March 30, 1965 A. F. VARNEY ETAL 3,175,499

PISTON PUMP HAVING A VARIABLE STROKE

Filed Dec. 31, 1962 4 Sheets-Sheet 1

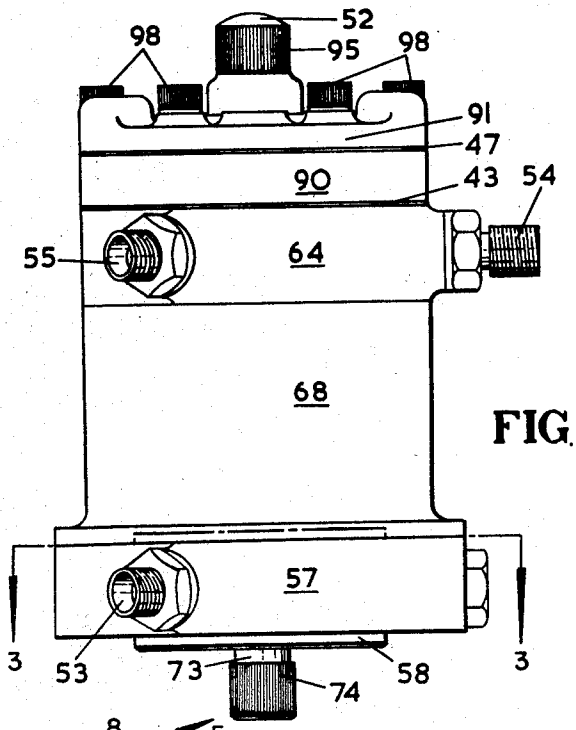



FIG. 1

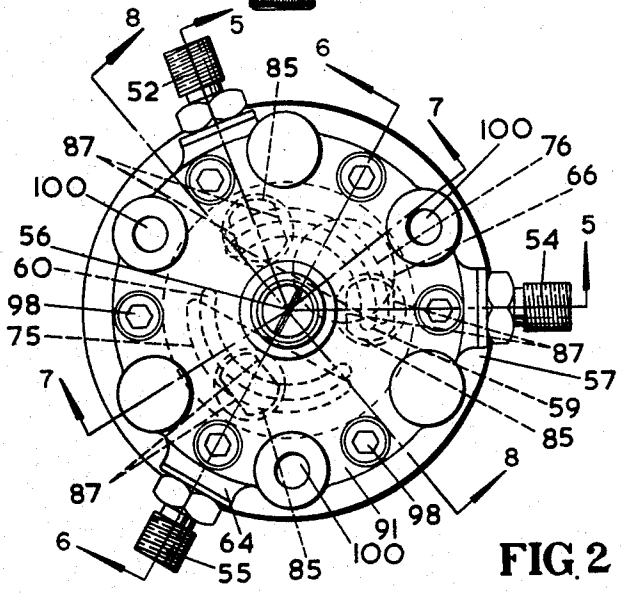



FIG. 2

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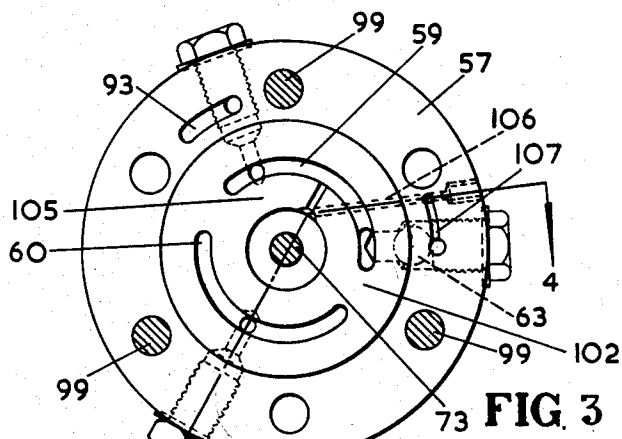



