June 24, 1958 J. N. MORRIS 2,840,066

FUEL INJECTION PUMPS

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FIG. 1.

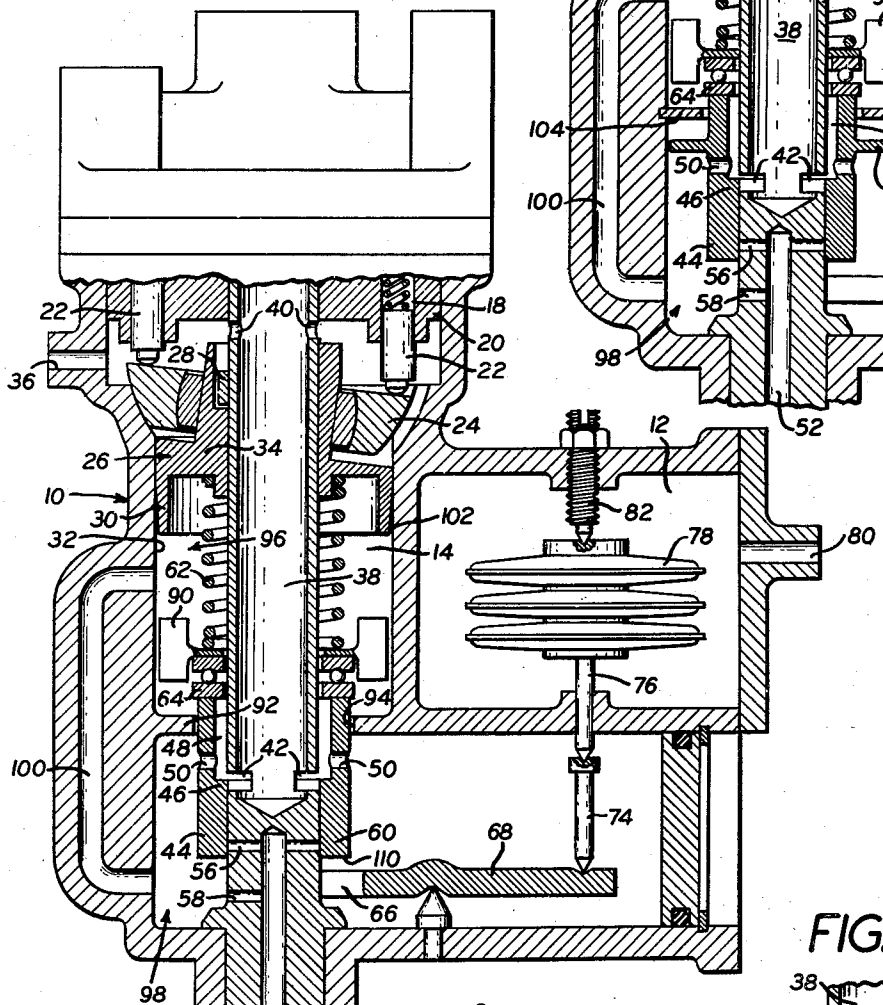



FIG. 4.




FIG. 2.

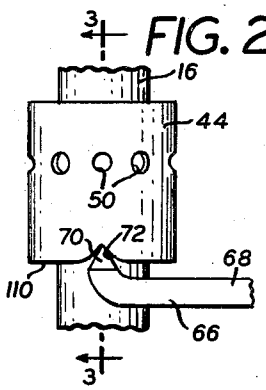
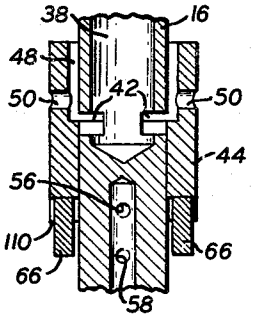



FIG. 3.




