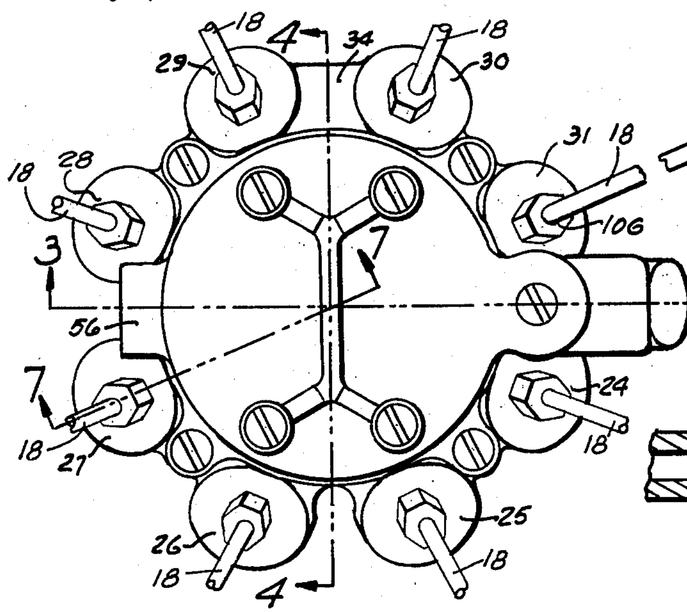
FUEL INJECTION

Filed May 1, 1957 6 Sheets-Sheet 1



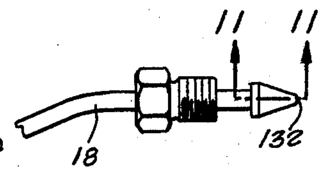


FIG. 1

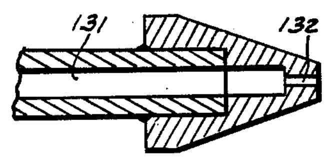


FIG. 11

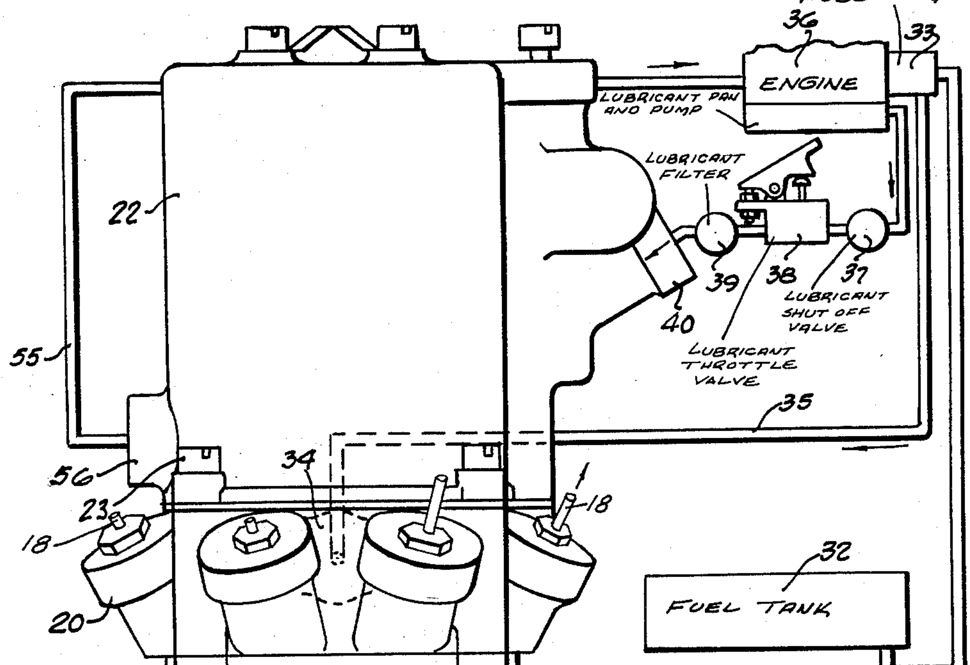


FIG. 2

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FUEL INJECTION

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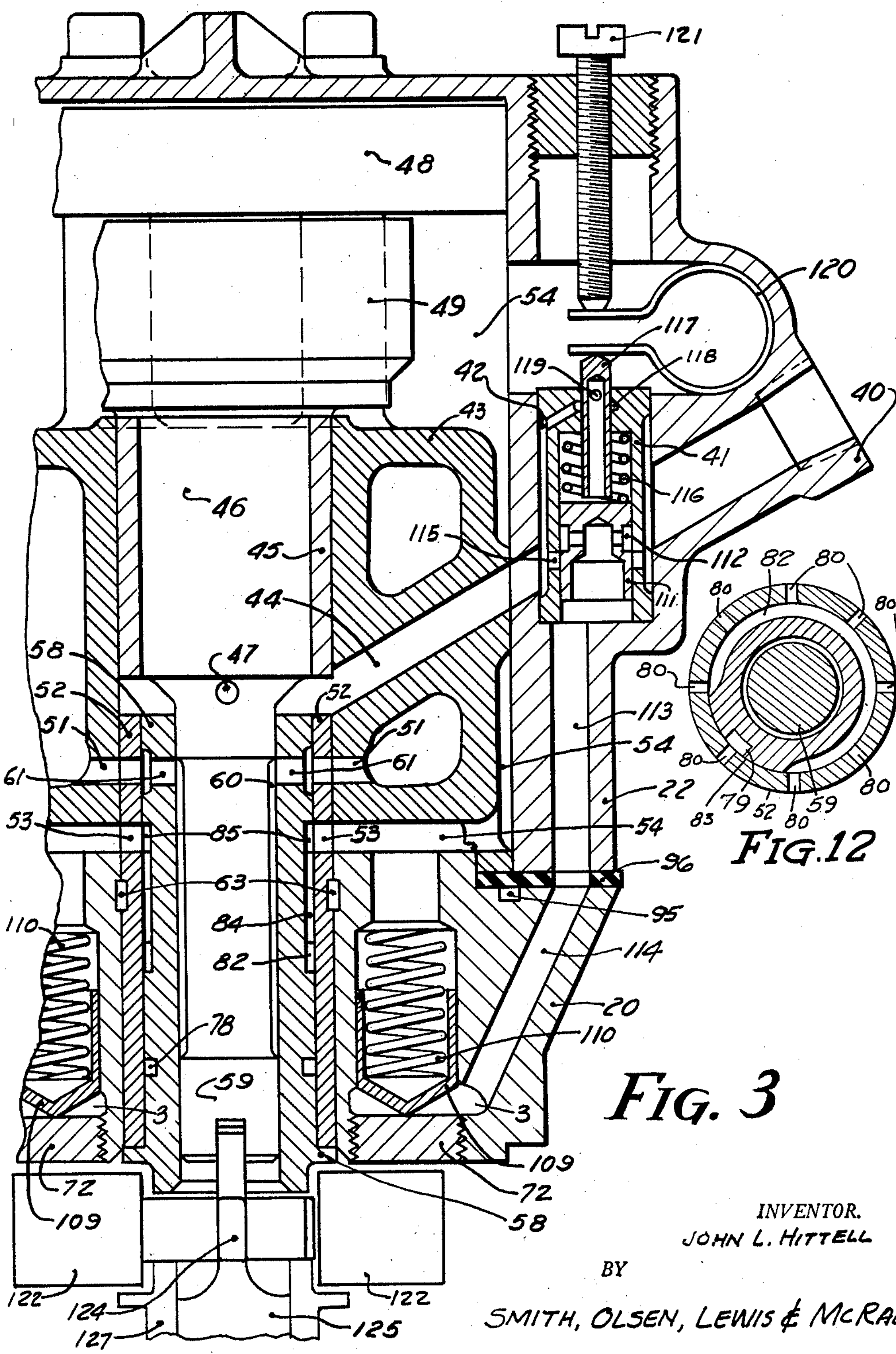

FIG. 3

FIG. 12

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FUEL INJECTION

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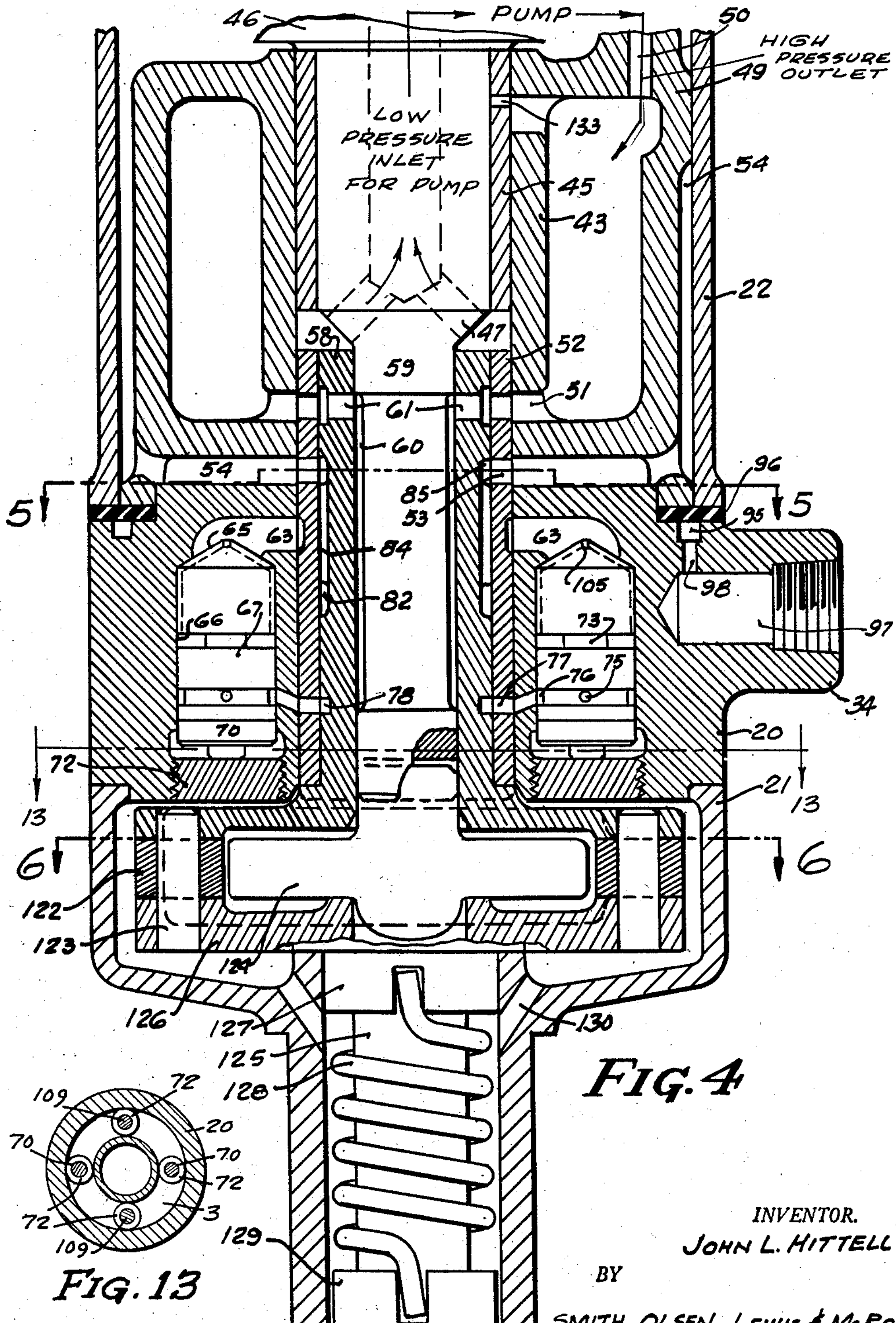

