Nov. 11, 1969 V. D. ROOSA 3,477,418

FUEL SUPPLY SYSTEM

Filed Nov. 20, 1967 2 Sheets-Sheet 1

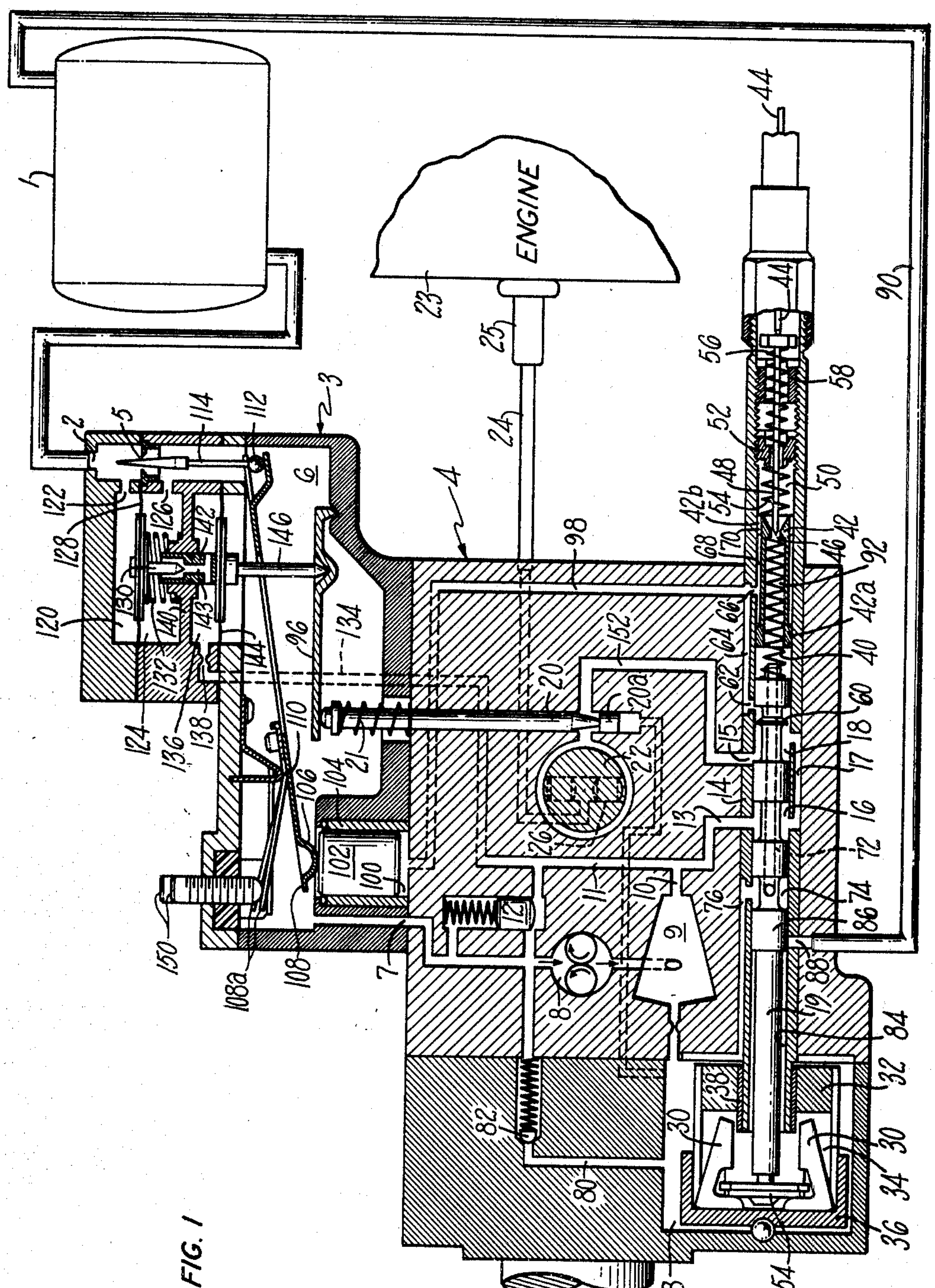

FIG. 1

INVENTOR.
VERNON D. ROOSA

BY
Lindsey, Prutzman and Hayes
ATTORNEYS

Nov. 11, 1969 V. D. ROOSA 3,477,418

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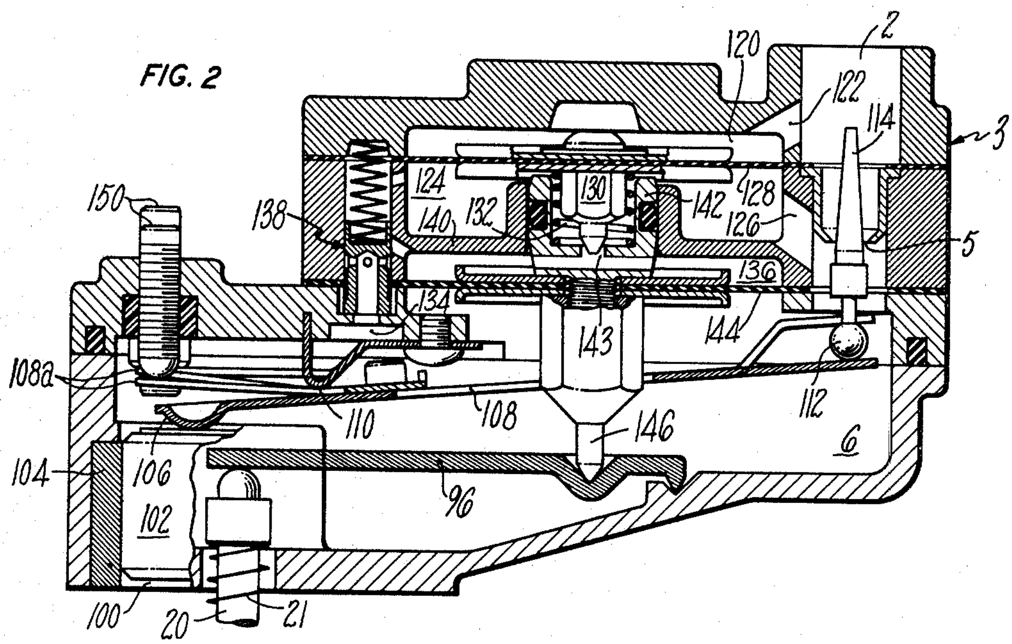
Filed Nov. 20, 1967 2 Sheets-Sheet 2

FIG. 2



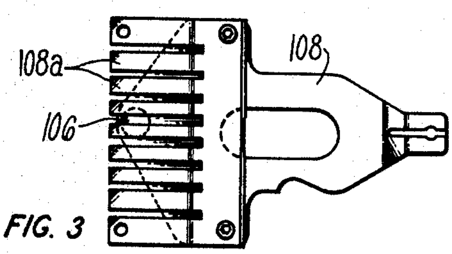


FIG. 3

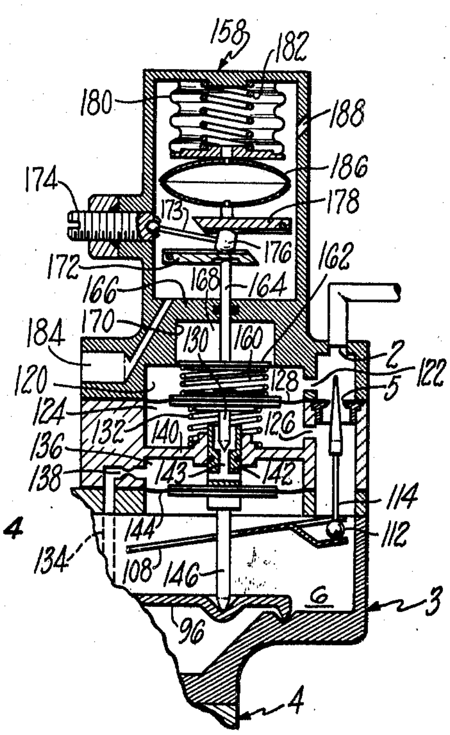


FIG. 4

Patented Nov. 11, 1969

1

3,477,418
FUEL SUPPLY SYSTEM

Vernon D. Roosa, West Hartford, Conn. (% Hartford Machine Screw Company, P.O. Box 1440, Hartford, Conn. 06102)
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U.S. Cl. 123—140 26 Claims

ABSTRACT OF THE DISCLOSURE

A fuel supply system for an internal combustion engine which includes a torque limiting valve in series with and actuated independently of a governor controlled metering valve. The torque limiting valve is inactive except when, at a given engine speed, the flow of fuel to the engine reaches a preselected level which varies with the throttle setting and initiates the operation of a servomotor to close the torque limiting valve. The fuel to the engine flows through a variable control orifice which includes a movable needle valve to vary the effective size of the control orifice with engine speed. A speed related control pressure, which operates through a piston and lever to move the needle valve, is developed in a chamber of the governor assembly wherein an axially shiftable ported governor spring seat cooperates with a hand throttle control rod to provide a variable spill port for the control pressure so that it varies with axial force of a centrifugal governor on the metering valve. An adjustable variable rate spring opposes the control pressure in adjusting the control needle. Adjustable manifold temperature and pressure compensators may be incorporated.

This invention relates to a fuel supply apparatus for an internal combustion engine and more particularly to such a system which will regulate the desired variable maximum fuel delivery to the engine at all speeds throughout the speed range of the engine and is readily adjustable to pattern such fuel delivery closely to the requirements for a wide range of engines.

It is an object of this invention to provide a fuel supply apparatus including means for readily and easily shaping the maximum fuel delivery at different engine speeds independent of the other features of the apparatus.

Another object of this invention is to provide a fuel system wherein means are provided for producing an accurate speed signal for actuating the regulating means for shaping the maximum fuel delivery or torque curve of the engine. Included in this object is the provision of an automatically variable flow orifice to provide a given pressure drop across the orifice at varying fuel flow rates to initiate the operation of an arrangement for regulating the maximum rate of flow of the engine throughout its speed range.