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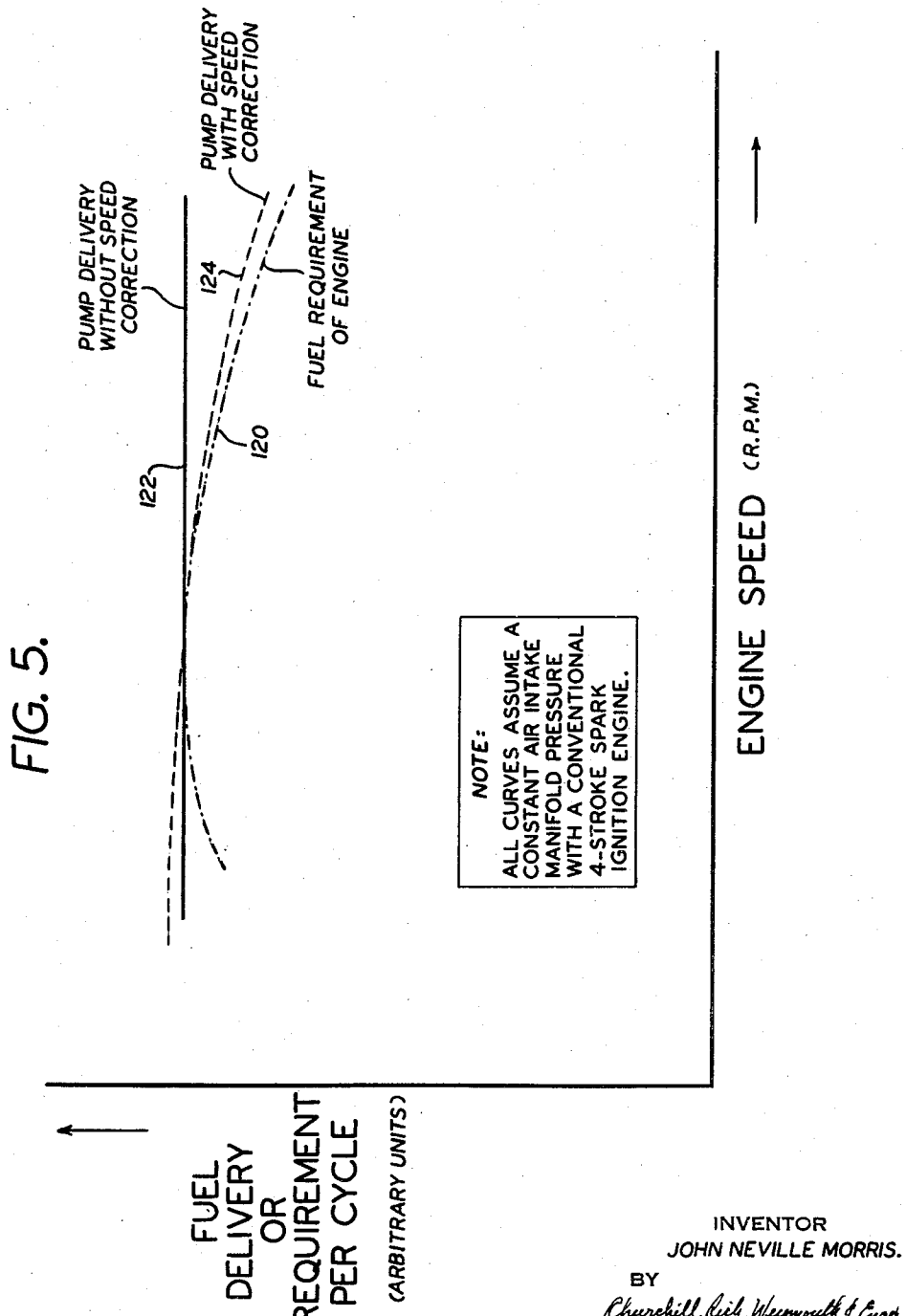



FIG. 5.

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# FUEL INJECTION PUMPS

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Application February 24, 1956, Serial No. 567,608

Claims priority, application Great Britain February 25, 1955

11 Claims. (Cl. 123—140)

This invention relates to a fuel injection pump for internal combustion engines. More particularly, it is directed to a pump of the type wherein means are provided for controlling the delivery of one or more pump members as a function of the air intake manifold pressure of the engine.

A pump of the above type will meter fuel correctly, i. e., in accordance with the air consumed per cycle by the engine, only so long as the manifold pressure affords a true criterion of the amount of air induced by the engine cylinders per cycle. That, however, does not hold true at high engine speeds because of the obstructive effect of the inlet ports and valves, which results in a deterioration in the filling of the cylinders with increasing speed for a given intake manifold pressure. As a consequence, an undesirable richening of the fuel/air ratio occurs with increasing engine speed. This is particularly objectionable when the injection pump is intended for use with spark-ignition engines of the type employed in automotive vehicles and in certain stationary installations where optimum fuel economy or operation is desired.

For convenience in explanation, the invention will be described with reference to one specific type of injection pump; however, as will appear from the description, the invention is applicable to all pumps of the general type referred to above.