FIG. 4

FIG. 13

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FUEL INJECTION

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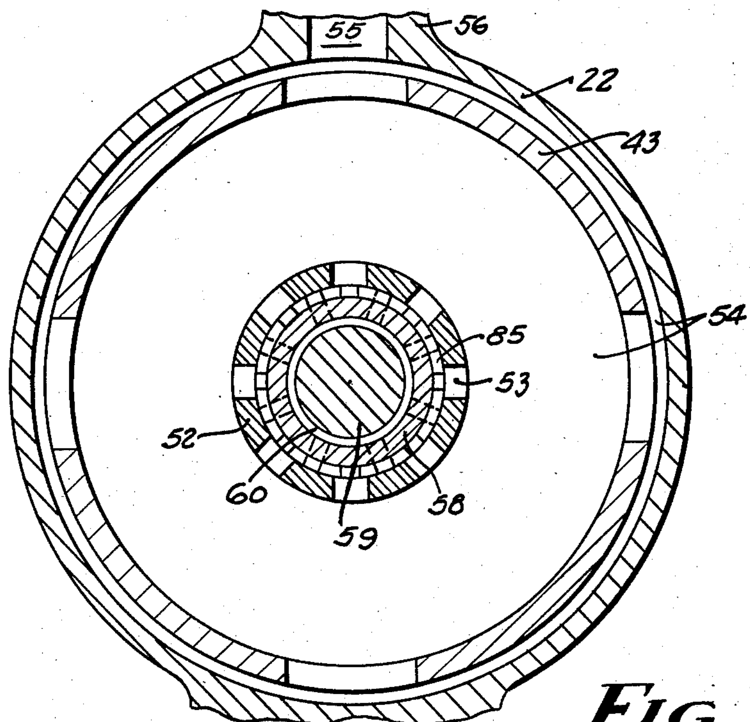

FIG. 5

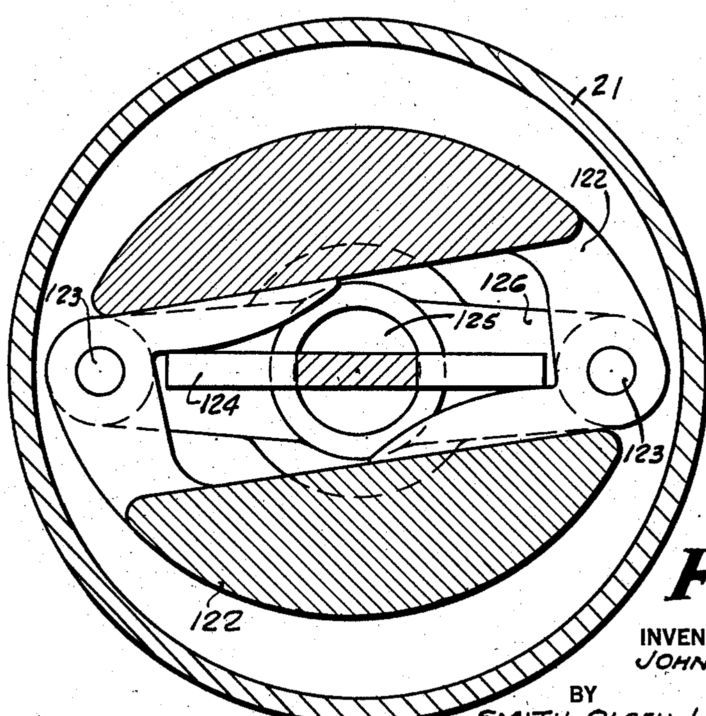

FIG. 6

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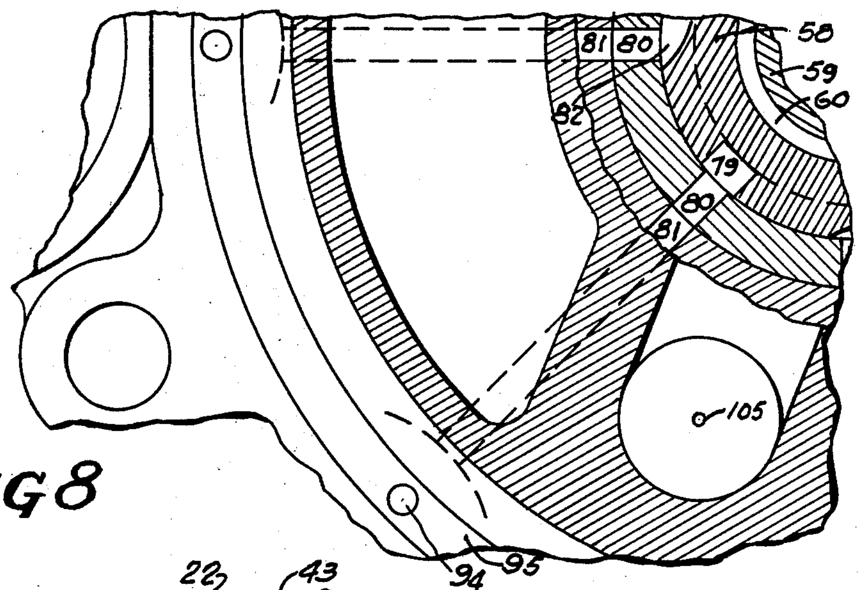

FIG 8

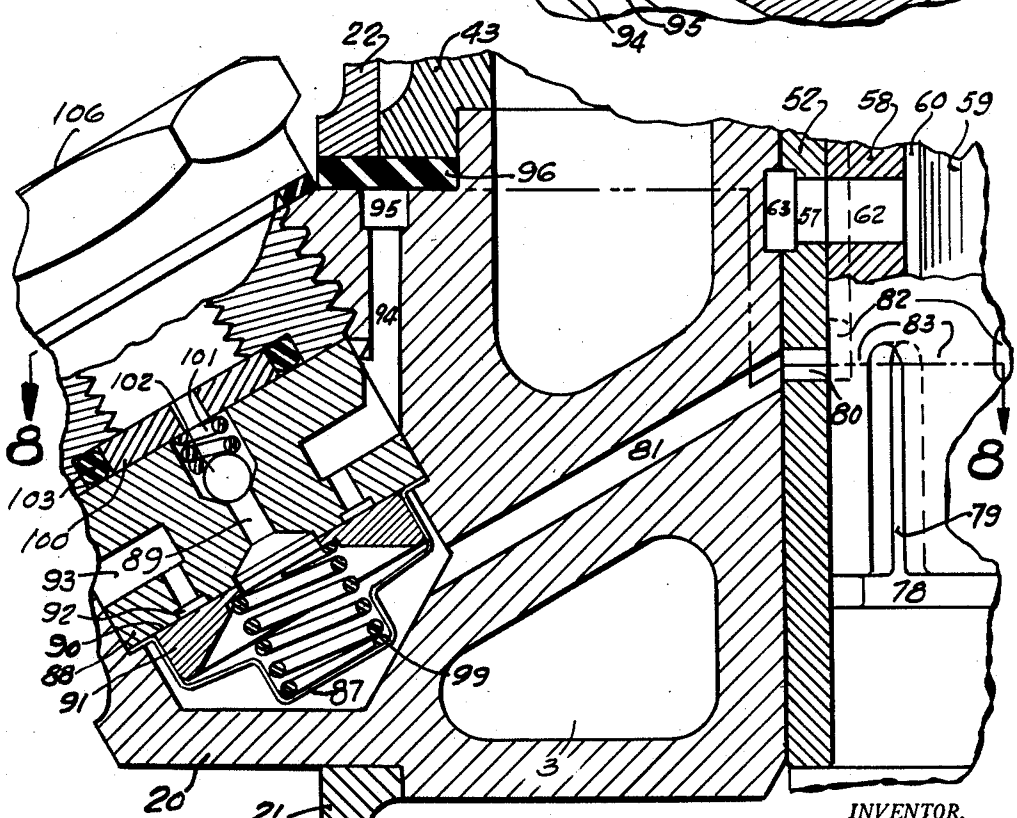

FIG. 7

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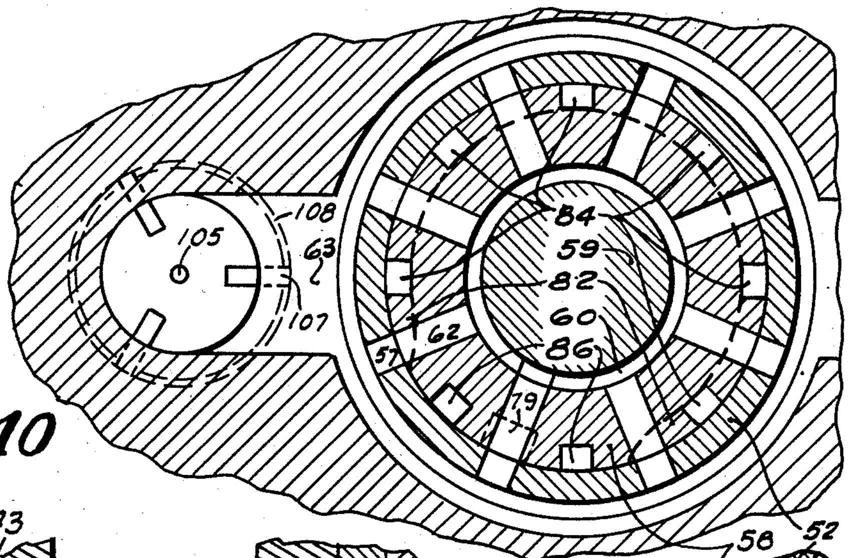

