

June 12, 1956

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2,749,844

PUMP

Filed Nov. 1, 1951

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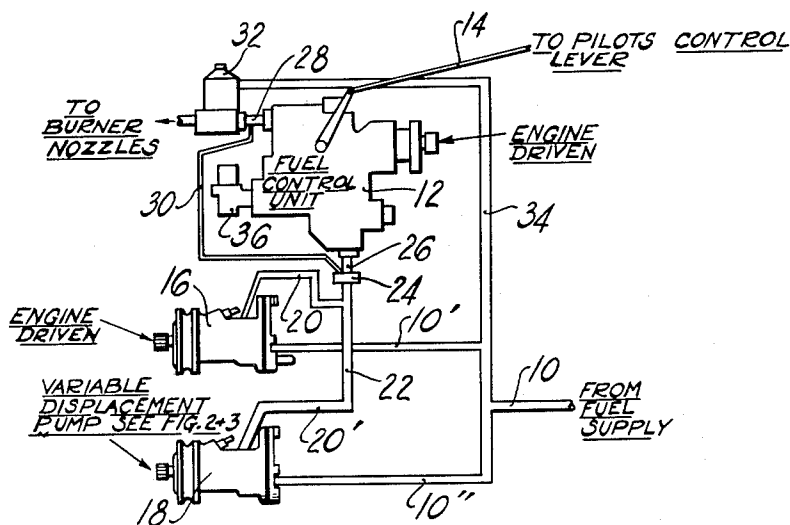


