Jan. 28, 1964 C. L. CUMMINS, JR 3,119,381

FUEL-SUPPLY SYSTEM

Filed Jan. 4, 1962

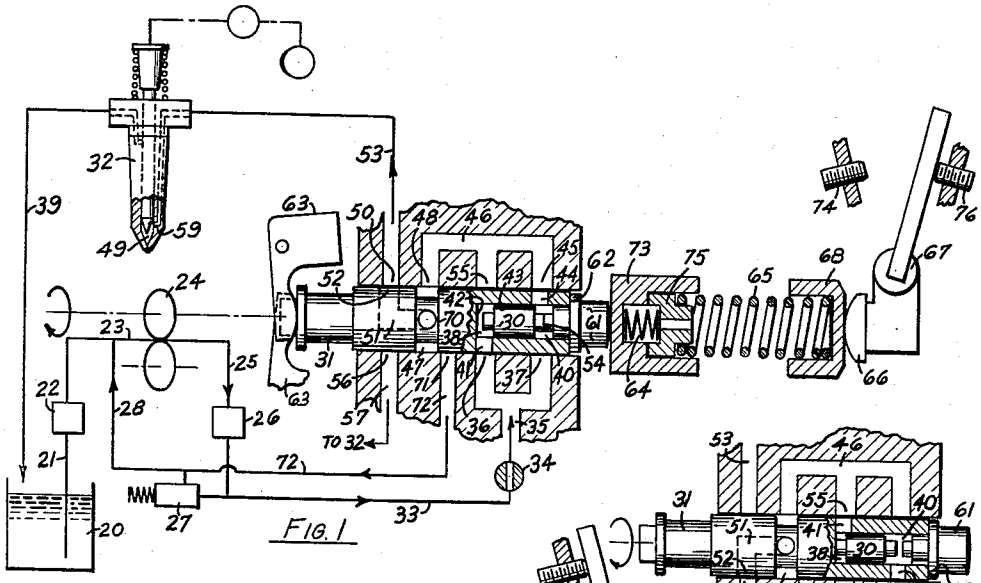


FIG. 1

FIG. 2

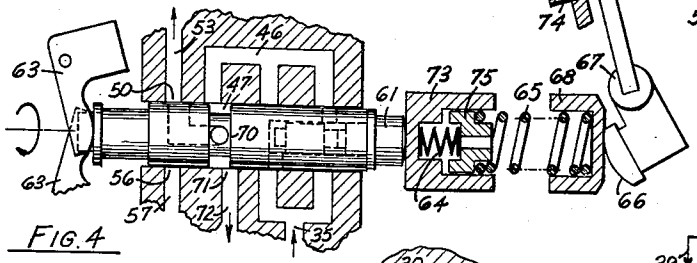


FIG. 4

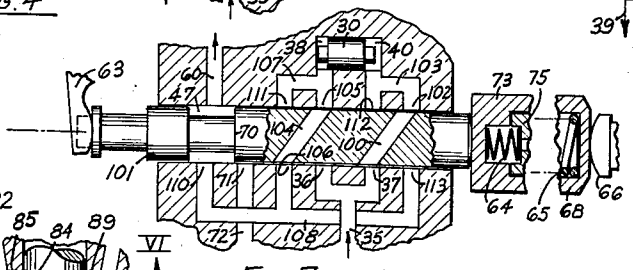


FIG. 7

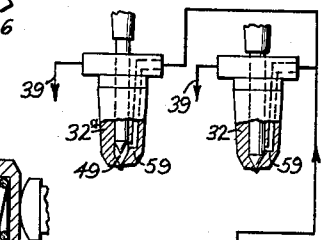


FIG. 3

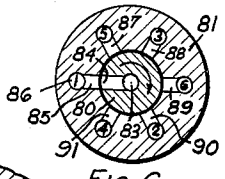


FIG. 6

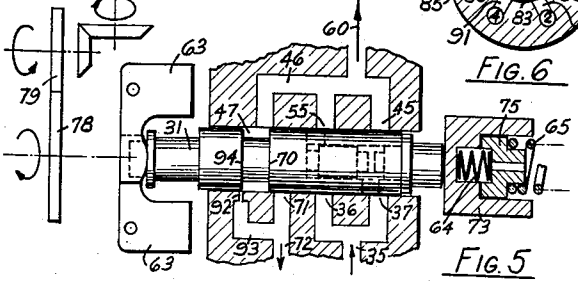


FIG. 5

INVENTOR
CLESSIE L. CUMMINS, JR.
BY ATTORNEY

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FUEL-SUPPLY SYSTEM

Clessie Lyle Cummins, Jr., Mill Valley, Calif., assignor to Clessie L. Cummins, Sausalito, Calif.

Filed Jan. 4, 1962, Ser. No. 164,284
19 Claims. (Cl. 123—140)

This invention relates to a fuel-supply system for compression-ignition internal-combustion engines and more particularly to a fuel-measuring pump of the positive displacement type for engines.

Even though the inherent economic and safety advantages of the compression-ignition or diesel-type engines are well recognized, there has been little success in developing a compact and reliable diesel engine which could be used, for example, in an outboard type of engine for marine use. One of the major reasons for this failure is the lack of an inexpensive but satisfactory fuel system. In the past diesel fuel systems have been relatively bulky and did not lend themselves to a reduction in size as the engine itself became smaller.