The specific form of pump with reference to which the invention is described in the following description is disclosed and claimed in my copending application Ser. No. 567,395 filed February 23, 1956. In the aforesaid application there is provided an adjustable device which is made up of an adjustable element for controlling the delivery of the pump members and a displaceable pressure responsive member for controlling the adjustable element. The displaceable member is disposed in a housing so as to form with said housing a first and a second pressure chamber on respective sides of the displaceable member. Means are provided for supplying a substantially unrestricted flow of fluid under pressure to the first pressure chamber. A servo-valve-controlled passage interconnecting the first and second pressure chambers is provided along with a servo-valve therefor. A restricted passage is provided for venting the second pressure chamber. The opening in the valve-open position of the servo-valve is chosen relative to the size of the restricted passage so as to permit a rapid build-up of fluid pressure in the second pressure chamber when the servo-valve is in its wide open position. This enables the pump to change its delivery rapidly in a first direction. In order to enable the pump to change rapidly in the reverse direction, an auxiliary venting means for the second pressure chamber is provided. This includes a vent passage of a size related to the restricted passage and to the size of the second pressure chamber such as to decrease rapidly the pressure in the second pressure chamber when unrestricted. The servo-valve is operatively associated with the last mentioned vent passage for restricting the flow



of fluid therethrough except when in the servo-valve-closed position.

In accordance with the present invention a fuel injection pump having its delivery controlled primarily as a function of the air induction manifold pressure of the engine is compensated for the deterioration in the mass of air induced per cycle with increasing speed by the provision of means to be driven by the engine for modifying the pump delivery in a manner tending to decrease the delivery with increasing speed for any given manifold pressure. More specifically the invention contemplates the provision of a centrifugal impeller to be driven by the engine and having a fluid output coupled to modify the control which pressure responsive means have over the delivery of the pump.

A better understanding of the invention will be had from a reading of the following detailed description with reference to the accompanying drawings in which:

Fig. 1 is an elevational view partly in section showing a fuel injection pump embodying the present invention;

Fig. 2 is a fragmentary elevational view showing the servo-valve and the bifurcated lever for coupling the input signal thereto;

Fig. 3 is a sectional view taken on line 3—3 in Fig. 2;

Fig. 4 is a fragmentary sectional view showing a modified construction of the servo-valve and baffle member forming a part of the present invention; and

Fig. 5 is a group of curves demonstrating the operating characteristics of the invention.

Referring now to the drawings, the fuel injection pump is provided with a cast housing **10** having a capsule pressure chamber **12** and an operating chamber **14**. A drive shaft **16** passes through the chamber **14** and has its upper end, which is not shown in the drawing, operatively connected to a distributing valve arrangement for sequentially connecting the pump cylinders, one of which is shown at **18**, to the fuel supply and to the injection nozzles for the engine. The other end of the drive shaft is arranged to be driven by the engine in a suitable manner. The cylinder **18** along with a suitable number of further cylinders, not shown, are positioned in the form of a circle in a cylinder block **20**. Each of the cylinders is provided with a pump member in the form of a plunger **22** of which two are visible in the drawing. The plungers **22** bear against a wobble-plate **24** which is mounted upon a combined skew shaft and servo-piston unit **26**. The unit **26** is slidably mounted upon the shaft **16** and keyed thereto by a key **28** for driven rotation therewith. The unit **26** is provided with a cylindrical apron **30** making a sliding fit in the cylindrical portion **32** of the housing **10**. The apron **30** along with the web portion **34** of the unit **26** forms a displaceable pressure responsive member within the aforementioned cylindrical portion **32**. The wobble-plate **24** has its peripheral region provided with a bearing surface which engages a complementary surface in the housing **10**. This surface may be generally spherical in nature. As the combined skew shaft and servo-piston unit **26** moves longitudinally along the drive shaft **16** the skew portion of the unit **26** will displace the center of the wobble-plate **24** therby increasing or decreasing its tilt or inclination. In a known manner rotation of the skew shaft will cause the wobble-plate to reciprocate the pump members **22**. As shown in the drawing, the stroke of the pump members will be a maximum when the unit **26** occupies its extreme upward position and will be a minimum when the unit **26** occupies its extreme lowermost position.

Provision is made for connecting a source of fluid under pressure to the housing **10** through the large port **36** for establishing a substantially unrestricted flow thereof. The pressure fluid might conveniently be the lubricating

oil employed in the engine. The fluid will fill up the portion of the chamber **14** above the unit **26** (a first pressure chamber) and pass into a servo-valve-controlled passage **38** formed in the drive shaft **16** by way of the ports **40**. It is noted that the ports **40** are located at a point on the drive shaft **16** above the upper limit of movement of the servo-piston unit **26**. A further group of comparatively large ports **42** are located in the walls of the drive shaft **16** in communication with the passage **38** in the lower portion of the chamber **14** at a point well below the lower limit of travel of the servo-piston unit **26**. A servo-valve in the form of a cylindrical sleeve member **44** is mounted for sliding movement longitudinally upon the drive shaft **16**. As shown in the drawing, the valve **44** is provided with a land area **46** which cooperates with the ports **42** to provide a variable obstruction therefor dependent upon the relative position of the servo-valve. As long as the ports **42** are not completely obstructed or closed the pressure fluid will leave the passage **38** through the ports **42** entering the enlarged internal portion **48** of the valve **44** and exiting through the ports **50** formed in the walls thereof. This fluid will now commence to fill up the portion of the chamber **14** located below the servo-piston **26** (a second pressure chamber).