Fig. 1

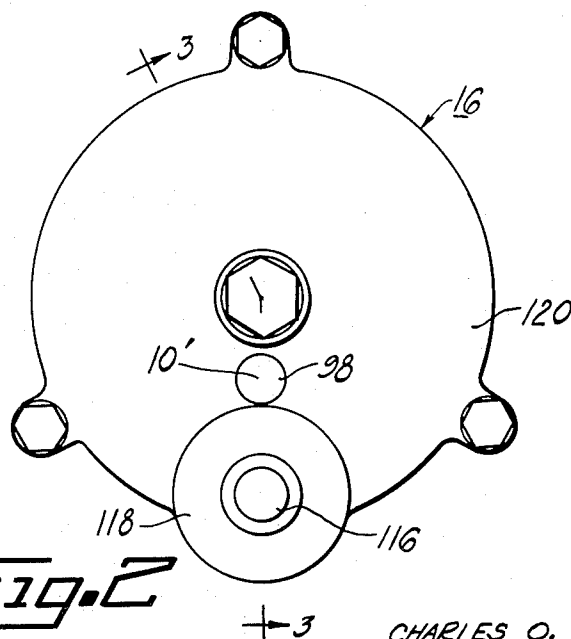


Fig. 2

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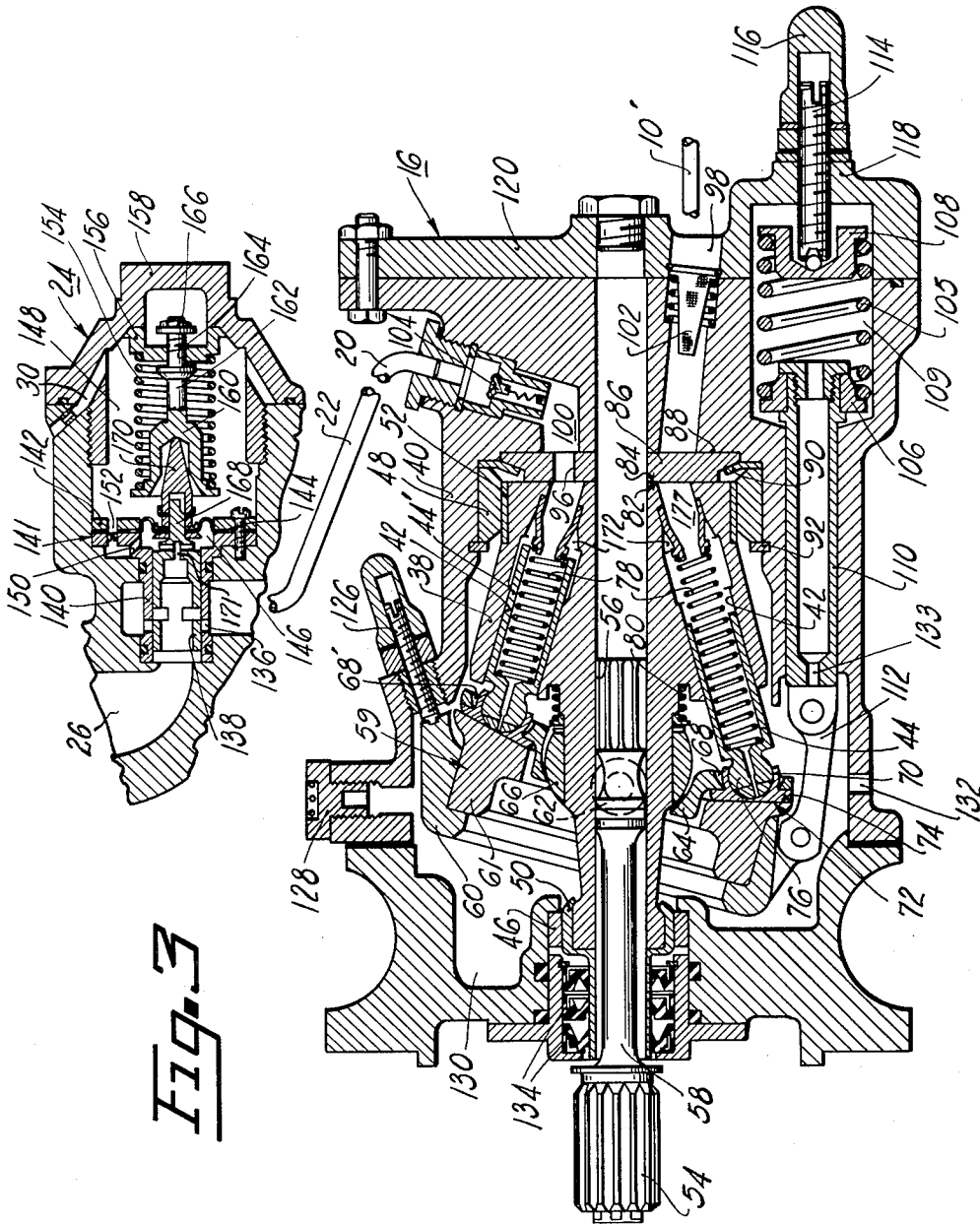
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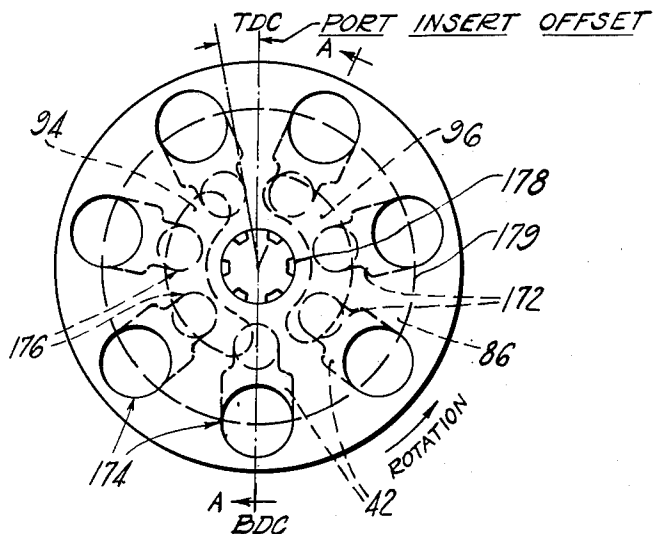