FIG. 3

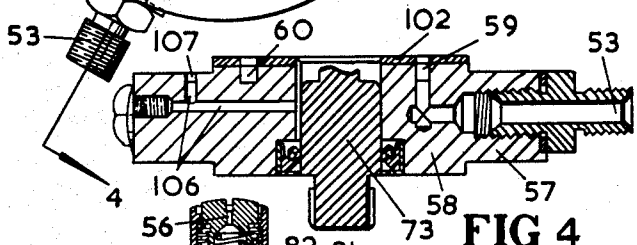



FIG. 4

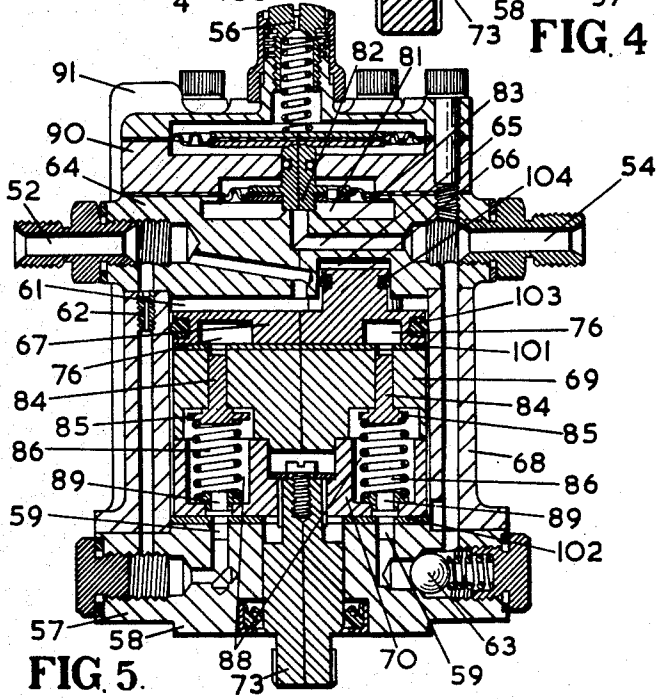



FIG. 5.

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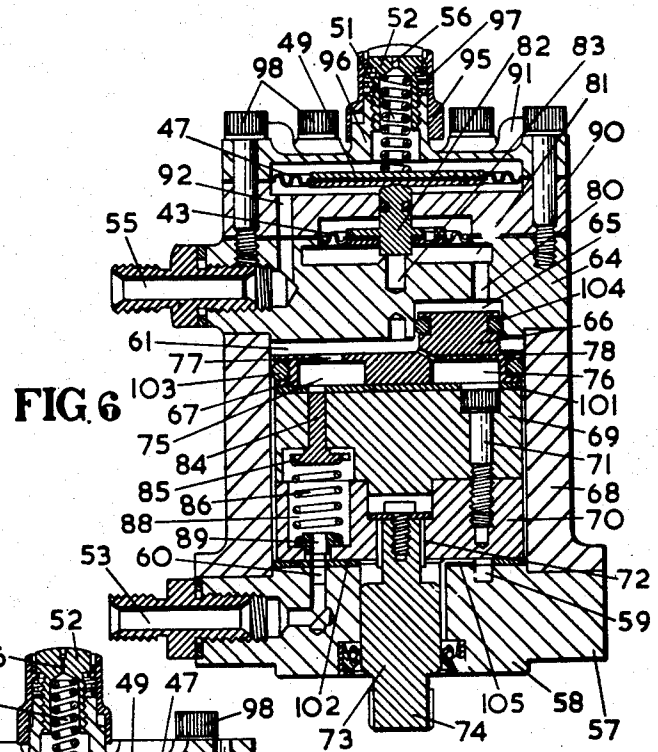