FIG. 10

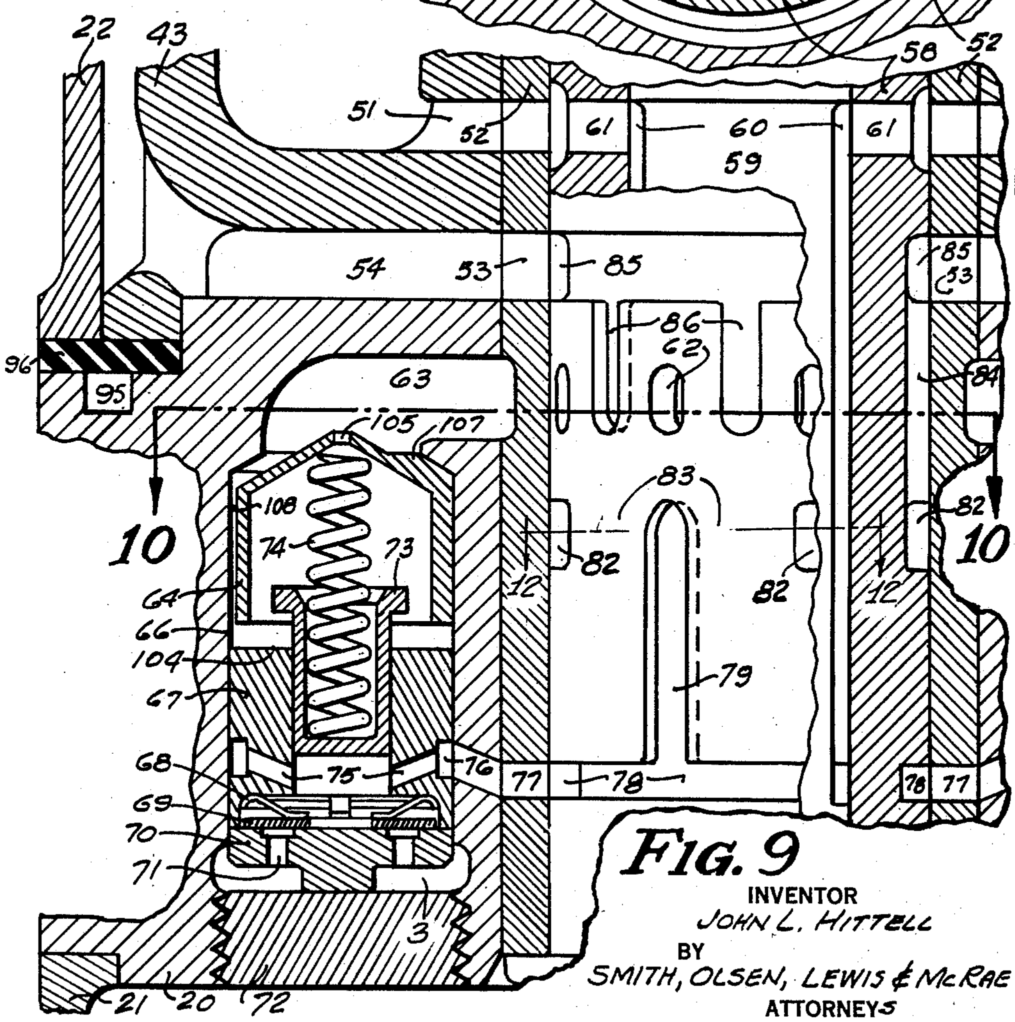

FIG. 9

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United States Patent Office

2,914,053
Patented Nov. 24, 1959

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FUEL INJECTION

John L. Hittell, Livonia, Mich.

Application May 1, 1957, Serial No. 656,244

15 Claims. (Cl. 123—139)

This invention concerns means which in their broadest sense constitute improved method and apparatus for metering and high pressure feeding of any low viscosity liquid and includes means for metering and feeding such a liquid in precisely repetitive quantities overcoming the usual leakage of metered quantity ordinarily accompanying high pressure feeding of low viscosity liquids.

Since such apparatus has widespread utility as means for metering and feeding gasoline to an internal combustion engine operating on the compression ignition principle, this type of use is considered as most suitable for illustration of a preferred embodiment of this invention, which is in many particulars an improved version of the device disclosed and claimed in applicant's Patent Re. 24,214.

A principal and general object of this invention is to achieve most of the objects and advantages of the patented device in new constructions, arrangements and revised operating principles whereby greatly simplified apparatus is used to achieve these results in a device rearranged to make it readily adaptable to engines of the in-line and widely popular V-8 types.

A further object is to provide such a system in an injection unit of the distributor type, attaining the above objects in a package fuel metering distributor which may be substituted for the usual spark ignition distributor, facilitating conversion of engines developed for spark ignition into compression ignition gasoline injection engines by simple change of distributor, substitution of nozzles for spark plugs, and by installation of pistons modified to secure adequate compression ratio and a suitable cavity for turbulent admixture of the finely atomized fuel charge.

An important object is to provide apparatus able to deliver additional fuel above that delivered by the carburetor type system when, and only when, peak power is required by the operator, thus permitting a substantial increase in peak engine power, particularly if the inlet manifold cross-sections are suitably increased, yet gaining this added peak power with reduced rather than increased fuel consumption under usual road travel conditions, by then utilizing the leaner mixtures which are well known to produce unparalleled efficiency at part load in compression ignition engines.

A further object is to provide fuel metering and feeding apparatus well suited to the use of inexpensive plain orifice injection nozzles, by always forcing the fuel at a controlled substantially constant pressure, whereby the flow rate through such nozzles is at a suitable constant velocity throughout the duration of any injection, regardless of changes in metered quantity of charge or in engine operating speed. This attains a further objective of always maintaining consistently fine atomization of the fuel injected, in sharp contrast to the usual result when fixed orifices are used with apparatus wherein injection

duration or time is set to equal a certain number of degrees rotation, resulting in very fast injection requiring harmful and needless excess injection pressure at high speeds, and causing slow injection, excessively low injection pressure and extremely poor atomization during operation at low speeds and in starting the engine.

Attainment of very fine atomoization at all engine speeds, particularly at cranking speeds, achieves a further objective of ability to utilize fuels of lower ignition quality and also provides much improvement in cold starting performance.

The present invention has also an important object to provide means for utilizing the same metering device to meter the charges to all cylinders, thereby obviating the need for providing a metering device for each cylinder and also the need for matching these devices so that the charge fed to each cylinder is alike under all conditions.

Another object is to provide means for metering the volumetric amplitude of oil pulses, and for utilizing these metered oil pulses to displace equal fuel charges by completely isolated hydraulic displacement. This permits handling a more viscous fluid having good lubricating properties, in the relatively moving parts of the metering and distributor units and distributing these metered oil pulses to oil-to-fuel pulse transfer units for each cylinder of the engine, with these pulses being transferred to fuel pulses of exactly equal amplitude at a controlled constant pressure maintained throughout the injection. Thereby the fuel charges are made alike to each cylinder in quantity, timing, atomization, and duration for any one operating condition.

Another object is to provide means for setting up and consistently maintaining a balance between the compressibility flow rate occurring behind a pressure wave being propagated in the injector line and the normal injection pressure flow rate of the fixed orifice nozzle, such that reflected pressure waves in the injection lines are almost completely eliminated by the balance achieved. This makes possible the maintenance of virtually constant velocity in the nozzle orifice throughout any duration of injection.

A further important object is to provide control means for governing the metered quantity of fuel per charge such that the maximum fuel charge per cylinder is limited to the fuel that can be properly burned in a cylinder filled with air, and to provide temperature responsive means for modifying this maximum to compensate for varying weight of air as the air and engine temperature varies.

A further object is to provide throttling control means that primarily act or tend to govern the fuel flow per unit of time, thereby providing increase of charge per cylinder as speeds lower and decrease of charge per cylinder as engine speed is increased, so tending to provide stable speeds and good lugging power at all throttled control settings and acting to provide a stable and even running when throttled to "idle" condition.

A further object is to provide timing advance governor control applied to both the timer and the distributor of the injection pulses.

An additional object is to so proportion, position and orient the various components as to provide working fluid spaces such as to be quickly and automatically self-purged of any air or vapors which may have accumulated during periods of non-operation.

Other objects of this invention will appear in the following description and appended claims, reference being had to the accompanying drawings forming a part of this specification wherein like reference characters designate corresponding parts in the several views.

The invention may be fully understood by the following description with reference to the drawings in which:

Fig. 1 shows an exterior plan view of a complete distributor unit;

Fig. 2 shows a side view of the exterior of the distributor unit and also shows in schematic form the connected apparatus for feeding oil and fuel to this unit and for throttling the feed of oil to the metering unit;

Fig. 3 is a partial vertical section along the line **3—3** of Fig. 1;

Fig. 4 is a partial vertical section along the line **4—4** of Fig. 1;

Fig. 5 shows a partial horizontal section along line **5—5** of Fig. 4;

Fig. 6 show a horizontal section along line **6—6** of Fig. 4;

Fig. 7 is an enlarged vertical partial section along line **7—7** of Fig. 1;

Fig. 8 is an enlarged partial horizontal section along line **8—8** of Fig. 7;

Fig. 9 is an enlarged partial section along line **4—4** of Fig. 1, showing some additional detail of a portion of Fig. 4;

Fig. 10 is a partial horizontal section along line **10—10** of Fig. 9, drawn to the same scale as Fig. 9.

