Feb. 12, 1935. J. PHILLIPS 1,990,565

INTERNAL COMBUSTION ENGINE

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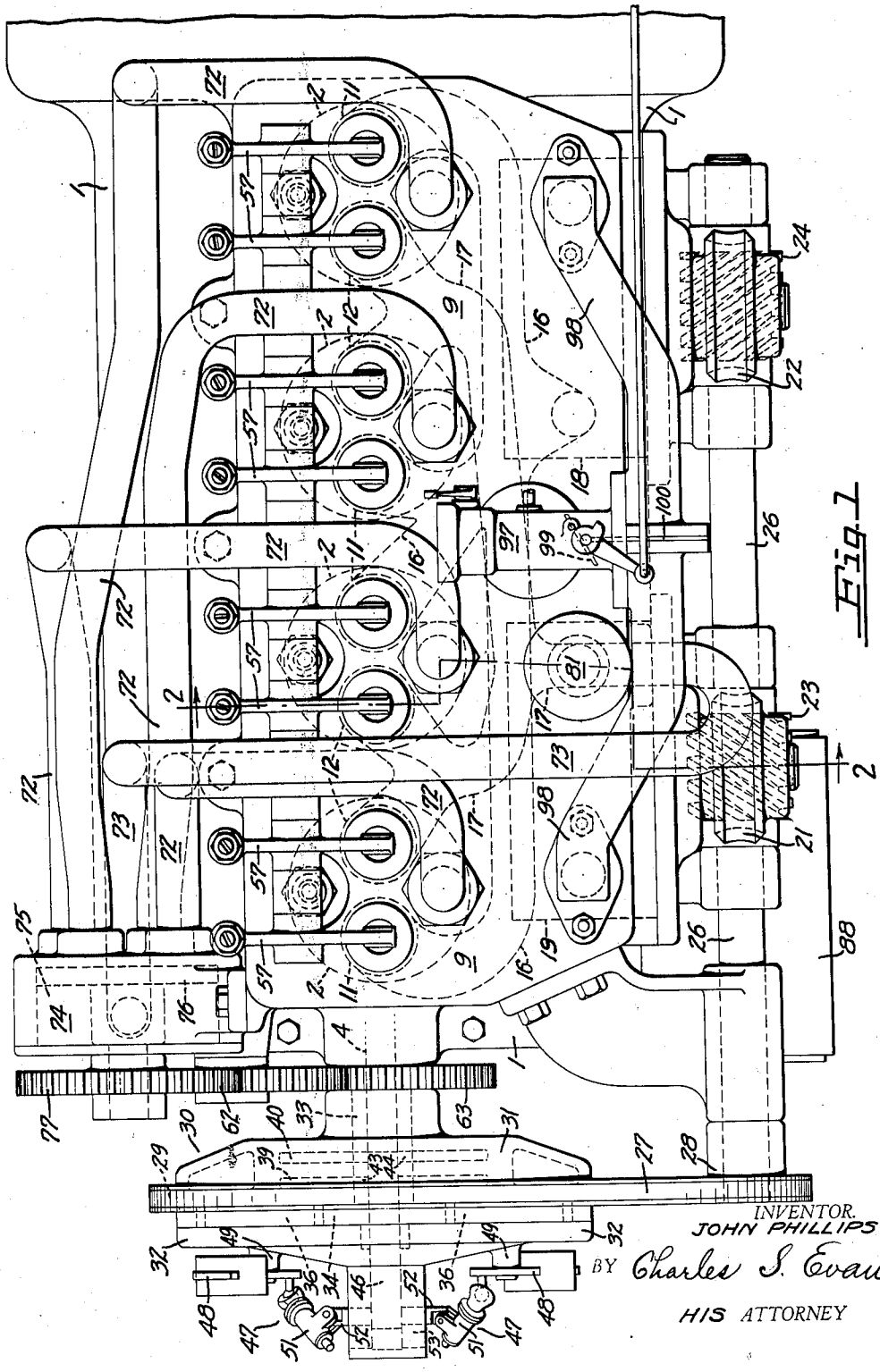


Fig. 1

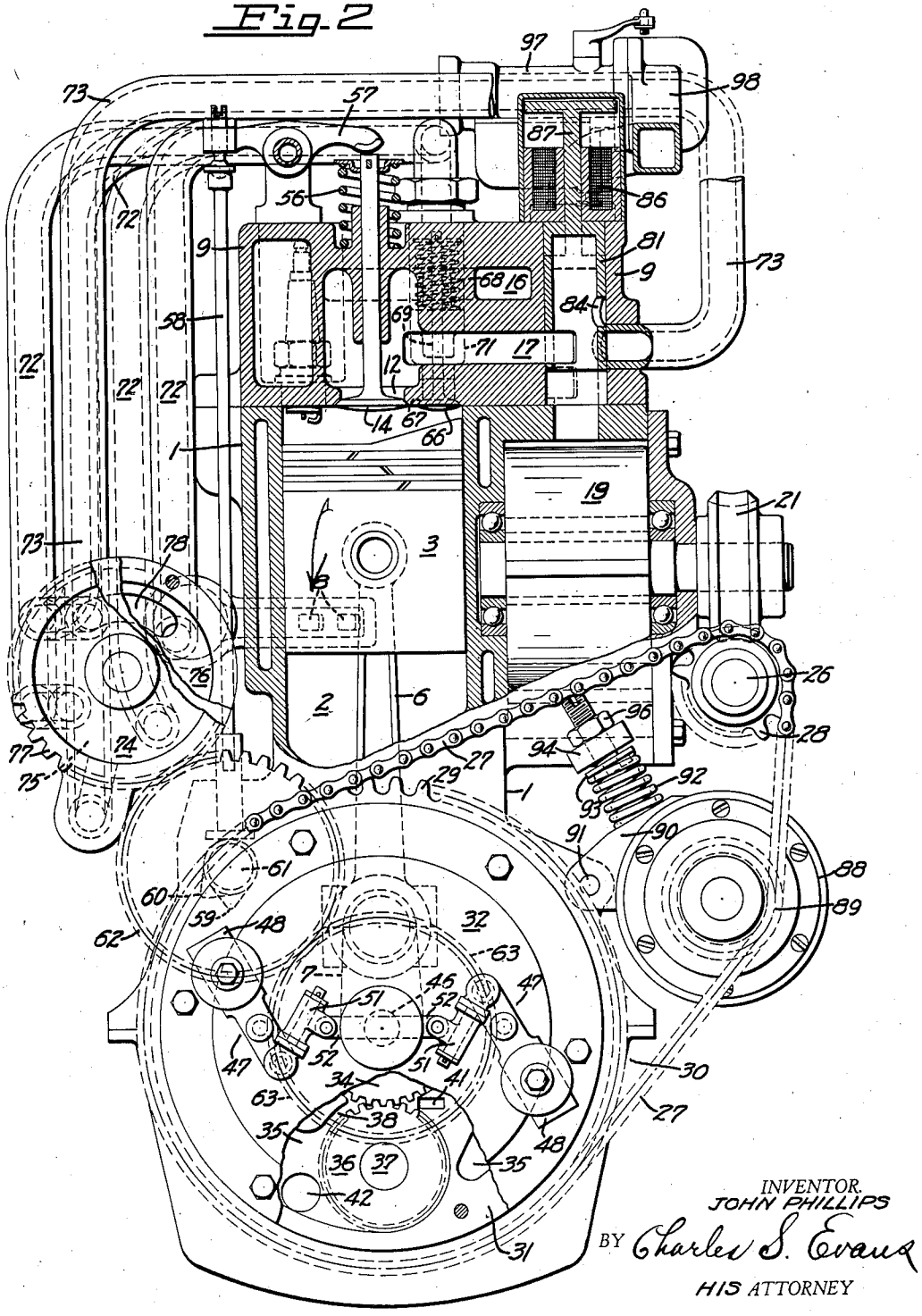
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Fig. 2