FIG. 6

FIG. 7

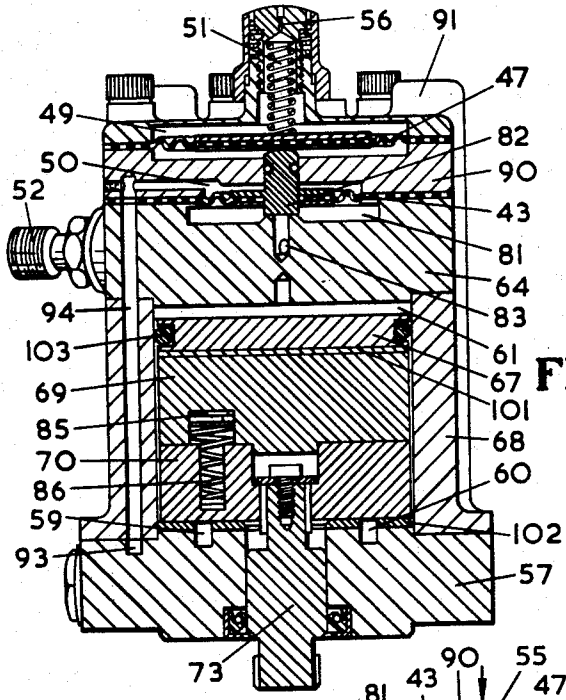

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73

FIG. 8

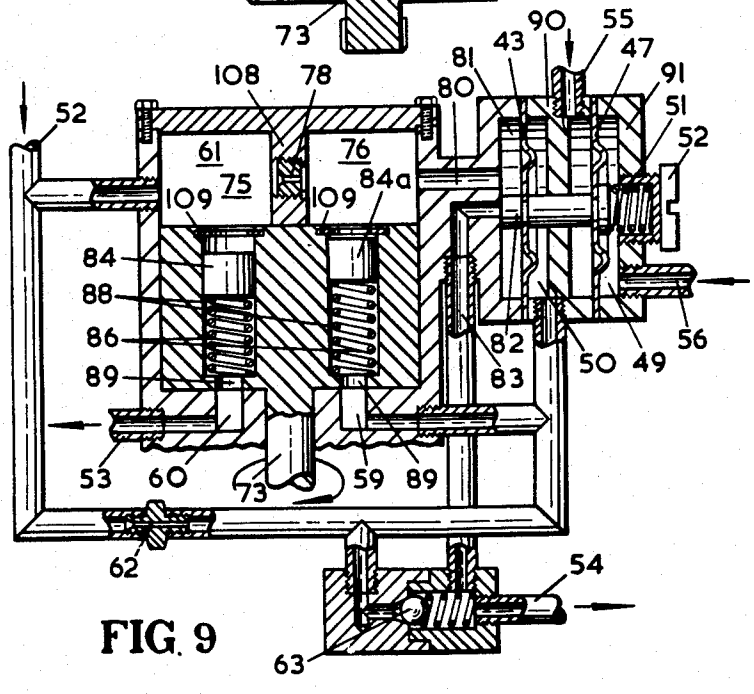

90
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73
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89
62
54
63

FIG. 9

United States Patent Office 3,175,499 Patented Mar. 30, 1965



3,175,499

# PISTON PUMP HAVING A VARIABLE STROKE

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3 Claims. (Cl. 103—38)

The invention relates to pumps of the kind in which a piston, working in a coacting cylinder, is associated with operating means which is adapted intermittently to move the piston against the action of a progressively increasing bias (e.g., a spring) whereby to cause the piston to reciprocate to give a pumping action. The invention is primarily intended for application to pumps of the kind set forth which are for supplying fuel to the induction system of an internal-combustion engine but has application in other fields.

According to the invention a pump, of the kind set forth, has a control fluid under pressure adapted to exert a force on the piston, in opposition to the bias, whereby variations in the pressure of the control fluid determine the length of the induction stroke and consequently the output of the pump at any given speed.

According to the invention, the piston has a head which is arranged to be acted on alternately by the control fluid to produce the induction stroke, and by the operating means, which is an operating fluid under such a pressure that the resultant force exerted on the piston produces the working stroke.

In the case where a pump is for supplying fuel to the induction system of an internal-combustion engine, the control fluid may be derived from the supply of lubricating oil for the engine. In the case where a pump is to be used as a metering device, the control fluid may be derived from the supply of fluid that is to be metered.

According to one feature, the piston may be reciprocable in a cylinder block which is supported for rotation in a casing provided with two chambers that are respectively for applying the pressure of the control fluid, and for applying the pressure of the operating fluid, to the head of the piston, according to the angular position of the cylinder block relative to the casing, the cylinder block and the casing being provided with an inlet port for supplying fluid for the piston to pump, and with an outlet port for the fluid that has been pumped by the piston, the ports being so arranged that the piston is connected to the inlet port whenever its head is in communication with the control fluid chamber, and that the piston is connected to the outlet port whenever its head is in communication with the operating fluid chamber.

In the case where the piston is arranged with its axis parallel with but radially-spaced from the axis of rotation of the cylinder block, the cylinder block may, according to another feature, have a circular end face coaxial with and lying in a plane normal to its axis of rotation, the end face having a communication with the piston head and being arranged to abut an axially-movable, but rotatively-stationary, piston which is guided by the casing and has the control fluid chamber and the operating fluid chamber defined in its end that abuts the cylinder block, the operating fluid chamber being supplied with operating fluid from a space situated between the end of the rotatively-stationary piston remote from the cylinder block and the casing, the effective area of the end of the rotatively-stationary piston remote from the cylinder block being such that the rotatively-stationary piston is urged into sealing contact with the cylinder block to check leakage between the operating fluid chamber and the control fluid chamber.