Fig. 4

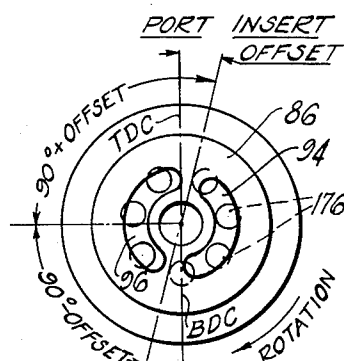


Fig. 5

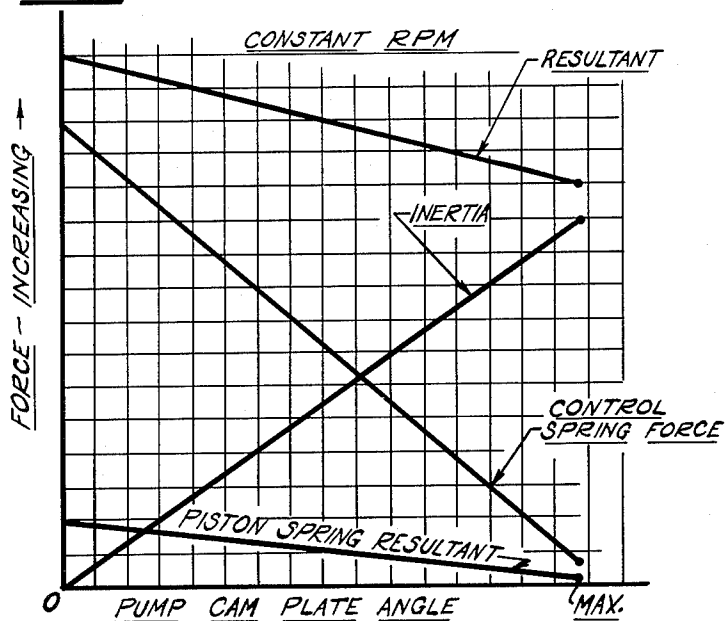


Fig. 6

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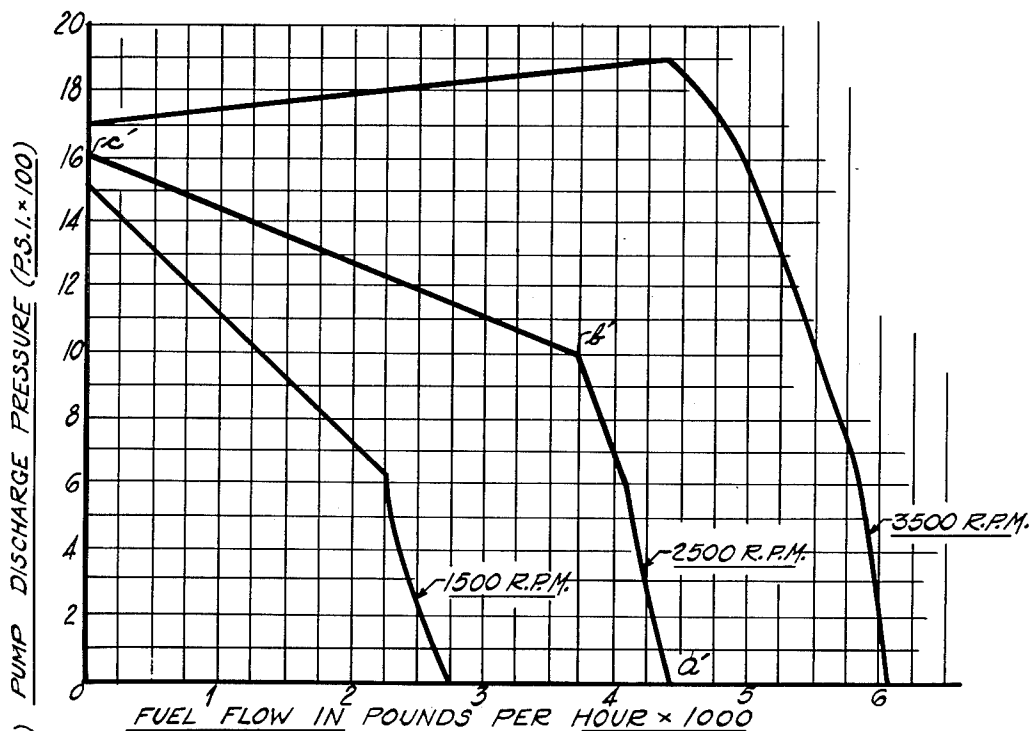