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Fig. 3

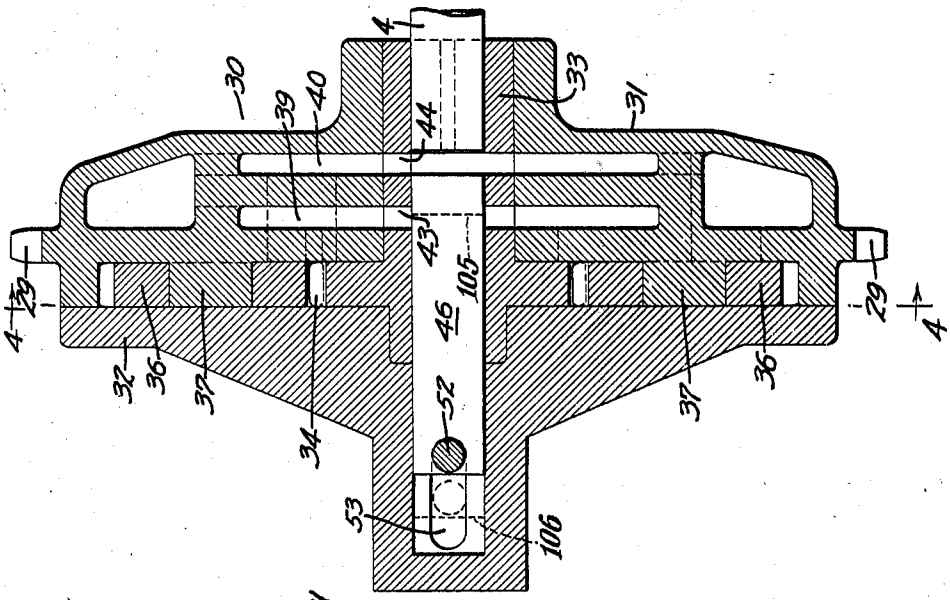
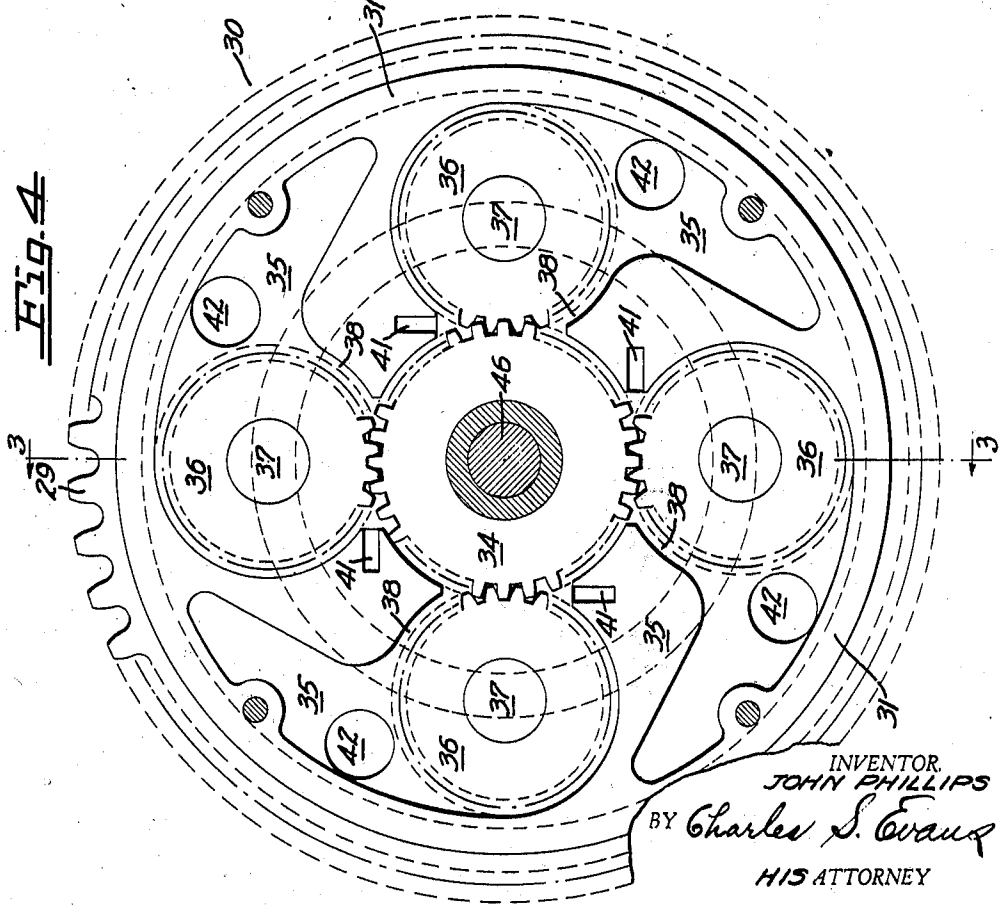


