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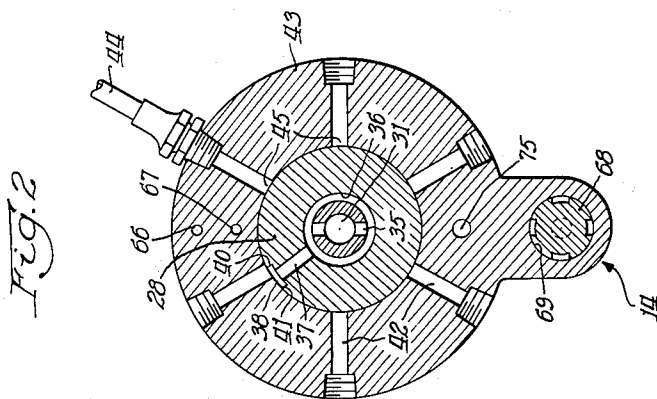
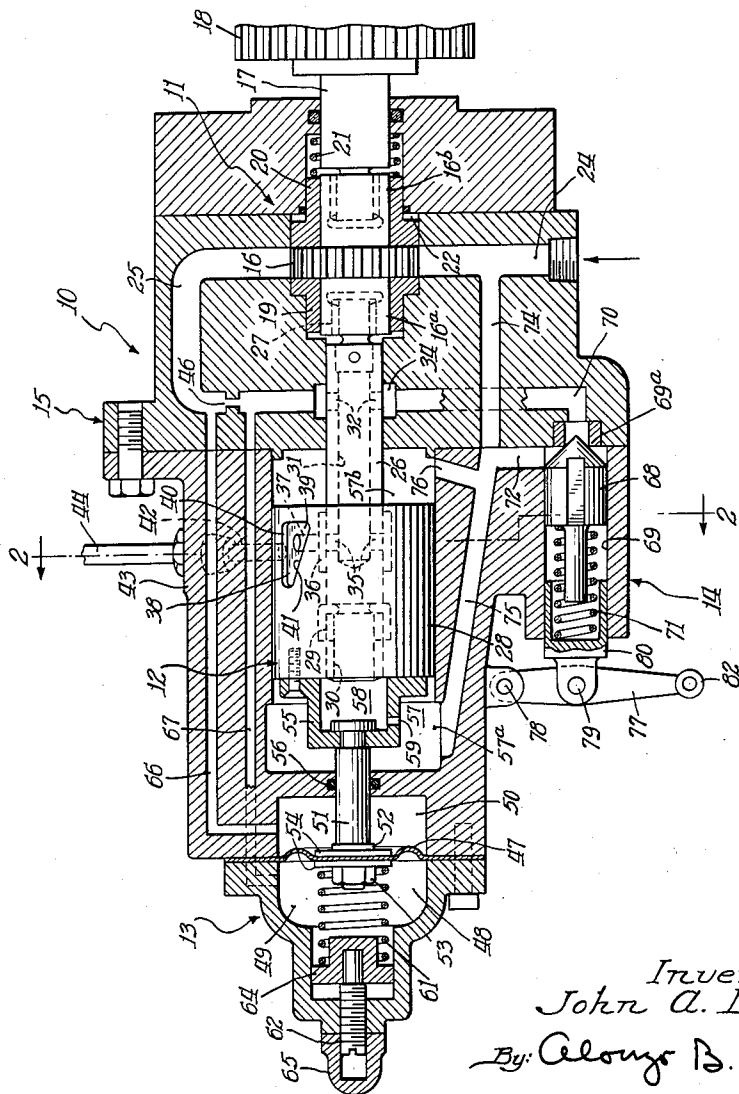
J. A. LAUCK