It is, therefore, an important object of this invention to provide a compact, inexpensive and easily serviced fuel-supply pump for diesel engines, including small, lightweight diesel engines.

While the fuel supply system lends itself to the small, marine engine field, it is to be understood that the invention can be applied also to diesel engines used in many other types of service, such as generator sets, earth-moving equipment, air compressors, automotive vehicles, etc.

Another object of the invention is to eliminate the need for rotating camshafts and complicated levers and rollers in positive displacement fuel pumps and also to get better results from a simplified device. The invention provides a metering mechanism utilizing a hydraulically actuated metering piston with a fixed stroke—the only oscillating part of the fuel feeding system. Costs are reduced, and part lives are extended.

Other objects and advantages will appear from the following description and the attached drawings in which:

FIG. 1 is a schematic view partly in cross-section of a fuel supply system embodying the principles of the invention;

FIG. 2 is a fragmentary view of a portion of the device of FIG. 1 showing the rotor-plunger rotated 180° from the position shown in FIG. 1 and distributing a fuel charge to the other of the two injectors;

FIG. 3 is a fragmentary view of a portion of a modified device generally similar to that of FIG. 1, but with the rotor-plunger in a modified type of housing and adapted to pump a fuel charge into a conduit leading to two paired injectors of a two-cycle engine;

FIG. 4 is a fragmentary view of a portion of the device of FIG. 1 showing the rotor-plunger in its idling speed position;

FIG. 5 is a schematic view partly in section of a portion of a modified form of fuel system embodying the principles of the invention;

FIG. 6 is a view in section of the distributor of FIG. 5 taken on the line VI—VI, showing a distributor for a six cylinder engine;

FIG. 7 is a fragmentary view of a portion of a modified form of fuel system also embodying the principles of the invention.

Important Characteristics of the Invention

Before giving a detailed explanation of the invention, I should like to summarize in general terms some of its outstanding characteristics.

The invention provides, in a fuel system for a diesel engine, a free-floating shuttle piston reciprocable between fixed stops in a cylinder. The cylinder is ported at each

end so that the pressurized fuel acts alternately on each end of the piston; so the pressurized fuel serves as the source of hydraulic power to operate the mechanism. There is no mechanical connection between the shuttle piston and the fuel-pump drive of the engine.

Fixed stops at each end of the cylinder limit the movement of the shuttle piston and so determine the maximum fuel charge delivered to the fuel injector. Therefore the maximum horsepower output of the engine is dependent on the distance between these fixed stops, and the maximum output can only be changed by increasing or decreasing this distance. One of the stops may be a replaceable unit (or, if desired, may be adjustable) so that, with the proper extension on the stop, various ratings of engines can be obtained from the same pump.

The invention provides in a two or four-cycle diesel engine of a compact cylindrical rotor-plunger, rotated at engine speed or at a speed bearing a fixed ratio to engine speed, the ratio being dependent on the number of engine cylinders and on the cycle type. The ported cylinder for the shuttle piston may be axially located within the rotor-plunger. This rotor-plunger is also slidably fitted in a bore in the pump housing and is positioned along the bore by engine-speed-created forces balanced against counterbalancing springs. The rotor-plunger rotates in order to connect the ported chamber on one end of the shuttle piston to the fuel-receiving chamber in a mechanically actuated injector at the same time that the ported chamber on the opposite end of the shuttle piston is connected to the pressurized fuel, so that the charge of fuel in the first-named ported chamber may be pumped to the fuel-receiving chamber in the injector. The rotor-plunger's axial position in its bore, determined by the balance of forces acting on each end of the rotor, controls the minimum and maximum speeds of the engine, as well as the rate of fuel delivery at all intervals between the two speed ranges.

In the above-stated combination the relationship between counterbalancing springs and a manually operated throttle stop acting against the end of the springs, enables the rotation of the throttle shaft to effect an increase or decrease in the force of the springs acting against the end of the rotor-plunger, whereby for a given engine speed the axial position of the rotor-plunger is determined by the position of the throttle shaft. The unwanted portion of the fuel charge normally sent from the fixed-volume ported chambers to the fuel receiving chamber in one of the injectors, is spilled off and returned to the fuel source.

The invention also provides, in combination with the slidably mounted and rotating cylindrical rotor-plunger a second rotor, turning at camshaft speed (for a four-cycle engine), which receives the charges from the rotor-plunger and distributes them to the fuel receiving chambers in mechanically actuated injectors in a properly timed sequence.

In some forms of the invention, there is the combination in a diesel engine having the mechanically actuated injectors, each with its fuel receiving chamber, of the slidably positioned cylindrical rotor-plunger, with its fuel-feeding ports connected so that one conduit from one port leads to two or more of the fuel receiving chambers; the rotor is in timed relation to the injectors so that only one of the fuel receiving chambers can receive a charge during the pumping stroke of the shuttle piston.

The Source of Pressurized Fuel for Combustion and for Hydraulic Operation of the Metering Mechanism (FIG. 1)

A fuel supply system embodying the principles of the invention is shown schematically in FIG. 1. This system is of such form that it may be mounted at any convenient location on the engine to obtain a drive synchronized to

the crankshaft. Fuel for the engine is adapted to be drawn from a tank 20 through a conduit 21, a primary filter 22, and a conduit 23 to an engine driven pump 24, herein illustrated as a gear pump turning at crankshaft speed.

