cylinder supplied with a pure combustible mixture undiluted by any spent gases which invariably remain in the cylinder in present types of crank-operated engines. The very quick compression stroke is very valuable, and the long power stroke spread over 120° of rotation of the main shaft is of extreme importance in increasing the volumetric and thermal efficiency of the engine. This long power stroke permits the gas in the cylinder to expand to a point heretofore impossible, and thus impart to the piston energy which is at present lost through the exhaust. This extended expansion also produces an exhaust which is much more nearly silent than the exhaust of the present crank-engines. Furthermore, the expansion of the gases decreases the temperature thereof, which makes the motor of my invention run much cooler than the motors running on cycles used in ordinary crank-engines. Higher compression is also possible due to low temperature, and this higher compression again reacts to increase the efficiency of the engine.

On the exhaust stroke I prefer to raise the piston to within .05″ of the cylinder head, and on the compression stroke the piston is advanced as far as possible, depending upon the permissible compression ratio of the fuel utilized. The main cam of my invention is so shaped that the power stroke utilizes as much as from 25% to 35% of the total travel of the main shaft, and expansion is carried down to as much as 40% more than the intake stroke, by a suitable design.

The particular design of the bearings 113 and 114 is an important auxiliary feature of

this invention. It is well established that the pressure which roller or ball bearings may carry is limited by the ball speed, so that if the ball speed is reduced, the pressure which the bearing is capable of supporting is proportionally increased. To eliminate high ball speeds, I have designed my bearings as illustrated in Fig. 1. It should be clear that the outer race of these bearings rotates at the same speed as the cam surfaces of the drive cam 30. Similarly, the inner races are held fixed on the pin structures 111 and 112. The intermediate race, however, rotates at half the speed of the outer race, and the balls between the intermediate and outer races, of course, rotate at a speed which is the mean peripheral speed of these races. Similarly, the balls between the inner race and intermediate race rotate at a speed which is a mean value of the peripheral speeds of the inner and intermediate races. In this manner the ball speed is very materially decreased and the type of bearing illustrated has been found to be particularly advantageous. Much larger bearings would necessarily have to be used if these bearings contained but one series of balls.

It will be noted from Fig. 1 that the bearing 114 is larger than the bearing 113. This is, of course, due to the fact that the bearing 114 must transmit to the drive cam 30 the pressure developed by the piston 105 while the bearing 113 ordinarily acts as a follower bearing and is not required to withstand high pressures.

Another type of bearing which may be substituted for the bearings 113 and 114 is illustrated in Fig. 9, each of the bearings illustrated having a central race 150 separated from an outer race 151 by a series of balls 152. The speed of the balls 152 is, of course, proportional to the radius of this series of balls or to the peripheral speed of the races. Thus, I have made the radius of this series of balls as small as possible, the outer race being made much larger in outer diameter than the inner race.

Another method of decreasing the ball speeds is to decrease the diameter of the drive cam 30. In the form of my invention illustrated in Fig. 1, the minimum diameter is determined by the size and number of cylinders. However, in Fig. 8 I have illustrated an alternative form of my invention wherein the drive cam 30 may be formed of a smaller diameter. In this form of the invention the axes of the cylinders 50 are not parallel to the axis of the main shaft 25, but are oblique thereto, as clearly illustrated. In this construction the piston rods 106 tend to converge toward the rear end of the motor, and thus it becomes possible to use a drive cam of smaller diameter. In this form of my invention the valve mechanism is somewhat different, the valves being operated by rocker-arms 151 connected to tappets 152 by ball-and-socket joints 153. The other ends of the tappets 152 are secured in sockets 154 of followers 155 which extend through the secondary support 12 and engage concentric surfaces of a cam 156, this cam defining both the intake and exhaust cam surfaces which are utilized for actuating the valves.

This form of my invention permits the use of a larger cylinder bore than the form illustrated in Fig. 1, thus producing a more powerful motor. The cylinders in this form are also exposed to a better air-blast for cooling purposes.