Fig. 11 is a sectional view taken on line **11—11** in Fig. 1 and showing to an enlarged scale a typical single orifice plain nozzle which may be used in practice of the invention.

Fig. 12 is a sectional view on line **12—12** in Fig. 9;

Fig. 13 is a sectional view on line **13—13** of Fig. 4 drawn to a reduced scale.

Referring to Figs. 1 and 2 a distributor body **20** has fastened below it a governor housing **21** which includes an integral boss extending downward which is suited to mounting of the distributor unit in the well known manner commonly used on spark distributors. A cover **22** forms another portion of the housing of the distributor unit, fastened by screws **23** through the cover and the body **20** and into the housing **21**. The body **20** has eight tapped angular bosses, **24** to **31** inclusive, seen equally spaced about a central point of Fig. 1. These may be connected by any known or desired type of fittings to tubes **18** leading to the injectors located in cylinders of an eight cylinder engine, for which the shown embodiment of this invention is arranged. This embodiment is also intended for clockwise distributor shaft rotation as viewed in Fig. 1, and in this case, the injections follow in a clockwise direction around from **24** to **31**, and the connected injector lines to each cylinder of the engine will be connected to the cylinders to fire them in the required firing order, in a manner well known to those skilled in the art. A fuel tank represented at **32** supplies fuel drawn by a low pressure pump **33**, and fed to a boss **34** on the body **20**, through a line **35**. The pump **33** is mounted on an engine, diagrammatically shown at **36**, and may suitably be a pump operating on the principles illustrated in Fig. 8 of applicant's reissued patent, in which case the pump is powered by oil pulses delivered through an internal pulse passage from a suitable valve such as shown in said patent.

Any fuel pump may however be substituted for use at **33** provided it will deliver adequate pressure and is prevented from delivering excess pressure when little fuel is being used. It is desirable also that this pump supply fuel at a steadily maintained pressure, free from ripples and surges.

A duct line supplies oil directly from the engine lubricant pump to a shut off valve **37** and then to a throttle valve **38**, and a filter **39** which is connected to supply filtered oil to a boss **40** of the cover **22**.

Referring now to Fig. 3, a valve sleeve **41** has an annular space **42** surrounding it which allows free passage of oil around it. A pressure reservoir **43** includes an isolated passage **44** for oil coming around sleeve **41**. A bushing **45** is press-fitted into reservoir **43** and oil from said passage is fed into the space below this bushing. A pump rotor **46** is supported by the bushing **45** which also journals a downwardly extending integral rotor shaft having ducts **47** for receiving the pump oil inlet at low pressure. Oil taken in here passes through the pump to a non-rotating pump portion **48** which completes the working portion of the pump, and is supported by upwardly extending bosses **49** on reservoir **43**, seen in Fig. 3. One of these bosses has a duct **50** (Fig. 4) for conveying high pressure oil discharged by the pump into the reservoir **43**. This pump is of the type commercially classified as pressure compensated variable displacement. It has a number of new features rendering it sufficiently compact, inexpensive and quiet operating to make it practical for use in this injection system, for which it has been specially devised. However, since such a pump has many other uses, it is made the subject of a separate patent application, Serial No. 656,837 filed May 1, 1957, now abandoned, in which it is described in detail.

This pump functions to deliver oil at a controlled preadjusted pressure and it automatically changes its displacement to adjust pump delivery according to the oil volume being used from the pump discharge, and it will go to just enough displacement to maintain the set pressure and overcome internal leakage if no oil is being used. Pressure is maintained at the set value as long as rotation is fast enough to supply the used volume with the pump at full displacement. The full displacement of this pump is made sufficient to always supply the maximum steadily used volume, plus ample allowance for losses, so pressure at the pre-adjusted value is always maintained in reservoir 43 during operation.

While oil is often said to be virtually incompressible, it actually compresses about 1½% to 2% of its volume at 4500 p.s.i. which is considered a suitable value for the operating pressure. This compressibility is sufficient to allow withdrawals from this reservoir in the amplitude required by the sudden demands of the injection system, within a pressure range of (plus or minus) 3%. The space within reservoir **43** is substantially annular as symmetrically shown in Fig. 4 where the pump rotor shaft and the bushing **45** are seen centrally within the reservoir cavity, which is also invaded by the pump inlet passage **44** and its surrounding walls as seen in Fig. 3. Four passages **51** communicate with the bottom of the reservoir space. A timing bushing **52** is pressed into reservoir **43** and into the distributor body **20**. This bushing has several different levels of ports. The top level has four ports communicating with passages **51** and reservoir **43**, the next level down has eight ports **53** communicating with an atmospheric pressure drain space **54** located between the reservoir and the distributor body **20**. This space **54** also communicates freely with additional space **54** between the reservoir **43** and the cover **22**. A drain tube **55** (Fig. 2) returns and drains oil from space **54** through a boss **56** to the engine crankcase. The third level of ports below the top of the bushing **52** are pulse timer ports **57** seen to a larger scale in Fig. 10. A timing advance sleeve **58** is rotatably fitted in the bushing **52**, and an integral shaft extension **59** of the rotor **46** is rotatably fitted inside the timing sleeve. Here rotation in this fit is only while timing advance is changing; otherwise shaft **59** and sleeve **58** rotate in unison. There is a relief **60** in the shaft **59** providing a duct for high pressure oil from passages **51** through four ports **61** on the same level through timing sleeve **58**, into the space provided by this relief. The pressure in this relief space is transferred to eight passages **62** (Fig. 10) through the timing sleeve **58** which, in the rotational position shown in Fig. 10, provide free communication through the ports **57** into a pulse pressure space **63** which surrounds the timing bushing at this level. Communicating with this space **63** freely are a pair of piston valves **64**, or a pair of piston valves **65** of an alternate construction as seen in Fig. 4. The valves **64** and **65** are slidably fitted in bores **66** in the body **20**. A pair of metering cylinders **67** are press fitted into these bores.

A pair of check valve springs **68** below the cylinders urge downwardly on a pair of check valves **69** which rest on a pair of valve plates **70** each having a plurality of ports **71** communicating with grooves which are sealed against outflow by the valves **69** seated on them. The valve plates **70** are also pressed into the bores in the body **20**, but the press fits of cylinders **67** and valve plates **70** is primarily to provide a tight seal against leakage, both being prevented from being forced downward by a plug **72**, which is secrewed into the body **20**.

Referring again to Fig. 9, a pair of metering pistons **73** are urged downward by a pair of springs **74**, which also act to urge the piston valves **64**, **65**, in an upward direction until they are seated against a conical shoulder as shown in Fig. 9 and Fig. 4. As seen in Fig. 9, the spaces within the cylinders **67** below the metering pistons **73** communicate, through drilled ports **75** and an annular groove **76**, with a pair of ports **77** in the timing bushing **52**. These ports communicate with a groove **78** extending completely around the the timing sleeve **58** and this groove has an upwardly extending communicating distributor groove **79** seen in Figs. 7 to 10 inclusive. At the time when the eight pressure ports **62** are in full registry with the pulse ports **57** as shown in Fig. 10, the top of groove **79** is in full registry with one of a set of eight ports **80** communicating with one of a set of eight passages **81** seen in Figs. 7 and 8. These passages **81** individually convey oil pulses to each of the eight pulse transfer pumps in sequence as controlled by the registry of groove **79** with one of the eight ports **80** in the timing sleeve **52**, communicating with the eight passages **81**. The passages **81** and ports **80** also act as return passages for oil pulses. At the time any one pulse passage **81** is in communication with the groove **79** and therefore under pressure, the other seven passages **81** communicate through ports **80** in bushing **52** into a groove **82** surrounding the timing sleeve **58** except at groove **79** and adjacent sealing surfaces **83**. The groove **82** communicates, through six vertical grooves **84** in sleeve **58**, with an atmospheric pressure annular groove **85** above them, which in turn communicates through the eight ports **53** with the atmospheric space **54** as has already been described and is seen in Fig. 5. The groove **85** also is in communication with two grooves **86**, similar to the grooves **84**, but shorter to leave additional sealing surface at **83**.

In Fig. 7, a flexible and stepped diaphragm **87** is gripped on its rim by a head **88**, which has a passage **89** through its center communicating with space between the diaphragm and the head **88**, which has a flat precision finished surface **90**. A check valve **91** seats upon this flat surface and covers and seals an annular groove **92** in the flat face. There is also an annular groove **93** around the outside of head **88**, and a plurality of passages interconnecting grooves **92** and **93**. The outer groove **93** communicates, through a passage **94** in the body **20**, with an annular groove **95** in the body **20**.

The groove **95** is sealed by a gasket **96** and is supplied with fuel from the boss **34** through a passage **97** and a small passage **98** (Fig. 4). A spring **99** acts to seat check valve **91**, and at its opposite end applies force to aid return of the flexible diaphragm **87**, which preferably is made of thin corrosion resistant metal such as stainless steel or beryllium copper. It is made thin enough to deflect to produce the required displacement at very light difference in pressure between its two faces, without being stressed highly enough to interfere with long fatigue life. A retainer **100**, holds a spring **101** which seats an outlet check valve **102** against the low pressure of fuel pump **33**. A gasket **103** seals against leakage of high pressure metered fuel into the low pressure passage **94**.

There are eight complete pulse transfer pump units as shown in Fig. 7 and above described, all of course communicating with the one annular groove **95** which receives fuel from pump **33** through passages as described and distributes fuel to these eight units.

Operations of timing, distribution and injection

Enough of the mechanism has been described in connection with the drawings to now explain the operation of the injection phase of an injection cycle, including timing and distribution of the injection.

At the start of an injection phase, the metering pistons **73** will have been lifted to positions such as shown in Fig. 9 by means which will be hereinafter described. Their position is such that the desired metered quantity of oil will be discharged by the two pistons through passages **77** into groove **78** when forced down to shoulder against the metering cylinders **67** at their top faces **104**.

The piston valves **64**—**65** are seated against top position shoulders as described, but are free to move downward, each opposed only by the spring **74**, which applies a force that is relatively insignificant in relation to the forces now to be applied.