Fig. 4



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Fig. 5

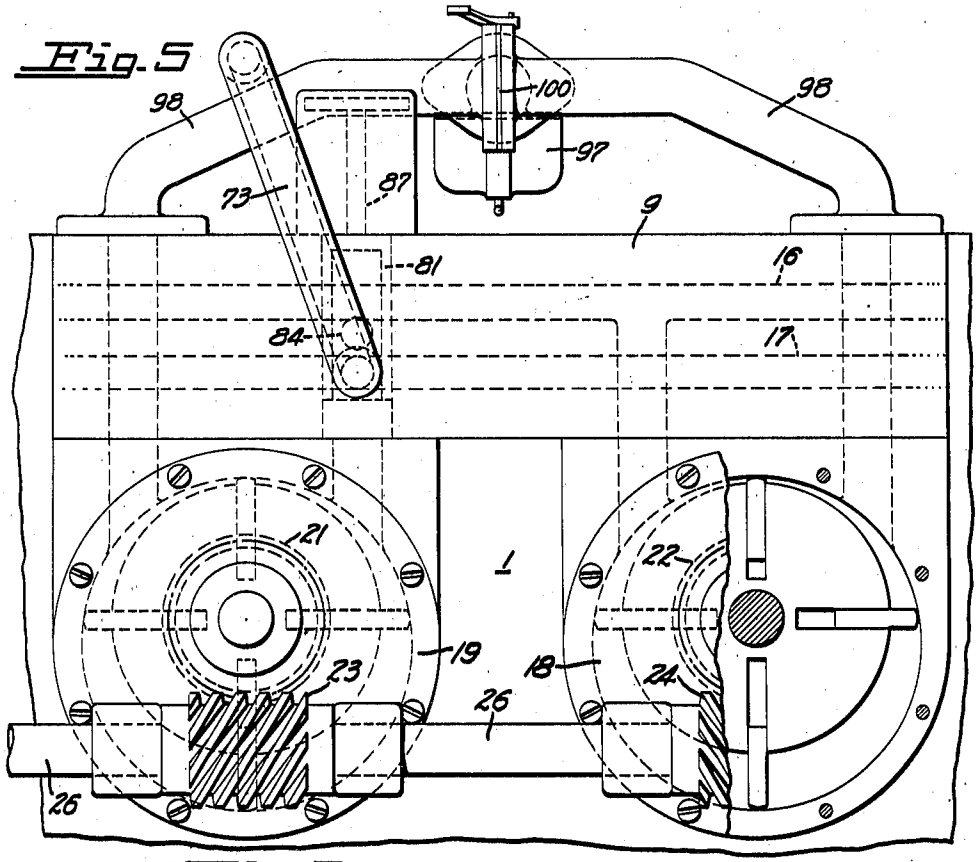


Fig. 6

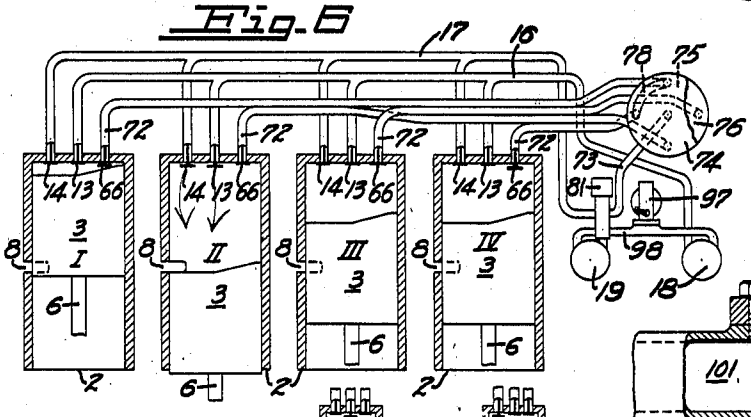


Fig. 9

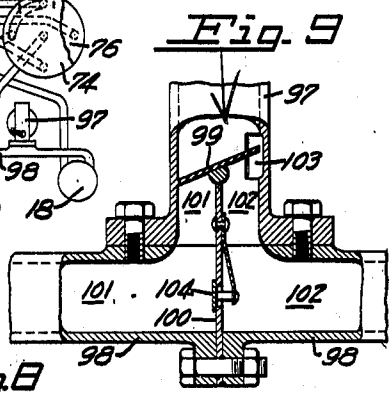


Fig. 7

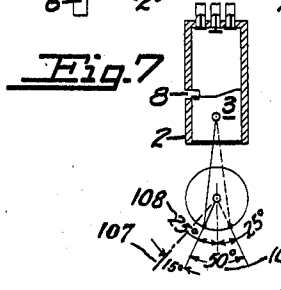
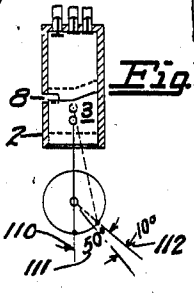


Fig. 8



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Patented Feb. 12, 1935 1,990,565

UNITED STATES PATENT OFFICE

1,990,565

INTERNAL COMBUSTION ENGINE

John Phillips, Oakland, Calif.

Application November 12, 1932, Serial No. 642,339

14 Claims. (Cl. 123—59)

My invention relates to internal combustion engines, and particularly to improvements in engines operating on the two-stroke cycle principle, one object being to provide a two-stroke cycle engine having increased efficiency, and which is flexible in operation to conform to varying loads and varying speeds.