In such a case and according to a further feature, the end face of the cylinder block remote from the piston head may also be circular and be coaxial with and lie in a plane normal to its axis of rotation, the end face remote from the piston head being arranged to abut an internal wall of the casing defining the inlet and outlet ports, the effective area of the said end of the rotatively-stationary piston remote from the cylinder block being such that the end face of the cylinder block remote from the piston head is urged into sealing contact with the said internal wall of the casing to check leakage between the outlet port and the inlet port.

According to a still further feature, the pressure of the control fluid may be altered by means, associated with the apparatus being supplied by the pump, which is actuated responsively to a supply requirement of the apparatus and which varies the stroke of the piston for the pump to satisfy that supply requirement.

In such a case the pressure of the control fluid may, according to another feature, be altered by a variable-pressure relief valve which is actuated by a movable operating member, arranged to be acted upon by an appropriate operating parameter of the said apparatus, whereby to adjust the pressure of the control fluid such that the delivery of fluid, from the pump to the said apparatus, at least, substantially matches the supply of fluid required by the apparatus. When the said appropriate operating parameter is a fluid pressure, the movable operating member may be fast with diaphragm means connected to be acted upon by the said parameter.

According to yet another feature, the movable operating member may be biased by a spring to close the variable-pressure relief valve when the said apparatus is not operating or is operating at very low speed, and means are provided for adjusting the bias exerted by the spring whereby to tune the pump.

In the case where the said apparatus is an internal-combustion engine and the pump is for supplying fuel at the necessary rate to the induction manifold of the engine, a pump may, according to still another feature, be adapted to be driven at a direct function of engine speed whereby to enable the pressure of the control fluid to be made independent of the engine speed.

The invention as specifically applied to pumps for supplying fuel to the inlet manifold of a reciprocating spark ignition internal-combustion engine is illustrated, by way of example, in the accompanying drawings, in which:

FIGURE 1 is an elevation of a pump;

FIGURE 2 is a plan of FIGURE 1;

FIGURE 3 is a section taken on the line **3—3** of FIGURE 1;

FIGURE 4 is a section taken on the line **4—4** of FIGURE 3;

FIGURE 5 is a section taken on the line **5—5** of FIGURE 2;

FIGURE 6 is a section taken on the line **6—6** of FIGURE 2;

FIGURE 7 is a section taken on the line **7—7** of FIGURE 2;

FIGURE 8 is a section taken on the line **8—8** of FIGURE 2, and

FIGURE 9 shows, diagrammatically, the operation of the pump illustrated by FIGURES 1 to 8.

The pump is provided with a supply of fuel under high pressure through pipe union **52** and has an outlet **53** for metered fuel and a spill outlet **54** for by-passing excess fuel. For control purposes, the manifold depression is conveyed to the pump through pipe union **55**, and the atmospheric pressure through an aperture **56**. It will be seen from FIGURES 1, 3 and 4 that the metered fuel outlet **53** is arranged in a lower casing portion **57** which is integral with a boss **58** serving to locate the pump relatively to the engine casing.

The lower casing portion **57** is formed with an arcuate

fuel inlet port 59, and with an arcuate fuel outlet port 60 which is permanently connected to the metered fuel outlet 53 as will be seen in FIGURES 4 and 6. As will be best seen from FIGURE 5, the fuel supply union 52 is permanently connected to supply space 61 with fuel under high pressure, and to supply arcuate fuel inlet port 59 through an orifice 62 with fuel under low pressure. The pressure of the fuel in the arcuate fuel inlet port 59 is determined by the pressure drop across the orifice 62 and, when the supply of fuel to the inlet port exceeds the flow required by the pump, the excess fuel is passed to spill outlet 54 through relief valve 63 to prevent the pressure of the fuel in port 59 from exceeding a predetermined value.

Fuel supply union 52, spill outlet 54, and pipe union 55 for manifold depression, are carried by an upper casing portion 64 which has an eccentric recess 65 for locating an eccentric boss 66 of a piston 67, whereby to prevent the latter from rotating relatively to the upper casing portion 64. A tubular, intermediate casing portion 68 is sandwiched between casing portions 57 and 64 to define a chamber for the piston 67 and for a composite rotor having an upper portion 69 and a lower portion 70.