2,750,933

FUEL INJECTION SYSTEM

Filed Feb. 29, 1952

2 Sheets-Sheet 1



Inventor:
John A. Lauck
By: *Alonso B. Knight*

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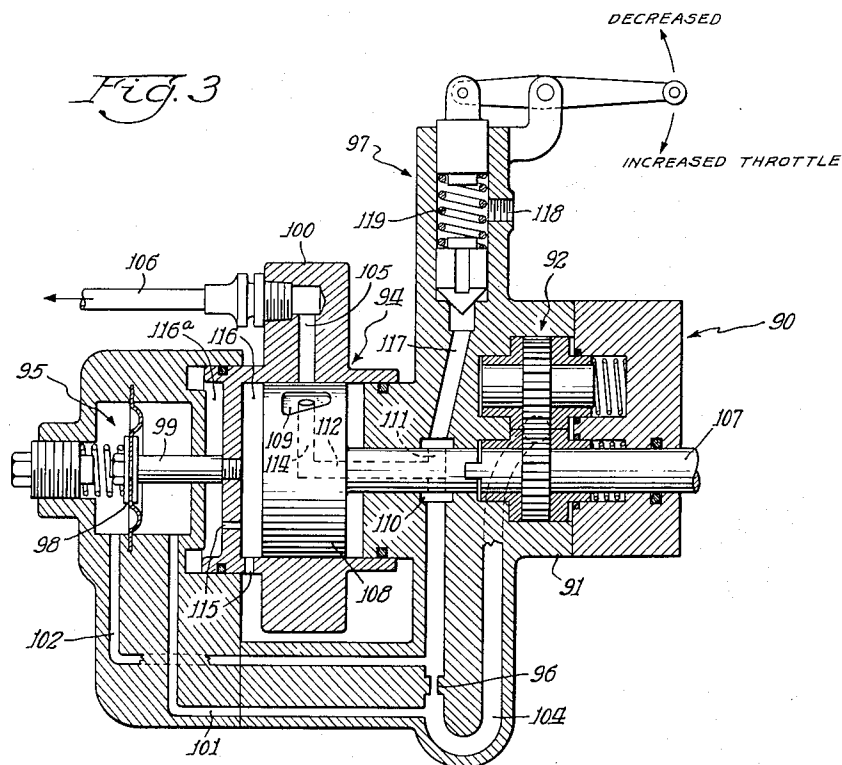
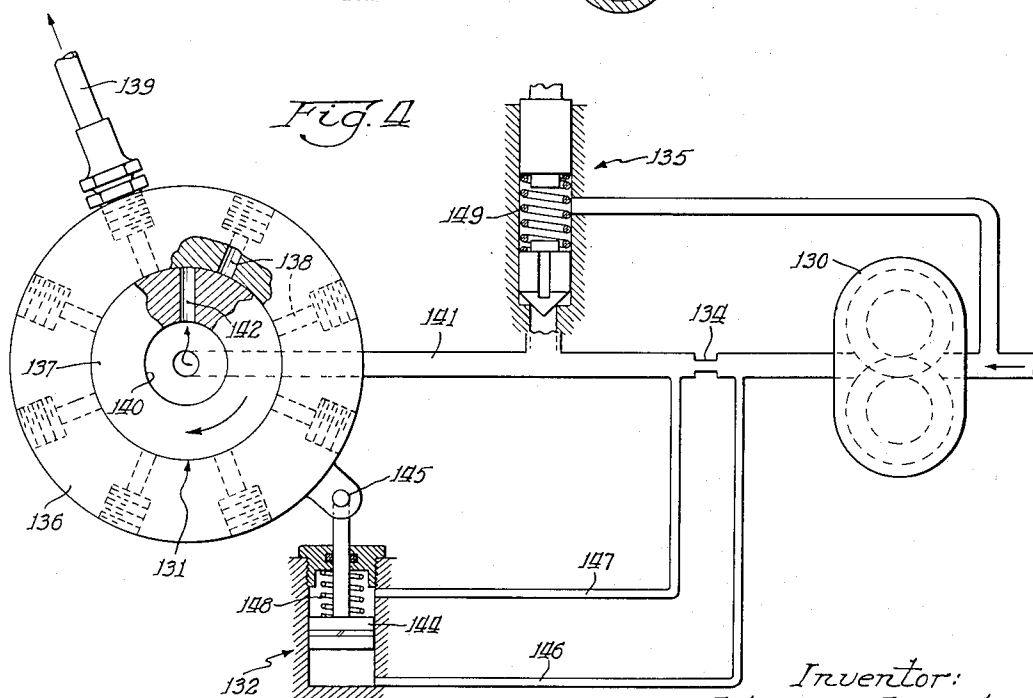
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2 Sheets-Sheet 2



Inventor:
John A. Lauck
By: *Alonso B. Kight*

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FUEL INJECTION SYSTEM

John A. Lauck, Shaker Heights, Ohio, assignor to Borg-Warner Corporation, Chicago, Ill., a corporation of Illinois

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11 Claims. (Cl. 123—139)

This invention relates to a fluid injection system and more particularly to a fluid fuel injection system for an internal combustion type engine.

Fuel injection systems have been known and utilized for many years. Such fuel injection systems provide several advantages over the ordinary carburetor system, among these advantages being better engine cooling and better fuel control during the combustion cycle. However, the known advantages of fuel injection have not led to general adoption in the automotive industry because of the complicated and expensive structures required to realize the advantages. Consequently, fuel injection has found widespread acceptance only in aircraft and other applications where cost is secondary to performance.

It will be readily understood that a fuel injection system intended for general automotive use must be competitive in price with the elements which it will replace and must perform with at least as great an efficiency and dependability.

Accordingly, an object of the present invention is to provide a simple and efficient fuel injection system, which can be produced at a low cost, and which is readily adapted for general automotive use.

An additional object of the invention is to provide a fuel injection system which can be economically utilized for replacing the fuel pump, carburetor, automatic choke and intake manifold ordinarily associated with a conventional automotive internal combustion engine.

A further object of the invention is to provide a fuel injection system of increased efficiency as compared with prior available systems and which can be produced at a competitive cost compared with the ordinary carburetion system of a conventional automotive engine.