Another object is to provide an engine having improved means for supplying fuel thereto to obtain a complete scavenging of the cylinders after each firing stroke, and to insure the introduction of a proper fuel charge to be compressed during each compression stroke.

Another object is to provide an engine having improved starting means whereby a compressed charge of fuel is introduced into successive cylinders independently of the normal intake valves to impart a starting impulse to the pistons, and to insure a prompt filling of the cylinders with a fuel charge.

Another object is to provide an improved means for regulating the speed of the pumps by which fuel is supplied to the engine cylinders in accordance with the speed of the engine.

The invention possesses other objects and features of advantage, some of which, with the foregoing, will be set forth in the following description of my invention. It is to be understood that I do not limit myself to this disclosure of species of my invention, as I may adopt variant embodiments thereof within the scope of the claims.

Referring to the drawings:

Figure 1 is a plan view of my improved internal combustion engine;

Figure 2 is a front elevation, with the upper portion shown in transverse section substantially corresponding to a line 2—2 of Fig. 1;

Figure 3 is a sectional view of the pump driving clutch and is taken on line 3—3 of Fig. 4;

Figure 4 is a sectional view of the main clutch member taken on line 4—4 of Fig. 3, a portion of the view being broken away;

Figure 5 is a fragmentary side elevation of the invention showing the fuel charging means with a portion of one pump head broken away to disclose the rotor, and with the end portion of the shaft therefor shown in section;

Figure 6 is a diagrammatic view illustrating the valve action of the several cylinders when the engine is being started;

Figures 7 and 8 are diagrammatic views showing the operation of the scavenging and supplemental intake valves respectively showing the different intervals of piston movement through which the valves are open;

Figure 9 is a broken sectional view of the fuel connection from the carburetor to the pumps.

In terms of broad inclusion, the engine of my invention comprises a plurality of cylinders each having a pair of intake valves arranged to admit portions of a fuel charge under pressure during different intervals of piston movement to expel spent gases from the preceding charge and to introduce a fresh charge of fuel into the cylinders.

My invention also contemplates the provision of separate pumps for introducing the charge into each cylinder in separate portions, and means for automatically controlling the speed of the pumps relative to the speed of the engine.

The invention also contemplates the provision of separate starting valves for each cylinder and means for introducing pressure into the cylinders at the start of the power strokes to the respective pistons independently of the intake valves for imparting an initial starting impulse to the pistons.

In terms of greater detail, the engine of my invention comprises a cylinder block 1 having a plurality of cylinders 2. In the preferred embodiment shown in the drawings, I have illustrated a four cylinder construction, but a greater or less number of cylinders may be used if desired. Pistons 3 are mounted in the cylinders 2, and are connected to a crankshaft 4 by means of connecting rods 6 engaging cranks 7 formed on the crankshaft 4. The cranks 7 are preferably disposed 90° apart.

Exhaust ports 8 are formed in each cylinder 2 adjacent the end of the power stroke of its piston 3. The exhaust ports are arranged to be uncovered by the piston as it approaches the end of its power stroke, and to be again covered as the piston starts back on the compression stroke thereof. The exhaust ports are so positioned that they will be uncovered through approximately 40° of crank movement before the piston reaches dead center at the end of its power stroke.

A cylinder head 9 is mounted upon the block 1. The cylinder head 9 is provided with intake ports 11 and 12 for each cylinder 2. Valves 13 and 14 are mounted upon the head 9 in association with the ports 11 and 12 respectively of each cylinder. An intake manifold passage 16 communicates with each of the cylinders through the ports 11, and a second intake manifold passage 17 communicates with each of the

cylinders 2 through the ports 12. The manifold passages 16 and 17 are independent of each other, and are preferably formed in the body of the cylinder head 9.

Fuel is delivered into the manifold passages 16 and 17 by suitable pumps 18 and 19 respectively. The pumps 18 and 19 may be of any desired type, rotary pumps of the form illustrated in Figure 5 being particularly well adapted for the purpose.

The pumps are preferably built into the cylinder blocks 1 and are driven by means of gears 21 and 22 engaged by worm gears 23 and 24 respectively. The worm gears 23 and 24 are carried by a shaft 26 driven by a chain 27, or other suitable driving connection, engaging a sprocket 28 upon the shaft 26 and a sprocket 29 carried by a clutch 30 upon the crankshaft 4.

The clutch 30 is preferably of the hydraulic type and comprises the main member 31 and an outer cover member 32. One suitable form of the hydraulic clutch is illustrated in Figures 3 and 4, though other forms of clutch may be substituted if desired. In the preferred form illustrated, the main member 31 is journaled on a sleeve 33 carried by a gear 34 keyed to the crankshaft 4. A plurality of recesses 35 are formed in the face of the member 31 to receive pinions 36 journaled upon lugs 37 formed upon the member 31, and meshing with the gear 34 which also seats within a recess in the face of the member 31. The gear 34 fits snugly between the main member 31 and the cover member 32 and against the walls of the recess except at points adjacent its point of engagement with the pinions 36. The recesses 35 are enlarged upon one side to form a chamber which communicates with the recess occupied by the gear 34 along a narrow passage 38.

Chambers 39 and 40 are formed in the main clutch member 31. A port 41 opens from the chamber 39 into the juncture of each recess 35 with the recess in which the gear 34 is seated at points immediately back of the points of engagement of the gear 34 with the pinions 36. A port 42 opens from the chamber 40 into the enlarged chamber portion of each recess 35. A plurality of ports 43 and 44 are formed in the sleeve 33 to match with the chambers 39 and 40. A plunger 46 is slidably mounted within the gear 34 and sleeve 33 for closing the ports 43 and 44 to normally prevent a flow of oil from one of the chambers 39 and 40 to the other.

