Aug. 13, 1963 E. S. DAHL 3,100,449

FUEL INJECTION PUMP

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Fig. 1

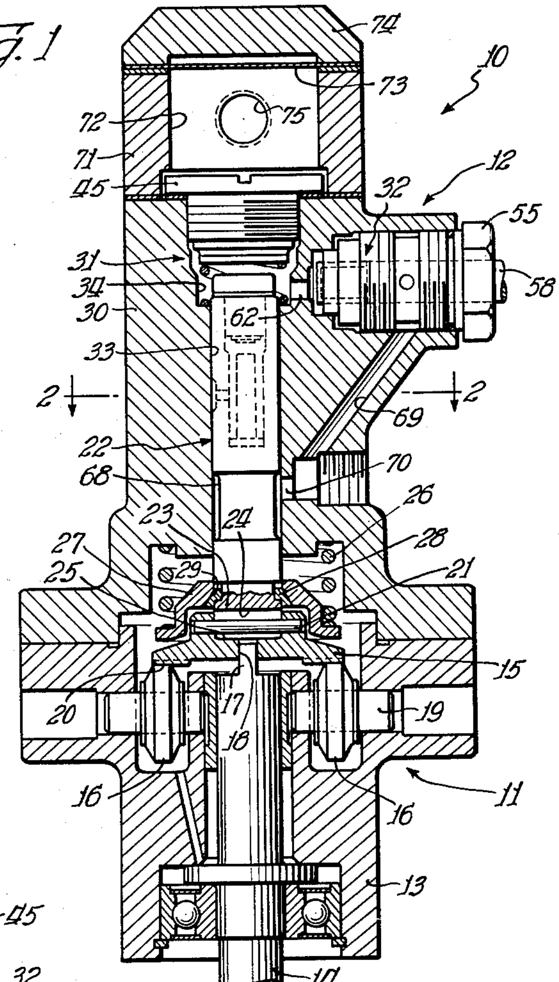


Fig. 2

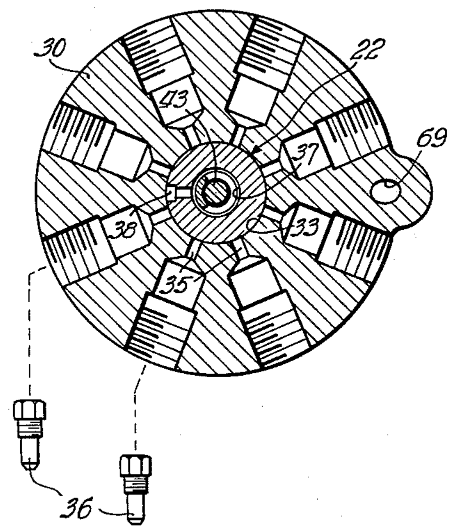
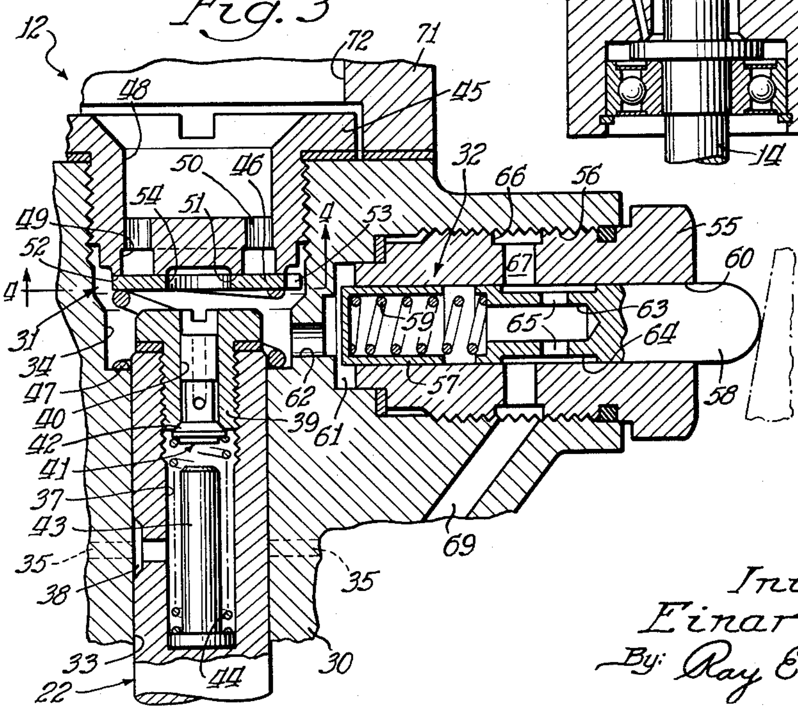


Fig. 3



Inventor:
Einar S. Dahl
By: Ray E. Snyder Atty.