Another object of the invention is to provide an improved fuel injection system which can be utilized with any type of fluid fuel in connection with an internal combustion engine.

In accordance with this invention, a fuel injection system is provided comprising a pump of high efficiency and relatively linear operating characteristics, which pump supplies pressure to the unique distributing valve arrangement whereby fuel under pressure is metered sequentially to the cylinders of an internal combustion engine with which the system is intended to be associated. The distributing valve arrangement includes a rotary injector valve driven in timed relation with the engine and located within a distributor structure which contains a plurality of ports connected with the respective engine cylinders. Means are provided for automatically advancing the time of initiation of injection into each of the cylinders as the speed of the engine increases. Speed control is achieved through the use of an accelerator controlled pressure relief valve in which valve the relieving pressure is manually variable for selectively by-passing a portion of the pump output. In some of the embodiments, means are also provided for maintaining the

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time of injection generally constant regardless of the speed of the engine.

Other objects, features and advantages of the present invention will be apparent from the following detailed description of several embodiments, by way of preferred examples only, taken in conjunction with the accompanying drawings in which:

Fig. 1 is a longitudinal sectional view through a preferred embodiment of an injector pump according to the present invention;

Fig. 2 is a sectional view taken along line 2—2 of Fig. 1;

Fig. 3 is a longitudinal sectional view of a second embodiment of the invention; and

Fig. 4 is a schematic view of a third embodiment of the invention.

In Fig. 1, a combination fuel pump and fuel injection distributor or injector system is generally designated by the reference numeral 10 and comprises a fuel pump 11, a distributor valve assembly 12, an injection advance servo motor 13 and an accelerator controlled pressure relief or throttle valve 14. These elements are disposed in operative relation in a casing or housing 15.

The pump 11 is of a type of high efficiency and relatively linear operating characteristics such as a pressure loaded gear pump shown in Roth and Lauck Patent No. 2,420,622, and includes a pair of meshing pump gears 16 (one shown) with one of the gears splined or otherwise secured to a pump drive shaft 17. The pump is driven through a gear 18 by an internal combustion engine (not shown) with which the fuel injector 10 is intended to be utilized. Each of the pump gears 16 has a hub 16a journaled in a relatively fixed sealing bushing 19 and a hub 16b journaled in an axially movable, pressure loadable sealing bushing 20. Initial loading springs 21 are provided for urging each of the bushings 20 toward the associated gear side faces to provide an initial seal. Fluid under pressure from the outlet side of the pump 11 is communicated to a pressure loading chamber 22 behind each of the bushings 20 to provide pressure loading of the bushings for sealing about the gears 16 in the manner shown by the aforementioned Roth and Lauck patent. The shaft 17 is rotated in a manner such that a passage 24, formed in the casing 15, is the pump inlet passage and a passage 25, also formed in the casing, is the pump outlet passage. It will be apparent that fluid fuel is drawn from a fuel tank or the like (not shown) by means of the positive displacement gear pump 11 and that the rate of flow of pressurized fluid into the outlet passage 25 will vary substantially directly with the speed of the pump drive shaft 17 and the internal combustion engine.

The distributor valve assembly 12 includes a pintle valve 26 which is splined within the hub 16a or otherwise secured for concurrent rotation with one of the pump gears 16 at 27. A rotary distributor or injector valve 28 is axially shiftably splined about the pintle valve 26 by means of external splines 29 formed on one end portion of the pintle valve and internal splines 30 formed within the rotary valve parallel to and equally spaced about the axis of rotation of the rotary valve. The pintle valve 26 is provided with an axially extending inner passage 31 which communicates with the pump outlet passage 25 by means of a plurality of radial ports 32 and an annular supply groove 34 formed in the casing 15 and with which the passage 25 is connected. The passage 31 is provided with a plurality of radial outlet ports 35 which communicate with an axially elongated annular groove 36 formed in the rotary valve 28 in such a manner that the ports 35 will always communicate with the

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groove 36 regardless of the axial position of the rotary valve 28 between the operative limits of axial movement.

For providing successive fuel injection into the respective cylinders (not shown) of the internal combustion engine, a passage 37 is formed in the rotary valve 28 and communicates between the annular groove 36 and an external circumferential discharge port in the form of an injection cavity or relief 38 formed in the rotary valve 28. In the present instance, the relief 38 is of triangular configuration with a vertex 39 formed in the direction of rotation of the rotary valve 28 so that a trailing edge 40 of the relief is substantially parallel to the axis of the rotary valve and a leading edge 41 is slanted forwardly in the direction of rotation from the trailing edge. A plurality of radially extending cylinder supply passages 42, herein shown as six in number (Fig. 2), are formed in a distributor structure portion 43 of the casing 15. The passages 42 correspond to and communicate with the respective cylinders of the internal combustion engine by means of respective cylinder supply conduits 44. Respective, radially equally spaced, injection inlet ports 45 are formed at the axially inward ends of the passages 42 and are adapted for successively communicating with the injection relief 38 as the valve assembly 12 is rotated.