Fuel under pressure from the pump 24 flows through a conduit 25 and a secondary filter 26 to a pressure regulator 27. Spill off from the regulator 27 passes back through conduits 28 and 23 to the suction side of the pump 24. It is the function of the regulator 27 to maintain a sufficient fuel pressure to operate a shuttle piston 30, which may be contained in a governor rotor-plunger 31, through its full permissible stroke for all desired engine speeds. Close control of pressure by the regulator 27 is unnecessary, for increasing the fuel pressure acting on the hydraulically operated piston 30 beyond the minimum required to move the piston through its full stroke at maximum engine speed can not increase the fuel delivery to injectors 32.

Fuel not bypassed by the regulator 27 passes through a conduit 33, a shutdown valve 34, and a branching conduit 35 to supply ports 36 and 37 at the governor rotor-plunger 31, which turns at crankshaft speed. The shutdown valve 34 may be either a manually operated valve or an electric solenoid valve.

Distribution of the Hydraulic Force to the Metering Mechanism and of Fuel to the Combustion Chamber (FIGS. 1 and 2)

The rotor-plunger 31 admits fuel under pressure, first to a metering chamber 38 at one end of the shuttle piston 30, and then to a metering chamber 40 at the other end, while the opposite end of the piston 30 is alternately connected to different injectors 32 (not all shown) in a timed sequence. A cycle of operations of the rotor-plunger 31 and piston 30 will clearly illustrate this. In FIG. 1 a rotor port 41 in the rotor 31 is indexed with the port 36, so that fuel under pressure can pass into the metering chamber 38 formed between a fixed end wall 42 and one end of the piston 30 sliding in its bore 43. At the same time that the port 36 is indexed with the rotor port 41, a rotor port 44 is indexed with an outlet port 45 leading into a conduit 46. A groove 47 in the rotor-plunger 31 is connected to the conduit 46 by a port 48 and also to a port 50 by an internal passage 51 (shown dotted) and its outlet port 52, which serves as a distributor port. The outlet port 50 leads to the injector 32 via a conduit 53. Thus, when the rotor port 41 aligns with the pressure port 36 and the rotor port 44 aligns with its outlet port 45, the fuel pressure in the chamber 38 pushes the piston 30 to the "right" against a fixed stop 54 and displaces the fuel in the chamber 40 through the conduit 46, the rotor groove 47, the rotor passage 51 and the conduit 53 to a fuel receiving chamber 49 in the mechanically actuated injector 32. (The injectors 32 all include fuel return lines 39 for carrying leakage fuel back to the fuel source or tank 20.)

In FIG. 2 the rotor-plunger 31 is shown turned through 180° to illustrate how the piston 30 is moved in the opposite direction to pump a fuel charge to another injector 32 (not shown). Now, the rotor port 44 aligns with the pressure port 37, and the rotor port 41 aligns with an outlet port 55, while the distributor port 52 aligns with an outlet port 56, so that the chamber 40 is connected to pressurized fuel, and so that the chamber 38 is connected by a conduit 57 with the fuel receiving chamber 49 of a second injector, and so that the fuel charge in the chamber 38 is pumped to the fuel receiving chamber of that second injector.

The fuel supply system shown in FIGS. 1 and 2 with two injectors (one not shown), and with the governor rotor 31 turning at crankshaft speed delivers two measured charges of fuel per revolution of the rotor 31. This combination of two distributor ports opening onto a rotor running at engine speed supplies the correct number of fuel charges for a two-cylinder, two-cycle engine. It is also easily possible to use this same rotor-plunger 31 for a four-cylinder, four-cycle engine in which four mechanically actuated injectors are employed. Then, however, two injectors must be paired as described below in connection with FIG. 3.

A Simplified Device (FIG. 3)

If the fuel system is to be employed on a two-cylinder, two-cycle engine only, the rotor-plunger 31 may be simplified as shown in FIG. 3. With a maximum of two injectors, and with the rotor still turning at crankshaft speed, the outlet ports 45 and 55 now remain permanently connected to a conduit 60 leading to two paired injectors 32, with a delivery port 59 adjacent to the fuel receiving chamber 49 in one of the paired injectors 32, cut off whenever the port 59 in the other injector 32*a* is open to its fuel receiving chamber 49. The rotor conduit 51, the distributor port 52, and the extra pump outlet 57 have thus been eliminated. By reducing the speed of the rotor 31 to one half engine speed, the construction of FIG. 3 may be used for a single cylinder, two-cycle engine or a two-cylinder, four-cycle engine with the two injectors paired. The operation of the piston 30, with its supply and outlet ports, is unchanged from that described for FIGS. 1 and 2, i.e., two fuel charges are pumped per one complete revolution of the rotor 31, regardless of the ratio of rotor speed to engine speed and whether the injectors are or are not paired.