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Fig. 5

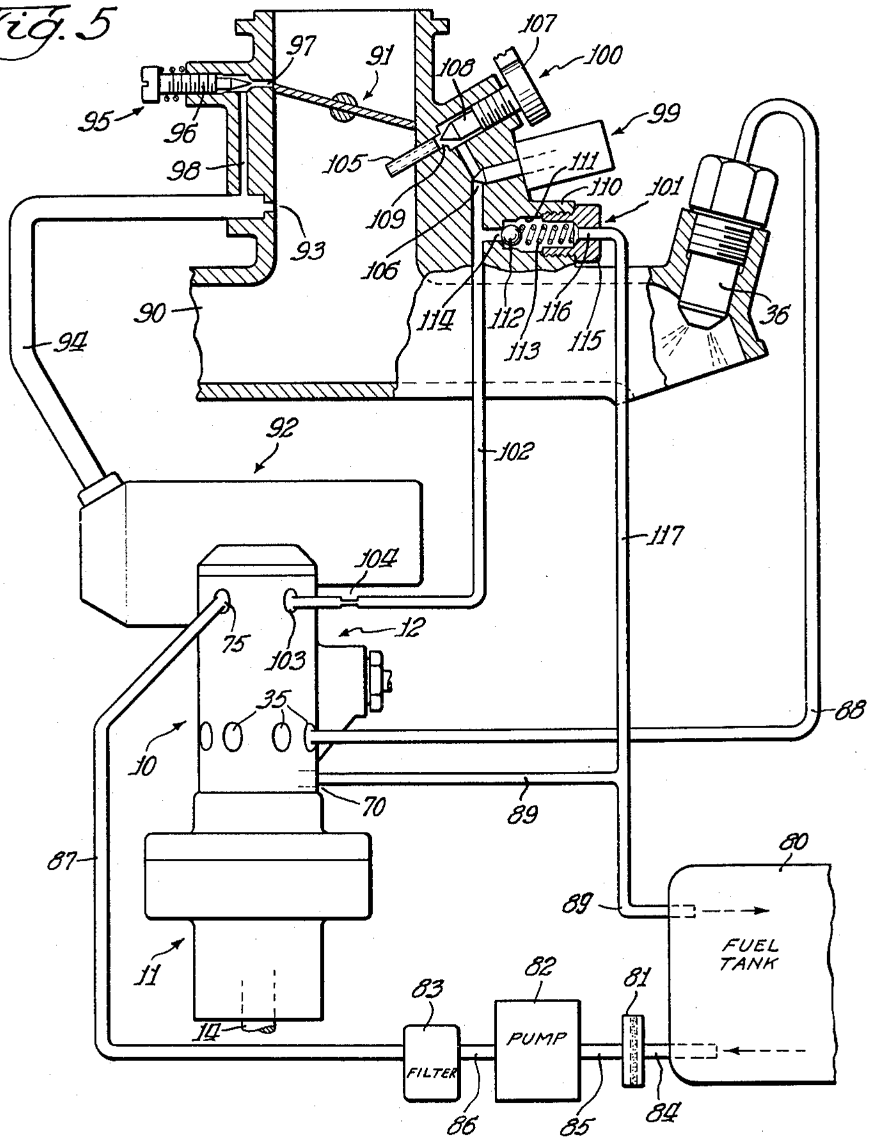
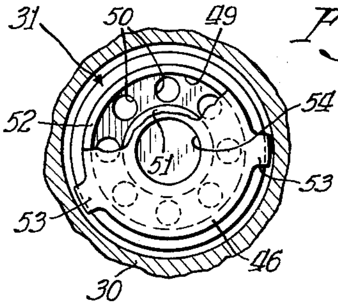


Fig. 4



Inventor:
Einar S. Dahl
By: Ray E. Snyder Atty.

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FUEL INJECTION PUMP

Einar S. Dahl, Decatur, Ill., assignor to Borg-Warner Corporation, Chicago, Ill., a corporation of Illinois

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5 Claims. (Cl. 103—2)

This invention relates to a fuel injection pump for supplying metered charges of fuel in timed sequence to the cylinders of an internal combustion engine.

It is an object of the present invention to provide an improved fuel injection pump employing a single plunger that is reciprocated for pumping fuel and is rotated for distributing the fuel and that is adapted to supply the correct amount of fuel to all of the cylinders for all conditions of load and speed of the engine.