If the fuel injector 10 is utilized with a four cycle internal combustion engine, the pump 11 and the rotary valve 28 are driven at one-half engine speed in order that the injection into each cylinder corresponds with the power strokes of the respective engine pistons. If the fuel injector is utilized with a two cycle internal combustion engine, the pump 11 and the rotary valve 28 are driven at full engine speed for the same reason.

It will be readily seen that the period of injection into each of the engine cylinders will be governed by the axial position of the rotary valve 28. Movement of the rotary valve to the right as shown in Fig. 1 results in a decrease in the injection period and movement to the left results in an increase in the injection period for each cylinder. Also, it will be seen that the point of initiation of injection into each cylinder will be advanced as the rotary valve 28 is moved toward the left in Fig. 1 because of the forwardly slanted leading edge 41, while the point of injection cut off will remain at substantially the same position regardless of the axial movement of the rotary valve 28 because of the configuration of the trailing edge 40. Of course, any desired relationship of injection initiation, injection cut-off and injection period with axial movement of the rotary valve 28 can be achieved by modification of the trailing and leading edges 40 and 41.

According to the present invention, means are provided for shifting the rotary valve 28 axially in response to changes in rotational speed of the pump drive shaft 17, and consequently operational speed of the internal combustion engine, in order to provide injection advance or retard in accordance with change in engine speed. Herein such means comprise the injection advance servomotor 13 and a fixed restriction or control orifice 46 provided in the passage 25. The injection advance servomotor 13 comprises a flexible diaphragm 47 secured about its periphery and disposed within a cavity 48 formed in the casing 15. The diaphragm 47 divides the cavity 48 into a sealed chamber 49 on one side of the diaphragm and another sealed chamber 50 on the other side of the diaphragm. A connecting member or rod 51 has one end portion fixedly secured to the central portion of the diaphragm 47 by means of a shoulder 52 on one side of the diaphragm and a nut 53 on the other side which serve to clamp the central portion of the diaphragm between a pair of washers 54. The other end portion of the rod 51 is rotatably secured in any suitable manner to a connection member 55 which is fixedly secured at one end of the rotary valve 28. The rotary valve 28, including the member 55, is rotatably and axially shiftably disposed in a cylindrical chamber 57 formed in the casing 15 so that axial shifting of the rotary

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valve 28 may occur for a distance substantially equal to the length of the triangular relief 38 without interference with the casing. A fluid seal 56 is provided about the rod 51 between the chambers 50 and 57. An axial clearance cavity 58 is provided in the member 55 in order to prevent interference between the end of the pintle valve 26 and the member 55 when the rotary valve shifts to the right as seen in Fig. 1. The cavity 58 is vented to the chamber 57 through an opening 59.

A compression spring 61 acts against one of the washers 54 to urge the rod 51 and rotary valve 28 toward the right as seen in Fig. 1 to bring the small portion of the triangular relief 38 into respective communication with the ports 45 as the valve 28 rotates. The compressive force exerted by the spring 61 may be adjusted by means of an adjusting screw 62 threadably inserted through the left end of the casing 15 and holding a spring seat member 64 against the opposite end of the spring 61 from the abutting washer 54. The adjusting screw 62 may be held in any adjusted position by means of a lock nut 65.

In order to provide a differential pressure bias on the diaphragm 47 in accordance with the fuel flow through the pump 11, a passage 66 imposes the pressure upstream of the orifice 46 within the chamber 50 and a passage 67 imposes the pressure immediately downstream of the orifice 46 within the chamber 49. Thus, the pressure differential acting on the diaphragm 47 is equal to the pressure drop through the orifice 46 which, in turn, varies as the square of the rate of fuel flow from the pump 11.

The spring 61 performs a two-fold function, first, it urges the rotary valve 28 toward injection retarded position when the engine is idling or shut off and, second, it imposes an increasing spring bias against movement of the rotary valve 28 as the engine speed and consequently the rate of fuel flow to the pump 11 increases. By properly adjusting the spring bias rate of the spring 61 the rate of injection advance can be selected to vary substantially linearly with the rate of increase in engine speed, or any desired relationship therebetween can be achieved.

For the purpose of varying the fuel injection pressure of the fuel immediately prior to injection into the engine cylinders, and consequently the operational speed of the internal combustion engine, the accelerator controlled pressure relief valve 14 is provided. The valve 14 comprises a valve member 68 slidably mounted in a bore 69 formed in the casing 15. The member 68 is adapted for seating against a valve seat 69a formed at one end of the bore 69 to block a passage 70 communicating with the annular supply groove 34. A compression spring 71 urges the member 68 into seating relation while fluid pressure in the passage 70 tends to move the member off of its seat to allow communication between the passage 70 and a bypass or pressure relief passage 72. The passage 72 communicates with the inlet passage 24 through another pressure relief passage 74. The passage 74 also has branches 75 and 76 communicating with respective chamber portions 57a and 57b of the cylindrical chamber 57 on opposite sides of the rotary valve 28 for preventing the formation of a hydraulic lock which would interfere with the shifting of the rotary valve.