To facilitate the cooling of the cylinders of my invention, I have provided the fins 55, previously described, these fins being formed integrally with or secured to the cylinder 50 in any desired manner. Similarly, I have provided fins 160 formed integrally with, or attached to, each of the cylinder heads 58. To further facilitate the cooling, I coat the external surfaces of the cylinders 50 and heads 58, together with the fins 55 and 160 after attachment thereto, with a layer of copper, bronze, aluminum, or any other suitable material having a heat conductivity which is greater than that of the metal forming the cylinders 50 and heads 58. This coating of material is indicated by the numeral 161 of Fig. 3, this material being preferably electroplated on the exposed surfaces of the cylinders 50 and heads 58. This coating of material of high heat conductivity draws the heat uniformly from the cylinders 50, heads 58, and fins 55 and 160. Ordinarily, fins extending outward from a body have a tendency to conduct the heat from the body only at the point of attachment. By providing the coating 161, heat is drawn from the body between the fins, and is readily conducted along the coating 161 and flows outward along that portion of the coating which surrounds the fins, whence it is removed by a blast of cooling fluid.

The propeller 200 on the main shaft 25 provides an adequate supply of air which is forced between the legs of the head-spider 63, and between the cylinders 50, as indicated by the arrows 162 of Fig. 1.

Another important advantage of mounting the cylinders in a position such as shown in Fig. 1 is that the pressure exerted on the drive cam 30 by each piston bearing its power stroke has a leftward component which tends to counterbalance the thrust imposed on the main shaft by the propeller, this thrust tending to force the main shaft to the right. The propeller thrust is thus offset by the pistons during their power strokes, this eliminating the necessity of a large thrust bearing and effecting a consequent saving in energy, inasmuch as such large thrust bearings ordinarily absorb considerable power.

A further important feature of the present

invention relates to the particular arrangement of the several cross-heads 107 and the cross-head guides 13 so as to transmit from one to the other the circular thrust of the several piston rods 106, which is due to the action of the bearings 113 and 114 with the drive cam 30. It is quite evident that upon a power stroke of a piston its bearing roller 114 exerts a circular thrust on the drive cam 30 to rotate the main shaft 25 in a counter clockwise direction (in Fig. 2), and it is also evident that an equal circuar thrust in the reverse direction will be exerted on the adjacent cross-head guide rod 13 which must serve as an abutment, and thus receives the full torque load of the associated cross-head. In other cam engines heretofore proposed, cross-heads have been guided in grooves in a heavy casing capable of withstanding this heavy circular thrust or on independent pairs of heavy guides each capable of individually withstanding such circular thrust. Such structures involve a very great deal of weight and are exceedingly costly, not only as to material, but also as to machining and assembling.

To lessen the cost and also to reduce the weight of the engine, a very important consideration in aviation motors, I have provided only six tubular guide rods 13 for the six cross-heads 107 with the cross-heads arranged between the rods so that each rod is engaged by two adjacent cross-heads, the cross-heads and guide rods thus forming a continuous circular ring, as clearly shown in Fig. 2, and although the several cross-heads are not diametrically aligned at any one time, each is sufficiently close to the adjacent ones that the several cross-heads form struts between the several guide rods, each backing up the other. This will be more fully apparent from an examination of Fig. 5 in which the several guide rods and cross-heads are illustrated in dotted lines. Here it will be seen that the three cross-heads involved in the power stroke, at 138, 142 and 141, overlap each other transversely, that the cross-head at 144 will overlap the cross-head of cylinder A, that the cross-head at 136 nearly overlaps the cross-head at 138, and that of cylinder A, while the cross-head at 144 is axially separated from the cross-head at 141 a distance of about one-half the length of the shoes 109 of the cross-heads.

Now considering the directions in which the circular thrusts are exerted on the cross-heads, it will be evident that there will be a heavy backward circular thrust on the cross-heads involved in the power stroke, and that the circular thrust on the cross-heads involved in the intake, the compression, and the exhaust strokes will be exerted in a forward direction, the circular thrust on the cross-heads involved in the intake and exhaust strokes being relatively light, while that exerted on the cross-head involved in the compression stroke will be of an intermediate degree. Thus, in this arrangement the circular or transverse thrusts of the several cross-heads are transmitted from one to the other through the intermediate guide rods in a complete circle with the thrusts of the power cross-heads opposed by the thrusts of the other cross-heads, whereby the guide rods are relieved of a great portion of the torque load and may therefore be of materially lighter construction than has heretofore been possible in cam engine construction.