It is an additional object to provide a pump in accordance with the preceding object which is designed to minimize leakage past the pumping plunger.

It is also an object to provide a single improved fill valve for the pump which is adapted to supply adequate amounts of fuel for all speeds of the operating range of the pump.

It is another object to provide an improved fuel injection pump incorporating improved metering means in the form of a spring loaded piston, the stroke of which is adjustable and which measures a portion of the fuel pumped during each stroke of the pumping plunger.

It is a more particular object to provide a rotating pumping plunger having an axial bore and a radial distribution slot in communication with said bore and a spring loaded check valve positioned within the bore and adapted to cooperate with the metering piston for metering each charge of fuel distributed through the distribution slot.

It is an additonal object to provide a fuel injection pump designed so as never to cut off a flowing column of fuel under pressure, thereby eliminating high pressure wear at the juncture of the cut-off ports.

It is another object to provide an improved fuel injection pump utilizing a relatively large fuel supply reservoir immediately adjacent the fill valve of the pump and means for maintaining a substantially constant pressure at the fill valve seat.

It is also an object to provide an auxiliary fuel circulating system for supplying additional starting fuel, for filling the fuel injection pump, and for purging air and vapor from the pump.

It is another object to provide a fuel injection pump having a rotating and reciprocating plunger, a reciprocating metering piston and improved leakage fuel return means for thereby salvaging the leakage fuel and eliminating the necessity for high pressure seals around the plunger and piston.

It is still another object to provide a fuel injection pump of a given predetermined capacity that is adapted to be used on engines within a certain size or power range and which pumps only the fuel required by the particular engine.

It is another object to provide an improved fuel injection pump having a single pumping plunger, a single inlet valve, a single metering piston, and a single delivery valve which cooperate to supply equal metered charges of fuel to a plurality of engine cylinders and which actually pumps only the fuel required to satisfy engine demand and to make up for leakage losses.

The invention consists of the novel constructions, arrangements, and devices to be hereinafter described and claimed for carrying out the above stated objects and such other objects as will appear from the following description of a preferred form of the invention illustrated with reference to the accompanying drawings wherein:

FIG. 1 is a longitudinal sectional view of the improved pump of the present invention;

FIG. 2 is a sectional view taken on line 2—2 of FIG. 1;

FIG. 3 is an enlarged sectional view of a portion of the pump shown in FIG. 1;

FIG. 4 is a fragmentary sectional view taken on line 4—4 of FIG. 3; and

FIG. 5 is a schematic illustration of the pump of FIG. 1 as it is used with an internal combustion engine.

Like characters of reference designate like parts in the several views.

Referring to FIG. 1, the improved pump of the present invention is designated generally by the numeral **10** and comprises a drive section **11** and a metering and distributing section **12**. The drive section **11** comprises a casing **13**, a drive shaft **14** journalled within the casing **13**, a cam **15**, and a plurality of rollers **16** which function as cam followers. The drive shaft **14** has a tang **17** formed on an end thereof which extends into a transverse slot **18** formed in the cam **15**. The rollers **16** are journalled on shafts **19** which are mounted in the casing **13** with their axes normal to the axis of rotation of the drive shaft **14**. The cam **15** is formed with an undulating cam surface **20** which is maintained in contact with the rollers **16**. A coupling pin **21** is mounted transversely in the cam **15** at right angles to the tang **17**. A pumping plunger **22** has a head **23** formed with a transverse slot **24** which engages the pin **21**. The head **23** is received within the cylindrical recess **25** formed in the cam **15**. A spring **26** and a spring retaining cap **27** surround the pumping plunger **22** and the cap **27** is fixedly attached thereto by means of a split ring washer **28** which fits into an annular groove **29** formed on the pumping plunger **22**. The spring **26** forces the retaining cap and head **23** of the plunger **22** into contact with the cam **15** which in turn forces the cam surface **20** into contact with the rollers **16**.

In operation, rotation of the drive shaft **14**, preferably at camshaft speed, causes the pumping plunger **22** to be driven at a 1:1 ratio therewith, and the undulating cam surface **20** in moving over the rollers **16** imparts a reciprocating motion to the plunger **22**. The cam surface **20** is provided with as many lobes as there are engine cylinders to be supplied. The coupling between the plunger **22** and drive shaft **14** by means of the cam **15** and particularly the tang **17** and the pin **21** corrects for any misalignment of the pumping plunger **22** with respect to the drive shaft **14**.