Determination of Volume of Fuel Charge Delivered to Injectors (FIGS. 1–3)

The maximum volume of the fuel charge pumped is determined by the stroke of the piston 30. By increasing the stroke, the volume of fuel pumped is likewise increased. Part-throttle operation will be described shortly in the explanation of FIG. 4. In FIGS. 1–3 the piston 30 is shown reciprocating between the stop 42, forming the endwall of the bore 43 in the rotor-plunger 31, and the stop 54 on one end of a plug 61 permanently inserted into the rotor 31 to form the endwall of the chamber 40. Changing the distance between the stop 54 and a locating shoulder 62 on the end of rotor 31 is all that is required to alter the maximum stroke of piston 30, and, in turn, the maximum output of the engine. This alteration in length, accomplished by changing the length of projection of the stub end 54 on the plug 61, can be made only by the engine builder at the time of the pump's manufacture, and thus effectively prevents any unauthorized tampering with the maximum fuel delivery to the injectors. More importantly, for a series of engines requiring the same basic rotor construction, the identical rotor-plunger 31 and piston 30 can be utilized. Only the simple plug 61 need be different. This interchangeability of parts effects a great cost savings in the manufacture of the pump.

Method of Throttling and Controlling the Idle and Overspeed (FIGS. 1 and 4)

Throttling and governing of idle and overspeed are controlled by the interaction of three elements to change the axial position of the governor rotor-plunger 31 in its bore (FIG. 1). Centrifugal governor weights 63 are keyed to the rotor-plunger 31 and both rotate it and push it to the right (in FIGS. 1–5) with an increasing force as the engine speed increases. Counterbalancing the governor weight force to the right are a pair of springs, namely, an idle control spring 64 and a maximum speed control spring 65, positioned between the plug 61 in the end of the rotor 31 and a lever 66 fastened to a manually operated throttle shaft 67 and in sliding contact with a rear spring guide 68. With this governor-throttle combination the piston 30 always reciprocates between its fixed end stops 42 and 54 and discharges to the outlet ports 45 and 55 a charge of fuel for full

load conditions. Control for any condition less than a maximum charge delivery is achieved by returning the undesired volume to the suction side of the pump 24. Thus at an engine idle speed most of the charge is spilled off; at full load (wide open throttle) none is spilled off; and at over speed the entire charge is returned to the suction side of the pump 24. The percentage of fuel charge spilled away is determined by the axial position of the rotor-plunger 31, and this axial position is a function of the position of the throttle lever 66 and of the speed-responsive force of the governor weights 63 opposing the throttle-cam-varied forces of the springs 64 and 65.

In FIG. 1, the throttle shaft 67 is shown in its full throttle position, and the rotor-plunger 31 is so positioned that the entire charge pumped by the piston 30 is delivered to an injector. In FIG. 4, the throttle 67 is shown in the closed position, and the rotor-plunger 31 is axially located where a shoulder 70, on the right (FIGS. 1–5) edge of the groove 47, uncovers a spilloff port 71 leading into a conduit 72. The conduit 72 returns the fuel spilled off through the port 71 back to the conduits 23 and 23 and to the suction side of the pump 24.

While the primary application of the fuel system is intended to be for marine service, the system can be used on an engine in any type of duty. Therefore, the four basic axial positions of the rotor 31 will be described. These are for idling speed, coasting, overspeed, and full load at any speed.

During the idling cycle of the engine the weak spring 64, encased in its guide 73, counterbalances the opposing force of the weights 63. For a given idle spring force the speed of the engine is determined by the axial position of the shoulder 70. If the engine speed increases, the weights 63 move the rotor 31 to the right, causing the shoulder 70 to uncover more of the port 71. Less fuel is then delivered to the injectors 32 because of the increased spilloff; so the engine speed drops. The spilloff port area is then, in turn, reduced, and the engine speeds up once more. The rotor-plunger 31 quickly reaches an equilibrium position, and the engine idles at a substantially constant speed.

To increase, for example, the idle speed, the position of an adjusting screw 74, limiting the angular movement of the shaft 67 in one direction, is altered so that the lever 66 can move the rear guide 68 slightly to the left. At low engine speeds the stiff spring 65 is essentially uncompressed and serves only as a spacer. The lever 66, acting through the guide 68, the spring 65 and an idle spring stop 75, directly controls the static compressive loading on the idle spring 64. With more initial "squeeze" now on the spring 64, the rotor-plunger 31 is forced to the left and reduces the spilloff to the pump suction. Engine speed then increases, and the governor weights 63 soon position the rotor-plunger 31 in a new equilibrium position which is slightly to the left of its previous position before the adjustment was made. The final amount of the rotor shift to the left is somewhat less than the shift of the lever-actuated guide 68, since the increased governor force at the higher speed tends to compress the spring 64 slightly more than at the lower idle speed.

Whenever a load drives the engine, as in a vehicle coasting down hill, all fuel flow to the injectors 32 is stopped, by spilling off even the idle charge to the suction side of the pump 24. With the throttle shaft 67 in its idle position, the governor force is strong enough to position the rotor 31 where it causes the entire fuel charge to spill through the port 71 whenever the engine is above an idle speed and the load is rotating the engine.

As the throttle shaft 67 is rotated away from the idle speed stop 74, or as the engine speed exceeds a "fast idle," the weak spring 64 collapses, and the forward guide 73 makes contact with the idle spring stop 75. The spring 64 no longer has control over engine speed.

Rotating the shaft 67 has the net effect of positioning the governor rotor 31 further to the left in exactly the same manner as was described for the adjustment of the idle speed. While in the idle speed range, the spring 65 could be considered at its free length, but now it begins to shorten as the engine speed increases to the normal operating range. As there was no measurable preload on the spring 65 at idle, it collapses at a rate directly proportional to its compressive force. This force, applied by the weights 63 on the end of the rotor-plunger 31 increases at a rate proportional to the square of engine speed. Therefore, in order to prevent any spilloff at the engine's maximum rated speed at full load, the movement of the lever 66 must be enough to keep the shoulder 70 on the rotor-plunger 31 from uncovering the port 71 even though the governor weight force has reduced the effect of this movement by the amount of deflection of the spring 65.