Another object of this invention is to provide a concentrically disposed in-line governor controlled valving arrangement for controlling the speed of the engine and for providing a speed signal which is accurate throughout the speed range.

Another object of this invention is to provide a fuel control system which provides excess fuel for cranking.

Another object of this invention is to provide a fuel control system as hereinbefore set forth which includes compensation for variations in fuel leakage within the system and fuel injection pressure.

Still another object of this invention is to provide a fuel control system of the type described incorporating means for compensating for variations in manifold pressure and temperature of the associated engine.

Other objects will be in part obvious and in part pointed out more in detail hereinafter.

2

The invention accordingly consists in the features of construction, combination of elemetns and arrangement of parts which is exemplified in the construction hereafter set forth, and the scope of the invention is indicated in the appended claims.

In the drawings:

FIG. 1 is a schematic representation of a fuel supply apparatus incorporating an illustrative embodiment of the present invention;

FIG. 2 is an enlarged cross-sectional view of the flow sensing and regulating subassembly incorporated in the illustrative embodiment of FIG. 1;

FIG. 3 is a top view of the adjustable biasing spring utilized in the subassembly of FIG. 2; and

FIG. 4 is a fragmentary cross-sectional view of a modified form of the illustrative embodiment of FIG. 1 incorporating a manifold pressure and temperature compensator.

Referring now in detail to the drawings, in which like numerals refer to like parts throughout the several figures, there is shown in FIG. 1 a schematic diagram of a fuel control apparaatus embodying an illustrative form of this invention. As shown, fuel passes from a tank **1** through an inlet port **2** of a flow sensing and control subassembly generally indicated at **3.** As shown, the flow sensing and control subassembly is directly mounted on fuel pump **4** by any suitable means (not shown).

Fuel pump **4** supplies regulated fuel through conduit **24** to delivery means, such as fuel injection nozzles **25,** of engine **23.**

In the illustrative form of the invention, fuel enters the inlet port **2** of flow sensing and control subassembly or control box **3,** passes through a flow sensing or control orifice **5,** which as best shown in FIG. 3 has essentially a knife edge so as to be viscosity insensitive, a chamber **6** of the control box **3,** and then into an inlet passageway **7** of the injection pump **4.** From the passageway **7** the fuel flows into the low pressure transfer pump **8** which discharges fuel into an air separator **9** from which the fuel passes into a branched outlet passage **10,** the branch **11** of which communicates with a pressure regulating valve **12** which recirculates a portion of the fuel to the inlet of the transfer pump **8** and regulates the outlet pressure of the fuel therefrom so as to provide a pressure which increases with engine speed.

Fuel from the transfer pump **8** also flows into a sleeve **14** through branch passage **13** from which it passes through metering port **15** via annulus **16,** passageway **17** and annulus **18.** The sleeve **14** mounts a rotatable and reciprocable metering valve **19** which regulates the opening of the metering port **15** in a manner ot restrict the flow therethrough to maintain a preset engine speed under normal operating conditions. From the metering port **15,** the metered fuel passes a torque limiting valve **20** which is biased in its full open position by a spring **21** and is disposed in series relationship with metering valve **15.** The fuel then passes into the rotor **22** of the pump **4** which includes the high pressure charge pump **26.** The charge pump **26** supplies measured charges of high pressure fuel in timed relation to an associated internal combustion engine through a plurality of delivery passages **24** and delivery means, which take the form of a plurality of injection nozzles **25** (only one of which is shown in FIG. 1 for convenience of illustration). The general details of construction of pump **4** may vary and for further information on the operation of one fuel pump to which this invention may be applied, reference is made to my copending application Ser. No. 505,840 filed Nov. 1, 1965 now Patent 3,411,365.

As shown in FIG. 1 the metering valve **19** is slidably mounted in the sleeve **14** mounted in a bore of pump **4** and is regulated by a centrifugal governor having a plurality of centrifugal flyweights **30** which are mounted in a cage formed by a gear **32** which is segmented to provide longitudinal fingers defining slots **34** to receive the weights **30**. The segmented fingers forming the slots **34** engage a cap member **36** which forms the end thrust member for the governor. The gear **32** is driven by a mating gear (not shown) mounted on the rotor of the pump **4** so that a centrifugal force which is correlated with engine speed is exerted on the governor weights **30**.

The gear **32** rotates on a fixed bearing **38** mounted on the sleeve **14** and the metering valve **19** is coupled to the governor case, as more fully described in my aforesaid copending application, so as to be driven for rotation with the governor. It will thus be seen that the centrifugal weights **30** under the influence of centrifugal force will exert a pressure to urge the metering valve **19** to the right as viewed in FIG. 1 by an amount which increases with engine speed. This centrifugal force tends to reduce the opening of the metering port **15** to reduce the amount of fuel passing through the port with increasing speed and accordingly regulates the speed of the engine at a preselected rate.

In addition to housing the metering valve **19**, the sleeve **14** further houses an independent means for providing, according to one aspect of this invention, a centrifugally regulated pressure having a level correlated with engine speed.