The plunger 46 is arranged to be actuated by means of a suitable governor designated in general by the numeral 47 and comprising a plurality of weighted governor arms 48 pivoted upon lugs 49 carried by the outer clutch member 32. A link 51 forms a toggle connection between the weighted arms 48 and extensions 52 carried by the valve member 46. The extensions 52 project outwardly through slots 53 formed in the clutch member 32, whereby pivotal movement of the weighted members 48 will impart axial movement to the plunger 46. The outward movement of the valve member 46 is such that the ports 43 and 44 will be kept closed until the clutch member attains a predetermined speed of rotation. When that speed is attained, the displacement of the weighted members 48 by centrifugal force causes the valve member 46 to uncover the passages 43 and 44 as indicated by the dotted lines 105 and 106, sufficiently to permit a restricted flow of oil between the chambers 39 and 40.

The operation of the clutch is briefly as follows: The chambers 39 and 40, and the recesses in which the gear 34 and pinions 36 are seated between the clutch members 31 and 32 are kept filled with oil or other suitable liquid. When the crankshaft 4 is rotated in a clockwise direction as indicated in Figure 4, its movement is imparted to the gear 34 which in turn tends to impart rotation to the pinions 36. As both the gear 34 and the pinions 36 are seated snugly between the clutch members 31 and 32 and fit closely in recesses formed in the member 29, and the outer member 32 seats against the outer faces of the gear and pinions, no leakage of oil can occur around the gears or along the sides thereof. As rotation starts, the oil filling the space between the teeth of the gear 34 and pinions 36 cannot escape and therefore forms an incompressible cushion between adjacent teeth as they move into meshing relation and thereby locks the gear and pinions together and causes the pinions and clutch member to be carried with the gear.

When a predetermined rate of rotation is attained, the plunger 47 is withdrawn sufficiently to permit an escape of oil through the ports 41 to the chamber 39 and thence through the passages 43 and 44 to the chamber 40 and back to the face of the clutch member 29 through the port 42. This permits a restricted circulation of the oil through the clutch member so that a limited movement of the pinion 36 will be permitted, thereby causing the clutch member 29 to be rotated at a speed less than the speed of rotation of the gear 34. The movement of the plunger 46 is controlled by the governor weight 48 in a manner such that no circulation will be permitted until a predetermined rate of rotation is attained, and that thereafter the speed of the clutch member will lag behind the rate of rotation of the crankshaft and gear 34 in proportion to the increase of the speed of rotation and the crankshaft. Thus, the pumps 18 and 19 will be driven in direct proportion to the speed of the engine up to a certain point, but thereafter the pumps will be driven at a speed which increases less rapidly than that of the engine. This arrangement avoids excessive pump pressure when the engine is operated at high speeds.

The intake valves 13 and 14 are normally held closed by light springs 56, and are actuated by means of tappets 57. Valve rods 58 are connected at one end to the tappets 57, and at their opposite ends engage cams 59 and 60 upon a camshaft 61. Gears 62 and 63 upon the camshaft 61 and crankshaft 4 respectively drive the camshaft at the same speed as the crankshaft.

The cams 59 which actuate the valves 13 are arranged to open the valves 13 as soon as the pressure within the cylinder is reduced by the escape of exhaust gases through the ports 8 to a point approximately equal to the pressure of the charge built up by the pump 13. This occurs in each cylinder preferably after the exhaust ports thereof have been opened through 15° of crankshaft movement, as indicated at 107, Fig. 7. In other words, each intake valve 13 is opened approximately 25° of crankshaft movement before the corresponding piston reaches dead center at the end of its power stroke, indicated at 108. The valve 13 is kept open through substantially 50° of crank move-

ment and is closed when the piston has moved approximately 25° past dead center, as indicated at 109.

The cams 60 actuate the valves 14 to open approximately 25° of crank movement after the valves 13 have opened, or when the piston reaches its lower dead center, as indicated at 110, Fig. 8, remaining open through substantially 50° of crank movement indicated at 111, and closing approximately 25° after the valves 13 have closed. The valves 13 and 14 of each cylinder are open at the same time for approximately 25° of crank movement (see Figs. 7 and 8). The valve 14 is closed as the pressure in the cylinder approximately equals the pressure of the charge delivered by the pump 19. This occurs at approximately 10° of crank movement after the exhaust ports 8 are closed, as indicated at 112.

In addition to the intake ports 11 and 12 and the valves 13 and 14 associated therewith respectively, I provide for each cylinder a starting valve 66 mounted in a passage 71 in the cylinder head 9 to normally close a starting port 67. Each valve 66 is normally held seated by a light valve spring 68, mounted within the passage 71 over a valve stem 69 on the valve 66. A separate supply duct 72 is connected to each of the passages 71. A by-pass 73 is arranged to be connected at one end to the intake supply passage 17 supplied by the pump 19. At the opposite end, the by-pass 73 is arranged to be connected to the ducts 72 successively by means of a distributor 74.

The distributor 74 comprises a chamber 75 into which the distributing ducts 72 open. A rotor 76 is mounted within the chamber 75, the rotor 76 being arranged to seat against the wall of the chamber 75 through which the ducts 72 open. The rotor 76 is driven by means of a gear 77 meshing with the cam shaft timing gear 62 and operating to drive the rotor at the same speed as the cam shaft. A slot 78 is formed in the rotor 76, rotation of the rotor operating to move the slot 78 to uncover the ducts 72 successively, thereby connecting the by-pass 73 to the ducts 72 in the order in which their respective cylinders are fired. The movement of the slot 78 to open the ducts 72 is timed with respect to the movement of the pistons within the respective cylinders so that each duct 72 will be opened as the corresponding piston starts its firing stroke. The length of the slot 78 is such that each duct 72 will be uncovered for approximately 120° of movement, thereby insuring that each duct will remain open until the rotor 76 is moved to uncover the duct to the next succeeding cylinder.

The by-pass 73 is rendered operative to deliver a compressed charge from the pump 19 to the cylinders by means of a plunger valve 81 mounted within the passage connecting the pump 19 to the passage 17. The plunger 81 is normally held in a position opening the passage 17 and closing the by-pass 73 as indicated in full lines in Figure 2, the plunger being maintained in this position by pressure from the pump 19. The plunger 81 is arranged to be moved to a starting position closing the passage 17 and opening the by-pass 73. For this purpose I prefer to provide a solenoid 86 which is energized preferably by the electrical circuit through a starting motor. When the solenoid is energized, the plunger 81 is moved to its starting position, indicated in dotted lines in Figure 2, by means of a stem 87 extending into the solenoid. In the starting position, a port 84 is moved to register with the by-pass 73, thereby directing the compressed charge from the pump 19 into the by-pass.