Fig. 7

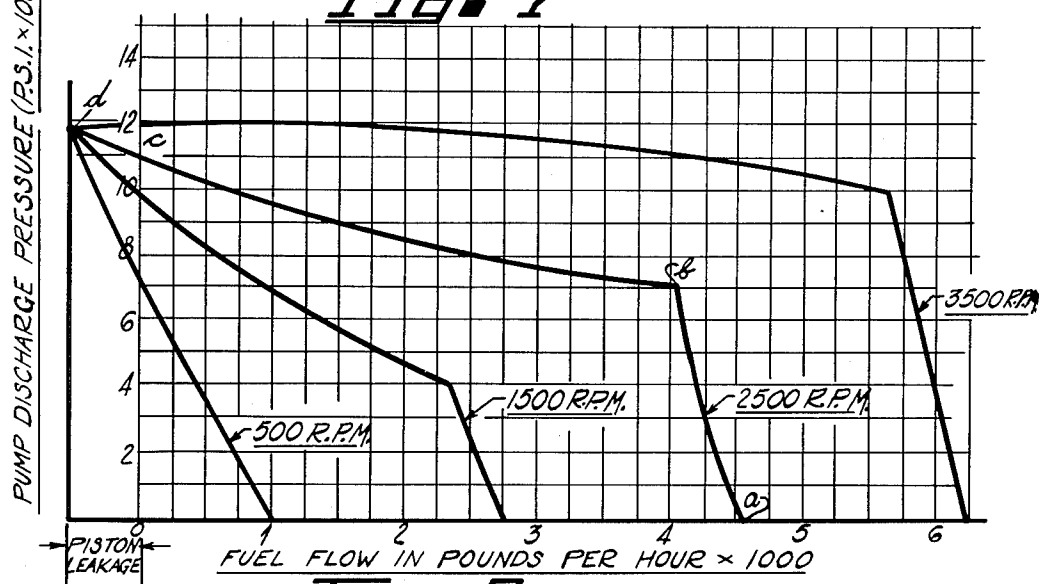


Fig. 8

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2,749,844

PUMP

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Application November 1, 1951, Serial No. 254,262

2 Claims. (Cl. 103—162)

This invention relates to pumps and more particularly to fuel pumps of the variable displacement, reciprocating plunger, swash plate type for pressurizing fuel to prime movers, adapted to respond automatically to a change in fuel flow and discharge pressure requirements. Certain types of engines, notably turbojet engines for aircraft, require extremely high delivery pressures at the burner discharge nozzles to obtain good fuel atomization and efficient combustion, and the nozzle pressure requirements may vary over a wide range between operation at sea level and at high altitudes. The reciprocating plunger, swash plate type of pump is well adapted to supply fuel at high pressures, but automatic regulation or variation of pump delivery in relation to engine fuel requirements has heretofore posed difficult problems, solved only by means of relatively complex control systems. It is therefore a primary object of this invention to provide in a pump of the type specified a simple, compact and self-contained regulator mechanism which acts to vary the angle of the swash plate as a function of pump discharge pressure.

A further object is to provide, in a pump of the type specified, simple automatic regulating means whereby great pump versatility is realized.

Another and more specific object is to provide, in a pump of the type specified, automatic regulating means wherein the pump displacement varies directly as a function of pump plunger inertia and inversely as a function of the pump discharge pressure, and wherein a resilient means tends to increase the pump displacement as an inverse function of pump swash plate angle.

A further object of this invention is to provide a pump of the variable displacement type especially adaptable for use in the fuel supply system of gas turbine engines for aircraft.

Other objects and advantages will be readily apparent from the following detailed description taken in connection with the accompanying drawings in which:

Figure 1 is a diagrammatic view of a fuel system for an aircraft gas turbine engine;

Figure 2 is a right hand end view of one of the variable displacement pumps in Figure 1;

Figure 3 is a sectional view of one of the variable displacement pumps shown in Figure 2 taken on line 3—3 of Figure 2 with a sectional view of a throttling valve shown connected thereto;

Figure 4 is a left hand end view of the pump rotor and port insert, shown in section in Figure 3, with the pump plungers removed;

Figure 5 is a right hand end view of the port insert, shown in section in Figure 3, with the ends of the rotor cylinders shown dotted and in the same relation to the ports of said insert as is shown in Figures 3 and 4;

Figure 6 is a graph illustrating force versus pump cam plate angle characteristics for a particular control spring rate and at a constant R. P. M.;

Figure 7 is a graph illustrating the discharge pressure

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versus flow characteristics when a relatively small port insert rotation is used; and

Figure 8 is a graph illustrating the discharge pressure versus flow characteristics when a relatively large port insert rotation is used.