At the start of injection the ports **57** shown in Fig. 10 are only slightly open to ports **62**, or "cracked" as such a condition is frequently called, the timing sleeve **58** having about 10° to travel before reaching the position shown in Fig. 10. However there are eight ports **57**, each open a slight amount which is rapidly increasing if the engine is running at high speed.

Although there are some equalizing passages, essentially four of the ports **57** feed oil to the space above each piston valve **64**, **65**. The function of these piston valves is in control of metering, as will be hereinafter described, but they move downward during the injection phase since oil is being supplied to their top surfaces far faster than it can flow through the small bleed orifice **105** at the top of these pistons. However, the relative area of piston valves **64**, **65** is approximately five times the area of the metering pistons **73** so that the oil under the piston valve would force the metering piston down its full stroke while the piston valve is moving down about 20% of this distance. Since the piston valves are very light in weight and also need move such a short distance, the pressure of the reservoir **43** is applied to the top of the metering pistons virtually unimpeded, producing a substantially equal pressure below the metering pistons which are also extremely light in relation to the applied fluid pressure.

The ports **57** shown in Fig. 10 have been called pulse ports, but they act also as timer ports, and the timing of the injections may obviously be advanced by advancing the rotational position of sleeve **58** in relation to the shaft **59** which is driven in direct relation to the engine crankshaft rotation by the usual drive means (not shown).

When these timer ports have opened ⅛ degree their combined open area is about 4 times the area of the injection nozzle orifice which would be normally used, and this ⅛ degree opening occurs in about .000008 second or 8 micro-seconds at full engine speed. The port orifice velocity of ¼ the nozzle orifice velocity produces only $\frac{1}{16}$ the pressure drop, consequently there is a virtually instantaneous pressure rise to about 93% of its full value in one-eighth degree rotation, as the timer ports **57** open, creating a pressure wave having (in oil) approximately 55,000 inches per second propagation velocity and substantially full reservoir pressure. Approximately .0004 second later full pressure is almost as instantly established at the nozzle.

This pressure wave, and the flow which follows it (permitted by oil compression at the wave front) travels from below the metering piston **73**, through the channels **75**, **76**, **77**, **78** described, to the groove **79** which is in registry with a port **80** connecting to one of the passages **81**; the oil then pressurizes one side of the diaphragm **87**, transferring pressure and expelling fuel through the passage 89 by movement of this diaphragm toward the head 88. (This fuel has previously entered from passages 94 and 95 by lifting the inlet check 91, having about 7 injection cycles time to enter from pump 33 through the channels that have been described.)

The pressurized fuel now lifts the ball-check 102 against force of the spring 101 and tightly seats the inlet check 91 against surface 90 by pressure against its exposed face. The pressurized fuel now passes into the injector line through a connection point 106 leading to the injection nozzle which may be of any of the known types, but preferably is a simple fixed orifice with which this system is particularly well fitted to cooperate, since constant pressure produces a constant flow rate and therefore a consistently fine atomization at all engine speeds and fuel charges. The fuel flow for injection may be caused to stop even more instantly than it started, since when the metering pistons 73 travel down until their shoulders strike the top surfaces 104 of cylinders 67 the pressure below the piston 73 immediately drops to zero or substantially so and a pressure drop wave starts from below the piston 73 toward the nozzle. This differs from the pressure wave in that there is no flow behind the advancing wave front, the expansion of oil which now occurs at the wave front now allows flow in the injection line to continue at all points in advance of the wave front, until when the pressure drop wave reaches the nozzle the flow through the nozzle suddenly stops and the entire line is now at low pressure. The open nozzle may now admit pressure from the cylinder gases, which may be at approximately 900 p.s.i The cylinder pressure causes a pressure wave to flow back in the line as far as the check 102, which has by now firmly seated. A small recompression of fuel in the line follows this wave front and admits a small amount of combustion gases to enter the nozzle orifice. However this only amounts to about .09 inch line flow and as the cylinder pressure drops toward atmospheric, the gas is expelled by re-expansion of the fuel in the injection line but no fuel follows since after expulsion of the gases, the line pressure is also down to atmospheric.

In the past, high pressure fuel injection systems have quite commonly been plagued with strong reflected pressure waves, which in some cases cause transient pressure rises effective to momentarily increase total pressure to about 200% of normal, accompanied by transient drops of pressure to virtually zero, resulting in an engine cylinder receiving its charge in several separate pulses. The timing of these transient pulses in respect to crank angle is uncontrolled and subject to wide scatter as engine speed, fuel charge and other conditions vary, causing unpredictable performance deficiencies under certain conditions and combinations. These scattered injections often cause poor combustion due to some of the fuel which is injected late being only partly burned, leaving a carbon core residue, due to burning before evaporation. These carbon cores represent wasted fuel and cause severe smoke.

One of the objects of this invention is to bring about virtually complete elimination of these reflected pressure waves which is accomplished by almost completely preventing the initial reflection which starts the extra waves. This is done by making the compressibility flow in the injector lines, which follows immediately behind the advancing main pressure wave front, equal to the outlet flow through the nozzle orifice or orifices at the established pressure.

The method of determining the flow through nozzle orifices is well known to those skilled in the art, who also know that pressure waves propagate at approximately 55,000 inches per second in light petroleum oils and fuels. It is equally true, but frequently not recognized, that a flow exists behind the propagating wave front, which occurs to keep up the pressure at the wave front as additional liquid contracts under compression as the wave front advances. The velocity of this flow is equal to the propagation velocity of 55,000 inches per second times the compressibility which is approximately .000004 times the pressure, or equal to 0.018 at 4500 p.s.i., resulting in a compression velocity of 990 inches per second behind a 4500 p.s.i. propagating wave front. This same pressure produces approximately 9900 inches per second in a typical orifice, requiring that the line area be 10 times the orifice area to attain the desired balance. Fig. 11 illustrates a typical nozzle and injection line relationship in which the inside diameter of the line is designated at 131, and the inside diameter of the orifice at 132. It has been found that a .055" line I.D. is suitable for use with a .017 I.D. single orifice at this pressure. It should be noted that a balance as above set up only holds true under a certain controlled operating pressure since the compressibility flow velocity is proportional to the pressure directly, while the orifice flow velocity is proportional to the square root of the pressure. The means for establishing a constant pressure have been hereinbefore described.

It is recognized that there is a very brief interval of time, estimated at 20 to 100 micro-seconds, which is required for the pressure to reach its full value after the first pressure increment arrives at the nozzle orifice. A similar period of time is estimated to be more than adequate for establishing normal nozzle orifice flow, since this flow is being established during the pressure rise, and only one micro-second approximately is required for a pressure wave to pass through the orifice. The time of pressure rise allows 20 to 100 pressure increment waves to pass through the orifice, each adding to orifice flow velocity, while pressure is building up to the normal value. Consequently, when full pressure is reached the balancing full nozzle flow has been established and no reflected wave is created.

Since full normal injection requires about 1200 micro-seconds, and the injection quantity which may occur during the pressure rise is less than proportional to the time proportion so might be from 10/1200 to 60/1200 of a full charge, it is likely that a maximum of 5% of the charge may be injected under sub-normal pressure conditions. It should be noted here that the sub-normal atomization and slight percentage of coarser particles enter at a favorable time, having the maximum time for evaporation to occur before ignition and also have been heated by being in the nozzle tip between injections to favor quick evaporation and full combustion without carbon cores.

Since the downward velocity of the metering pistons 73 may be about 60 or 70 inches per second due to the short duration of the injection stroke, means have been provided to check the last portion of this stroke to eliminate any audible noise resulting from sudden shouldering of the metering pistons. To accomplish this, the outlets 75 from under the metering pistons have been placed at the sides thereof and so positioned vertically in respect to the bottom of the pistons that the oulets are almost entirely covered as the metering pistons shoulder. This builds up some extra pressure in the spaces immediately below the metering pistons which extra pressure naturally decelerates the pistons before they shoulder against the cylinders 41. This arrangement can be proportioned to secure any desired amplitude of decelerating effect, in fact could be proportioned so that the metering pistons never shouldered in normal operation, but this is not a preferred construction, since it would give a more variable metering, due to lack of precision of stopping position.

The groove 79 acts as a distributor port and is made slightly wider than the timing ports 57 and 62 as seen in dotted lines in Fig. 10 so that, whatever the advance of the timing sleeve, the distributor port always opens slightly before the timer ports and closes slightly after they close.

The sealing surface 83 is made wide enough (Fig. 7) to completely seal pressure in the one port 80 until this port is completely cut off from groove **79**. This leaves approximately seven injection cycles for the oil to return to the atmospheric pressure chamber under pressure induced by the diaphragm **87**, spring **99** and the pressure of fuel filling the chamber on the opposite side of the diaphragm. This will generally admit more fuel to be contained in the space between the diaphragm **87**, and the head **88** than will be expelled by the metered oil pulse, but this space is made as large as needed for a maximum fuel charge for a cold engine, to prevent the oil pressure collapsing diaphragm **87**. Even if there is no fuel in the system, the diaphragm **87** and spring **99** will return a maximum oil pulse in the seven injection cycle time available.

From the foregoing it is apparent that the timing of injections may readily be advanced by advancing the rotated position of the timing sleeve **58** in respect to the rotated position of shaft **59**, and that all injections occur at a substantially constant flow rate, regardless of amount of charge being injected, or the speed of the engine, since injections occur at constant pressure without reflected pressure waves and the only significant resistance normally existing is the orifice discharge resistance which insures a constant flow rate when a constant pressure is supplied. As a corollary to this, the constant high pressure effective at the nozzle orifice insures uniformly fine atomization throughout the major part of all injections.