As shown in FIG. 1, the main governor spring **40** engages the right end of the metering valve **19** to provide a biasing force which opposes the force applied to the metering valve **19** by the flyweights **30** and the equilibrium position of the metering valve longitudinally in the sleeve **14** occurs when these opposing forces are equal. It will be observed that elongated hollow seat **42** for governor spring **40** is not fixed or directly connected to the throttle rod **44** which is adjustable to set the desired speed of the engine. The seat **42** is provided with peripheral lands **42*a*** and **42*b*** at its ends to slidably mount the seat **42** in the sleeve **14** coaxially of the metering valve **19**. Seat **42** is provided with a central aperture or orifice **46** which cooperates with governor spill needle **48** to regulate the effective opening of orifice **46**.

Since the metering valve **19** is coupled for rotation with the centrifugal governor and the governor spring **40** mechanically couples the seat **42** for rotation with the sleeve **14**, it will be apparent that the seat **42** is driven for rotation while at the same time is axially movable relatively thereto. Since as indicated above, the metering valve **19** assumes a position of equilibrium when the biasing force of the spring **40** equals the opposing force of flyweights **30**, it will be seen that the axial position of the spring seat **42** must be axially fixed at any preselected speed. Since the spring seat **42** is axially slidable with respect to the spill needle **48**, it will be apparent that its position will be determined by the fluid pressure within the chamber **50** between axially adjustable low idle stop **52** and the spring seat **42**.

The chamber **50** houses the idle spring **54** which serves to provide a light bias on the spring seat **42** when the throttle plunger **44** is moved to its full right position and the spill needle **48** is biased to its full right position by spring **56**. A second adjustable stop **58** is likewise threadedly engaged with the mating thread of the sleeve **14** to limit the maximum movement of the spill needle **48** (and the throttle rod **44**) toward the left to fix the top speed of the engine due to the bottoming of the needle **48** thereon.

Fuel may enter the chamber **50** by flowing from the annulus **18** around the metering valve **19**, past an annular rib **60** which provides a fixed restriction to the flow of fuel into chamber **50** by defining an annular clearance with the bore of the governor sleeve **14** of about ½ mil. The fuel then flows through a variable orifice **62**, the effective area of which varies in accordance with the amount which the metering port **15** is opened and inversely therewith. The fuel then flows into chamber **50** by longitudinal passageway **64** provided in the outer wall of the governor sleeve **14**, port **66**, annulus **68**, and the slots **70** through the land **42*b*** of seat **42**.

Fuel may be discharged from the chamber **50** through the variable flow restriction offered between the orifice **46** of the spring seat **42** and the spill needle **48** and then passes through the isolated longitudinal passageway **72** of the metering valve **19**, annulus **74**, passageway **76**, and the governor chamber **78** to be returned to the inlet of the transfer pump **8** through a passageway **80** and housing pressure regulating valve **82** which maintains a constant housing pressure within the pump on the order of say, 3 pounds.

With the foregoing construction, it is readily apparent that the fuel pressure in the chamber **50** is regulated by the automatic variation of the inlet and outlet flow restrictions offered at the variable inlet port **62** and the variable spill port **46**. Moreover, since the pressure in the chamber **50** must balance the opposing force of governor spring **40** (and housing pressure within the chamber **92** which is fixed at a constant level by the housing pressure regulating valve **82**) and this force, in turn, must be equal to the force imposed on the metering valve **19** by the centrifugal weights, it is readily apparent that the spring pressure in the chamber **50** will be regulated automatically at a level correlated with engine speed.

It will be observed that the left end of the metering valve **19** is spaced from the bore of the metering valve sleeve **14** to provide an annular passageway terminating with the land **84** of the metering valve **19**. Any air which might be entrained in the fuel in the housing of the pump **4**, being lighter than the fuel, will tend to be centrifugally separated from the fuel by the rotation of the governor and concentrated adjacent the spin axis thereof and in annular passageway **84**. The land **86** of metering valve **19** is axially disposed with respect to the vent port **88** so as to provide communication between the port **88** and the annular passageway **84** only under light engine load conditions or when the metering valve is moved substantially to its full right position to vent the air through conduit **90** back to the tank **1**, this being the only return path between pump **4** and the tank **1**. Since port **88** is open under conditions of very light load only, it will be apparent that, except for any slight seepage at a uniform rate past land **86**, all the fuel flowing past the orifice **5** at the inlet of control box **3** under all other conditions is delivered to the engine.

From the foregoing, it will be apparent that this invention provides a governor in which the metering valve **19**, throttle rod **44**, the axially adjustable valve seat **42** for providing a control pressure in the chamber **50**, which is correlated with engine speed for purposes hereinafter more fully described, are coaxially arranged in the bore of the governor sleeve **14** to provide a unique, compact and simplified assembly. Moreover, the members providing for the variable restriction of the metering port **15**, the variable inlet port **62** and the spill port **46** are positively driven for relative rotation to prevent sticking in an arrangement by the driving connection between the metering valve **19** and the use of spring **40** for coupling valve **19** to slidable valve seat **42** without interference with their assumption of independent axial positions.