Referring to the drawings, and first to Figure 1, fuel from a fuel supply tank (not shown) supplies the fuel system shown in Figure 1 through conduit 10 to the burner nozzles (not shown) in the combustion chambers of a gas turbine engine for aircraft. The fuel supply is under automatic regulation by means of a fuel control unit 12 which may be of any type suitable for the function which it is designed to perform. A fuel control unit of the type illustrated in the copending application Serial No. 74,322, filed February 3, 1949, of Dray and Kuzmitz, now Patent No. 2,628,472, may be used if the control amplifier unit, and its associated hydraulic circuitry, be deleted therefrom. The fuel is pressurized to the fuel control unit by means of variable displacement pumps, shown generally at 16 and 18, which receive fuel from inlet conduits 10' and 10'' and discharge fuel at high pressures through discharge conduits 20 and 20' into common conduit 22. Interposed between said pumps and fuel control unit 12 in conduit 22 is a throttling valve shown generally at 24 which maintains a constant head at all times across fuel control unit 12 and is sensitive to the control unit 12 pressure drop between control inlet conduit 26 and metered fuel pressure conduits 28 and 30. A pressurizing valve and shut-off cock unit is shown generally at 32 interposed between control unit 12 and the burner nozzles, not shown, in metered fuel conduit 28. The pressurizing valve proportions the fuel flow to the primary and secondary fuel manifolds, not shown, while the shut-off cock, when closed, enables the manifolds to be drained to atmosphere while relieving the fuel pressure in conduit 28 to the low pressure side of the fuel pumps through conduit 34. The unit indicated at 36 is a barometric element designed to automatically vary the fuel feed as a function of changes in entering air density. This unit prevents the fuel-air ratio from becoming too rich as altitude is gained thereby preventing excessively high burner temperatures and over-speeding of the engine.

Referring to Figures 2 and 3, the pump 16 comprises a rotor body 38 arranged in a casing 40 and having a plurality of annularly disposed bores 42 which are angularly symmetrical with respect to the axis of the rotor body, each of said bores containing a reciprocable hollowed plunger or piston, two of which are shown at 44 and 44'. The rotor body 38 is journaled in bearings 46 and 48 along sleeve sections 50 and 52 respectively of the rotor, said rotor being engine driven from driving spline 54 which is connected to an internally splined drive 56 by shaft 58. A nonrotatable assembly 59 consisting of a substantially cylindrically-shaped control ring 60 and a swash or cam plate 61 is centrally pivoted on a transverse axis thereof on trunnion pin 62. A rotatable auxiliary cam plate 64 is pivotal about a spring loaded thrust bushing 66, and has a series of holes, such as 68 and 68', therein which are adapted to receive cup-shaped plunger or piston slippers 70. The auxiliary cam plate 64 insures continuous sliding contact between cam plate face 72 and piston slippers bearing surface 74 when the pump is in operation. A universal jointure is formed between each piston slipper 70 and the balled end 76 of each pump plunger. Springs 78 insure pump plunger return after each pumping stroke and aid thrust bushing spring 80 in maintaining pressure contact between surfaces 82 and 84 of the rotor and a port insert member 86 respectively. When the control ring and cam plate assembly 59 pivots on trunnion 62, as during a period of change in demand on the pump 16, the auxiliary cam plate 64 pivots about thrust bushing 66, said auxiliary cam plate being pivotally

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driven to follow the movements of assembly 59 by the action of the pump plungers 44 and 44' and the plunger slippers 70. A port insert 86 is held rigidly against surface 88 of casing 40 by retainer 90, bearing 48 and annular key member 92. Arcuate-shaped inlet port 94 and outlet port 96 (Figures 4 and 5) in port insert member 86, offset with relation to the axis of trunnion 62 as projected on port insert 86, and also offset in relation to the vertical axis of said member as shown in Figure 5, provide communicating means between fuel inlet and outlet conduits 98 and 100, and rotor body cylinders 42. Filter means 102 is preferably inserted in inlet conduit 98, and a check valve 104 is included in discharge conduit 100 to prevent reversal of flow downstream thereof.

A control compression spring 105 mounted between retainers 106 and 108 in chamber 109, is operatively connected to the control ring and cam plate assembly 59 by a control shaft 110 and a rod 112 pivoted at one end to shaft 110 and at the other end to control ring 60. Spring 105, which may be manually adjusted by a screw 114 in cover 116 and boss 118 of the end plate 120 of the pump, tends to increase the angle of the control ring and cam plate assembly 59.

An adjustment screw 126 limits the maximum pump flow by limiting the maximum angle to which the control ring and cam plate assembly 59 can go. A vapor vent 128 is connected to housing cavity 130 and a fuel vent 132 connects cavity 130 to pump inlet pressure (connection not shown) thereby preventing fuel pressure buildup in cavity 130 due to leakage past the pump plungers. A vent 133, connecting cavity 130 to chamber 109 through control shaft 110, maintains pump inlet fuel pressure in said chamber thereby insuring hydraulic balance of said shaft. The drive shaft seals are indicated at 134.

