Oct. 8, 1968 F. EHEIM ET AL 3,404,668

FUEL INJECTION PUMP

Filed March 28, 1966 2 Sheets-Sheet 1

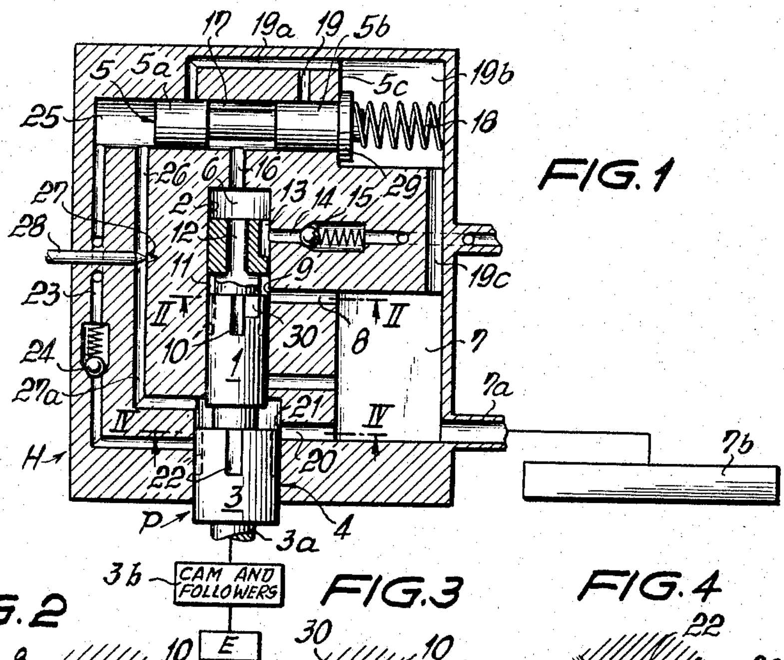

FIG. 1

FIG. 2

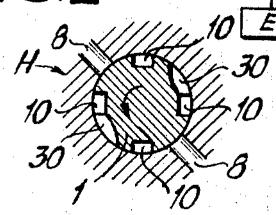

FIG. 3

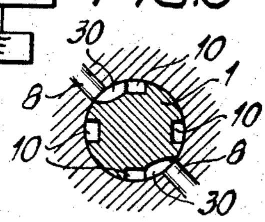

FIG. 4

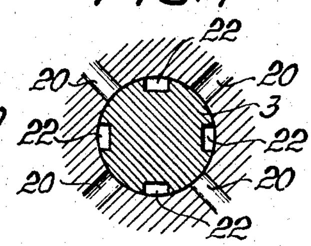

FIG. 5

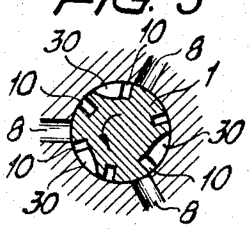

FIG. 6

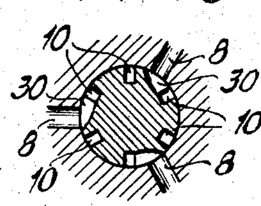

FIG. 7

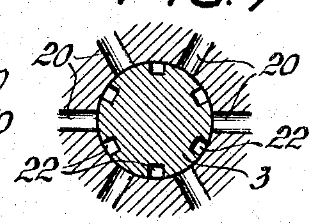

FIG. 8

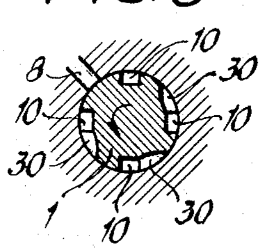

FIG. 9

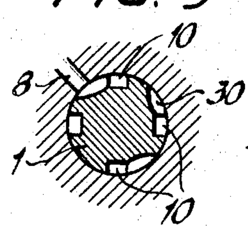

FIG. 10

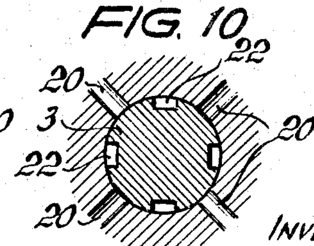

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FIG. 11

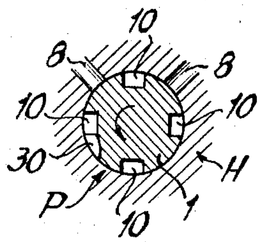


FIG. 12

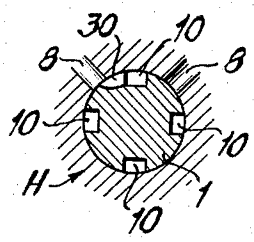


FIG. 13

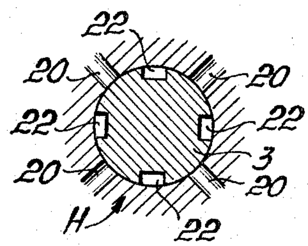


FIG. 14

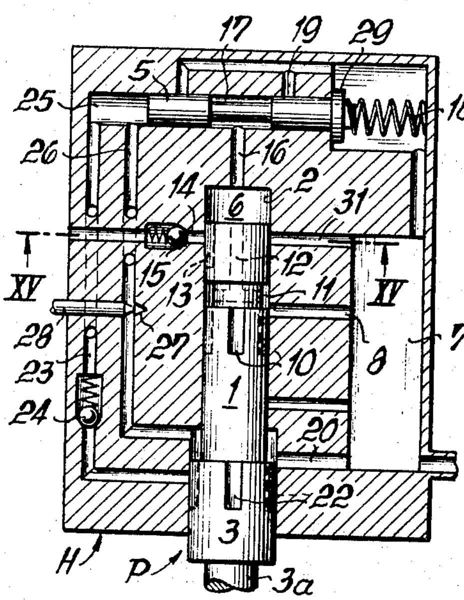


FIG. 15

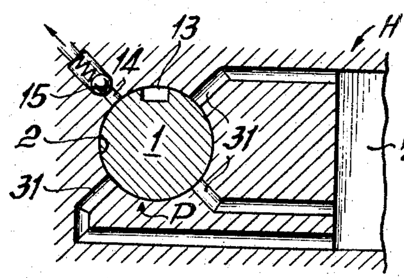


FIG. 16

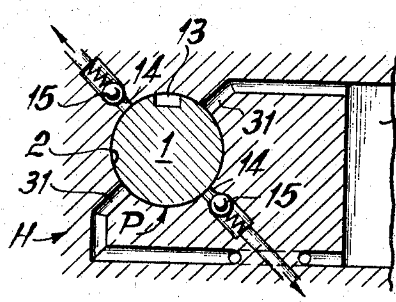


FIG. 17

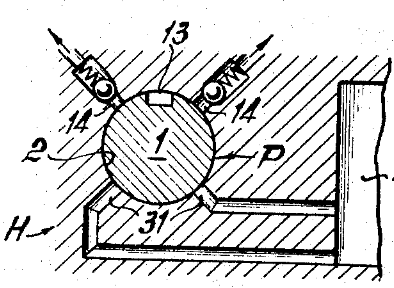
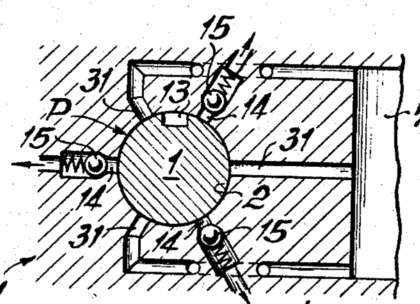


FIG. 18



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FUEL INJECTION PUMP

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7 Claims. (Cl. 123—139)

ABSTRACT OF THE DISCLOSURE

In a fuel injection pump for supplying fuel to an internal combustion engine having a first number of cylinders, a piston turnably and reciprocatably arranged in a working chamber of the pump, drive means for rotating the piston about its axis and for reciprocating the same during each revolution along second number of working and suction strokes greater than the above mentioned first number and including a first number of effective working strokes during which fuel is fed from the working chamber to the cylinders of the engine and a number of ineffecting working strokes during which fuel is fed from the working chamber to a source of fuel connected to the working chamber.

The present invention relates to fuel injection pumps for internal combustion engines. More particularly, the invention relates to improvements in a fuel injection pump of the type which is capable of supplying accurately metered quantities of liquid fuel to one or more cylinders of an internal combustion engine.

It is well known that the demand for four-, six- and eight-cylinder engines considerably exceeds the demand for one-, two- or three-cylinder engines. Therefore, it is highly desirable to construct fuel injection pumps for use in connection with more popular internal combustion engines in such a way that their component parts, eventually with minor alterations, may be used in pumps for less popular engines, for example, in pumps which are used to inject fuel into the cylinders of one-, two- or three-cylinder engines. This is advisable for reasons of economy since a supply of pumps for four-, six- or eight-cylinder engines is more likely to be available