A second passage **52** is formed within the drive shaft **16** and communicates through a lateral passage **54** with the exterior of the housing **10**. The other end of the passage **52** communicates with a plurality of port means **56** and **58** which are located in the shaft **16** within the lower portion of chamber **14**. The port means **56** is located with respect to the servo-valve such that the lower portion **60** thereof completely obstructs the port means **56** for all positions of the servo-valve except that position wherein the land area **46** completely obstructs or closes off the ports **42**, i. e. for all positions other than the servo-valve-closed position. The reason for this will be described below. The other port means **58** is located at a point which is always unobstructed by the servo-valve and forms with the passages **52** and **54** a permanent restricted passage for venting the aforementioned second pressure chamber in the lower portion of chamber **14**. The ports **42** as well as the clearance **48** and ports **50** in the servo-valve should be of such size and proportion that the opening in the valve-open position of the servo-valve relative to the size of the restricted passage will permit a rapid build-up of pressure in the second pressure chamber below the servo-piston **26**.

A helical compression spring **62** surrounds the drive shaft **16** and is disposed between the web **34** of the servo-piston **26** and the upper face of the servo-valve **44**. A thrust race **64** is provided between the spring **62** and the upper face of the valve **44**. The spring **62** urges the valve **44** downwardly against the bifurcated ends **66** of an articulable link or see-saw lever **68** which straddles the drive shaft **16**. A seen in Figs. 2 and 3 the ends **66** of the lever **68** are provided with tapered bearing surfaces **70** engaging V-notches **72** formed in the lower edge of the servo-valve **44**. This prevents rotation of the servo-valve and at the same time applies the controlling force thereto. The other end of the lever **68** is acted upon by a thrust pin **74** which in turn is engaged by a second thrust pin **76** fixed to one face of a pressure responsive device **78** situated in the chamber **12**. The device **78** may take the form of a sealed multi-capsule stack. The chamber **12** is provided with a port **80** for connection with the air induction manifold of the engine. The opposite face of the device **78** is positioned centrally by an adjustable abutment pin **82**.

The device as described so far corresponds with the disclosure in my above-mentioned copending application. The elements now to be described are those forming a basis of the present invention. The pressurizing means takes the form of a centrifugal impeller **90** positioned between the spring **62** and the thrust race **64** to be rotated by the drive shaft **16**. A baffle member **92** is formed, as best seen in Fig. 1, integral with the housing **10** and positioned around the intermediate portion of the servo-valve **44**. The baffle member **92** has an opening **94** therein which allows the servo-valve to move unobstructed therethrough with a slight clearance. The clearance also permits a somewhat restricted flow of the controlling fluid between the first and second subsidiary chambers **96** and **98** formed on respective sides of the baffle member. A passage **100** formed in the walls of the housing **10** interconnects the two subsidiary chambers **96** and **98**. For a purpose to be described, the apron **30** of the unit **26** has the lower portion of its outer surfcae provided with a taper **102**.

In a modified form of the invention shown in Fig. 4 the integral baffle member of Fig. 1 has now been replaced by an annular ring **104** which is positioned within the bore **32** of the housing **10** so as to leave the portion of the bore **106** below the ring in the lower or second subsidiary chamber **98**. It should be noted that the walls of the bore are joined to the baffle **104** perpendicular thereto. In this embodiment the servo-valve is provided with an annular radial extension or flange **108** fitting within the bore portion **106** with a slight clearance therebetween. The clearance between the flange **108** and the portion of the bore **106** should be less than the clearance existing between the baffle ring **104** and the servo-valve. The reason for this will be apparent when the operation of this embodiment is described below.

Returning to Figs. 1, 2 and 3 the operation of the pump will be discussed first assuming that the impeller **90** along with baffle **92** and passage **100** is omitted.