The maximum governed speed of the engine is determined by the strength of the spring 65 and by a second throttle adjusting screw 76. The screw 76 only provides a "fine" adjustment over a very limited speed range. If the throttle is in its fully open position and the engine should begin to speed up due to a decrease in load, the governor force overcomes the predetermined maximum spring force and moves the rotor-plunger 31 to uncover the spill port 71 and divert the fuel charge, dependent on engine load, to maintain the engine at its maximum governed speed. If the load is further decreased, more fuel is spilled, etc.

One of the features of this governor, a modified form of a "variable speed" governor, is that the engine will also self-govern at any speed below its maximum rated speed. For example, if the rated full-load speed is 2,100 r.p.m., and the engine is operating a piece of equipment at a part-load speed of 1,800 r.p.m., it is unnecessary to touch the throttle shaft to maintain a constant speed if the load on the engine should suddenly increase. As soon as the governor weights 63 sense a drop in speed, their force lessens and the spring 65 pushes the rotor-plunger 31 to the left and reduces the volume of fuel spilled off. Speed then is returned to the original 1800 r.p.m. As long as the ability of the engine to maintain this new load is not exceeded (determined by the stroke of the piston 30), the engine speed will remain constant within a few percent. This degree of regulation thus makes the fuel supply system applicable for electric power generation.

This form of governor also lends itself to marine use, because the fuel delivered to the injectors is automatically reduced if the boat's propeller should lift out of the water as in a heavy sea or if, with an outboard engine, the propeller should be knocked out of the water when hit by an obstruction. The self-regulating speed feature under a suddenly decreased load works on the same principle but conversely to that described above for a suddenly increased load. With the throttle held constant, an increase in engine speed due to a load decrease causes a rotor shift to the right and more fuel spills off through the port 71. Obviously this feature applies to any variable load application where a constant speed must or should be maintained regardless of the load applied to the engine.

Method of Fuel Distribution by Means of a Distributor Rotor Synchronized with the Governor Rotor-Plunger (FIGS. 5 and 6)

In FIGS. 5 and 6 I show a modification to the fuel supply system of FIGS. 1–4, whereby the system becomes adaptable to any multi-cylinder diesel engine using mechanically actuated injectors. The governor rotor-plunger 31 now rotates at a multiple of engine speed, and the piston 30 pumps the fuel charges to a distributing rotor 80 which distributes them to the injectors in a timed sequence. The rotor 80 is inside a housing 81

and turns at crankshaft speed for a two-cycle engine and at one half speed for a four-cycle engine. However, the speed of the rotor-plunger 31 is a function of the number of engine cylinders in addition to the cycle. This ratio is fixed by the gears 78 and 79. For example, a six cylinder, four-cycle engine requires three fuel charges per engine revolution. Therefore, the rotor-plunger 31, which delivers two charges each revolution, must turn at one-and-one-half times engine speed to pump the three charges per crank revolution. The fuel charge is pumped from either chamber 38 or 40 through the conduit 60 to a groove 82 on the rotor 80. It then travels upward through a rotor passage 83, leading to a distributor port 84, which aligns with one of the radial discharge ports 85 and then into an outlet conduit 86 leading to an injector 32. Next, the governor rotor-plunger 31 rotates 180° and the distributor rotor 80 rotates 60° connecting the port 84 with another port 87, next with a port 88, then with a port 89, then with a port 90, and finally with a port 91. In one complete revolution of distributor rotor 80 a fuel charge is delivered to each of the six injectors in the firing order sequence, for example, 1-5-3-6-2-4 in FIG. 6.

If the modification of FIGS. 5 and 6 is adapted to an eight-cylinder, four-cycle engine the rotor-plunger 31 would turn two times engine speed to deliver four fuel charges per crank revolution. Also, the distributor housing 81 would necessarily contain eight radial discharge outlet ports rather than the six as shown in FIG. 6. If a six-cylinder, two-cycle engine were to use this fuel supply system (six fuel charges per crank revolution), the governor rotor-plunger 31 would need to turn three times engine speed. Pairing the injectors would reduce the number of ports by one half. Thus by the proper number of distributor ports and rotor speeds the system is readily adaptable to multi-cylinder diesel engines of both two and four-cycle design.

The System With the Rotor-Plunger in a Housing Separate From the Rotor-Plunger (FIG. 7)

While the piston 30 is shown in FIGS. 1–5 operating between fixed stops within the rotor-plunger 31, it should be understood that it is not necessary for the rotor 31 to contain piston 30. By making additional ports in the housing and rotor, the piston 30 can just as well be contained in the housing adjacent to the rotor. Control by the throttle shaft 67 and governor weight 63-springs 64, 65 combination are identical for either piston location.