As will be seen from FIGURE 6, the portions 69 and 70 of the rotor are held together by bolts 71 and the lower portion 70 is arranged to be driven through splines 72 by a power input shaft 73 which is journalled in the lower casing portion 57 and has splines 74 for being driven by the camshaft of the engine.

The piston 67 is provided with two chambers 75, 76 which are of arcuate shape as will be appreciated from FIGURE 2. Chamber 75 communicates with space 61 through a port 77, shown in FIGURE 6, and contains fuel under high pressure. An orifice 78 allows fuel to escape from space 61 to chamber 76 which is connected to spill outlet 54 through bore 79 in eccentric boss 66, as will be seen in FIGURE 7, through port 80 to chamber 81 in the upper casing portion 64, see FIGURE 6, and through pintle valve 82 to spill outlet 83 as will be seen in FIGURE 5. It will be appreciated that the pintle valve 82 will control the flow of fuel from chamber 76 to spill outlet 54 and, consequently, will control the flow of fuel through orifice 78 and the pressure of the fuel in chamber 76 and chamber 81.

The upper portion 69 of the rotor supports three pistons 84 which are placed symmetrically about the axis of the rotor and have enlarged heads 85 which are acted upon by compression coil springs 86 and have chordwise flats 87, see FIGURE 2, to prevent the heads 85 from damping the motion of the pistons 84.

Each piston 84 has a cylinder 88, for containing the fuel to be pumped, which is permanently in communication with a port 89 in the lower portion 70 of the rotor.

The piston 67 is arranged relatively to the arcuate ports 59 and 60 so that the pistons 84 are acted upon by the high pressure fuel in chamber 75 whenever their respective ports 89 are registering with fuel outlet port 60, and are acted upon by the low pressure fuel in chamber 76 whenever their respective ports 89 are registering with fuel inlet port 59. Thus, when the rotor 69, 70 carries a piston 84 from high pressure fuel chamber 75 to low pressure fuel chamber 76, the piston 84 will perform an induction stroke and fuel will flow from fuel inlet port 59 into cylinder 88 and, when the rotor 69, 70 carries a piston 84 from low pressure fuel chamber 76 to high pressure fuel chamber 75, the piston 84 will perform a working stroke and will expel fuel from its cylinder 88 into fuel outlet port 60. The working stroke is limited by the head 85 of the piston 84 abutting the lower portion 70 of the rotor and, under normal operating conditions, the pressure of the fuel in high pressure fuel chamber 75 is arranged to be sufficient to cause the heads 85 to abut portion 70 of the rotor. The induction stroke is limited by the head 85 of the piston 84 abutting the upper portion 69 of the rotor and this prevents the pistons 84 fouling the piston 67. However, the induction stroke is controlled by the pressure of fuel in low pressure fuel chamber 76 which exerts a force on the piston 84 in opposition to the force of its spring 86. Thus, the induction stroke of the pistons 84, and consequently the output of the pump at any given speed, may be controlled by operating the pintle valve 82 to control the pressure of the fuel in low pressure fuel chamber 76. It will be appreciated that, when pintle valve 82 is fully open, the pressure of the fuel in low pressure fuel chamber 76 will be substantially zero, and the output of the pump at any given speed will be a maximum. Conversely, when the pintle valve 82 is closed, the pressure of the fuel in low pressure fuel chamber 76 will be equal to the pressure of the fuel in high pressure fuel chamber 75, and the output of the pump will be zero irrespective of the speed of the pump.

Chamber 81 is defined on one side by a diaphragm 43 which is sealingly anchored at its periphery between a plate 90 and upper casing portion 64, and is axially fast with pintle valve 82. A diaphragm 47 bears on pintle valve 82 and is sealingly anchored at its periphery between an end plate 91 and plate 90. Atmospheric pressure is applied to a chamber 49 through aperture 56 to act on one side of diaphragm 47, and the manifold depression is applied through pipe union 55 and passage 92, see FIGURE 6, to act on the other side of diaphragm 47. The pressure of the fuel in arcuate inlet port 59 is applied through arcuate passage 92 in lower casing portion 57, see FIGURES 3 and 8, and passage 94 to chamber 50 to act on the side of diaphragm 43 opposite to chamber 81.