The metering and distributing section **12** comprises a casing **30**, the plunger **22**, a fill valve **31**, and a metering portion **32**. The casing **30** is fixedly attached to the casing **13**, and the pumping plunger **22** is disposed within a longitudinal cylindrical bore **33** formed in the casing **30**. The cylindrical bore **33** is enlarged at **34** to define a pumping or compression chamber adjacent the free end of the plunger **22**. The casing **30** is also formed with a plurality of radial fuel delivery ports **35**, each of which are connected to a fuel injection nozzle **36**.

The plunger **22** is formed with an axial central bore or fuel passage **37** and a radial distribution port or slot **38** communicating with the bore **37** and which is adapted to be aligned successively with the ports **35** as the plunger **22** rotates. A plug **39** is threaded into the open end of the axial bore **37**. The plug **39** is formed with a central bore **40** and a fuel delivery or discharge check valve **41** having a tapered head **42** tends to form a fluid tight seal on the inner end of the bore **40**. A pin **43** is mounted axially within the bore **37**, and a spring **44** surrounding the pin **43** acts against the valve **41** tending to maintain it in a fluid sealing position.

The fill valve **31** comprises a plug **45** threaded into the pumping chamber **34**, a ring shaped sealing member

46, and a spring 47. The plug 45 is formed with an enlarged central cavity 48, an annular groove 49, and a plurality of holes 50 connecting the annular groove 49 with the cavity 48. The annular groove 49 is bounded by concentric lapped surfaces 51 and 52 defining a valve seat formed on the inner end of the plug 45. The sealing ring 46 is adapted to form a fluid tight seal against the surfaces 51 and 52 closing the annular groove 49 from the pumping chamber 34. The ring 46 is formed with a plurality of radial spacing tabs 53 and a central hole 54. The spring 47 is disposed under compression within the pumping chamber 34 and forces the sealing ring 46 into contact with the surfaces 51 and 52.

The metering portion 32 of the pump 10 comprises a plug 55 threaded into a radial bore 56 formed in an extension of the casing 30, a movable piston or shuttle 57, an adjustable metering stop pin 58, and a spring 59. The piston 57 and pin 58 are slidably disposed in a cylindrical bore 60 formed in the plug 55. The spring 59 is compressed between the piston 57 and the pin 58 and tends to force the piston 57 to the limit of its motion toward the left. The inner end of the bore 56 defines a metering cavity or chamber 61 which is in communication with the pumping chamber 34 through a port 62. The stop pin 58 is formed with an internal central bore 63, an annular groove 64, and radial passages 65 connecting the central bore 63 with the annular groove 64. The grooves and passages collectively constitute a leakage fuel return means for the pin 58. The plug 55 is also formed with an annular groove 66 and radial passages 67 connecting the annular groove 64 with the groove 66.

The plunger 22 is also formed with an annular leakage fuel return groove 68. The groove 68 is connected with the annular groove 66 through a channel 69 formed in the casing 30, and both grooves are connected through a portion 70 to a fuel return line which carries the leakage fuel back to the fuel supply tank.

The pump 10 also includes a cylindrical casing portion 71 formed with a relatively large cylindrical cavity 72 which defines a fuel supply reservoir. A flexible diaphragm 73 is provided on top of the casing 71 and is held in place by an end cap 74. The reservoir 72 is in communication with the bore 48 of the plug 45 and is connected through a fuel inlet port 75 to a primary fuel supply pump. The primary fuel supply pump supplies fuel to the reservoir 72 under pressure of, for example, 40 p.s.i. and this pressure is maintained substantially constant by means of the diaphragm 73, as will be described hereinafter.

In operation, the pump 10 functions as follows:

The drive shaft 14 is driven at a fixed speed ratio with respect to the crank shaft of the engine to which fuel is to be supplied and as the drive shaft 14 rotates, the cam 15 is driven and it moves downward for one portion of its cycle of operation. The spring 26 forces the plunger 22 downward, thereby reducing the pressure within the pumping chamber 34 for the fill stroke of the plunger 22. Fuel under pressure within the reservoir 72 forces the ring 46 off its seat and fuel is allowed to enter the pumping chamber 34 from the annular groove 49. As the drive shaft 14 rotates further, the cam 15 forces the plunger 22 upward, closing the valve 31 and compressing the fuel within the chamber 34. A portion of the fuel within the chamber 34 is forced through the port 62 into the cavity 61 where it acts upon the piston 57 forcing it to the right against the action of the spring 59. When the piston 57 reaches the limit of its motion against the stop 58, the remainder of the fuel within the pumping chamber 34 acts against the valve 41, forcing it off its seat and the fuel enters the bore 37. The fuel distribution slot 38 is in register with one of the ports 35 during the entire compression stroke of the plunger 22 and when the valve 41 opens, fuel within the passage 37 is forced out through the slot 38 and the port 35 to a respective nozzle 36. As the plunger 22 rotates still further, communication with the particular port 35 is cut off and the plunger 22 moves downward again and the cycle is repeated for the next port 35.