Under normal conditions, the metering port **15** under the influence of the governor serves as a sole means for regulating the flow of fuel to the engine. In accordance with another aspect of this invention, means independent of the metering valve are provided to regulate the maximum delivery of fuel to the engine at a varying rate dependent upon the engine speed set by the governor, thereby providing the desired torque curve for the engine. As shown, this torque limiting means takes the form of the torque limiting valve **20** which is positioned in the inlet passage for the charge pump **26** in series with the metering valve **15**.

Normally the torque limiting valve **20** is biased by its return spring **21** to its uppermost position so that it has no effect upon the regulation of the flow of the fuel to the rotor **22**. However, as hereinafter more fully described, when the desired maximum fuel flow to the engine at a given engine operating speed is approached, the torque limiting valve **20** is moved downwardly by the pivoted torque beam **96** and assumes exclusive control of the delivery of fuel to the engine.

As hereinbefore described, the pressure of the fuel within the chamber **50** is automatically regulated so as to produce a pressure which is closely correlated with engine speed. Because of the open communication between the chamber **50** and the port **66**, the pressure in the branch passageway **98** will likewise be at a level correlated with speed. The passageway **98** communicates with the cylinder **100** in the control box **3** and exerts a pressure urging the piston **102** upwardly. Desirably, the piston **102** is formed of carbon so as to have low inertia and low frictional resistance with the glass cylinder sleeve **104** of the cylinder **100**. The upper end of the piston **102** engages the follower **106** of the spring biased first class lever **108** which is pivoted on a fixed fulcrum **110**.

As best shown in FIG. 2, a flow needle **114** is secured to the free end of the lever **108** by means of a ball joint **112**. The free end of flow needle **144** is of non-uniform cross section and shown as being tapered and cooperates with the orifice **5** to provide a variable restriction to the flow of fuel. Since the control pressure in the cylinder **100** is correlated with engine speed, it will be apparent that the force applied to the piston **102** and hence its position in the cylinder **100** will be dependent upon the speed of the engine. Hence, the axial position of the flow needle **114** relative to the orifice **5** will likewise be determined by the speed of the engine and the resistance offered by the orifice **5** to the flow of fuel will vary with engine speed.

The decreased resistance offered by the orifice **5** to the flow of fuel is translated into a means for actuating the torque limiting valve **20**. As shown in the illustrative embodiment of the invention, the means for actuating the torque limiting valve **20** takes the form of a mechanical or hydraulic servomechanism.

A chamber **120** in the control box **3** communicates with the upstream side of orifice **5** through port **122** and a chamber **124** communicates with the downstream side of the orifice **5** through a port **126**. Thus, the pressure differential between the chambers **120** and **124** corresponds with the pressure drop across the orifice **5**. A diaphragm **128** isolates the chamber **120** from the chamber **124** and carries a servo valve plunger **130**. A weak constant rate spring **132** acts on the diaphragm **128** in an upward direction so as to oppose the hydraulic force caused by the pressure differential acting on the diaphragm **128**.

Branch passageway **11** connected to the outlet of the air separator **9** of pump **4** is also connected through a duct **134** in the control box **3** to deliver fuel to a chamber **136** through a fixed throttling orifice **138** which serves to throttle any fluctuations in the pressure of the fuel delivered and to stabilize the flow of fuel entering the chamber **136**. The chamber **136** is isolated from the chamber **124** by a rigid wall **140** which also serves to slidably and sealably mount an axially movable servo valve seat **142** coaxially of valve plunger **130**. The valve seat **142** is ported at **143** and cooperates with valve plunger **130** to control the rate of discharge of fuel from the chamber **136** into the chamber **124** for dumping fuel discharged from the chamber **136** into the inlet of the transfer pump through port **126** donwstream of orifice **5**. Fixed to servo valve seat **142** is a power diaphragm **144** and a depending rod **146** which engages torque beam **96**.

During normal operation, while the output of the pump is under the exclusive control of metering valve **19**, the pressure drop across the orifice **5** is such that the valve plunger **130** is positioned so as to provide only a slight restriction to the flow of fuel through the port **143** of the valve seat **142** so that pressure does not build up in chamber **136** and therefore little force is applied to the torque beam **96** by power diaphragm **144**. Accordingly, the torque limiting valve **20** remains at its top position where it is ineffective in regulating the flow of fuel to the engine. However, as the load on the engine increases, the flow past the variable orifice **5** also increases to increase the pressure drop across the orifice. This in turn causes a corresponding pressure differential across the diaphragm **128** to move the plunger **130** downward against the bias of the spring **132**. This increases the restriction offered by the servo valve **142** and causes a build up of pressure in the chamber **136**. This in turn moves the power diaphragm **144** downwardly and applies a force on the rod **146** to urge the torque beam **96** downwardly against the bias of spring **21**.