Spring 51 biases the pintle valve 82 to close spill outlet 83 when the pump is at rest. A screwed adjuster 52 is a datum setting device which alters the setting of spring 51 for tuning the pump. As will be seen from FIGURE 6, the adjuster 52 is surrounded by a sleeve 95 which is held rotatively fast with end plate 91 by dog 96, and is urged into engagement with the adjuster 52 by a compression coil spring 97 whereby to prevent the adjuster 52 from rotating accidentally due to vibration from the engine to which the pump is mounted.

When the pump is running, any unbalance of the pressures acting on diaphragms 43, 47 will cause the pintle valve 82 to move, thus altering the flow of fuel through orifice 78 until the pressure of the fuel in low pressure fuel chamber 76, and thus chamber 81, has changed to such a value that equilibrium is regained.

The plates 90 and 91 and the diaphragms 43 and 47 are secured to the upper casing portion 64 by six bolts 98, and the casing portions 57, 64 and 68 are held together by three bolts 99 as will be seen in FIGURES 3 and 7. Three holes 100, see FIGURES 2, 3 and 7, pass through the pump and are for receiving studs for holding the lower casing portion 57 to the engine casing.

The pistons 84 are preferably made of hardened steel and the upper portion 69 of the rotor is preferably made of bronze to give a suitable bearing surface for the pistons 84. Lubrication difficulties occur when petroleum spirit is pumped and, for this purpose, the upper portion 69 of the rotor has a facing 101 of carbon graphite for bearing against the piston 67 which is of hardened steel. Similarly, the lower casing portion 57 has a facing 102 of carbon graphite for bearing against the lower portion 70 of the rotor which is also hardened steel.

The piston 67 is axially movable in the intermediate casing portion 68 and the effective areas of its face defining space 61, of its eccentric boss 66, and of the chambers 75 and 76 are so arranged that the various fuel pressures acting on them produce a downwardly acting force which urges the piston 67 into contact with the graphite facing 101, to prevent leakage between chambers 75 and 76, and urges the rotor into contact with the graphite facing 102 to prevent leakage between ports 59 and 60. Leakage past the piston 67 is prevented by a seal 103, and leakage from space 61 to eccentric recess

65 is prevented by a seal **104**. It will be apparent to a person skilled in the art that the pump is self-compensating for wear and obviates the need for a high degree of accuracy in many of its components.

As will be seen from FIGURES 3 and 4, the power input shaft **73** is cooled and lubricated by fuel which passes from fuel inlet port **59** through groove **105** in graphite facing **102** to the shaft **73**, and is returned to the spill side of relief valve **63** through passage **106** and arcuate port **107** in the lower casing portion **57**.

It will now be described with reference to FIGURE 9 how the output of the pump, at any given speed, is varied to suit the operating conditions of the engine it supplies. Two pistons **84** have been shown but, for the purpose of distinguishing between them, one has been denoted **84***a*. Due to the diagrammatic arrangement of FIGURE 9, piston **67** has been omitted and chambers **75** and **76** are separated by a diametral web **108**. Also, the pistons **84**, **84***a* have been simplified and are retained by circlips **109** as shown, and space **61** is coincident with chamber **75**.

The piston **84***a* is shown in a position such that the pressure of the fuel in chamber **76** balances all other forces acting on it and, accordingly, prevents it from making a complete induction stroke by not allowing the head of piston **84***a* to abut circlip **109**. If the pressure of the fuel in chamber **76** is increased, the stroke of the piston will decrease and vice versa. Thus, at any operating speed of the pump its output can be varied from zero, to a maximum for the speed at which it is operating, by varying the pressure of the fuel in chamber **76** which is achieved by controlling pintle valve **82** to adjust spill outlet **83**.

The diaphragms **43** and **47** are so arranged that the pressures of the atmosphere, fuel supply and induction depression are balanced against the pressure of the fuel in chamber **81**, by varying the rate at which the fuel escapes through spill outlet **83**.

When the induction pressure is decreasing and the power output of the internal-combustion engine being supplied by the pump is being reduced, the atmospheric and fuel supply pressures remaining constant, the pintle valve **82** moves, as a result of the unbalanced pressures, so as to reduce the escape of fuel through spill outlet **83** whereby to increase the pressure of the fuel in chamber **76** until balance is regained by a higher fuel pressure which in turn reduces the stroke of the pistons **84** and thus reduces the fuel supply to the engine. When the induction pressure is increasing, the power output of the engine will be increasing and the unbalanced pressures will cause the fuel pressure in chamber **76** to be reduced whereby to increase the output of the pump.