A motor-generator 88 having a sprocket 89 is mounted upon a bracket 90 pivoted upon the cylinder block 1 by means of a pivot pin 91. A spring 92 mounted over an arm 93 carried by the bracket 90 and extending through a lug 94 on the block 1 operates to resiliently hold the motor-generator in operative position with the sprocket 89 engaging the pump driving chain 27. A nut 96 is threaded on the arm 93 for adjusting the position of the bracket to take up slack in the chain 27. The motor-generator 88 is arranged to operate as a motor when a starting circuit is directed therethrough in the ordinary manner. When the engine is in operation, the motor-generator is driven by the engine and thereafter functions as a generator.

In starting the engine, an electric circuit is directed in any suitable manner through the motor-generator 88 to energize the same as a motor and thereby drive the pumps 18 and 19 by means of the chain 27. At the same time, the solenoid 86 is energized to close the passage 17 and open the by-pass 73. While the pumps 18 and 19 are being driven by the motor-generator 88, the clutch 30 turns freely upon the crankshaft 4 so that the motor does not turn the engine. This is due to the fact that, when the main member 31 is driven by the chain 27, oil carried between the teeth of the pinions 36 may be displaced by the teeth of the gear 34 and forced back along the passages 38 to the enlarged chamber portions of the recesses 35. The circulation of oil so permitted allows the pinions 36 to turn freely over the gear 34 so that the pumps 18 and 19 may be driven independently of the engine.

The fuel supply is drawn into the pumps 18 and 19 through a suitable carburetor 97 connected to the intake sides of the pumps through an inlet connection 98. The carburetor is provided with the usual butterfly valve 99 by which the supply of fuel may be throttled as desired. A partition 100 is mounted in the inlet connection 98 to divide the same into two passages 101 and 102, one of which connects with the pump 18 and the other with the pump 19. The partition 100 extends to a point immediately next to the axis of the butterfly valve 99. A relatively large idling passage 103 is provided to permit fuel to be drawn into the passage 102 connecting with the pump 18 while the butterfly valve is closed. The valve 99 will substantially close the passage 101 to the pump 19.

A relief valve 104 is preferably mounted in the partition 100. The valve 104 is adjusted to open when the vacuum formed in the passage 101 while the pump 19 is in operation with the butterfly valve 99 closed reaches a predetermined point in order to avoid an excess of suction, and to borrow a portion of the idling charge from the passage 102 for delivery by the pump 19.

The operation of my improved two-stroke cycle internal combustion engine is as follows:

To start the engine, an electrical circuit is closed to energize the starting motor and the solenoid 86, thereby moving the valve 81 to its starting position and causing the motor to drive the pumps 18 and 19 as above described. The pumps 18 and 19 compress a combustible charge of air and fuel drawn through the carburetor 97 and into the pumps 18 and 19 through the pas-

sages 101 and 102. The compressed charge from the pump 19 is delivered into the by-pass 73 while the charge from pump 18 is delivered into the intake passage 16.

The charge delivered through the by-pass 73 passes to the distributor 74 and thence through the ducts 72 to those cylinders in which the pistons 3 happen to be at the start of a power stroke. The pressure of the charge delivered through a duct 72 forces the starting valve 66 associated therewith to an open position against the light resistance of its spring 68. In this manner, an initial pressure is delivered into the cylinders successively and imparts a starting impulse operating to move the pistons within the cylinders. The initial charge so introduced is not ignited but serves merely to displace the pistons by pressure.

At the same time, the compressed charge from the pump 18 is delivered through the connection 16 to one of the cylinders 2 through whichever intake valve 13 is open. The butterfly valve 99 is preferably kept open during the starting operation so that a full charge may be delivered by both pumps.

In Figure 6 I have illustrated diagrammatically the action that takes place. Assuming that the firing order is 1—3—2—4 and that the piston in cylinder I is at the start of its power stroke, the piston in cylinder II will be at the start of its compression stroke; and the pistons in cylinders III and IV will be midway of their compression and power strokes respectively. The intake valves 13 and 14 of cylinders I, III and IV will be closed. The intake valve 13 of cylinder II will be fully opened; and valve 14 will be just starting to open, but as the passage 17 is closed by valve 81, no fuel charge from pump 19 will be delivered past valve 14.

The compressed charge from pump 19 is delivered first to cylinder IV, and as its piston is displaced by the pressure of the charge, the distributor 74 is actuated to open the duct 72 to cylinder I and to close the duct 72 to cylinder IV. The charge from pump 19 is now directed to cylinder I and the piston therein is displaced along its power stroke by the pressure of the charge. The distributor then delivers the compressed charge from pump 19 to successive cylinders as their respective pistons start their power strokes. In the meantime, cylinder II will have received an initial compressed charge delivered by valve 13 from pump 18.

The initial charge admitted into cylinder II by valve 13 will be compressed during the compression stroke of the piston and will be fired in the ordinary manner as the piston reaches the end of its compression stroke. In the meantime, the piston in cylinder IV will have started its compression stroke and a charge of fuel from the pump 18 will have been admitted by valve 13 so that cylinder IV will be charged and will fire immediately following cylinder II. As the piston in cylinder I reaches the end of its initial power stroke, the exhaust port 8 will be uncovered, the initial pressure in cylinder I being released, and a rich charge from pump 18 will be delivered to the cylinder by valve 13. In normal starting operation, the initial charge admitted to cylinder II will be fired when the crankshaft has completed a half revolution so that normally only a half turn of the crankshaft is necessary to charge the cylinder and obtain the initial firing impulse. Thereafter, the several cylinders will fire in order in the ordinary manner.