The quantity of fuel displaced into the metering cavity 61 is determined by the position of the metering pin 58 which is adjustable in accordance with the engine demand. It is important that the spring 44 tending to maintain the valve 41 in its fluid sealing position be stronger than the spring 59 which acts on the metering piston 57. With this provision, the valve 41 remains seated until the piston 57 has been forced to the limit of its motion to the right. The pumping plunger 22 has a constant displacement for each stroke, and the portion of the fuel compressed within the pumping chamber 34 that is not displaced into the cavity 61 is delivered to a nozzle 36. Since the operation of the fill valve 31, the check valve 41 and the metering piston 57 is the same for each pumping stroke, for a constant setting of the metering pin 58, the charges of fuel delivered to all of the nozzles 36 should be identical.

It should be noted that during the fill stroke of the plunger 22, the filling of the pumping chamber 34 is facilitated by the return of fuel from the metering chamber 61 by the spring loaded piston 57. It is necessary, therefore, only to draw in sufficient fuel through the fill valve 31 to supply the amount delivered to the engine cylinder and to make up for leakage losses past the metering piston 57 and the plunger 22. In this regard, the pump of the present invention should be compared with constant displacement pumps of other designs wherein a portion of the fuel is spilled during each pumping stroke.

In order to obtain accurate metering, another detail of some importance is the fact that the port 62 connecting the metering cavity 61 with the pumping chamber 34 is preferably located at the lowermost part of the pumping chamber 34. This location is provided to minimize the amount of new fuel entering the metering cavity 61. It has been found that fuel having air bubbles mixed therein does not have the same pumping characteristics as fuel from which the air bubbles have been removed. The location of the port 62 enhances the accuracy of the metering function because the fuel used for metering purposes is recycled and only a minimum amount of new fuel is allowed to enter the metering cavity 61.

It is necessary to remove the leakage fuel from behind a metering piston 57 in order to prevent an accumulation of fuel there that would interfere with the normal reciprocation of the piston 57. The pressure within the leakage release conduit 69 should preferably be quite small, that is, only sufficient to return the leakage fuel to the supply tank. A very low pressure in the leakage conduit 69 eliminates the necessity for high pressure seals around the lower end of the pumping plunger 22 and around the stop pin 58, which seals would interfere with the normal reciprocation of these members. The removal of leakage fuel also prevents dilution of lubricating oil within the drive section 11.

The pressure developed within the pumping chamber 34 establishes a pressure gradient along the length of the plunger 22 between the chamber 34 and the annular groove 68, and also along the length of the shuttle piston 57. This pressure gradient causes a small amount of leakage fuel to drift through the bores 33 and 60 to the fuel release port 70. A certain amount of leakage fuel is necessary for lubrication of the plunger 22 and piston 57. It should be noted, however, that the leakage past the pumping plunger 22 has been minimized by the fact that only a single opening 38 in the plunger 22 is provided. In this regard, the pump of the present invention should be compared with similar pumps having a plurality of fuel inlet and outlet or spill ports in the pumping plunger and wherein leakage becomes a problem of considerable importance.

Fuel is supplied to the pump 10 from a primary fuel supply pump at a pressure of approximately 40 p.s.i. This fuel is supplied into the reservoir 72 which is comparatively large. This primary supply pressure also tends to flex or distort the diaphragm **73**. It had been found previously that rapid opening of the fill valve tended to reduce the pressure instantaneously in the fuel supply line. This sudden reduction in pressure inhibited proper filling of the pumping cavity **34**. To overcome this problem, the large volume within the cavity **72** is provided and the flexible diaphragm **73** functions to maintain the pressure within the cavity **72** substantially constant when the valve **31** opens.

It should be noted also that at high speed of operation, for example, with an eight cylinder engine operating at 5,000 r.p.m., the fill valve **31** is open for only $\frac{1}{40,000}$ of a minute. At this speed or a comparable speed, it is necessary that sufficient fuel be admitted to fill the pumping cavity **34** as rapidly as possible. Merely increasing the size of the fill valve so as to increase its flow capacity is not a logical solution because this also increases its mass. To solve this problem, the configuration of the fill valve **31** is provided which allows fuel to flow from the annular groove **49** through the central hole **54** in the valve ring **46** and also around its outer periphery. As compared to a solid flat disc or poppet-type valve of comparable size, this configuration of the valve ring **46** provides a substantially greater flow capacity to mass ratio.