I preferably form the guide rods of cylindrical stock as a matter of economy since these cylindrical rods may be formed and finished in a lathe or other similar machine with much greater facility than rods of other shapes which would have to be planed.

The peculiar shape of the profile cam illustrated in Figs. 1 and 5 is necessitated by the fact that the center line of the bearings 113 and 114 is always parallel to the axes of the cylinders 50, thus causing these bearings to engage the cam surfaces 120 and 121 at points which tend to lie along this center line of the bearings.

In Figs. 6 and 7 I have illustrated an alternative form of cam wherein the distance between the cam surfaces 120 and 121 is substantially constant throughout the length of the cam. This is made possible by mounting the bearings 113 and 114 on a pivot structure 165 which is journalled in the cross-head 107 by means of a bearing 166. Thus, the pivot structure may turn in the cross-head.

In Fig. 6, I have illustrated the development of this cam, the bearings 113, 114, and 166 being diagrammatically illustrated in different positions throughout a cycle. The scale of Fig. 6 is substantially the same as the scale of Fig. 5 so that the different events may be approximately determined by a comparison of these two figures. It will be noted in Fig. 6 that the center line of the bearings 113 and 114 always is perpendicular to the cam surfaces 120 and 121, and that the point of contact between these cam surfaces and the bearings 113 and 114 always lies along this center line. The cam illustrated in Fig. 6 is adapted to follow through substantially the same cycle as that illustrated for the profile cam illustrated in Fig. 5.

The particular ignition system, oiling system, etc. necessary to the operativeness of my engine are not a part of this invention, such systems and devices being well-known in the art. I have, however, shown a gear 168 meshed with the gear 35 of the main shaft, and having a shaft 169 extending from a plate 170 secured to the outer end of the primary supporting structure 11, this shaft being useful in operating a distributor, oil pump, or other device.

It should be clear that my invention is not

limited to a four-cycle internal-combustion engine, but finds utility in other types of engines, including compressors or other types of pumping devices. The embodiment herein illustrated and described has thus been used only for the purpose of illustration, and I am not in any way limited thereto.

Furthermore, variations of the particular cycle disclosed herein may be made without departing from the spirit of the invention as defined in the appended claims.

I claim as my invention:

1. In a cam-engine, the combination of: primary and secondary supporting structures; a plurality of longitudinal cylindrical members holding said supporting structures in spaced relationship; a plurality of cylinders secured to said secondary supporting structure between said longitudinal members; a piston in each of said cylinders; a cross-head for each of said pistons disposed between two of said members and having opposed arcuate surfaces engaging said members, said cross-head being guided by said longitudinal members; and means for moving said cross-heads.

2. In a cam-engine, the combination of: a drive shaft; a cylinder longitudinally disposed at one side of said shaft; a piston slidable in said cylinder; a drive-cam on said shaft having a radial cam wall disposed substantially at right angles to the axis of said cylinder; and a drive roller carried by said piston and engaging the radial cam wall of said drive-cam, said roller being of frusto-conical form and disposed so that the apex of an imaginary cone formed by a continuation of its peripheral walls will intersect the axis of the drive shaft in the plane of that portion of said radial cam wall engaged by said roller.

3. In a cam-engine, the combination of: primary and secondary supporting structures; a plurality of longitudinal members holding said supporting structures in spaced relationship; a plurality of cylinders secured to said secondary supporting structure between said longitudinal members; a piston in each of said cylinders; a cross-head for each of said pistons, said cross-head being guided by said longitudinal members; and means for moving said cross-heads, there being an equal number of said cross-heads and members so that each of said members is engaged on opposite sides by two adjacent cross-heads whereby transverse thrusts on any of said cross-heads are transmitted to adjacent cross-heads through the intermediate guide members.

In testimony whereof, I have hereunto set my hand at Los Angeles, California, this 13th day of November, 1928.

HERBERT W. HOUSTON.