Metering unit operation

As has been mentioned, the amount of fuel to be injected per charge is metered by governing the lift of the metering pistons **73** which in the present invention are lifted by oil pressure. In static conditions, or very low speed operation, the piston valves **64** or **65** are in their seated positions before metering starts, and the springs **74** oppose the lift of the metering pistons. The lift position reached is then determined by the force created by pressure of oil in the space below the piston as related to the force applied by the spring **74**. This simplified statement is only strictly true at extremely low speeds. Many other factors must be considered as speed is increased, but it remains the basic condition, basic in the further sense that the amount of lift so produced is desired to be maintained over a wide range of speeds so that the quantity actually metered remains according to the pressure supplied to the spaces below the metering pistons over a large range of speed. It is known that a certain degree of damping, here supplied by resistance to the rate of flow of the oil, must be in effect to prevent over-run of piston **73** under the influence of momentum of the piston and the oil which is moving with it. This flow resistance can supply sufficient damping, when suitably proportioned, to reach what is called critical damping, in which case no over-run takes place. However, when such an amount of damping is applied, it is too much to permit the required lift to occur in the very short period of time available in high speed operation. In the present invention wherein pulses from an oil metering device are distributed in sequence to insure like metering to each cylinder, it has become necessary to complete the lift stroke of the metering pistons **73** in less than ½ of a complete injection cycle or about one fifteenth the time available in the invention described in U.S. Re. 24,214. This has required the applicant to devise a number of means of speeding operation and reducing effective momentum as well as ingenious new means of controlling and timing the changes of damping required as the speed of operation is increased. The problem has been made more severe by a further aim to keep the required metering pressure at all times less than the probably available oil pressure from the usual engine oil pump, which has limited the metering pressure to about 45 p.s.i. maximum under extreme cold conditions and 22 p.s.i. maximum under normal running temperatures. These are merely figures used as a basis for recent work

and can be varied by reproportioning the various elements.

The new means include simplified new types of piston valves **64**, **65** which cause substantially critical damping conditions to be maintained over the lower range of speeds, and as speeds increase, cause this damping to be partly by-passed by a displacement up-stroke of the piston valves **64**, **65** which take a down-stroke during every injection cycle.

This down-stroke is followed by upward movement of the piston valves under the influence of springs **74**, causing oil to enter the space between the metering pistons and the piston valves. At low engine speeds, this up-stroke is completed before the metering stroke of the metering pistons starts, while at higher speeds there is an increasing proportion of the piston valve up-stroke remaining to occur during the metering piston stroke, and the displacement of this remaining stroke of the piston valves allows a stroke of equivalent displacement to be taken by the metering pistons free of the damping resistance. Then when the piston valve has completed its up-stroke the remaining portion of the metering piston stroke occurs under the main damping flow resistance and is checked adequately to prevent over-run occurring before the timer cuts off the lift by sealing the pulse chamber ports **57** from atmospheric pressure grooves **84**—**86** preparatory to applying injection pressure from ports **62**. As the engine speed is lowered more of the metering stroke occurs after the piston valve has seated, so more is under the primary damping resistance, while a compensating increase of available time for completion of the metering stroke is also in effect.

It has been found that this outflow from between the metering pistons and piston valves must occur in approximately .0004 second at maximum operating speed while at the same speed, the inflow to this space (which may be called the trap space) may occupy .0024 second approximately out of a total of .0031 second assumed available for one complete metering and injection cycle.

This causes, due to the time ratio of .0024/.0004 second or 6 to 1, an equivalent flow rate ratio to exist, as the same quantity of oil must pass each way. Taking this flow through a standard orifice would normally result in velocity squared flow resistance, making nominally 36 to 1 ratio of outlet pressure difference to inlet pressure difference. The springs **74** have forces which must be determined to suit the metering piston area and the range of metering pressure supply to be used, consequently, the springs set up a certain pressure differential to cause inflow into the trap space. It has been found that 36 times this pressure would produce severe braking of the up-stroke of the metering pistons such that the time used to stop would reduce from .0004 second to possibly .0001 second, tending to produce a still higher braking pressure.

Two different remedies for this situation have been devised, one illustrated on metering pistons **64** and the other on the style seen at **65**.

In the case of the style **64** metering pistons, they have each been provided with three guiding lands **107** which are a sliding fit in the bore in which the pistons are mounted. Between these lands a shallow relief **108** is provided, which allows suitable oil flow in and out to occur under purely viscous flow conditions, where the flow resistance is then proportional to the first power of velocity. This was found to make such a drastic difference that too little brake effect occurred, resulting in over-run before the timer could cut off the up-stroke. A remedy for this was devised by enlarging the small orifice **105** at the top of the piston to make some substantial part of the flow follow the velocity squared law, reducing the viscous flow clearance to suit. By suitable proportioning of the two types of flow, the desired degree of braking may be secured. The top orifice **105** is initially provided to bleed any air or vapors out of the trap space.

With metering pistons **65**, a different means of remedy is secured. Here, the in and out flow is all through the orifice at the center of the top, except for relatively negligible flow through the side clearance fit. The tendency for excessively fast stop of the metering piston upstroke is overcome in this case by having the top of the piston-valve come to a seat on a considerable area, such that .0003 second approximately is required for the piston valve to reach a substantially metal to metal seat, due to the oil trapped between the top surface of the piston valve and the mating seat area retarding the last .002 to .003" of piston valve uptravel. The pressures involved in stopping the metering piston is then spread over a longer period of time and can be limited to on the order of 60 to 100 p.s.i. No appreciable delay of down-stroke results when pressures of about 4500 p.s.i. are applied. The orifice in the top of piston valve **65** of course also serves to purge any air or vapors as with the piston valves **64**. This construction is also such that any desired stopping rate for the metering pistons can be secured by suitable proportioning.

The use of two metering piston assemblies and their accompanying parts accomplishes two important things. First, the time required for the two pistons to reach the then required one-half of the full metered volume and stroke is reduced, and second, each can be placed close to four of the timer ports **57** instead of having four ports on the far side in the case of using the eight ports with a single metering unit. This reduces the effective momentum of the oil moving with the metering pistons, as does also the larger total area of passages conveying the oil which exists when two pistons are used.

Another new device which aids greatly in reducing the effect of oil mass moving with the metering piston or pistons is introduced. Two pistons **109** (Fig. 3), are provided for this purpose. Two springs **110** apply downward force to the pistons **109**. This is not basically different from the usual spring loaded accumulator, but the use is markedly different in that these are employed as what may aptly be called exchange accumulators. The spaces under pistons **109** are part of a generally annular passage **3** as seen in Fig. 13. This passage surrounds the central portion of the distributor unit and has cross-sections varying at different points as seen in the space under the metal around passages **81** in Fig. 7, and in the spaces under the pistons **109** in Fig. 3. A minimum cross-section exists in spaces under valve plates **70** as seen in Fig. 9, but the passage provides ample two-way communication between these spaces and the spaces under the pistons **109**. During the critical metering or lift stroke of the metering pistons oil (at the metering pressure in effect) is taken from below the pistons **109** to supply most of the oil used to lift the metering pistons, while the oil being displaced above the metering pistons **73** and piston valves **64** or **65** is equally accommodated in the space increase which has resulted above the pistons **109**. This permits rapid free exchange of volume through short passages of ample cross-section and a much slower steady flow of oil in the much longer channels supplying the space below the pistons **109** and **73** with oil at metering pressure, and the much longer line **55** leading atmospheric pressure oil to the crankcase. All the extra flow over the steady average value, is handled by very short paths interconnecting the spaces below the metering pistons **73** and below the accumulator pistons **109** on the metering pressure side, and by the short paths interconnecting the space above the piston valves, through the timer ports **57** and vertical grooves **84** and **86** to the atmospheric pressure chambers above pistons **109**. Since the major part of metering piston lift stroke occurs in as little as .001 second the importance of these short paths of ample area is evident. The area of the pistons **109** as shown in about five times the area of the metering pistons, so their stroke in action is about one-fifth as much. They are also made as light as possible, to further minimize inertia. The reasons for using two pistons **109** are similar to the reasons for using two metering assemblies, but the advantages reinforce each other due to the cooperative effect in shortening paths and enlarging area of paths interconnecting the alternately positioned devices.

It should be here noted that accurate matching of these units is not required, since their combined delivery is fed to each cylinder in sequence, the sum of the quantities delivered by the plural metering units equaling the fuel charge fed to each cylinder.

Metering control means and operation

The metering means which has been described, is responsive to control by reducing or increasing the oil pressure under the pistons **73**. Two means of control of this pressure are provided; throttling, and a maximum fuel limit control achieved by limiting the metering pressure to a temperature compensated maximum, which cannot be exceeded regardless of throttle settings.

The maximum fuel limit control comprises a valve spool **111** (Fig. 3) slidably fitted in a bore in the sleeve **41**. The spool **111** has an annular groove **112** and passages to an open center which communicates with its lower end area and with a passage **113** through the cover **22** to a passage **114** in the body **20**. A plurality of inlet ports **115** in the sleeve **41** may be in or out of communication with the annular groove **112** around the spool **111**, depending on its position. A spring **116** urges the spool **111** downward and is so proportioned that when the desired pressure is established in the space below the valve spool it comes to the position shown in Fig. 3, where flow through the inlet ports **115** has just been cut off. The flow of oil to the outer part of sleeve **41** has been described, so with oil here at pressure, a suitable lower pressure can be automatically maintained under the valve spool **111** regardless of reasonable outlet quantity. The pressure so maintained is proportioned to establish a value which will limit the metering to the maximum fuel charge that can be properly utilized in a cylinder of the engine at normal operating temperature. This pressure may be called the fuel metering limit pressure.

A small pilot valve spool **117** is slidably fitted in a bore in the sleeve **41** and a small annular groove **118** in this bore communicates with the space **42** around sleeve **41**, containing oil from the low pressure supply means. The spool **117** has a central passage open to the bottom of the spool communicating with cross passage ports **119** which are shown just out of communication with aforesaid small annular groove, and also just out of communication with the atmospheric pressure space above the sleeve **41**. A thermally responsive element **120** applies increasing force to induce downward movement of the pilot spool **117** as its temperature lowers, and a screw **121** is provided to adjust the temperature at which the thermal element **120** starts to apply force downward on the pilot spool **117**. When the temperature is below this value, spool **117** is forced down slightly and oil under pressure passes down and applies pressure to the top of valve spool **111**. This results in the pressure below this spool being maintained at a pressure equal to its original value plus the pressure above. This total pressure is the cold compensated fuel metering limit pressure.

The thermally responsive element **120** is heated by the heat of the oil passing through the distributor assembly and in part by surrounding air so that when suitable operating temperatures of the engine oil and water are reached the spool **117** is free to rise under the influence of any pressure below it, exhausting the space between spool **111** and spool **117** to atmospheric pressure.