In FIG. 7, fuel under pressure in the conduit 35 is conducted to the chamber 40 by an angling conduit 100 in a new plunger-rotor 101, which connects the port 37 to a rotor port 102 at one end of a branching conduit 103. At the same time, another angling rotor conduit 104 connects a rotor port 105 to a port 106. The fuel charge is then pumped from the chamber 38 through a second branching conduit 107, the rotor conduit 104, a third branching conduit 103 and to a port 110 leading into the groove 47 on the plunger-rotor 131. The fuel is then conducted to an injector through the conduit 60 as before. The position of the shoulder 70, controlling the area of the port 71, determines the volume of charge spilled off and returned to the suction side of the pump 24 through the conduit 72. When the rotor 101 is rotated 180°, the angling conduit 104 connects a port 111 on the conduit 107 with fuel under pressure at the port 36, and the angling conduit 100 connects a port 112 on the conduit 103 with a port 113 on the conduit 108. The piston 30 is then forced to the "right" by pressurized fuel entering the chamber 38, and the fuel in the chamber 40 is pumped to an injector through the newly connected conduits 103, 100, 108 and 60.

Prevention of Overfueling Upon Sudden Movement of Throttle to Maximum Position (FIG. 5)

If the throttle 67 is rotated suddenly from a part throttle position, corresponding to light load and low engine speed, to a "full throttle" position, it may be desirable in some applications to add a control to prevent momentary overfueling and/or overloading of the engine. This control is provided by the addition of a small spilloff port 92 and a connecting conduit 93 leading into the return conduit 72, as seen in FIG. 5. Normally, when the throttle is rapidly opened at a low engine speed, the rotor-plunger 31 is shifted leftward, with the shoulder 70 completely cutting off the port 71, so that the maximum fuel charge is delivered to the injectors momentarily. The governor weights 63 in the lower speed ranges do not have enough force to prevent this entire shutoff of the spill port 71. However, by means of the spill port 92, this transient overloading condition can not occur. When the sudden shift takes place, a left shoulder 94 of the rotor groove 47 uncovers the port 92 and spills off part of the charge. As the engine speed increases, the larger governor force causes the shoulder 94 to gradually close off the port 92 and let more of the fuel charge pass to the injectors. By the time the port 92 is completely cut off, the speed of the engine has reached the point where a maximum fuel charge will not cause overloading. If the engine is operating in the higher speed ranges and the throttle shaft 67 is suddenly rotated to its maximum position, the shoulder 94 never uncovers the port 92 because the governor force is strong enough to prevent such a large leftward shift of the rotor-plunger 31. This is permissible because the engine can now absorb its maximum fuel charge without being overloaded. If the operator applies the throttle moderately, the spill port 92 is not uncovered even at low engine speeds.

Summary of Operation

Fuel under pressure from the gear pump 24 is alternately directed into the metering chambers 38 and 40 at the opposite ends of the shuttle piston 30. When the fuel pressure is acting on one end of the piston 30, the chamber at the opposite end is connected to the fuel receiving chamber 49 of an injector 32. The secondary distributor rotor 80 of FIGS. 5 and 6 is used if necessary to accommodate any multi-cylinder engine.

The metering piston 30 shuttles between the fixed stops 42 and 54, its full stroke determining the maximum desirable fuel charge. During part-throttle, idle and overspeed conditions, the unwanted portion of the fuel charge is returned to the suction side of the pump 24 through the port 71. The force due to the governor weights 63 is counterbalanced by the force of the springs 64 and 65, in conjunction with the throttle 67, to axially position the rotor-plunger 31 and in turn the spilloff control shoulder 70.

In a multi-cylinder engine the additional injectors are supplied by the same rotor design by changing the speed of the rotor 31. The ratio of rotor speed to engine speed is a function of the number of engine cylinders and of type of cycle, either two or four.

Control of Runaway

A very important advantage obtained by adopting this form of overspeed control is the prevention of the engine from "running away" if the suction line 21 between the tank 20 and gear pump 24 should rupture or if the tank should become empty while the engine is running. Without some form of "runaway" control a diesel engine can destroy itself under these conditions by overspeeding. This occurs if any significant amount of air fills the supply passages between the governor rotor 31 and the injectors 32. For all practical purposes fuel oil is incompressible, and under normal conditions the fuel in the supply passage is "batched" through by a new "slug" pumped either from the metering chamber 38 or 40 (FIGS. 1–5). But air is highly compressible and will continue to force fuel to the injectors in a manner similar to a pneumatic ram as long as the injector plunger supply

port 59 indexes with passages leading to the plunger chamber 49 (FIGS. 1 and 3). Thus, many times the fuel charge required will be injected into the engine causing an extremely rapid acceleration of the engine to speeds which are destructive. The engine will only slow down when the expanding air behind the column of fuel in the lines to the injectors has shoved all of the fuel into the plunger chambers and then injected into the engine or when the engine has destroyed itself. Even with the engine under heavy load there is a period of uncontrolled high speed, although not as dangerous as when under light load. By opening one end of the conduits between the governor rotor 31 and the injectors 32 to the suction side of the pump 24, any air which should pass into the conduits 53 (FIG. 1) or 60 (FIGS. 3, 5) can be sucked back to the supply pump 24 and eliminate the possibility of the pneumatic ram effect. At the same time the air can no longer be effectively pumped by the shuttle piston 30 because the rotor ports 41, 44 have almost passed out of index with their connecting fuel supply ports 36 and 37. The engine simply comes up to governed speed and remains at that speed until all of the fuel in the lines has been burned with no harm to the engine.

To those skilled in the art to which this invention relates, many changes in construction and widely differing embodiments and applications of the invention will suggest themselves without departing from the spirit and scope of the invention. The disclosures and the description herein are purely illustrative and are not intended to be in any sense limiting.