When the flow past the orifice **5** reaches the maximum desired level for the specific operating speed of the engine, the pressure differential against the orifice **5** and the diaphragm **128** is sufficient to cause the servo needle valve to restrict the flow through the valve **142** sufficiently to force the power diaphragm or servo motor **144**, the torque beam **96**, and the torque limiting valve **20** downwardly so that the torque valve offers a restriction to the flow of fuel. Since the metering valve **15**, under the influence of the governor, acts to maintain the engine at a constant speed, it simultaneously assumes its full open position and the torque valve **20** serves as the exclusive means for regulating the flow of fuel to the engine.

It will be noted that as the power diaphragm or servo motor **144** is lowered to actuate the torque limit valve **20**, it also lowers the servo valve seat **142** to tend to reduce the restriction to the flow of fuel from the chamber **136**. However, any increase in the flow past orifice **5** will tend to cause a corresponding movement of plunger **130** to produce an equilibrium condition. This servo follow-up, or feedback action, of the valve seat **142** relative to valve plunger **130** in arriving at the equilibrium position improves system stability and regulates the torque control valve **20** with high sensitivity without hunting.

Since the shape of the portion of the torque limiting valve **20** which offers a restriction to the flow of fuel to the engine as well as the amount of axial movement of the valve **20** are variables in the regulation of fuel by valve **20**, it can be seen that the shaping of the tip of valve **20** as well as the spring bias on lever **108** may be utilized in shaping of the torque curve (i.e. the maximum flow of fuel at any speed of the engine throughout its operating range) to match that desired for the particular engine with which the control is used.

Preferably, the tip of valve **20** is provided with an extension **20*a*** of the same size as the valve stem to equalize the axial forces on the valve due to the pressure of the fuel in passage **152** which is being regulated. With this construction the closed end of the bore receiving the valve tip extension **20*a*** communicates with governor chamber **78** so as to provide fixed housing pressure therein.

FIG. 3 is a top view of the operating lever and spring subassembly **108** used in the embodiment of FIG. 2. As shown, the spring thereof has in effect a variable rate which is adjustable to match the torque curve of a plurality of engines with which the invention is used. One end of the spring is fixed to the lever **108** and the other end is provided with a plurality of spring leaves **108*a***. Threadably mounted in the control box **3** so as to adjustably and selectively cooperate with the several spring leaves **108*a*** are a plurality of adjusting screws **150**. These adjusting screws **150** are respectively adjusted so as to sequentially engage an increasing number of the spring leaves 108a as the lever 108 pivots clockwise to provide the desired level of biasing force opposing the piston 102. As shown in FIG. 2, which illustrates the condition at low speed, only one of the screws may be adjusted to engage a spring leaf 108a, for example, and the other adjusting screws 150 are spaced different distances from their associated leaves 108a so that they sequentially engage a spring leaf 108a at different amounts of rotation of pivoted lever 108 biasing the lever at different spring rates to oppose the centrifugal pressure on the piston 102 as the engine speed changes. Thus, the screws may be adjusted so as to position the control needle 114 to produce a given pressure drop across the orifice 5 to effect the initiation of the control of the torque control valve 120 at different fuel flow rates throughout the speed range.

Referring again to the speed control governor which regulates the speed of the engine under normal opening conditions, it will be apparent that the centrifugal pressure required in chamber 50 to balance the flyweight force at any given speed will be greater when the metering valve is moved to its closed limit (i.e., when the flyweights 30 have their greatest radius of gyration) than when the metering valve is at the open limit. This is of no consequence when flow is controlled by the metering valve; however, if the centrifugal, or control pressure in chamber 50 changes during the regulation of fuel flow by the torque limit valve 20, the controlled flow level will change because a change in the control pressure will change the restriction offered by the flow orifice 5 and the needle 114 will change as a result of the axial movement of the control needle 114.

In order to compensate for this condition, and as shown in FIG. 1, the toe of the flyweights are curved so that the point of toe contact with thrust washer 154 moves as the radius of gyration of the flyweights increases to increase the ratio of the length of the lever arm from the toe contact to the flyweight pivot to that of the lever arm from the center of gyration of the flyweights to the flyweight pivot. By shaping the toe, such compensation is effected at least during part of the metering valve (and flyweight) travel as the metering valve approaches its full open position. In this manner, the control pressure acting on the piston 102 is unchanged due to the movement of the flyweights to full open position when the torque limit valve 20 is controlling the rate of fuel flow to the engine.

As indicated above, since the bleed port 88 is open to return fuel and any entrained air back to the tank 1 only at light engine loads and the metering valve 15 is fully open when fuel flow is regulated by torque limit valve 20, any return flow to the tank under torque control conditions is fixed at a constant level dependent upon any leakage past land 86 of metering valve 19. Thus, the fuel which passes the orifice 5 is an accurate measure of the fuel delivered to the engine.