For controlling the compressive force exerted by the spring 71 to vary the pressure at which the valve 14 relieves, a lever arm 77 is pivotally secured to the housing 15 at 78 and is also pivotally secured at 79 to a piston 80 which, in turn, is slidably disposed in the bore 69. The compression spring 71 is disposed between the piston 80 and the valve 68. An end portion 82 of the lever 77 is adapted to be secured by any suitable linkage (not shown) to the accelerator pedal (not shown) of the internal combustion engine. The accelerator linkage may be arranged so that depression of the accelerator pedal will cause movement of the end portion 82 of the lever 77 to the right, as seen in Fig. 1, so that the compression force exerted by the spring 71 on the valve member 68 will be increased to increase the pressure at which communication will oc-

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cur between passages 70 and 72. As a result, the fuel injection pressure will be increased, due to an increase in pressure in the passages 70, 25, 31 and 42 and the ducts 44, to cause an increase in the amount of fuel injected into each engine cylinder.

Summarizing the operation of the fuel injector pump 10, the pressure loaded gear pump 11 is driven at one-half engine R. P. M. for a four cycle general combustion engine and directly at engine R. P. M. for a two cycle internal combustion engine through the gear 18 and the shaft 17. Fluid fuel under pressure passes into the discharge passage 25, thence through the orifice 46 into the annular groove 34, through the ports 32 into the passage 31, through the ports 35 into the annular groove 36 and through the passage 37 into the triangular relief 38. From the triangular relief 38 the fluid fuel is injected successively, as the valve 28 is rotated, into the cylinders of the internal combustion engine through the respective passages 42 and ducts 44. It will be seen that driving the pump 11 and consequently the rotary valve 28 at one-half engine speed, in the case of a four cycle engine, will inject fuel at substantially the beginning of the power stroke in each of the engine cylinders when the timing is adjusted properly, and driving the rotary valve 28 directly at engine speed in the case of a two cycle engine will achieve the same result.

The rotary valve 28 will be shifted to the left as seen in Fig. 1 by an increase in fuel flow through the orifice 46 by means of the increased pressure differential across the diaphragm 47 acting against the bias of the compression spring 61. Such a shifting of the rotary valve 28 to the left provides both an increase in the injection period to each of the engine cylinders and an advance in the point of initiation of injection by means of the triangular relief 38. It will be understood that a decrease in speed of the engine resulting in a decrease of fuel flow through the orifice 46 will decrease the pressure differential across the diaphragm 47 to permit the rotary valve 28 to move to the right, due to the action of the compression spring 61, to decrease the injection period to each cylinder and to retard the point of initiation of injection.

In initial construction of the device the spring rate of the spring 61, the size of the orifice 46, the size and configuration of the diaphragm 47, the size and configuration of the triangular relief 38 and the size of the pump 11 can all be varied to achieve any desired relationship between engine speed, injection period and injection advance. For example, the relief 38 might well be of some configuration other than triangular or if triangular, the vertex might be formed in the trailing side. In a stationary engine which is operated at constant speed, it may be desirable to have the injection period extend farther past top dead center when the engine is operating under full load while at the same time retaining a constant advance of injection throughout the load range. Adjustments in the compression exerted by the spring 61 can be made in the field through use of the adjusting screw 62. Preferably, the arrangement is such that the injection period varies directly as the speed of the engine in order that the time of injection remains substantially constant regardless of engine speed. For example, the arrangement may be such that the injection period is six times as great at 3600 R. P. M. of the engine as it is at 600 R. P. M., so that the actual time of injection at both of these speeds would be substantially the same.

An example of variation of injection advance which might be utilized is the provision of an initial advance of 5 degrees before top dead center on the power stroke of each piston at idle R. P. M. with a substantially linear rate of change of advance to a maximum of 35 degrees before top dead center at maximum engine R. P. M.

Control of engine speed and power is achieved through the accelerator control valve 14 by changing the compressive force exerted by the spring 71 in order to change the engine power by changing the amount of fuel by-

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passed back to the pump inlet and the amount of fuel injected into the engine cylinders for any given R. P. M. of the engine. Of course, the speed of the engine will be determined by the load thereon for any given amount

of fuel injected into the cylinders.

A somewhat similar embodiment is disclosed in Fig. 3 wherein the distributor structure is shifted axially instead of the rotary valve as is the case in Figs. 1 and 2. As shown in Fig. 3, a combination fuel pump and fuel injection distributor or injector pump 90 comprises generally a body member or casing 91, a pressure loaded gear pump 92, an injection distributor valve assembly 94, an injection advance servomotor 95, a control orifice 96 and a pressure relief type throttle valve 97.