For optimum operation of the pump **10**, it has been found that the valve **41** should have an opening pressure of approximately 150 p.s.i. and the piston **57** should respond to a displacement pressure of approximately 130 p.s.i. The nozzles **36** used with the pump **10** should be of the spring loaded valve type and preferably should have an opening pressure of approximately 200 p.s.i. The size of the springs **44** and **59** provided to establish these operating pressures is dependent upon the size of the valve **41** and of the piston **57**, respectively. It is important that the opening pressure of the valve **41** and of the nozzles **36** exceed the displacement pressure of the piston **57**.

The dstribution slot **38** has a width of approximately $\frac{1}{16}$ inch and is in communication with one of the ports **35** from the beginning of the compression stroke until approximately 6°–8° of plunger rotation past the end of the compression stroke. The flow of fuel under pressure through a port **135**, therefore, is never cut off by the slot **38**. This feature obviates the possibility of high pressure wear at the edges of the ports **35** and slot **38**. It should be understood that the structure and the operating pressures described are by way of example only and are not intended to limit the scope of the present invention.

Referring now to FIG. 5, there is illustrated a schematic diagram of the fuel injection pump **10** as used with an internal combustion engine. The complete fuel supply system for the pump **10** comprises a fuel tank **80**, a primary filter **81**, a primary fuel supply pump **82**, a secondary filter **83**, and the fuel injection pump **10**. The primary filter **81** is connected to the fuel tank **80** through a conduit **84**; the supply pump **82** is connected to the outlet of the filter **81** through a conduit **85**; the secondary filter **83** is connected to the supply pump **82** through a conduit **86**, and the port **75** of the pump **10** is connected to the outlet of the filter **83** through a conduit **87**. The fuel delivery ports **35** are connected to the respective nozzles **36** through nozzle lines **88**, and the port **70** is connected to a fuel return conduit **89** which leads back to the fuel tank **80**.

The fuel injection system also includes an air intake manifold **90** for the engine and a butterfly valve **91** positioned in the throat of the air intake manifold. The nozzles **36** are mounted in the manifold **90** at points adjacent each of the intake valves of the engine cylinders. A control mechanism **92** is mounted on the pump **10** and is adapted to control the position of the metering pin **58** in accordance with engine demand. For this purpose, manifold vacuum is utilized and is supplied to the

mechanism **92** through a restricted orifice **93** and conduit **94**. The orifice **93** is located in the manifold wall at a point below the throttle valve **91**. The manifold vacuum supplied to the control mechanism **92** is modified somewhat by means of an idle adjustment valve **95**. The valve **95** includes a spring loaded needle valve **96** and an orifice **97** which is adapted to be closed by the needle valve **96**. The orifice **97** is located in the manifold wall adjacent one edge of the blade of the throttle valve **91** and is connected through a channel **98** to the conduit **94**. The needle valve **96** is adjustable for controlling the amount of air bleed past the throttle valve **91** into the conduit **94**. The control mechanism **92** may be of a type capable of transforming this manifold vacuum into a mechanical motion for adjusting the position of the metering pin **58**.

An auxiliary fuel circulating system for the pump **10** is provided and which comprises a solenoid operated valve **99**, a temperature responsive valve **100**, and a pressure regulator valve **101**. The valves **99**, **100**, and **101** are connected by means of a conduit **102** to a fuel outlet port **103** formed in the casing portion **71**. A restriction **104** of approximately .025 inch is provided in the conduit **102** for limiting the quantity of fuel flow therethrough. The conduit **102** terminates at a jet **105** which opens into the air intake manifold **90** at a point below the throttle valve **91**.

The solenoid valve **99** is connected to the electrical starting circuit of the engine, and normally blocks the conduit **102** at a port **106** except when the solenoid is energized.

The temperature responsive valve **100** comprises a bimetallic element **107** and a needle valve **108** which is adapted to restrict a port **109** leading to the jet **105**. The valve **100** is responsive to ambient temperautre conditions and controls the amount of fuel flowing through the port **109** and jet **105** while the engine is being started. The pressure regulator valve **101** comprises a casing portion **110** formed with a cylindrical cavity **111**, a ball **112**, and a spring **113**. The casing portion **110** is formed with a fuel inlet port **114** which is connected to the conduit **102**. A spring retaining plug **115** is threaded within the cavity **111** and is formed with a fuel outlet port **116**. The fuel outlet port **116** is connected through a conduit **117** to the fuel return line **89** which leads back to the tank **80**. The spring **113** acts against the ball **112** which tends to block the port **114** and functions to regulate the pressure within the conduit **102** at a predetermined value.