Referring now to FIG. 4, there is shown a modified form of the embodiment in FIG. 1 which incorporates a manifold pressure and temperature compensator. In this embodiment, a trimmer spring 160 having a constant spring rate is positioned in the chamber 120 with its lower end bearing against diaphragm 128. The upper end of the trimmer spring 160 engages an axially movable loading plate 162 which is connected to compensator rod 164 so as to be movable therewith. In this embodiment, the upper wall 166 of the chamber 120 is provided with an annular recess 168 which is apertured to mount the compensator rod 164 for reciprocating movement coaxially of the servo valve 130. Desirably, the compensator loading plate 162 is slightly spaced from the walls 170 of the recess 168 to serve as a dashpot thereby to stabilize the movement of the compensator rod 164 against rapid fluctuations.

The upper end of the compensator rod 160 engages a lever 172 which is pivoted on the side wall of the compensator 158. Above the lever 172 is an adjustable trimmer control arm 173 having a universal or ball joint connection with an adjusting screw 174. The free end 176 of the trimmer control arm 173 is provided with spherical surfaces engageable with lever 172 and a second lever 178 which is likewise pivotally mounted on the side wall of the compensator. On the opposite side of the spherical surface 176 from the lever 172 so that the adjustment of the trimmer adjusting screw 174 changes the lever arms of both levers 172 and 178 to change the biasing force applied to the trimmer spring 160 through the compensator rod 164.

As shown, the compensator 158 includes an evacuated bellows member 180 which is held in its extended position by an internal spring 182 so that increasing pressure in the compensator 158 external of the bellows member 180 tends to shorten the bellows. The compensator 158 is provided with a manifold pressure inlet 184 which communicates with the manifold of the associated engine to equalize the pressure within the compensator 158 with that in the manifold. It will thus be seen that with increasing manifold pressure, the biasing force exerted on the trimmer spring 160 is decreased to allow the servo control valve 130 to move upwardly. Since the biasing force of trimmer spring 160 aids the pressure differential across the orifice 5 in closing the servo valve, a greater pressure differential across the orifice 5 is required to actuate the torque limiting valve 20. Accordingly, where the manifold pressure is increased to provide a greater quantity of air in the cylinders of the associated engine, a higher rate of flow of fuel across orifice 5 is required to actuate the torque limiting valve 20. In effect, this arrangement varies the maximum flow of fuel with manifold pressure to provide a fixed maximum fuel-air ratio in the cylinders regardless of variations in manifold pressure.

The illustrated embodiment, as shown, includes a pair of bimetal thermal elements 186 which tend to collapse when subjected to increased temperatures to reduce the biasing force of the trimmer spring with increased temperatures. Where temperature compensator elements 186 are incorporated in compensator 158, it is necessary to provide bleed port 188 to assure the flow of manifold air across the thermal elements 186. An increase in air temperature has the opposite effect of increasing manifold pressure on the desired biasing force of trimmer spring 160 and in the assistance it gives the pressure differential across orifice 5 in actuating torque limiting valve 20. Moreover, it will be apparent that the adjustment of the trimmer control 173 may be used to regulate the biasing force of trimmer spring 160 in conjunction with the adjustment of the biasing spring for lever 108 to position the needle valve 114 which adjusts the effective size of the orifice 5 and hence the position of servo valve 130 (and the actuation of torque limting valve 20) under any operating condition.

From the foregoing, it will be apparent that this invention provides a viscosity insensitive torque control arrangement which is readily adjustable to match the desired torque curve throughout the operating range of an associated engine for the plurality of flow rates required for different engines with which it may be used.

It is further apparent that the invention provides for the actutaion of a torque limiting valve independently of the metering valve which regulates fuel flow to the engine under less than maximum torque conditions and provides a flow sensing signal of the same magnitude for actuating the torque valve regardless of the flow rate of the fuel or the speed of the engine.

Further, the invention provides for the development of a reliable hydraulic control pressure by the speed control governor which acts through a unique arrangement for positioning a flow control needle with respect to an orifice to provide an accurate speed signal for use in the actuation of the torque limiting valve independent of other variables.

As will be apparent to persons skilled in the art, various modifications and adaptations of the structure above-described will become readily apparent without departure from the spirit and scope of the invention, the scope of which is defined in the appended claims.

I claim:

1. A fuel supply apparatus for an internal combustion engine comprising means for supplying fuel, delivery means adapted to be connected to the engine for conducting fuel thereto, a duct interconnecting said supply means and said delivery means, and a pair of valves in series in said duct, each of said valves having variable orifices for controlling the flow of fuel through said duct, first automatic control means for controlling one of said valves to increase its orifice opening upon increase of load on the engine to promote constant speed operation of the engine and second automatic control means for controlling the other valve to reduce its orifice opening for limiting the maximum flow of fuel to the engine at a different level of flow for each engine speed throughout the operating speed range of the engine.

2. An apparatus as recited in claim 1 wherein said other valve limits maximum flow of fuel to the engine at a varying level which increases with engine speed.

3. An apparatus as recited in claim 1 wherein said first and second automatic control means are independent of each other.

4. An apparatus as recited in claim 3 wherein each of said valves operates to control the flow of fuel through said duct only when the other is at is maximum effective open position.

5. An apparatus as recited in claim 2 wherein said second automatic control means includes means for generating a control pressure which increases with increased engine speed and means responsive to said control pressure to control said other valve.