When the throttle valve **38** is partly closed, the resistance causes a pressure drop to exist under the flow that is passing through the valve. This pressure drop is increased with higher engine speeds and decreased with lower speeds so that the pressure going into the oil filter and boss **40**, which is equal to the oil pump pressure minus the pressure drop due to throttling, undergoes a marked increase as engine speed is lowered and a similar decrease as engine speed increases.

This directly governs oil metering, since the valve spool **111** lowers, opening free passages from the ports **115**, through the groove **112** and passages **113** and **114** whenever the inlet pressure is less than the fuel metering limit pressure for which this valve assembly is set. This allows the above described pressure changes to bear on the bottoms of metering pistons **73** and exchange accumulator pistons **109** and to have full control of metering when the throttle valve **38** has any appreciable resistance to fluid flow. This arrangement gives the engine on which this system is used a very high lugging effect in part throttle and reduces engine torque as engine speed increases when there is any appreciable throttling by valve **38**.

When this valve is well open, so that little pressure drop occurs in it, the pressure limit valve and the maximum fuel limit pressure take over to limit the fuel to the quantity that can be efficiently burned, causing the engine to develop substantially constant torque over a wide speed range.

This torque and fuel feeding can be held to and beyond a speed where fuel feeding by carburetor and manifold air travel starts to reduce the change per cylinder, thereby increasing the peak horsepower output by enriching the mixture when peak power is needed. The extra power can be further increased by enlarging manifold cross sections. The throttle valve **38** is held slightly open by an adjustable stop, which comes into effect to govern idle speed when the throttle valve is released, while the engine can be stopped by shutting the valve **37**, cutting metering pressure off or down to a value where no lift of the metering pistons occurs.

Timing advance means and operation

Advancing the rotational position of the advance sleeve **58** in relation to shaft **59** advances the action of both timer valve and distributor valve ports, as has been described. The apparatus will be proportioned and assembled in such manner that, when no advance is applied, the injections will occur at approximately top dead center at the end of the compression stroke in each cylinder of the engine. Figs. 4 and 6 show the advance governor in this non-advanced position. Flyweights **122**, pivoted on pins **123**, have integral cam portions which extend inward and abut against the opposite flyweight in the manner shown in Fig. 6, when the governor is in the non-advanced position, which is held until a certain low speed is exceeded. At speeds below this, the governor action is stabilized by the effect of these abutting surfaces. A flat driver plate **124**, has integral therewith a drive shaft **125** and an upward central extension of the flat portion which engages a slot in the pump drive shaft **59** (Fig. 3). A yoke **126** has an integral hub **127** rotatably fitted to the diameter of the shaft **125** and two arms carrying bores in which the pins **123** are firmly press-fitted. A similar yoke is an integral part of the advance sleeve **58**, but in this case the pins **123** are slidably fitted in the bores in the arms. The cams on the flyweights extend from their pivot bosses into the space between these yokes, and contact the flat drive plate **124** with their cam contour surfaces. A helically coiled torsion spring **128** (Fig. 4) has a bent end which engages a slot in the hub **127** of the yoke **126**. A drive collar **129** which is affixed to the shaft **125** also has a slot to receive a second bent end of the spring **128**. Passages **130** drain oil from housing **21**, and the direction of coiling of spring **128** causes these coils to carry oil down to drain past collar **129**. The spring **128** applies sufficient torque to yoke **126**, in a counter-clockwise direction in Fig. 6, to cause the reaction at the cam surfaces against the driver plate **124** to swing flyweights **122** into the position shown in Fig. 6, and to hold them in this position until a certain suitable minimum speed is exceeded, at which time centrifugal force starts to swing the flyweights outward. At low speeds the cam surfaces contact the plate **124** near its ends, so a small torque around the pins **123** resulting from a low centrifugal force occurring at low drive speed has a large lever advantage. At maximum speeds (which may be from 10 to 20 times the minimum speed, producing from 100 to 400 times the centrifugal force) the cams contact the plate **124** near its center and at the tips of the cams so that here the torque around pins **123** is at a large leverage disadvantage, permitting the balancing retarding torque of the spring to be only 5 to 20 times its low speed torque instead of 100 to 400 times it as would be necessary if this leverage ratio did not change. It has been found that with suitable proportion of the springs, cam contours and weights that a substantially straight line advance, that is, a certain number of degrees per each hundred r.p.m. can be achieved if required. It has also been found that a true arc contour of cam surface of suitable proportions on the cams can produce a similar result with slightly less advance in the upper range of the speed, which may be a desirable condition.

The need for advance of injection timing arises from an accumulation of a number of delay factors. One major factor is the line delay mentioned hereinbefore as approximately .0004 second. There is also a spray formation and penetration delay, an ignition lag of a certain period varying with compression temperature, fineness of atomization and with fuel characteristics, and the time after ignition required for about ½ the fuel to be burned, developing ½ the available heat of the charge. For the reason that it is difficult to predetermine all of these factors and they will vary also with changes in engine design, each engine may require its own special cam contour to be developed to attain highest efficiency. However, it is believed more likely that a well chosen contour will serve 99% as well in virtually all suitably designed engines. It has been calculated that an advance of 18° at the governor or 36° at the crankshaft will be adequate for a suitably designed engine at a speed of 4800 r.p.m., and the construction shown is practical to secure an advance of this magnitude. It is also able to hold about 70% of the full normal advance against the drag of reasonably cold oil in the clearance between the timing sleeve **58** and the bushing **52**. The distributor may be driven by either the shaft **125** or the collar **129**, by the means usually provided for the distributor drive.

This disclosure, showing the invention as arranged for clockwise rotation as seen in Figs. 1, 5, 6, 8 and 10, may obviously be rearranged by anyone skilled in the art to provide for operation in a reverse direction of rotation at unimpaired efficiency, and the construction shown has been contrived to permit such reversal with great ease.

It is apparent that the device herein disclosed does provide greatly simplified means for achieving most aims of Patent Reissue 24,214 which pertain to fuel injection and that the device has been re-arranged to make it very readily adaptable to standard 8 cylinder engines and shows means and principles adaptable to all other numbers of cylinders. The easily applied and mounted distributor type package has also been achieved, means provided for obtaining increased peak power output when required by the operator, yet automatically returning to extremely efficient lean mixture operation when less than full torque is required.

The means and methods for using the inexpensive plain orifice nozzles under conditions attaining the highest performance from them in respect to atomization fineness and constancy, injection pressure and durations, constant flow velocity and atomization conditions throughout any one injection and all injections have been shown and described fully. The means and methods shown and described in connection with these things and with the avoidance of transient pressure variations by virtually complete elimination of reflected pressure waves are believed to all be of great importance in securing sustained good performance at low first cost and low upkeep expense, as well as insuring the very essential quality of good cold starting performance.

The means shown for metering the charges to all cylinders from a single device, thereby insuring like fuel charges to all cylinders without requiring any matching of metering devices, is also deemed of great importance and is believed to be achieved in an entirely new and highly efficient manner. It should be noted that nowhere in the system is there any point where fuel can leak through a sliding or moving fit to mingle with oil and that nowhere is fuel relied upon for lubrication of any parts rubbing or sliding upon one another under pressure or a close fit. No leakage of the fuel metered can occur in normal operation of the system except at the flat surface where check valve **91** seats on the surface **90**. Both of these surfaces can be inexpensively flat lapped to about 8 micro inches finish such that leakage is virtually nil, and the parts can easily be re-lapped for reconditioning since precision of dimension is not a factor. Any leakage that might occur here is retained in the fuel supply.

In the oil metering units, the pistons **73** need not be particularly closely fitted to cylinders **67** since the pressure difference above and below the pistons is limited to about 22 p.s.i. even when delivering at 4500 p.s.i., making oil leakage here insignificant as to its effect on the metered quantity. The valve **69** mates to its seat **70** on flat lapped surfaces making this leakage insignificant, especially when the fluid is oil.

There is however slight leakage at the oil distributor from grooves **78** and **79** to atmospheric pressure oil collecting drain spaces. This requires that to hold this leakage to a harmless value the hot running clearance between timing sleeve **58** and timing bushing **52** must be held to about .0004 inch on the diameter under operation at normal running temperature and pressure. Materials for these parts and the body **20** into which the bushing **52** is press-fitted, may be selected to reduce the cold clearance by about .0001 on the change from 70° ambient to about 190° under normal operation. A further and more potent means has been provided to produce some reduction of clearance here as the operating pressure comes up to normal. As seen in Fig. 3, the recess **60** exposes most of the inside of the advance sleeve **58** to the full working pressure of the reservoir **43**, consequently the sleeve **58** may be expanded .0003 to .0004 as this pressure comes up. This reduction of clearances under pressure and temperature allows larger tolera ces in manufacture if these parts are run-in under gradual increase of pressure and/or temperature.

It should be noted that the arrangement, shape and orientation of the various elements has been so contrived that all the working fluid chambers will purge themselves of any contained air or vapors by the combined action of gravity separation and fluid sweeping, within a few cycles of operation, except the reservoir **43**, where presence of air or vapors would not cause any malfunction, but would be purged by absorption, bleed, and sweeping action in some additional time. A bleed passage **133** (near the top of reservoir **43** in Fig. 4) provides slow bleed through the bearing clearance.

The words "oil" and "lubricant" have been used interchangeably and are intended to apply to any liquid having suitable lubricating properties, regardless of its origin or manufacture, and the word fuel can be considered to apply to any low viscosity liquid, whether combustible or not, in use where its combustion is not required.

Having shown and described specific embodiments of this invention, including its method of operation, objectives and advantages, the spirit and scope of my invention is defined in the following claims.

I claim:

1. Means for minimizing creation of reflected pressure waves in an injection system for an internal combustion engine, comprising means for delivering injection volumes at constant pressure throughout substantially the entire duration of each injection, a line for receiving these injection volumes, and a fixed orifice discharge nozzle at the end of said line, the flow areas of said nozzle and line being such that the compressibility flow behind a propagating wave front in said line is balanced to substantially equal the flow through said fixed orifice discharge nozzle while said constant pressure is maintained at said orifice.