In this embodiment, the servomotor 95 includes a diaphragm assembly 98 secured through a rod 99 to an axially shiftable distributor ring 100 for shifting the ring in response to changes in pressure differential across the diaphragm 98 corresponding to changes in pressure drop across the control orifice 96 as referenced to the opposite sides of the diaphragm through the passages 101 and 102. The control orifice 96 is located in a pump outlet passage 104 as is the case with the first embodiment.

The distributor ring 100 is provided with a plurality of equally radially spaced internal passages 105 (one shown) which are connected to the respective cylinders of the internal combustion engine through a plurality of ducts 106 (one shown).

The pump 92 is driven by the internal combustion engine through a pump drive shaft 107 which is, in turn, arranged for driving a rotary valve 108 which is not shiftable axially. The rotary valve 108 is provided with a discharge port in the form of a triangular cavity or relief 109 similar to the relief 38 of the valve 28 in Fig. 1. The triangular relief 109 is connected to the pump outlet passage 104 through an annular supply groove 110, a plurality of ports 111, an axial passage 112 in the rotary valve 108 and a passage 114. As is the case with the embodiment shown in Figs. 1 and 2, the triangular relief 109 successively communicates with the passages 105 as the rotary valve 108 and the pump 92 are driven by the internal combustion engine to inject fuel into the respective cylinders of the engine. Pressure bleed passages 115 are provided for bleeding a chamber 116 formed between the distributor ring 100 and the rotary valve 108 and another chamber 116a formed between the distributor ring and a portion of the casing 91 containing the diaphragm to prevent the formation of pressures in the chambers to interfere with the axial shifting of the ring.

The accelerator controlled valve 97 operates in a manner similar to the valve 14 of Fig. 1 to by-pass pressure from the annular groove 110 through a passage 117 to a pressure relief passage 118 leading to the pump inlet or to the fuel tank in accordance with the force exerted by an adjustable compression spring 119.

The operation of this device is identical with that of the device of Fig. 1 except that the distributor ring 100 is shifted instead of the rotary valve 108. However, the operational effects are the same as will be understood.

The embodiment shown in Fig. 4 is arranged for increasing the fuel injection advance in accordance with increase in engine speed without concurrently increasing the injection period. The arrangement which is shown schematically includes an engine driven pressure loaded gear pump 130, an injection advance mechanism 131, an injection advance servomotor 132, a control orifice 134 and an accelerator pedal controlled pressure relief type throttle valve 135.

The mechanism 131 comprises a radially shiftable distributor ring 136 and a rotary valve 137 coupled for being driven by the pump 130 in any suitable manner (not shown). The ring 136 is provided with a plurality of equally radially spaced passages 138 which are connected with the respective engine cylinders through a plurality of ducts 139. The rotary valve 137 contains an axial

passage 140 which is connected with the outlet side of the pump 132 through a pump outlet passage 141. Fluid fuel is successively injected into the passages 138 and consequently the respective engine cylinders by means of a radially extending discharge passage 142 which communicates with the passage 140 and is adapted to successively communicate with the passages 138 as the rotary valve is rotated.

The servomotor 132 may be similar to the servomotor 13 of Fig. 1 or the servomotor 95 of Fig. 3, but in the present instance is shown as comprising an axially movable piston 144 which is pivotally connected to the periphery of the distributor ring 136 at 145 so that axial movement of the piston will shift the distributor ring radially to advance or retard injection into each of the cylinders as the case may be. One side of the piston is referenced to the pressure upstream of the orifice 134 by means of a passage 146 while the other side of the piston is referenced to the pressure downstream of the orifice 134 through the passage 147. A compression spring 148 biases the piston against the pressure introduced by the passage 146. If the rotary valve is arranged to rotate in a clockwise direction as indicated, an increase in flow through the passage 141 and the orifice 134 due to an increase in engine speed and consequently pump speed will result in a counter-clockwise radial shifting of the distributor ring 136 due to the increase in the pressure differential on the piston 144 acting against the bias of the spring 148. Such a counter-clockwise shifting of the distributor ring 136 will result in an advance of fuel injection into each of the engine cylinders, but since no triangular relief is provided as is the case with the first two embodiments and since there is no relative axial shifting between the distributor ring 136 and the rotary valve 137, there will be no consequent increase in the injection period.

The accelerator controlled valve 135 operates in a manner similar to the previous two embodiments to bypass fluid fuel from the passage 141 back to the inlet side of the fuel pump with an adjustable compression spring 149 being provided to vary the relieving pressure of the valve and, consequently, the amount of fuel bypassed to vary the amount of fuel injected into the engine cylinders.