As soon as one cylinder has fired, the starting circuit may be broken, and the valve 81 will be moved back by pressure from pump 19 to its normal position for closing the by-pass 73 and opening the passage 17. Thereafter, the distributor 74 will be driven in time with the operation of the motor, but will perform no function as no charge will be delivered thereto from the pump 19.

In the normal operation of the motor, fuel is charged into the cylinders as indicated diagrammatically in Figures 7 and 8. As each piston approaches the end of its power stroke, it uncovers the exhaust port 8, thereby permitting the substantially spent charge to escape from the cylinder. As soon as the pressure in the cylinder is reduced to approximately the pressure of the charge compressed by pump 18, the valve 13 is opened to admit a charge of fresh fuel from the pump 18 while the exhaust port is open. The inrushing charge displaces the spent residue of the preceding charge, and effects a thorough scavenging of combusted gases from the cylinders. At the same time the rush of gases through the exhaust port sweeps away any carbon or foreign matter in its path.

The valve 13 remains open after the piston passes dead center and starts its compression stroke. In the meantime, the intake valve 14 will have opened and a further charge will be admitted into the cylinder. The valve 14 remains open until the combined charge portions are compressed to substantially equal the pressure under which the charge from the pump 19 is injected into the cylinder which, as above stated, is at approximately 10° of movement after the exhaust ports 8 are closed.

The effective pressure under which charge is introduced into the cylinder from the pump 19 is substantially higher than obtained by pump 18. This is due to the fact that the valves 13, which admit the charge from the pump 18, open while the exhaust ports are open and while the pistons are moving along their power strokes. As a result, the pressure in the cylinder drops very rapidly and the pressure built up by the pump 18 is quickly dissipated. The valves 14 operating to admit charge from the pump 19 however open as the pistons start their compression stroke, and the pressure of the incoming charge is met by the pressure due to compression of the charge already trapped in the cylinder as the piston starts its compression stroke; and as a result the pressure built up by the pump 19 is not dissipated to as great an extent as in the case of the pump 18 and hence a higher effective pressure is built up and maintained.

By introducing the charge into the cylinders past separate intake valves, and during different intervals of the piston stroke, the first or scavenging portion of the charge effects a complete scavenging of the cylinders, and the second or surcharging portion of the charge is added to the low pressure charge which is trapped in the cylinder as the exhaust ports are closed, thereby insuring the delivery of an efficient charge into each cylinder. The combined charge portions are compressed during the compression stroke of the piston, and firing occurs in the ordinary manner as the piston ends its com-

pression stroke and starts the ensuing power stroke.

The provision of separate pumps 18 and 19 supplying portions of the charge through separate valves 13 and 14 is of particular value as a means for obtaining a quick pick-up from a slow or idling speed. While the engine is idling, the butterfly valve 99 is closed and only the fuel admitted through the idling jet passage 103 to the passage 102 is charged into the cylinders by the pump 18. While the valve 99 is closed, the pump 19 builds up a high degree of suction in the passage 101 so that when the valve 99 is opened a relatively rich charge of fuel is drawn from the carburetor and charged into the cylinders by the pump 19.

Pumps 18 and 19 are driven at a speed which increases in direct proportion to the increase in speed of the engine until a predetermined rate of rotation is obtained. Thereafter, the governor actuated valve 46 is moved to cause a lag in the rotation of the pumps thereby avoiding excess pressure when the motor is driven at high speed, and insuring an efficient operation of the pumps regardless of the speed of the engine. For installations wherein the engine operates under a substantially uniform load and at substantially constant speed, the governor actuated valve 46 may be omitted, as this feature is primarily useful where the load and speed varies through a wide range such as occurs in automotive practice.

The capacity of each of the pumps 18 and 19 is preferably such that when driven from the crankshaft each pump will deliver free air equal in volume to the piston displacement for each revolution of the crankshaft. When operating at high speed, each pump will build up and maintain sufficient pressure to insure a complete scavenging and an efficient charging of the engine cylinders. If the speed of the engine drops under an increase of load below the speed at which the governor 47 causes the pumps to lag, the pumps will build up and maintain a higher pressure. By thus introducing the fuel charge under a higher initial pressure, the charge and the final degree of compression within the cylinders is correspondingly increased. In other words, for a given throttle opening, the pumps will deliver the fuel charge at a higher pressure when the engine is operating at a slow speed than when the engine is operating at a high speed. The result is that the engine will operate smoothly and efficiently at slow speed under load, and the power output at low speeds is materially increased.

The air supplied through the carburetor may be preheated, and a relatively high degree of heat may be applied in any convenient manner to heat the mixture of air and fuel in the manifold passages between the carburetor and the pumps. The relatively large volume of air drawn through the carburetor prevents overheating at the carburetor and hence the inlet connection 98 may be heated to a high degree to vaporize liquid fuel drawn through the carburetor. Any liquid fuel which may pass into or be condensed in the pumps will be thoroughly atomized and mixed with the air by the action of the pumps. The engine of my invention is therefore particularly adapted for efficient operation with low grade fuels.

The pressure upon the pistons is always in the same direction, and hence the crankshaft bearings do not need to be finished and maintained with the high degree of accuracy necessary in a four cycle engine wherein the pressure upon the bearings is in one direction during the power stroke and in the opposite direction during the suction stroke of the piston. The provision of four cranks disposed 90° apart lends itself to a smooth continuous operation which tends to eliminate vibration and obtain operative efficiency as each of the cylinders will fire once during each full rotation of the crankshaft.

The valves require little or no attention. The exhaust ports may be made relatively large so as to facilitate a rapid scavenging of the cylinder. The seats of the intake valves are kept clean by the inflow of fresh charge thereby, and the internal pressure of the cylinder tends to keep the valves firmly seated.

I claim:

1. A two-stroke cycle internal combustion engine comprising a plurality of cylinders provided with pistons, exhaust ports arranged to be uncovered by the pistons during a portion of each stroke thereof, a pair of intake valves for each cylinder, a pump for introducing a portion of a fuel charge into each cylinder past one of the valves thereof after the exhaust port thereof is opened, and a second pump for introducing a separate portion of the charge past the other valve during a different interval of piston movement before the exhaust port is closed.