In operation, the primary fuel supply pump **82**, which is preferably electrically driven, draws fuel from the tank **80** and transports it through the filters and the conduit **87** to the fuel injection pump **10**. The primary supply pump **82** establishes a pressure of approximately 40 p.s.i. within the supply reservoir **72**. A portion of the fuel within the supply reservoir **72** is allowed to pass through the restricted orifice **104** into the conduit **102**. The pressure within the conduit **102** is regulated by means of the valve **101** at a desired predetermined pressure of approximately 10 to 25 p.s.i. The fuel released by the regulator valve **101** passes through the port **116** and conduit **117** and **89** back to the fuel supply tank **80**.

The fuel under pressure within the conduit **102** is available for supplying additional fuel to the engine for starting it. The solenoid operated valve **99** is energized by the starting circuit of the engine which opens the port **106** and allows fuel to pass through the port **109** and jet **105** into the air intake manifold **90**. The quantity of fuel passing through the port **109** and jet **105** is controlled by the valve **100**. In cold weather, when additional fuel enrichment is desired, the valve **100** allows a greater quantity of starting fuel to pass through the jet **105** than in warm weather.

The fuel circulated through the conduit **102**, the valve **101**, and the conduit **117** serves the additional functions of initially filling the reservoir 72 of the fuel injection pump 10 and also of purging the reservoir 72 of any accumulated air or vapor therein. The restriction 104 in the conduit 102 limits the quantity of fuel flowing therethrough and therefore does not substantially increase the load on the primary fuel supply pump 82.

It is to be understood that this invention is not to be limited to the specific constructions and arrangements shown and described except only insofar as the appended claims may be so limited, as it will be apparent to those skilled in the art that changes may be made without departing from the principles of the invention.

I claim:

1. In a fuel injection pump for an internal combustion engine, the combination of a casing formed with a longitudinal bore and a plurality of fuel delivery ports opening into said bore, a combined pumping and distributing plunger disposed within said bore, fill valve means for supplying fuel to said bore, means for reciprocating said plunger to compress fuel within said bore and rotating said plunger to distribute fuel through said ports, metering means comprising a spring loaded piston in direct communication with said bore and adapted to be displaced a limited amount by fluid pressure developed in said bore with each stroke of said plunger and including control means for limiting the stroke of said piston in accordance with engine demand, whereby metered charges of fuel are formed and distributed by said plunger through each of said ports.

2. In a fuel injection pump for supplying charges of fuel to a plurality of engine cylinders, the combination of a casing formed with a longitudinal cylindrical bore and a plurality of spaced radial fuel delivery ports in communication with said bore, said bore defining a pumping chamber in one end thereof; fill valve means for supplying fuel to said chamber; a plunger rotatively and reciprocatively disposed within said bore and formed with an axial central fuel passage open to said pumping chamber and a radial fuel delivery port for connecting said passage in succession to said casing ports as said plunger rotates; a single delivery valve disposed in said passage; a single metering piston movable between two limits and directly connected to said pumping chamber and adapted to be displaced by fluid pressure developed within said chamber and with the stroke of said piston being limited in accordance with engine demand; and drive means for reciprocating said plunger to pump fuel and rotating said plunger to distribute charges of fuel; said valve being effective during each stroke of said plunger to prevent the release of fuel from said pumping chamber until said piston has been displaced to one of its limits of motion, whereby metered charges of fuel are formed and delivered by said plunger through said casing ports to the engine cylinders.

3. In a fuel injection pump for supplying metered charges of fuel to a plurality of engine cylinders, the combination of a casing formed with a longitudinal cylindrical bore and a plurality of spaced radial fuel delivery ports in communication with said bore, a pumping plunger rotatively and reciprocatively disposed within said bore and formed with an axial central fuel passage and a radial fuel delivery port for connecting said passage to said casing ports, means defining a fuel supply reservoir for supplying fuel under pressure to said bore; means defining a fill valve including a ring-shaped valve member separating said reservoir from said bore; metering means including a spring loaded piston in communication with said bore and adapted to be displaced to an adjustable limit by fluid pressure developed within said bore and including control means for adjusting said limit in accordance with engine demand; and a single spring-loaded check valve positioned in said passage, said check valve being effective during each compression stroke of said plunger to prevent the release of fuel through said radial port until said metering piston has been displaced to its limit of motion, whereby metered charges of fuel are formed and delivered by said plunger through said casing ports to the engine cylinders.