In all of the embodiments shown, it will be noted that the control orifice is located between the fuel pump and the pressure relief passage controlled by the throttle valve so that the injection advance servo motor operates in response to changes in total fuel flow delivered by the pump. Since this total fuel flow is directly proportional to the speed of rotation of the positive displacement pump which is coupled to the engine, the injection advance will be a given value for each speed of the engine regardless of the power output of the engine. If desired the operation of the injection advance servo motor can be made responsive to the flow of fuel to the engine only by placing the control orifice downstream of the pressure relief connection. With this latter arrangement, the injection advance would be generally responsive to the power output of the engine instead of the speed of the engine.

From the above description it will be readily understood that the present invention provides an improved and substantially simplified fuel injection system which can be readily utilized for general automotive applications. With this fuel injection arrangement, the fuel pump, carburetor, the automatic choke and the intake manifold of a conventional automotive carburetion system are replaced by the compact combination fuel pump and fuel injection distributor of this invention. This fuel injector is relatively simple and inexpensive to produce compared with the fuel injection systems of the prior art, and furthermore, the system is sufficiently foolproof to give the dependability required by the automotive industry.

It will be understood that modifications and variations may be effected without departing from the scope of the novel concepts of the present invention.

I claim:

1. Mechanism for injecting fuel under pressure into the cylinders of an internal combustion engine comprising a rotary valve having a discharge port therein, distributor means associated with said valve and having a plurality of ports connected to respective cylinders of said engine and adapted for successive registration with said discharge port when said valve is rotated, means for rotating said valve in timed relation with said engine, means for supplying fluid fuel under pressure to said valve for successive injection into said engine cylinders, a fluid flow restriction between said supply means and said valve, and injection advance means in fluid communication with said flow restriction and responsive to the pressure drop across said flow restriction for advancing the instant of initiation of injection into each cylinder in accordance with increase in pressure drop past said restriction.

2. Mechanism for injecting fuel under pressure into the cylinders of an internal combustion engine comprising a rotary valve having a discharge port therein, distributor means associated with said valve and having a plurality of ports connected to respective cylinders of said engine and adapted for successive registration with said discharge port when said valve is rotated, means for rotating said valve in timed relation with said engine, means for supplying fluid fuel under pressure to said valve for successive injection into said engine cylinders, a fluid flow restriction between said supply means and said valve, and a servo motor associated with said valve and said distributor means and responsive to changes in pressure drop past said restriction for changing the relative relationship between the valve and the distributor means whereby the instant of initiation of injection to each cylinder is advanced in accordance with increase in pressure drop past the restriction.

3. Mechanism for injecting fuel under pressure into the cylinders of an internal combustion engine comprising a rotary valve having a discharge port therein, distributor means associated with said valve and having a plurality of ports connected to respective cylinders of said engine and adapted for successive registration with said discharge port when said valve is rotated, means for rotating said valve in timed relation with said engine, means for supplying fluid fuel under pressure to said valve for successive injection into said engine cylinders, a fluid flow restriction between said supply means and said valve, a shiftable member having axially opposite surfaces and associated with said valve and said distributor means for changing the relative relationship therebetween to change the instant of initiation of injection into each cylinder, conduit means for referencing the respective fluid pressures upstream and downstream of said restriction to the respective opposite surface of said shiftable member, whereby the time of injection into each cylinder is advanced in accordance with increase in pressure differential between the pressures upstream and downstream of said restriction.

4. In a fuel injector, distributor structure having a plurality of spaced distributor ports therein, a rotary valve associated with said distributor structure and having a discharge port therein adapted for successive registration with said distributor ports when said valve is rotated, means for supplying fluid fuel under pressure to said valve for successive discharge into said distributor ports, discharge advance means responsive solely to the flow of fluid fuel under pressure to said valve for advancing the instant of initiation of discharge into each of said distributor ports in accordance with increase in fuel flow from said supply means, and a throttle control including a pressure relief valve fluid connected between said supply means and said rotary valve and means for selectively

changing the relieving pressure of said relief valve for varying the pressure of the fluid supplied to said rotary valve.

5. In a fuel injector, distributor structure having a plurality of spaced distributor ports therein, a rotary valve associated with said distributor structure and having a discharge port therein adapted for successive registration with said distributor ports when said valve is rotated, means for supplying fluid fuel under pressure to said valve for successive discharge into said distributor ports, a fluid flow restriction between said supply means and said rotary valve, discharge advance means in communication with the pressures through said flow restriction and responsive to said flow therethrough for advancing the instant of initiation of discharge into each of said distributor ports in accordance with increase in pressure drop past said restriction, and a throttle control including a pressure relief valve fluid connected between said supply means and said rotary valve and means for selectively changing the relieving pressure of said relief valve for varying the pressure of the fluid supplied to said rotary valve.

6. In a fuel injector, distributor structure having a plurality of spaced distributor ports therein, a rotary valve associated with said distributor structure and having a discharge port therein adapted for successive registration with said distributor ports when said valve is rotated, means for supplying fluid fuel under pressure to said valve for successive discharge into said distributor ports, a fluid flow restriction between said supply means and said rotary valve, a servomotor associated with said rotary valve and said distributor structure and responsive to changes in pressure drop past said restriction for changing the relative relationship between the rotary valve and the distributor structure whereby the instant of initiation of discharge into each distributor port is advanced in accordance with increase in pressure drop past the restriction, and a throttle control including a pressure relief valve fluid connected between said supply means and said rotary valve and means for selectively changing the relieving pressure of said relief valve for varying the pressure of the fluid supplied to said rotary valve.