2. A two-stroke cycle internal combustion engine comprising a plurality of cylinders provided with pistons, exhaust ports arranged to be uncovered by the pistons during a portion of each stroke thereof, a pair of intake valves for each cylinder, a pump for introducing a portion of a fuel charge into each cylinder past one of the valves thereof after the exhaust port thereof is opened, a second pump for introducing a separate portion of the charge past the other valve during a different interval of piston movement before the exhaust port is closed, and driving means actuated by the engine for driving the pumps.

3. A two-stroke cycle internal combustion engine comprising a plurality of cylinders provided with pistons, exhaust ports arranged to be uncovered by the pistons during a portion of each stroke thereof, a pair of intake valves for each cylinder, a pump for introducing a portion of a fuel charge into each cylinder past one of the valves thereof after the exhaust port thereof is opened, a second pump for introducing a separate portion of the charge past the other valve during a different interval of piston movement before the exhaust port is closed, and means actuated by the engine for driving the pumps at speeds varying with the speed of the engine.

4. A two-stroke cycle internal combustion engine comprising a plurality of cylinders provided with pistons, exhaust ports arranged to be uncovered by the pistons during a portion of each stroke thereof, a pair of intake valves for each cylinder, a pump for introducing a portion of a fuel charge into each cylinder past one of the valves thereof, a second pump for introducing a separate portion of the charge past the other valve during a different interval of piston movement, means actuated by the engine for driving the pumps at speeds varying with the speed of the engine; and means for automatically reducing the speed of the pump relative to the speed of the engine above a predetermined rate of movement.

5. A two-stroke cycle internal combustion engine comprising a plurality of cylinders provided with pistons, exhaust ports arranged to be uncovered by the pistons during a portion of each stroke thereof, a pair of intake valves for each cylinder, a pump for introducing a portion of a fuel charge into each cylinder past one of the valves thereof, a second pump for introducing a separate portion of the charge past the other valve during a different interval of piston movement, means including a hydraulic clutch actuated by the engine for driving the pumps at speeds varying with the speed of the engine; and governor means in connection with the clutch for controlling the speed of the pumps relative to the speed of the engine.

6. A two-stroke cycle internal combustion engine comprising a plurality of cylinders provided with pistons, exhaust ports arranged to be uncovered by the pistons during a portion of each stroke thereof, a pair of intake valves for each cylinder, and separate pumps for delivering independent portions of a fuel charge from a common source to the cylinders past separate intake valves.

7. A two-stroke cycle internal combustion engine comprising a plurality of cylinders provided with pistons, exhaust ports arranged to be uncovered by the pistons during a portion of each stroke thereof, a pair of intake valves for each cylinder, a carburetor, and separate pumps for delivering independent portions of a fuel charge from the carburetor to the cylinders past separate intake valves.

8. In a two-stroke cycle internal combustion engine, a pump driven by the engine for delivering fuel under pressure into the cylinders past a set of intake valves, a second pump driven by the engine for delivering fuel under pressure into the cylinders through a second set of intake valves, and means for automatically controlling the speed of the pumps relative to the speed of the engine.

9. In a two-stroke cycle internal combustion engine, a pump driven by the engine for delivering fuel under pressure into the cylinders past a set of intake valves, a second pump driven by the engine for delivering fuel under pressure into the cylinders through a second set of intake valves, and means including a hydraulic clutch and governor for automatically controlling the speed of the pumps relative to the speed of the engine.

10. The method of charging fuel into the cylinders of a two-stroke cycle engine which comprises introducing a scavenging charge portion into a cylinder through one valve after the opening of an exhaust port in the cylinder and while the piston is ending its power stroke, and introducing a supplemental charge portion through a different valve and from a different source of pressure while the piston is beginning its ensuing compression stroke.

11. A two stroke cycle internal combustion engine comprising a plurality of cylinders each provided with a piston, an exhaust port of each cylinder arranged to be opened by the piston during approximately 80° of piston movement at the end of the power stroke thereof, an intake port for each cylinder arranged to be opened approximately 15° of piston movement after the exhaust port is opened, a second intake port for each cylinder arranged to be opened approximately 25° of piston movement after the first intake port is opened, and means for introducing separate portions of a fuel charge into each cylinder under pressure through the first and second intake ports thereof.

12. A two stroke cycle internal combustion engine comprising a plurality of cylinders each provided with a piston, an exhaust port of each cylinder arranged to be opened by the piston during approximately 80° of piston movement at the end of the power stroke thereof, an intake port for each cylinder arranged to be opened approximately 15° of piston movement after the exhaust port is opened, a second intake port for each cylinder arranged to be opened approximately 25° of piston movement after the first intake port is opened, and separate pumps for delivering independent portions of a fuel charge to each cylinder under pressure through the intake valves thereof.

13. A two stroke cycle internal combustion engine comprising a plurality of cylinders each provided with a piston, an exhaust port of each cylinder arranged to be opened by the piston during approximately 80° of piston movement at the end of the power stroke thereof, an intake port for each cylinder arranged to be opened approximately 15° of piston movement after the exhaust port is opened, a second intake port for each cylinder arranged to be opened approximately 25° of piston movement after the first intake port is opened, a pump for delivering a portion of a fuel charge to each cylinder under pressure through the first intake port thereof, and a separate pump for delivering a second portion of the fuel charge to each cylinder under greater pressure through the second intake port.

14. A two stroke cycle internal combustion engine comprising a plurality of cylinders each provided with a piston, an exhaust port of each cylinder arranged to be opened by the piston during approximately 80° of piston movement at the end of the power stroke thereof, an intake port for each cylinder arranged to be opened approximately 15° of piston movement after the exhaust port is opened, a second intake port for each cylinder arranged to be opened approximately 25° of piston movement after the first intake port is opened, a pump for delivering a portion of a fuel charge to each cylinder under pressure through the first intake port thereof, a separate pump for delivering a second portion of the fuel charge to each cylinder under greater pressure through the second intake port, and means for controlling the speed of the pumps relative to the speed of the engine.

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