Connected to pump discharge conduits 20 and 22 is the automatic throttling valve shown diagrammatically at 24 in Figure 1 and in cross section in Figure 3. It functions to maintain a constant pressure drop across fuel control unit 12, as hereinbefore mentioned, by area control of port 136. The valve itself, shown at 138, is shown as of cylindrical form mounted to slide in a sealed bushing 140 formed with a mounting flange 141, between which and a plate 142 is secured a diaphragm 144, the latter constituting a movable wall between chambers 146 and 148. Chamber 146 is vented to chamber 148 by vent 150 and passage 152. A spring 154 has one end abutting a fixed bushing 156 seated in a cap or cover 158, and the opposite end engaging a cup-shaped member 160 having a stem 162 slidably projecting through an opening 164 in the bushing 156, movement of the member 160 in a valve opening direction being limited by an adjustable nut 166. The valve 138 has connected thereto a stem 168 which projects through the center of the diaphragm 144 and carries a cone-shaped abutment member 170 which projects into the cup 160. Metered fuel pressure in conduit 28 (Figure 1) is communicated to chamber 148 through conduit 30, whereas throttling valve outlet pressure (fuel control inlet pressure) is communicated to chamber 146 by means of passages 171 in the body of valve 138, whereby the fuel pressure in conduit 26 will be maintained at a substantially constant value over and above the fuel pressure in chamber 148 as determined by the effective force of spring 154, or in other words, there will be a substantially constant pressure drop across the fuel control unit 12 under all conditions of operation.

Referring now to Figures 4 and 5, rotor body bores 42 decrease in size along conical sections 172 (Figure 3), shown diagrammatically in Figure 4, the plunger ends of the bores being indicated at 174 and the ports at 176, said ports being connected to the cylinders by passages 177. Inner and outer circumferential boundaries of port insert 86 are indicated at 178 and 179, respectively. To further clarify the relation of Figure 4 to Figure 3, a cross section taken on line A—A of Figure 4 would yield the

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view of the rotor body 38 and the port insert 86 seen in Figure 3. Fuel inlet port 94 of insert 86, which is curved to coincide with the path of the revolving cylinder ports 176 during the intake stroke of the plungers, is spaced farther from the top dead center position (TDC—position of cylinder when plunger is at end of discharge stroke) than from the bottom dead center position (BDC—position of cylinder when plunger is at end of intake stroke). The distance between the top dead center and port 94 in the direction of rotor rotation will normally be greater than the width of ports 176. In this construction, as the cylinders pass from top dead center to the point at which communication is established between ports 94 and 176, the pressure in said cylinders remains at substantially fuel outlet pressure. The moment communication between said ports is established the pressure in the respective cylinders immediately becomes equal to the fuel inlet pressure in conduit 98. The high pressure in the cylinders passing from top dead center to the point of communication between ports 94 and 176 exerts a force through the pump plungers urging cam plate 61 toward neutral position. This force will increase and decrease as the pump outlet pressure increases and decreases in response to variations in fuel demand. Fuel outlet port 96, which is curved to coincide with the path of ports 176 during the discharge stroke, is spaced farther from bottom dead center than from top dead center. The distance between bottom dead center and port 96 in the direction of rotor rotation will normally be greater than the width of ports 176. Thus, as the cylinders pass from bottom dead center to the point at which communication is established between ports 96 and 176, the pressure in the cylinders remains substantially equal to fuel inlet pressure. Ports 94 and 96 are preferably so disposed in relation to the bottom and top dead center positions, respectively, that they will be in communication with ports 176 of the cylinders in bottom and top dead center positions. The space or portions of metal between top dead center and port 94 and between bottom dead center and port 96 may be varied from one pump to another to satisfy the requirement of any particular installation or it may be varied in any particular pump to meet special operating requirements by rotating insert 86.

Operation

In the operation of the pump, drive shaft 58 rotates rotor body 38 at some predetermined ratio of engine speed, whereby pump plungers 44, 44' reciprocate in their respective cylinders as they are caused to follow non-rotatable cam plate face 72 on the piston slippers 70 by the combined action of springs 78, fuel inlet or discharge back pressure on the pump plungers, and auxiliary cam plate 64. As the control ring and cam plate assembly 59 variably tilts, due to a combination of forces to be hereinafter described in detail, the displacement of the plungers 44, 44' varies, and therefore the fuel flow varies, increasing as the cam plate angle increases relative to the vertical axis of trunnion pin 62, a maximum flow position as determined by the setting of manual adjustment stop screw 126, and decreasing to zero as the vertical axis of the cam plate assembly becomes perpendicular to the axis of the rotor body.