So long as the operating conditions of the engine remain unchanged (i. e. while the induction manifold pressure applied to the capsule unit in the chamber **12** remains constant), the mechanism described above is in a stable state. This arises from the circumstance that, under the conjoint influences of the control spring **62** and the load transmitted by the linkwork **68**, **74** and **76** from the capsule unit **78**, the servo-valve **44** assumes a position in which its land area **46** so regulates the efflux of fluid through the outlet ports **42** that, in conjunction with the restricted provision for escape of fluid from the second pressure chamber, the differential fluid pressure acting on the servo-piston unit **26** exactly balances the load exerted by the control spring **62**. Consequently, any given load exerted upon the servo-valve **44** by the capsule unit **78** and its associated linkwork is necessarily exactly counterbalanced by the load exerted by the control spring **62**, and, therefore, corresponds to a particular length of that spring and hence a definite position of the combined skew shaft and servo-piston unit **26**. Thus, any given value of the induction manifold pressure results in a corresponding stroke being performed by the fuel-pumping plungers **22**.

Let it be assumed that the pump is in a stable fuel-metering condition with a relatively low pressure obtaining in the chamber **12**; which, of course, corresponds to the engine either idling or being on light load (that is to say, when the throttle admitting air into the induction manifold is partially shut). If now the throttle is abruptly opened, the induction manifold pressure rises very rapidly, for example to atmospheric pressure or to some pressure intermediate between the low pressure previously existing and atmospheric pressure. The capsule unit **78** has a relatively low elasticity and, therefore, this sudden increase in its ambient pressure not only has the effect of diminishing momentarily the thrust exerted by the capsule unit on its associated linkwork, but also enables the capsule unit to undergo a substantial contraction. This permits the servo-valve **44** momentarily to slide under the influence of the control spring **62**, it being borne in mind that at the instant of opening the throttle and consequently

applying the sudden increase in pressure to the capsule unit **78** the pump was operating at a relatively small stroke; that is to say, the control spring **62** was substantially compressed and exerting a relatively large load upon the servo-valve.

The momentary downward displacement of the servo-valve **44** brings about a momentary large increase in the effective opening of the associated outlet ports **42**, and thus permits high-pressure fluid from the passage **38** in the drive shaft **16** to flow through these ports at a high rate into the relatively lower-pressure region constituted by the second pressure chamber. This momentary high flow of fluid temporarily raises the fluid pressure behind the servo-piston unit **26** to such a value that, acting on the rear face of the servo-piston and thereby assisting the control spring, the latter is enabled to expand and rapidly move the servo-piston unit. A condition of stability is eventually established, with the control spring **62** now more distended and the servo-piston unit **26** having moved to a position corresponding to a greater stroke of the fuel-pumping plungers **22**. The load exerted by the control spring on the servo-valve eventually exactly counterbalances the new load exerted by the capsule unit **78**.

It will be appreciated from the foregoing description that the provision of outlet ports **42** in the drive shaft which are of substantial depth and peripheral extent, taken in conjunction with the relatively low spring rate of the capsule unit **78**, permits the stroke-varying mechanism to respond very rapidly to an abrupt increase in the induction manifold pressure.

Now consider the operation of the device when the throttle is suddenly closed or partially closed from a formerly open position. For this purpose assume, initially, that the port means **56** has been omitted. The concomitant sudden reduction in the induction manifold pressure causes the capsule unit **78** momentarily to distend, it being free to do so since at the instant of the change of pressure in the induction manifold the control spring was relatively distended, corresponding to a substantial displacement of the servo-piston unit towards the full-stroke position. The control spring **62** was, therefore, at the instant under consideration, exerting a relatively small load upon the servo-valve. Now the sudden distension of the capsule unit momentarily causes the servo-valve to be moved by the lever **68** against the control spring **62**, and this movement results in the valve completely closing the outlet ports **42** in the drive shaft, thus precluding temporarily any further flow of fluid from one side of the servo-piston unit to the other. Under the action of the high-pressure fluid on its front face, the servo-piston unit now proceeds to move towards the position at which the amplitude of the wobble-plate **24** is diminished, forcing the fluid out of the second pressure chamber by way of the restricted vent passage referred to earlier. This the servo-piston unit is free to do since the fluid behind it is momentarily no longer being replenished as it is normally by the passage of fluid past the land area **46** on the servo-valve **44**. The rate at which this movement of the servo-piston unit takes place is, however, limited by the necessarily relatively small size of the vent passage by which fluid can escape from the second pressure chamber. The vent passage has to be relatively small in order to limit the amount of fluid supplied by way of the relatively large port **36** to the front face of the servo-piston unit. In consequence the response of the mechanism to a sudden closure of the throttle is very sluggish and a considerable interval of time may elapse before the mechanism has readjusted itself to the new pressure obtaining in the induction manifold. During this interval the fuel delivered to the engine is excessive, and misfiring and temporarily excessive fuel consumption result.