2. In a fuel injection system for an internal combustion engine, a housing structure; shaft means rotatably mounted in said housing structure; a passage in said shaft means for continuously receiving oil under pressure; first distributor ports radiating from said passage; second ports in the housing structure in the path of the first ports; a chamber in the housing structure communicating with said second ports; movable wall means closing said chamber; an oil discharge passage in the housing; third ports in the shaft means having a path crossing the discharge passage; said second and third ports being alternately opened to the chamber so as to cyclicly develop a high oil pressure on the movable wall means; a second chamber in communication with the movable wall means; passage means for introducing low pressure oil into said second chamber; check valve means closing against back flow out of said second chamber; an annular passage in the shaft in communication with the second chamber; whereby movement of the movable wall means causes oil in the second chamber to be intermittently forced into the annular passage at high pressure; a fourth port in the housing structure; a second oil discharge passage in the shaft means; a third chamber in the housing structure communicating with the fourth port; second movable wall means closing said third chamber; a third passage in the shaft means extending from the annular passage across the fourth port; said third passage and second oil discharge passage being alternately open to the fourth port whereby a high oil pressure can be intermittently developed on the second movable wall means; a fuel chamber in communication with the second movable wall means; inlet and outlet ports for said fuel chamber; and check valve means closing against back flow from said fuel chamber; whereby said second movable wall means is enabled to force quantities of fuel out of the fuel chamber at a high constant pressure throughout the discharge phase.

3. In a fuel injection system, means for creating oil pulses of accurately and controllably metered volumetric amplitude delivered under constant pressure of predetermined value, said means including a pulse chamber defined partially by a movable wall, and rotary shaft means having inlet and discharge passages alternately connected to said chamber for moving the movable wall to develop oil pulses; a plurality of oil-to-fuel pulse transfer devices; and valve means for distributing said oil pulses to said transfer devices in sequence and in timed relation to the oil pulse delivery, said valve means comprising a discharge duct and passages in the rotary shaft alternately connecting the transfer device with the first mentioned means and discharge duct.

4. A liquid metering unit and high pressure delivery means for cyclicly repeated metering of a predetermined volume, comprising cylinder means, a metering piston reciprocable therein, inlet passage means for controllable low pressure supply of the liquid to be metered, a check valve closing said passage means against back flow, a piston valve within the cylinder means and having a greater diameter than the piston; spring means between

the piston and piston valve urging them apart; stop means for limiting movement of the piston and piston valve under the influence of the spring means, an orifice through the head of said piston valve, a high pressure liquid supply source and means for intermittently applying and releasing liquid from this source to the exposed face of the piston valve to produec metering cycles, and passage means for accepting high pressure liquid discharged by said metering piston.

5. Means for minimizing creating of reflected pressure waves in an injection system for an internal combustion engine, comprising means for delivering injection volumes at constant pressure throughout substantially the entire duration of each injection, a line for receiving these injection volumes, check valve means preventing back flow in the line; and a fixed orifice discharge nozzle at the outlet end of said line, the product of compressibility flow velocity behind the propagating wave front and line area being equal to the product of the nozzle orifice velocity and orifice area, whereby to balance the two flows and prevent creation of primary reflected waves.

6. Charge distributing means comprising a plurality of separate means for utilizing metered charges, passage means for receiving a series of separate metered charges, and distributor valve means having a plurality of ported members with ports positioned to provide intermittent fluid passage from said passage means to each of said separate means in sequence as one of said ported members is rotated, means for providing a very small substantially fixed running clearance between the cooperating ported surfaces of said ported members, and pressure recess-forming means formed and positioned to reduce the running clearance between said rotated ported member and another of the formed ported members when said recess is supplied with fluid under high pressure to thereby cause said substantially fixed clearance to be reduced by deflection of said rotated ported members as said pressure is increased.

7. In a fuel injection system, mechanism for creating oil pulses of controllably metered volumetric amplitude delivered under constant pressure of a predetermined value; a plurality of oil-to-fuel pulse transfer pumps, each having its own oil inlet and outlet connection, valve means connected between the oil pulse metering mechanism and each of said connections for distributing the oil pulses to said pulse transfer pumps in sequence, said valve means operating in timed relation to said oil pulse mechanism to thereby distribute the oil pulses to the transfer pumps in timed relation to the oil pulse delivery from the oil pulse metering mechanism.

8. A liquid metering unit and high pressure delivery means for cyclically repeated metering of a predetermined volume, comprising a source of liquid at substantially constant high pressure, cylinder means, a metering piston reciprocable in said cylinder means, inlet passageway means for controllable low-pressure supply of liquid to said cylinder means to be metered thereby, valve means closing said passageway means against back flow, cooperating stop means on said piston and cylinder means limiting motion of said metering piston in one direction, a piston valve in said cylinder means, stop means for said piston valve limiting its motion in a direction away from said metering piston, a metering spring between the piston valve and metering piston applying a separating force to each of them, an orifice through said piston valve, passageway means for accepting high pressure metered liquid discharged from the end of said metering piston remote from the piston valve, pressure liquid passageway means for supplying high pressure liquid to the face of the piston valve remote from the metering piston, discharge means for receiving liquid returned from said face of the piston valve, and means alternately connecting said pressure liquid passageway means and discharge means to said remote face of the piston valve to produce metering cycles.

9. In a cyclic liquid metering and distribution system, a plurality of charge receiving devices, pumping means for continuously supplying liquid at substantially constant pressure, a source of liquid at lower pressure, manually controllable means for governing the flow of said lower pressure liquid, cyclically operable means for metering a volume of liquid subject to control by said manually controllable means, said cyclically operable means comprising two pressure expansible chambers, one of said chambers receiving and metering the lower pressure liquid in the first phase of each cycle and discharging the metered quantity of liquid in the second phase of the cycle by admission of liquid from said pumping means to the other of said chambers, and valve means having timing ports to cause repeated timed cycles of said cyclically operable means and having a distributor port to distribute metered volumes of liquid from said cyclically operable means in sequence to said plurality of charge receiving devices in synchronism with the discharge phases of said cyclically operable means.

10. In a fuel injection system for internal combustion engines, a series of liquid receiving devices, engine driven drive means, pumping means directly driven by said drive means and delivering liquid at substantially constant high pressure, means for supplying liquid at low pressure, manually controllable means governing the pressure of the low pressure liquid, cyclically operable means for repeated metering of volumes of liquid, said cyclically operated means comprising two co-extensive pressure-expansible chambers, one of said chambers receiving and metering the lower pressure liquid in the first phase of each cycle and discharging the metered quantity of liquid in the second phase of the cycle by the admission of high pressure liquid to the other of said chambers, speed responsive means for advancing the cycle time of the cyclically operated means, a member of tubular character surrounding said drive means and speed responsively advanced in rotational position relative thereto by said speed responsive means, said member having timing ports to cause repeated timed cycles of said cyclically operable means for metering and discharge of liquid and having a distributor port to distribute the metered discharges in sequence to said series of liquid receiving devices in synchronism with the discharge phases of said cyclically operable means.

11. A liquid metering unit and high pressure delivery means for cyclically repeated metering of a controllable volume, comprising cylinder means, a metering piston reciprocable therein, inlet passage means for controllable low pressure supply of the liquid to be metered, a check valve closing said passage means against back flow, a piston valve within the cylinder means and having a greater diameter than the piston; spring means between the piston and piston valve urging them apart; stop means for limiting movement of the piston and piston valve under the influence of the spring means, an orifice through the head of said piston valve, a high pressure liquid supply source and means for intermittently applying and releasing liquid from this source to the exposed face of the piston valve to produce metering cycles, plural passage means for accepting high pressure liquid discharged by said metering piston, and distributor valve means for distributing one cyclic discharge to each of said plural passage means cyclically in sequence.

12. The device defined in claim 11, in combination with speed responsive advance means for advancing the timing of said means for intermittently applying and releasing liquid and for simultaneously advancing the timing of said distributor valve means.

13. In a cyclic liquid metering and distribution system, a plurality of charge receiving devices, pumping means for continuously supplying liquid at substantially constant

high pressure, a source of liquid at lower pressure, manually controllable means for governing the flow of said lower pressure liquid, cyclically operable means for metering a volume of liquid subject to control by said manually controllable means, said cyclically operable means comprising two pressure expansible chambers, one of said chambers receiving and metering the lower pressure liquid in the first phase of each cycle and discharging the metered quantity of liquid in the second phase of the cycle by contraction caused by admission of liquid from said pumping means to the other of said chambers, valve means having timing ports to cause repeated timed cycles of said cyclically operable means and having a distributor port to distribute metered volumes of liquid from said cyclically operable means in sequence to said plurality of charge receiving devices in synchronism with the discharge phases of said cyclically operable means, spring loaded accumulator means for said lower pressure liquid, and common housing means for said valve means, said cyclicly operable means for metering, and said accumulator means.

14. The device defined in claim 13, with said common housing means having said valve means centrally disposed therein, and said accumulator means and said cyclicly operable means for metering both being in plural, and alternately positioned around said valve means.

15. In a fuel injection system, mechanism for creating

oil pulses of controllably metered volumetric amplitude delivered under pressure; a plurality of oil-to-fuel pulse transfer pumps, each having its own oil inlet and outlet connection, valve means connected between the oil pulse metering mechanism and each of said connections for distributing the oil pulses to the pulse transfer pumps in sequence, said valve means operating in timed relation to the oil pulse mechanism to thereby distribute the oil pulses to the transfer pumps in timed relation to the oil pulse delivery from the oil pulse metering mechanism.

References Cited in the file of this patent

UNITED STATES PATENTS

1,558,081	Gano et al.	Oct. 20, 1925
1,683,317	Wakefield	Sept. 4, 1928
1,816,157	Scott	July 28, 1931
2,101,064	Hautzenroeder	Dec. 7, 1937
2,170,413	Johansson	Aug. 22, 1939
2,196,360	Kamenarovic	Apr. 9, 1940
2,598,528	French	May 27, 1952
2,674,236	Humber	Apr. 6, 1954
2,689,527	Emerson	Sept. 21, 1954
2,803,234	Mansfield	Aug. 20, 1957
2,816,533	Reggio	Dec. 17, 1957