I claim:

1. In a fuel system for use with a compression ignition engine having fuel injectors, said system having

a housing with a bore having fixed stops, a shuttle piston in said bore, with its ends defining first and second ported chambers, one in each end of said bore between said piston and said fixed stops, means for charging fuel under pressure from a fuel source to each said chamber alternately, thereby to move said shuttle piston a fixed stroke and discharge fuel from the opposite chamber, and conduit means for connecting each said chamber at discharge to a fuel injector on the engine,

the combination therewith of:

speed-responsive means in said conduit means for reducing the amount of fuel delivered to said fuel injector at predetermined engine speeds by said shuttle piston at discharge, without affecting the fixed stroke of said shuttle piston.

2. The system of claim 1 having

a manually controlled throttle, and

yieldable means connecting said throttle to said speed-responsive means for varying the effect thereof on the fuel delivered to said fuel injector according to the position of said throttle.

3. The system of claim 2 having additional means for reducing the amount of fuel delivered to said fuel injector by said shuttle piston at discharge, said additional means being actuated by said yieldable means overcoming said speed-responsive means at low engine speeds.

4. The system of claim 1 in which each fuel injector is adjacent a combustion chamber and includes

a fuel injector housing having a bore therein,

a plunger reciprocable in said bore,

a fuel-receiving chamber formed in said bore upon retraction of said plunger, and

engine timed means for moving said plunger in said bore to inject the fuel from said fuel-receiving chamber into said combustion chamber, and in which

the means connecting each said ported chamber at discharge to a fuel injector includes a conduit connected at one end to said ported chamber and at its other end to two or more of said fuel injectors which inject fuel at different times from each other, and

means synchronizing said shuttle piston movement and the metering of fuel thereby with the means for moving said plungers of said injectors,

whereby said reciprocable plunger in each of said fuel-injector housings controls the time of deposit of fuel in each thereof from said ported chamber and functions as part of the fuel distributing mechanism for the engine.

5. The system of claim 1 as applied to a four-cycle engine and in which each fuel injector is adjacent a combustion chamber and includes

a fuel-receiving chamber,

a plunger mechanically actuated in said fuel-receiving chamber to force a fuel charge pumped to said fuel-receiving chamber into said combustion chamber,

a single conduit connecting a said ported chamber at discharge to

a selected pair of fuel injectors, which inject their charge of fuel at different times so that when the passage to the fuel-receiving chamber in one is open, the other is closed, and

means synchronizing said shuttle piston movement with the means for moving said plungers of said injectors,

whereby said injector plungers function as part of the fuel-distributing mechanism for the engine.

6. In a fuel system for use with a compression ignition engine with fuel injectors and a governor, said system having

a housing with a bore having limiting members fixed at each end, a shuttle piston in said bore between said fixed limiting members, with its ends and said limiting members defining first and second ported chambers, one in each end of said bore, a fuel reservoir, a fuel pump having a suction side connected to said reservoir, means for charging the fuel under pressure from said fuel pump to each said chamber alternately, thereby to move said shuttle piston a fixed stroke and discharge a fixed volume of fuel from the opposite chamber, and conduit means for connecting each said chamber at discharge to a fuel injector on the engine,

the combination therewith of:

spill-off means in said conduit means controlled by said governor for reducing the amount of fuel delivered to said fuel injector at predetermined engine speeds by returning a controlled portion of the fixed volume of fuel from the discharging chamber to said suction side of said fuel pump.

7. The system of claim 6 having

a manually controlled throttle, and

spring means connecting said throttle to said governor controlled means for partially overcoming the fuel-reducing effect of said governor controlled means by an increase in throttle opening.

8. In a fuel system for use with a compression ignition engine with fuel injectors and a governor, said system having

a housing with a bore having limiting members fixed at each end, a shuttle piston in said bore between said fixed limiting members, with its ends and said limiting members defining first and second ported chambers, one in each end of said bore, a fuel reservoir, a fuel pump having a suction side connected to said reservoir, means for charging the fuel under pressure from said fuel pump to each said chamber alternately, thereby to move said shuttle piston and discharge fuel from the opposite chamber, and means for connecting each said chamber at discharge to a fuel injector on the engine,

the combination therewith of:

means controlled by said governor for reducing the amount of fuel delivered to said fuel injector at predetermined engine speeds by returning a controlled portion of the fuel from the discharging chamber to said suction side of said fuel pump,

a manually controlled throttle,

spring means connecting said throttle to said governor controlled means for partially overcoming the fuel-reducing effect of said governor controlled means by an increase in throttle opening,

said spring means acting on said governor controlled means to prevent said reduction of fuel and said return of fuel at speeds below a first predetermined speed when said throttle is wide open, and

second means for reducing the amount of fuel delivered to said fuel injector by returning a second controlled amount of said fuel from said discharging chamber to said suction side of said pump,

said second means being actuated by said spring means overcoming the action of said governor at fully open throttle at speeds below a second predetermined speed lower than said first predetermined speed.