What we claim as our invention and desire to secure by Letters Patent of the United States is:

1. A pump for use as a metering device and comprising a cylinder and a coacting piston, conduit means adapted to receive a supply of fluid under pressure, a first pressure-reducing means, a second pressure-reducing means, an inlet valve means and an outlet valve means for said cylinder, said inlet valve means connected to be supplied with fluid through said first pressure-reducing means from said conduit means, said piston when moved in one direction in said cylinder adapted to induce fluid into said cylinder through said inlet valve means to provide an induction stroke, and said piston when moved in the opposite direction in said cylinder adapted to educe fluid from said cylinder through said outlet valve means to provide a working stroke, biasing means acting on said piston to exert thereon a force for producing said induction stroke, said biasing means such that the force it produces decreases progressively during said induction stroke, means for alternately applying the pressure of a control fluid and the pressure of an operating fluid to said piston in opposition to said biasing means, said operating fluid derived from the supply of fluid under pressure in said conduit means, said control fluid derived from said operating fluid through said second pressure-reducing means, the length of said induction stroke to be determined by equilibrium of the force exerted by said control fluid and the force produced by said biasing means, the pressure of said operating fluid such that the resultant of the forces acting on said piston produces said working stroke, and control means for varying the pressure of the control fluid whereby to vary the length of said induction stroke.

2. A pump comprising a rotor defining a cylindrical bore with its axis parallel with but radially-spaced from the axis of the rotor, a piston coacting with said cylindrical bore, a casing surrounding said rotor, an inlet valve means and an outlet valve means for said cylindrical bore, said piston when moved in one direction in said cylindrical bore adapted to induce fluid into said cylindrical bore through said inlet valve means to provide an induction stroke, and said piston when moved in the opposite direction in said cylindrical bore adapted to educe fluid from said cylindrical bore through said outlet valve means to provide a working stroke, a compression coil spring acting between said rotor and said piston to exert thereon a force for producing said induction stroke, said rotor remote from said inlet valve means and said outlet valve means having an end face, said casing having an end face for coaction with said end face of said rotor, said rotor defining a fluid commuication between said piston and said end face of said rotor, said end face of said casing defining a chamber for a control fluid under pressure and a separate chamber for an operating fluid under pressure whereby during rotation of said rotor said piston is alternately subjected through said fluid communication to the pressure of said control fluid and to the pressure of said operating fluid, the length of said induction stroke to be determined by equilibrium of the force exerted by said control fluid and the force produced by said compression coil spring, the pressure of said operating fluid such that the resultant of the forces acting on said piston produces said working stroke, and control means for varying the pressure of the control fluid whereby to vary the length of said induction stroke.

3. A pump comprising a rotor defining a cylindrical bore with its axis parallel with but radially-spaced from the axis of the rotor, a piston coacting with said cylindrical bore, a casing surrounding said rotor, an inlet valve means and an outlet valve means for said cylindrical bore, said piston when moved in one direction in said cylindrical bore adapted to induce fluid into said cylindrical bore through said inlet valve means to provide an induction stroke, and said piston when moved in the opposite direction in said cylindrical bore adapted to educe fluid from said cylindrical bore through said outlet valve means to provide a working stroke, a compression coil spring acting between said rotor and said piston to exert thereon a force for producing said induction stroke, said rotor remote from said inlet valve means and said outlet valve means having an end face, a rotatively-stationary piston guided by said casing and having an end face for coaction with said end face of said rotor, said rotor defining a fluid communication between said piston and said end face of said rotor, said end face of said rotatively-stationary piston defining a chamber for a control fluid under pressure and a separate chamber for an operating fluid under pressure whereby during rotation of said rotor said piston is alternately subjected through said fluid communication to the pressure of said control fluid and to the pressure of said operating fluid, the length of said induction stroke to be determined by equilibrium of the force exerted by said control fluid and the force produced by said compression coil spring, the pressure of said operating fluid such that the resultant of the forces acting on said piston produces said working stroke, a bias reacting between said rotatively-stationary piston and said

casing whereby to urge said end face of said rotatively-stationary piston into sealing engagement with said end face of said rotor to prevent leakage from said chamber of said operating fluid to said chamber for said control fluid, and control means for varying the pressure of the control fluid whereby to vary the length of said induction stroke.



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LAURENCE V. EFNER, *Primary Examiner.*