4. In a fuel injection system the combination comprising:

(1) a pump housing formed with a fuel inlet and a plurality of fuel outlet means;
(2) means defining first and second bore means in said housing;
(3) means defining a pumping chamber formed in said housing at one end of said first bore means;
(4) a fuel pumping and distribution plunger disposed in said first bore means;
(5) means for reciprocating said plunger in said first bore for pumping fuel;
(6) means in said plunger including valve means providing fluid communication between said outlet and inlet means;
(7) means for rotating said plunger for distributing said fuel to said plurality of outlet means;
(8) means in said housing defining a fuel supply reservoir for supplying fuel under pressure to said pumping chamber;
(9) means defining a fill valve disposed between said reservoir and said pumping chamber;
(10) means including a flexible diaphragm for maintaining a substantially constant pressure within said reservoir when said fill valve opens;
(11) fuel metering means in fluid communication with said pumping chamber for metering the amount of fuel pumped to said outlet means, said fuel metering means comprising a piston disposed in said second bore in direct communication with said pumping chamber and adapted to be displaced by fluid pressure developed within said pumping chamber, adjustable stop pin means adapted to be moved axially in said second bore in response to a manifold vacuum actuated control control mechanism, and biasing means disposed between said piston means and adjustable stop pin means for limiting the movement of said piston means to thereby control the volume of fuel distributed to said outlet means.

5. In a fuel injection system the combination comprising:

(1) a pump housing formed with a fuel inlet and a plurality of fuel outlet means;
(2) means defining first and second bore means in said housing;
(3) means defining a pumping chamber formed in said housing at one end of said first bore means;
(4) a fuel pumping and distribution plunger disposed in said first bore means;
(5) means for reciprocating said plunger in said first bore for pumping fuel;
(6) means in said plunger including valve means providing fluid communication between said inlet and outlet means;
(7) means for rotating said plunger for distributing said fuel to said plurality of outlet means;
(8) means in said housing defining a fuel supply reservoir for supplying fuel under pressure to said pumping chamber;
(9) means defining a fill valve disposed between said reservoir and said pumping chamber;
(10) means including a flexible diaphragm for maintaining a substantially constant pressure within said reservoir when said fill valve opens;
(11) and fuel metering means in fluid communication with said pumping chamber for metering the amount of fuel pumped to said outlet means, said fuel metering means comprising a piston disposed in said second bore in direct communication with said pumping chamber and adapted to be displaced by fluid pressure developed within said pumping chamber, adjustable stop pin means adapted to be moved axially in said second bore in response to a manifold

vacuum activated control mechanism, and biasing means disposed between said piston means and adjustable stop pin means for limiting the movement of said piston means to thereby control the volume of fuel distributed to said outlet means and fuel leakage passage means in communication with said second bore to prevent a build-up of pressure behind said piston.

References Cited in the file of this patent

UNITED STATES PATENTS

204,747	Maxim	June 11, 1878
1,076,915	Setz	Oct. 28, 1913
1,116,494	Setz	Nov. 10, 1914
1,135,476	Watts	Apr. 13, 1915
1,799,704	Riley	Apr. 7, 1931
1,993,369	Goldberg	Mar. 5, 1935
2,055,578	Hurst	Sept. 29, 1936
2,102,117	Goldberg	Dec. 14, 1937
2,237,347	Grannan	Apr. 8, 1941
2,267,479	Sturm	Dec. 23, 1941
2,297,942	Collins	Oct. 6, 1942
2,301,435	Mercier	Nov. 10, 1942
2,318,128	Tabb	May 4, 1943
2,543,828	Brown	Mar. 6, 1951
2,593,316	Kraft	Apr. 15, 1952
2,603,159	Johnson	July 15, 1952
2,619,907	Paterson	Dec. 2, 1952
2,711,697	Gibbs	June 28, 1955
2,728,351	Cooper	Dec. 27, 1955
2,739,643	Voit	Mar. 27, 1956
2,780,173	Herbrich	Feb. 5, 1957
2,794,397	Burman	June 4, 1957
2,810,375	Froelich	Oct. 22, 1957
2,810,376	Aldinger	Oct. 22, 1957
2,833,218	Evans	May 6, 1958
2,839,999	Shallenberg	June 24, 1958
2,846,995	Foltz	Aug. 12, 1958
2,851,026	Dahl et al.	Sept. 9, 1958
2,918,008	Thompson	Dec. 22, 1959
2,922,369	Nystrom	Jan. 26, 1960
2,928,352	Aldinger	Mar. 15, 1960
3,000,369	Bischoff	Sept. 19, 1961

FOREIGN PATENTS

28,191	Great Britain	June 6, 1912