Now consider the effect of the port means **56** which, under normal steady metering conditions (i. e. when the land area **46** of the servo-valve **44** is very close to its shut off position with respect to the associated ports **42**), are just covered by the valve **44** and are thus inoperative as a leakage or vent path. During the momentary phase when the capsule unit **78** is substantially expanded temporarily, and consequently the valve **44** has been moved against the spring **62** and has shut off the associated ports **42**, the additional port means **56** becomes exposed and thus affords additional paths, in parallel with the permanently opened passage of relatively small size previously mentioned, for the fluid to escape from the lower portion of chamber **14** to the exterior of the housing **10**.

As will be appreciated from the above description, the effect of the provision of the auxiliary venting passage is momentarily to increase the rate of fluid venting or spillage from the low pressure side of the servo-piston unit **26**, thus increasing the speed at which the servo-piston unit **26** can move in the downward direction to diminish the amplitude of the wobble-plate **24** and the stroke of the pump plungers **22**. This facility is provided for a brief interval of time following upon a sudden decrease in the air induction manifold pressure.

Now consider the effect of the elements **90**, **92** and **100** of Figs. 1, 2 and 3. In operation the fluid in the first subsidiary chamber **96** is set in motion by the centrifugal impeller **90** and a centrifugal pressure head is created which results in circulation of the fluid through the passage **100** into the second subsidiary chamber **98**. The centrifugal pressure head is proportional to the square of the peripheral velocity of the fluid and thus is higher at the boundary of the chamber **96** than it is at a smaller radius such as at the opening **94** in the baffle member **92**.

Since only a small clearance is provided between the baffle member **92** and the servo-valve **44** which it surrounds, a fluid pressure differential is established between the two subsidiary chambers and is effective upon the end face **110** of the servo-valve **44** urging the latter in a direction towards the servo-piston unit **26**. In effect, this is equivalent to an increase in the metering force exerted by the capsule stack upon the servo-valve **44** through the associated linkwork. That is to say, the setting of the mechanism tends to alter in such a way as to reduce the fuel delivery of the pump. Thus a weakening effect is achieved upon the delivery of the pump per cycle with increasing speed, this effect following approximately a square law. Such a weakening effect conforms approximately to the rate of reduction of air consumption of an orthodox four-cycle engine with increasing speed.

Referring to Fig. 5, the fuel requirement of an ordinary four-cycle spark-ignition engine at different speeds has been plotted in curve 120 for an arbitrarily assumed value of air induction manifold pressure. Since it has also been assumed in determining the fuel requirement that the fuel/air ratio is constant, curve 120 also represents the mass of air induced in the cylinders per cycle for varying speeds. Therefore, from Fig. 5 it can be seen that the air mass induced per cycle, for a given manifold pressure, attains a maximum value at some intermediate point in the speed range of the engine, and falls off both above and below this speed.

Curve 122 in Fig. 5 shows the output of an injection pump as being constant as long as the manifold pressure is constant and in the absence of the fluid pressurizing means of the instant invention.

Curve 124 shows how the fluid pressurizing means or centrifugal impeller causes the pump delivery to fall off as the engine speed increases. Note particularly how closely the curve 124 follows or approximates the engine requirement curve 120 for the upper half of the speed range. As shown in Fig. 5 a slight error has been deliberately introduced in the speed correction curve at low engine speeds. This error was introduced in order to have perfect correction in the optimum air filling region represented by the maximum point in curve 120. This error in certain instances may in fact be desirable because it provides an enriched mixture at low speeds or

near idling. The error was introduced by increasing the pump delivery for the particular manifold pressure prior to including the correction of the centrifugal impeller.

A further feature of the invention is that the weakening effect as described above is correlated with the output of the pump per revolution; that is to say, with the stroke of the pump plungers and, therefore, with the axial displacement of the servo-piston unit **26**. In this connection it will be realized that the arrangement so far described depends for its effectiveness upon free circulation of fluid through the passage **100** interconnecting the two subsidiary chambers. Consequently, if this passage is partially obstructed the fluid pressure differentially which acts upon the control means or servo-valve **44** will be reduced. Such an effect is produced by the taper **102** on the apron **30** of the servo-piston unit **26** which is arranged progressively to overrun the mouth of the passage **100**. The selection of the taper **102**, or alternatively the profiling of the mouth of the passage **100**, can be so selected as to afford any required correlation between the freedom of fluid circulation from the one subsidiary chamber to the other. Thus the weakening effect can be correlated with the stroke of the plungers.