7. In a fuel injector, distributor structure having a plurality of spaced distributor ports therein, a rotary valve associated with said distributor structure and having a discharge port therein adapted for successive registration with said distributor ports when said valve is rotated, means for supplying fluid fuel under pressure to said valve for successive discharge into said distributor ports, a fluid flow restriction between said supply means and said rotary valve, a shiftable member having axially opposite surfaces and associated with said rotary valve and said distributor structure for changing the relationship therebetween to change the instant of initiation of discharge into each distributor port, conduit means for referencing the respective fluid pressures upstream and downstream of said restriction to said opposite surfaces of said shiftable member, whereby the instant of initiation of injection into each cylinder is advanced in accordance with increased pressure differential between the pressures upstream and downstream of said restriction, and a throttle control including a pressure relief valve fluid connected between said supply means and said rotary valve and means for selectively changing the relieving pressure of said relief valve for varying the pressure of the fluid supplied to said rotary valve.

8. Mechanism for injecting fuel under pressure into the cylinders of an internal combustion engine comprising a rotary valve having a discharge port therein, distributor means associated with said valve and having a plurality of distributor ports connected to respective cylinders of said engine and adapted for successive registration with said discharge port when said rotary valve is rotated, means for rotating said valve in timed relation with said

engine, means for supplying fluid fuel under pressure to said valve for successive injection into said engine cylinders, means responsive solely to the flow of fluid fuel to said valve for adjusting the relative positions of the valve and the distributor structure, said discharge port and said distributor ports being formed such that change of the relative positions between the rotary valve and the distributor structure in one direction advances the instant of initiation of registration of the discharge port with the successive distributor ports, and a throttle control including a pressure relief valve between said supply means and said rotary valve and means for selectively changing the relieving pressure of said relief valve for varying the pressure of the fluid supplied to said rotary valve.

9. Mechanism for injecting fuel under pressure into the cylinders of an internal combustion engine comprising a rotary valve having a discharge port therein, distributor means associated with said valve and having a plurality of distributor ports connected to respective cylinders of said engine and adapted for successive registration with said discharge port when said rotary valve is rotated, means for rotating said valve in timed relation with said engine, means for supplying fluid fuel under pressure to said valve for successive injection into said engine cylinders, said discharge port and distributor ports being formed such that change of the relative positions of the rotary valve and the distributor structure in one direction advances the instant of initiation of registration of the discharge port with the successive distributors ports, a fluid flow restriction between said supply means and said valve, a shiftable member having axially opposite surfaces and associated with said valve and said distributor means for changing the relative position therebetween, conduit means for referencing the respective fluid pressures upstreams and downstream of said restriction to said opposite surfaces of said shiftable member, whereby the time of injection into each cylinder is advanced in accordance with increased pressure differential between the pressures upstream and downstream of said restriction, and a throttle control including a pressure relief valve between said supply means and said rotary valve and means for selectively changing the relieving pressure of said pressure relief valve for varying the pressure of the fluid supplied to said rotary valve.

10. Mechanism for injecting fuel under pressure into the cylinders of an internal combustion engine comprising a rotary valve having a discharge port therein, distributor means associated with said valve and having a plurality of ports connected to respective cylinders of said engine and adapted for successive registration with said discharge port when said valve is rotated, means for rotating said valve in timed relation with said engine, means for supplying fluid fuel under pressure to said valve for successive injection into said engine cylinders, said means for supplying fluid fuel under pressure comprising a gear pump of the pressure loaded type having at least one set of pressure loadable bushings, a fluid flow restriction between said supply means and said valve, and injection advance means in fluid communication with said flow restriction and responsive to the flow of fluid therethrough for advancing the instant of initiation of injection into each cylinder in accordance with increase in pressure drop past said restriction.

11. Mechanism for injecting fuel under pressure into the cylinders of an internal combustion engine comprising a rotary valve having a discharge port therein, distributor means associated with said valve and having a plurality of ports connected to respective cylinders of said engine and adapted for successive registration with said discharge port when said valve is rotated, means for rotating said valve in timed relation with said engine, means for supplying fluid fuel under pressure to said valve for successive injection into said engine cylinders, said

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fluid pressure responsive means including a gear pump of the pressure loadable type having at least one set of pressure loadable bushings, a fluid flow restriction between said supply means and said valve, and a servomotor associated with said valve and said distributor means and responsive to changes in pressure drop past said restriction for changing the relative relationship between the valve and the distributor means whereby the instant of initiation of injection to each cylinder is advanced in accordance with increase in pressure drop past the restriction.

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