9. The system of claim 6 in which each fuel injector is adjacent a combustion chamber and includes

a fuel injector housing having a bore therein,

a plunger reciprocable in said bore,

a fuel-receiving chamber formed in said bore upon retraction of said plunger, and

engine-timed means for moving said plunger in said bore to inject the fuel from said fuel-receiving chamber into said combustion chamber, and

in which the means connecting said discharging ported chamber to a fuel injector includes a conduit connected at one end to said discharging ported chamber and at its other end to two or more of said fuel injectors which inject fuel at different times from each other, and

means synchronizing said shuttle piston movement and the metering of fuel thereby with the means for moving said plungers of said injectors,

whereby said reciprocable plunger in each of said fuel-injector housings controls the time of deposit of fuel in each thereof from said discharging ported chamber and functions as part of the fuel distributing mechanism for the engine.

10. The system of claim 6 as applied to a four-cycle engine and in which each fuel injector is adjacent a combustion chamber and includes

a fuel-receiving chamber,

a plunger mechanically actuated in said fuel-receiving chamber to force a fuel charge pumped to said fuel-receiving chamber into said combustion chamber, and

a single conduit connecting said discharging ported chamber to

a selected pair of fuel injectors, which inject their charge of fuel at different times so that when the passage to the fuel-receiving chamber in one is open, the other is closed, and

means synchronizing said shuttle piston movement and the metering of fuel thereby with the means for moving said plungers of said injectors,

whereby said injector plungers function as part of the fuel-distributing mechanism for the engine.

11. The system of claim 6 wherein one of said limiting members is replaceable by like members of different length to change the maximum stroke of said piston and thereby change the maximum fuel output of said system.

12. In a fuel system for use with a compression ignition engine with fuel injectors, said system having

a housing with a bore, a shuttle piston in said bore, with its ends defining first and second ported chambers, one in each end of said bore, a fuel source, a fuel pump having a suction side connected to said fuel source,

first conduit means for charging fuel under pump pressure to each said chamber alternately, thereby to move said shuttle piston and discharge fuel from the opposite chamber, and second conduit means for connecting each said chamber at discharge to a fuel injector on the engine,

the combination therewith of:

fixed stops for determining the extent of movement of the shuttle piston in each direction, so that said shuttle piston has a constant stroke in both directions,

engine-speed responsive means,

a plunger-rotor rotating at a speed synchronized with the speed of the engine to open and close alternately the connection to each said chamber from said fuel pump via said first conduit means, and alternately to open and to close said second conduit means from each said chamber, all in synchronism with said fuel injectors, and

third conduit means for returning spilled-off fuel from said chambers at discharge to said suction side, when connected to said chambers,

said plunger-rotor being connected at one end to said engine-speed responsive means for moving axially toward its other end under control of said engine-speed responsive means to connect the discharging chamber to said third conduit means and thereby reduce the amount of fuel delivered to said fuel injector at predetermined engine speeds.

13. The system of claim 12 having

a manually controlled throttle lever adjacent the other end of said plunger-rotor and

spring means between said other end and said lever opposing the axial movement of said plunger-rotor toward said other end when so urged by said engine-speed responsive means,

so that movement of said throttle lever against said spring means increases the resistance thereof to movement of said plunger-rotor toward said other end and consequently the resistance to the reduction in the amount of fuel delivered to said fuel injector.

14. The system of claim 13 having

fourth conduit means for the prevention of overfueling at low engine speeds by return to said suction side of a fraction of the fuel being discharged from said chambers,

said plunger-rotor having means for connecting said fourth conduit means to said chambers at discharge at times when said spring means has prevented connection of said third conduit means to said chambers at discharge due to the engine speed being lower than a first predetermined speed at which said engine-speed-responsive means causes said plunger-rotor to connect said third conduit means at discharge and

when, in addition, said engine is under substantially full throttle and at a second predetermined speed slower than said first predetermined speed.

15. The system of claim 13 wherein said spring means comprises two springs in series, one of said springs being a light spring used for yielding readily under idling conditions of the throttle and the other a relatively stiff spring yielding less readily.

16. The system of claim 12 wherein said housing is part of said rotor plunger, with the bore in said rotor plunger and said shuttle piston moving therein.

17. The system of claim 12 wherein said housing is a stationary member separate from said plunger-rotor.

18. The system of claim 17 wherein the plunger-rotor has a pair of conduits with axially offset ends that perform the opening and closing of the connection to said chambers of said first and second chambers, in alternation, each rotor conduit first connecting one of said ported chambers to said first conduit means and, second, half a revolution of said plunger-rotor later, connecting that said ported chamber to said second conduit means.

19. In a fuel system for use with a compression ignition engine with fuel injectors, said system having

a fuel source and a fuel pump connected to said fuel source,

the combination therewith of:

engine-speed responsive means,

a plunger-rotor rotating at a speed synchronized with the speed of the engine and moving axially under control of said engine-speed responsive means, said

plunger-rotor having a bore therein and stationary ends for said bore,

a shuttle piston in said bore, with its ends and the bore ends defining a ported chamber in each end of said bore,

conduit means for charging fuel under pump pressure to each chamber alternately, thereby to move said shuttle piston a fixed stroke and to discharge a fixed volume of fuel from the opposite chamber, and

a conduit means for connecting each said chamber at discharge to a fuel injector on the engine,

the rotation of said plunger-rotor opening and closing the connection from said fuel pump by said conduit means for charging fuel to each said chamber alternately, to open and to close the conduit means from

each chamber to fuel injectors alternately, all in synchronism with said fuel injectors, and the axial movement of said plunger-rotor controlling the amount of fuel delivered to said fuel injectors at predetermined engine speeds.

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