By means of the feature just described, the force operative upon the end face **110** of the servo-valve **44** can, for a given pump speed, be controlled in such a way that the weakening effect bears any relationship (e. g. a linear relationship) to the pump output per revolution and therefore, as would normally be required, to produce a weakening effect which would represent a constant percentage of the pump output per stroke for any given speed.

In the embodiment shown in Fig. 4 the differential force exerted upon the servo-valve **44** is augmented by the flange **108**. As a result of the difference in clearance between the valve **44** and the baffle **104**, and the flange **108** and the wall of the bore portion **106**, substantially the whole of the fluid pressure differential occurs across the flange **108**. Since the flange increases the surface area upon which the fluid pressure differential is effective, the force developed is greater than that developed in the structure of Fig. 1.

The invention has been described with reference to a specific embodiment thereof. Nevertheless, it is intended to encompass the numerous changes and modifications which will appear obvious to those skilled in the art to which is appertains and which come within the scope of the appended claims.

What I claim is:

1. A fuel injection pump for internal combustion engines having an adjustable device controlling the delivery of one or more pump members, and a displaceable pressure responsive member for controlling the adjustable device disposed in a housing and forming with said housing a first and a second pressure chamber on respective sides of said member, comprising in combination, means for supplying a substantially unrestricted flow of fluid under pressure to said first chamber, a servo-valve-controlled passage interconnecting said two chambers, a servo-valve therefor, a restricted passage for venting said second chamber, fluid pressurizing means responsive to the speed of said pump for providing fluid under pressure with the pressure varying as a function of said speed, and means coupling said pressurizing means to said servo-valve for applying a force thereto proportional to said pressure in a manner tending to change the position of said servo-valve, whereby the delivery of said pump members is controlled as a function of the speed of said pump.

2. A fuel injection pump for internal combustion engines having an adjustable element controlling the delivery of one or more pump members, a fluid operated servo-piston for controlling the adjustable element, and a servo-valve for controlling the fluid pressure on opposite faces of said servo-piston, comprising in combination a centrifugal impeller, drive means for coupling said impeller in driven relationship to the internal combustion engine, means for supplying fluid to said impeller, first fluid coupling means coupled to said impeller at a high pressure point therein, second fluid coupling means coupled to said impeller at a low pressure point therein, and means coupling said first and second fluid coupling means for exerting a differential force upon said servo-valve in a manner tending to change its position, whereby the delivery of said pump members is controlled as a function of the speed of said engine.

3. A fuel injection pump for internal combustion engines having an adjustable wobble-plate for actuating a plurality of pumping plungers of variable stroke, a combined skew shaft and servo-piston unit keyed to and longitudinally slidable on a drive shaft for altering the amplitude of the motion of the wobble-plate, and a housing for said combined skew shaft and servo-piston unit forming with said unit a first and a second pressure chamber on respective sides of said unit, comprising in combination, means for supplying a substantially unrestricted flow of fluid under pressure to said first chamber, a servo-valve-controlled passage interconnecting said two chambers, a servo-valve therefor, a restricted passage for venting said second chamber, a fluid pressurizing means responsive to the speed of said pump for providing fluid under pressure with the pressure varying as a function of said speed, and means coupling said fluid pressurizing means to said servo-valve for applying a force thereto proportional to said pressure in a manner tending to change the position of said servo-valve, whereby the stroke of said pump plungers is controlled as a function of the speed of said pump.

4. A fuel injection pump for internal combustion engines having an adjustable device controlling the delivery of one or more pump members, control means for positioning said adjustable device, and pressure responsive means for coupling said control means to the air induction manifold of the engine for regulating the control means in response to the air pressure in said manifold with increased delivery of said pump members corresponding to a rise in air pressure, and vice versa, said engine being of the type wherein the mass of air induced per cycle decreases with increasing speed above a given value for any given induction manifold pressure, comprising in combination a centrifugal impeller, drive means for coupling said impeller in driven relationship to the engine, means for supplying fluid to said impeller, and coupling means for coupling the output of said impeller for exerting a differential force upon said control means in a manner tending to change its position towards one calling for less delivery from the pump members with increase in said force, and vice versa, the relationship between said impeller output and the regulating effect of said pressure responsive means being such as to cause the delivery of said pump members to be progressively decreased with increasing speed for any given induction manifold pressure at such a rate as to match said decrease in mass of induced air at speeds above said given value to prevent thereby an undesirable richening of the fuel/air ratio.