In Figure 3 plunger 44, shown at bottom dead center, has just completed its fuel intake stroke having sucked fuel from conduits 10' and 98 through inlet port 94 and is ready to begin discharging fuel at high pressure through discharge port 96. Plunger 44', shown just prior to reaching top dead center, is shown completing its discharge stroke and will remain substantially at pump discharge pressure until it reaches the approximate position past top dead center presently occupied by the plunger immediately adjacent to it in the direction of rotation (Figures 4 and 5).

Three distinct primary forces are present which tend to increase the angle of the control ring and cam plate assembly 59 when the pump is pumping fluid. Firstly, the

control spring 105 exerts a force in a cam angle increasing direction as do the springs 78 of the pump plungers which are moving across the lower half of the cam plate. Secondly, a relatively large motional force of inertia is acting on the cam plate to increase its angle as a result of the change in direction of movement of the pump plungers as they pass through the plane of bottom dead center just after termination of the suction stroke. This inertial force varies in a direct linear relation to the cam plate angle and directly as the square of the pump R. P. M. In addition to these primary forces the relatively small force of inlet fuel pressure, acting on the plungers traversing the lower half of the cam plate, acts in a cam plate angle increasing direction, the fuel at inlet pressure contained within casing 40, due to the venting of the casing cavity through port 132 to the inlet side of the pump, having a zero resultant force about trunnion 62.

Two primary forces are present which tend to return the cam plate assembly to a no-flow or neutral position. Firstly, the plunger springs 78, which are in a condition of maximum compression at top dead center, tend to return the cam plate to neutral while the plungers are traversing the top half of the cam plate. Secondly, the carry-over effect of pump discharge pressure, due to the offset porting arrangement, results in a large force which tends to return the cam plate to a neutral position. For example, as best illustrated in Figures 4 and 5, a plunger passing top dead center at pump discharge pressure remains substantially at said pressure until vented to inlet port 94 thereby exerting a neutralizing force on the cam plate 61 which is equal to discharge pressure multiplied by the effective area of the plunger times the moment arm of the cam plate about trunnion 62. This carry-over effect may be varied as required by varying the port insert offset angle. On the other hand, a plunger passing bottom dead center remains substantially at pump inlet pressure until such time as it breaks into the discharge port 96.

Referring now to Figure 6, the characteristic curves of those forces tending to increase the cam plate angle are shown. It is seen that at any given R. P. M. the force of inertia increases linearly with increasing cam plate angle and that the control spring force decreases linearly with increasing cam plate angle at a rate greater than the increase in inertia force so that a resultant negatively-sloped curve is insured. It has been found that, for best pump stability, a resultant cam plate angle increasing force should be used which decreases appreciably with increase in the angle of the cam plate. The control spring force and the pistons spring forces are, of course, substantially constant for all pump R. P. M.'s while, as previously mentioned, the force of inertia varies as the square of the R. P. M. Therefore, the resultant force curve shown in Figure 6 will, as the R. P. M. increases and decreases, decrease and increase its negative slope respectively, so that a fan of resultant force curves all emanating from the same point on the coordinate of force results. At each R. P. M. and cam plate angle the corresponding resultant force tending to increase the cam plate angle is balanced by the forces tending to return the cam plate to a neutral position. This balance of cam plate angle increasing forces automatically results from the action of the throttling valve 24 (Figure 2) and the effect of the pump discharge pressure on cam plate position as determined by the port insert offset angle. As, for example, the pilot moves lever 14 (Figure 1) to an increased power setting, the fuel flow to the engine increases due to the action of fuel control unit 12, resulting in an immediate decrease in fuel pressure in chamber 146 of throttling valve 24 and an immediate increase in fuel pressure in chamber 148 of said valve. As a result, valve 138 opens resulting in a momentary decrease of pump discharge pressure whereby the forces tending to increase the angle of cam plate 61 overcome the forces acting to decrease said angle and the cam plate moves to a new position resulting in the desired increase in fuel flow to the fuel

control unit 12 as the new discharge pressure acts, due to the porting arrangement, to balance the decreased control spring 105 force and the increased force of inertia.