6. An apparatus as recited in claim 5 wherein said responsive means includes means for varying the hydraulic resistance of a control orifice in said duct and said other valve is actuated by a substantially constant drop in pressure across said control orifice regardless of the rate of flow of fuel therethrough at varying engine speeds and loads.

7. An apparatus as recited in claim 6 wherein said means for varying the hydraulic resistance of said control orifice includes a spring biased needle valve which cooperates with said control orifice to provide a hydraulic resistance across said control orifice which decreases with increased engine speed.

8. An apparatus as recited in claim 7 wherein said needle valve is biased in a direction to reduce the effective size of said control orifice by a spring having a varying effective spring rate.

9. An apparatus as recited in claim 8 wherein said control pressure acts on said needle valve to position the same to produce a fixed pressure drop across said control orifice at a rate of flow of fuel therethrough which increases with increased engine speed.

10. An apparatus as recited in claim 9 wherein said control pressure opposes the biasing force of said spring and said spring is adjustable to provide an effective spring rate which is adjustable to position the needle valve so as to produce the desired rate of flow of fuel at the fixed pressure drop across the orifice to match the desired torque curve of a plurality of engines.

11. An apparatus as recited in claim 10 wherein a servo valve responsive to the pressure drop across said orifice controls the force acting on said other valve to actuate the same when the pressure drop across the orifice reaches a prescribed fixed level.

12. A device as recited in claim 11 wherein the servo valve has movable seat and plunger members, one of said members being connected to a lever to actuate the other valve against the bias of a spring.

13. In a fuel supply apparatus for an internal combustion engine, a duct, said duct having a control orifice and a torque limit valve connected in series therein, a control needle cooperating with said control orifice to provide a variable hydraulic restriction therethrough, means biasing the control needle in a direction to reduce the effective size of said control orifice, means providing an opposing force to vary the position of the control needle relative to said control orifice so as to reduce the hydraulic restriction therethrough as engine speed increases, an hydraulic servomotor to actuate said torque limit valve to limit the maximum rate of flow of fuel through said duct at a different level of flow at each engine speed throughout the operating speed range of the engine, and a control valve responsive to the pressure drop across said control orifice to initiate the operation of said servomotor.

14. A device as recited in claim 13 wherein said control valve is provided with movable seat and plunger members, one of said members being responsive to the pressure drop across said control orifice to assume a position which changes with engine speed and the other being operably connected to said servomotor.

15. A device as recited in claim 14 wherein said movable seat and plunger members are each mounted on diaphragms in chambers independent of each other, one of said diaphragms being acted upon by the pressure drop across said control orifice for fixing the position of one of said members and the other of said diaphragms being responsive to a pressure which increases when the control valve approaches its closed position to initiate the operation of the servomotor.

16. A device as recited in claim 15 wherein the first diaphragm is biased in a direction opposing the pressure drop across the orifice by a constant rate spring.

17. A device as recited in claim 16 wherein the valve plunger is connected to the first diaphragm.

18. A device as recited in claim 15 wherein means are provided for modifying the position of said one of said members in accordance with the manifold pressure of the associated engine.

19. A device as recited in claim 15 wherein means are provided for modifying the position of said one of said members in accordance with the manifold temperature of the associated engine.

20. In fuel supply apparatus for an internal combustion engine, means for producing a control pressure which increases with increasing engine speed comprising a mechanical governor having an axially movable valve member which rotates therewith, a governor spring engaging valve member, said governor spring having an apertured valve seat closing a chamber, a control needle cooperating with the aperture in said valve seat for controlling the effective hydraulic restriction provided thereby, means for providing a hydraulic pressure in said chamber acting on said valve seat to urge the same in a direction to increase the biasing force of said spring, said aperture and said control needle providing a variable orifice for controlling the pressure in said chamber to regulate the same at a level which increases with engine speed, and a throttle for adjusting the axial position of said control needle for setting the desired speed of the engine.

21. A device is recited in claim 20 wherein said axially slidable valve member, said valve seat and said control needle are coaxially arranged within a bore.

22. A device as recited in claim 21 wherein said valve seat is mounted for slidable movement with respect to said valve member and is provided with a driving connection therewith to rotate the valve seat.

23. A device as recited in claim 21 wherein the fluid entering said chamber passes through a restriction which is automatically regulated by the axial position of said valve member in said bore.

24. A device as recited in claim 23 wherein the fluid entering said chamber also passes through an annular passage between said valve member and said bore, and said valve member having an annular rib to provide a

fixed restriction limiting the flow of the fluid into said chamber.

25. A device as recited in claim **21** wherein said mechanical governor comprises a plurality of flyweights having lever arms connected to said valve member for exerting an increased force in a direction to move said valve member towards said valve seat upon increased speed, said flyweights being constructed and arranged to compensate for the increased center of gyration of the flyweights as the valve member moves towards the valve seat during at least the part of movement of the valve member as it approaches its maximum travel.

26. A device as recited in claim **25** wherein the lever arms of the flyweights are provided with curved tips to vary the effective lengths thereof.

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LAURENCE M. GOODRIDGE, Primary Examiner

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103—41