5. A fuel injection pump according to claim 4, wherein said adjustable device is provided with valve means directly coupled to said coupling means for modulating the fluid output of said impeller as a function of the delivery of said pump members, whereby the progressive decrease in delivery from said pump members is caused to vary as a function of the presssure in said air induction manifold in addition to varying as a function of the speed of the engine.

6. A fuel injection pump for internal combustion engines having a plurality of individual pump members for supplying fuel to the cylinders of said engine, an adjustable element for controlling the delivery of all of said pump members, and a fluid operated servo-system including a fluid chamber for regulating said adjustable ele-

ment in direct response to the air induction manifold pressure of said engine, with increased delivery of said pump members corresponding to a rise in air pressure, and vice versa, said engine being of the type wherein the mass of air induced per cycle decreases with increasing speed above a given value for any given induction manifold pressure, comprising in combination an impeller disposed within said fluid chamber, drive means for coupling said impeller in driven relationship to the engine, and means for causing said impeller when driven by said engine to vary the output of said servo-system in a direction tending to decrease progressively said delivery of the pump members with increasing engine speed, and vice versa, whereby the delivery of fuel is caused to decrease with increasing speed so as to match said decrease in the induced air above said given value.

7. A fuel injection pump according to claim 6 wherein said impeller is constructed and arranged for developing a fluid pressure which varies as a direct function of the square of its speed of operation whereby the decrease in said delivery of fuel substantially follows a square law.

8. A fuel injection pump for internal combustion engines having an adjustable element for actuating a plurality of pumping plungers of variable stroke, a servo-piston unit longitudinally slidable on a drive shaft for causing the adjustable element to alter the stroke of said plungers, a housing for said servo-piston unit forming with said unit a first and a second pressure chamber on respective sides of said unit, means for supplying a substantially unrestricted flow of fluid under pressure to said first chamber, a passage formed within said drive shaft and extending beyond the limits of movement of said servo-piston unit on both sides thereof, one or more ports formed in the wall of said drive shaft on each side of said servo-piston unit for establishing communication between said passage and said first and second chambers, respectively, a servo-valve slidably mounted on said drive shaft within said second chamber for operative valving cooperation with the drive shaft ports therein, and means for coupling said drive shaft in driven relation to said engine; comprising in combination a baffle member surrounding an intermediate portion of said servo-valve with a slight clearance therebetween and dividing said second chamber into first and second subsidiary chambers with movement of said servo-valve into said first subsidiary chamber causing a decrease in the stroke of said plungers, a centrifugal impeller mounted on said drive shaft for driven rotation therewith within said first subsidiary chamber, and a passage in said housing establishing communication between both of said subsidiary chambers such that fluid tends to flow from the first to the second subsidiary chamber in response to a pressure differential being developed across said impeller, thereby to urge said servo-valve into said first subsidiary chamber with a force varying as a function of the speed of said shaft, whereby the stroke of said plungers is decreased as a function of increasing engine speed.

9. A fuel injection pump according to claim 8, wherein said servo-piston unit is provided with a portion making sliding contact with the walls of said housing within said second pressure chamber, said portion being constructed to variably obstruct the mouth of said passage which establishes communication between said subsidiary chambers as a function of the position of said servo-piston unit, to modify thereby the amount by which the stroke of said plungers is decreased.

10. A fuel injection pump according to claim 8, wherein said housing has a straight bore portion surrounding said servo-valve within said second subsidiary chamber with the walls thereof joined to said baffle member perpendicular thereto, and said servo-valve is provided with an annular radial flange fitting with a slight clearance within said bore portion, said last mentioned clearance being less than the clearance existing between the servo-valve and the baffle member so as to increase the force acting upon said servo-valve in response to a given rise in pressure across said impeller, thereby to increase the sensitivity of the pump.

11. A fuel injection pump for internal combustion engines having its delivery controlled primarily as a function of the air induction manifold pressure of the engine and compensated for the deterioration in the mass of air induced per cycle by the engine with increasing speed above a given value for any given induction manifold pressure, comprising in combination a fuel pumping device, a pressure responsive means for controlling the delivery per cycle of said pumping device as a function of said manifold pressure with increased delivery corresponding to rise in manifold pressure, and vice versa, a centrifugal impeller, means for coupling said impeller in driven relationship to said engine, means for supplying said impeller with a fluid, and means coupling an output of said impeller to said pressure responsive means for modifying its control of the delivery of said pumping device in a manner tending to decrease the delivery with increasing speed, the relationship between impeller output and the control exercised by said pressure responsive means being such as to cause the delivery of fuel to deteriorate with increasing speed at such a rate as to match said deterioration in the induced air at speeds above said given value to prevent thereby an undesirable richening of the fuel/air ratio.

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