Referring now to the characteristic pump curves shown in Figures 7 and 8, it is to be noted that the same control spring rate is used with reference to each set of curves, the differences in curve contour at any given R. P. M. being solely due to variation in the angle of port insert offset. The characteristic curves at the indicated R. P. M.'s in Figure 7 result from the use of a small port insert offset whereas those shown in Figure 8 result from the use of relatively large port insert offset. To further clarify the differences in pump operation as illustrated by these curves a comparison of the 2500 R. P. M. curves in each case will be made. With no restriction in the discharge conduits 20 and 22 the pump discharges fuel at a rate indicated by points a' and a in Figures 7 and 8 respectively, at which points the cam plate angle is maximum. As the discharge pressure is increased from atmospheric, as by means of the action of a valve such as that shown at 24, it follows curves $a'b'c'$ and abc , in each case. From a' to b' and from a to b the pump control is in a position of maximum displacement whereas from b' to c' and from b to c , pump displacement is steadily decreasing due to the action of discharge pressure carry-over effect on the port insert which returns the cam plate to neutral at points c' and c respectively, at which points the discharge line restriction is fully closed and no fuel is flowing. The breaking of the curves at points b' and b is a result of the discharge pressure action overcoming the control spring force and the force of inertia at maximum cam plate angle, and the lower pressure at which curve abc breaks is due to the greater port insert offset angle. Curve portions ab and $a'b'$ would be vertical but for fuel leakage past the pump plungers as discharge pressure increases. The upward negative slopes of curve portions $b'c'$ and bc are due to the control spring rate, which may be varied as desired. Point d of Figure 8 represents a theoretical point of intersection of the family of curves of Figure 8 if there were no piston leakage to the inner cavity of pump casing 40. The decreasing negative slope with increasing R. P. M. of the curves of Figures 7 and 8 respectively, after the respective points of breaking, is caused by the increasing inertial force with increasing R. P. M., so that a curve drawn through the respective break points of the curves of Figure 7, for example, would be, theoretically, a square curve. It is seen that the portion of the 3500 R. P. M. curve of Figure 7 after the break point assumes a positive slope due to a greater decrease in inertial forces that increase in control spring force as pump cam plate angle decreases (Figure 6), which is undesirable, and that therefore a greater port insert offset, such as used with Figure 8, will result in better pump stability at that speed. Desired variations in the characteristic curves of the pump can be easily obtained with this invention by simply changing the control spring rate used and/or the port insert offset. It has been found that, with proper selection of a control spring and optimum port insert offset, the stability of this type of pump is greatly improved as is the response time to changes in engine operating conditions. It has also been found that the subject pump control system is ideal for pumps operating in tandem (Figure 1), particularly where equal distribution of pump load is desirable under all conditions of operation.

While we have herein shown and described a preferred embodiment of our invention, it will be evident that the invention may be embodied in other forms. For example, the rotor body, as described in the preferred embodiment, may be made the stator with the engine drive connected to the swash plate and port insert, in which case a non-rigid connection between the control spring shaft and the swash plate would be used. Likewise, rotor body cylinders may be parallel to the axis of the pump. Also, the swash plate, as shown in the preferred embodiment, may be

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axially offset with respect to the trunnion, in either direction, thereby providing a larger or smaller moment arm on which the reactive forces of pump discharge pressure may act and in connection with which a smaller or larger port insert offset angle would be used respectively. Other modifications will likewise be apparent to those skilled in the art from the foregoing description taken in connection with the accompanying drawings.

We claim:

1. In a pump, an inlet and an outlet port, a plurality of cylinders adapted to register with said ports, a piston in each of said cylinders, a thrust member formed to engage said pistons, said member being centrally pivoted on a transverse axis thereof, resilient means connected to said member on one side of a plane through said axis for urging said member in one direction about said axis, said outlet port having one part disposed on said one side of said plane and having a second larger part disposed on the other side of said plane whereby a hydraulic reaction force is directed against a portion of said pistons to urge said member in a direction opposite to said one direction.

2. In a pump, an inlet and an outlet port, a plurality of cylinders adapted to register with said ports, a piston in each of said cylinders, a thrust member adjacent said

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cylinders to control the movement of said pistons, a pivot axis for said member passing substantially through the center thereof, resilient means connected to said member on one side of a plane through said axis for urging said member in a direction to increase piston stroke, at least a major portion of said outlet port being located on the side of said plane opposite to said one side whereby the reaction force against said pistons urges said member in a direction to decrease piston stroke.

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