and it is clear that the manufacturer or assembler or less popular engines can save considerable money, storage space and time if parts of a more frequently demanded fuel injection pump can be converted for assembly in a pump which is useful with less popular internal combustion engines. Of course, greatest economies can be achieved if a pump for use with four- or six-cylinder engines can be simply converted for use with one-, two- or three-cylinder engines.

Accordingly, it is an important object of the present invention to provide a novel and improved fuel injection pump for use with engines having a low number of cylinders which may be produced by effecting minor changes in fuel injection pumps which are constructed and assembled for use in connection with more popular engines having a relatively large number of cylinders.

Another object of the invention is to provide a fuel injection pump which, though originally produced for use in connection with a four-, six- or eight-cylinder engine, can be rapidly converted for use in connection with one-, two- or three-cylinder engines without necessitating any major alterations in its appearance, operation and/or space requirements.

A further object of the invention is to provide a pump of the above outlined characteristics wherein the amounts of metered fuel may be regulated with utmost precision so that the pump invariably delivers to the cylinder or cylinders of an internal combustion engine fuel at a rate which is an accurate function of the rotational speed of the engine.

Still another object of the invention is to provide a novel pump housing or body which may be utilized in a pump of the above defined type.

An additional object of the invention is to provide a novel pumping member which can be used in the improved pump to draw, meter and distribute jets of fuel to one or more cylinders of an internal combustion engine.

A further object of the invention is to provide a novel pump which comprises a so-called liquid stop and wherein the liquid stop may be adjusted or regulated with the same degree of precision as in a pump which is used in connection with an engine having a relatively large number of cylinders despite the fact that the novel pump serves to meet the requirements of an engine having a relatively small number of cylinders.

A concomitant object of the invention is to provide the improved pump with a novel system of passages or channels which convey fuel to and from the working chamber of the pump.

Still another object of the instant invention is to achieve unexpectedly high economies in storage facilities, manufacturing cost and assembly cost of fuel injection pumps for engines having a relatively small number of cylinders.

A further object of the instant invention is to provide a novel method of converting fuel injection pumps for use with engines having a first number of cylinders into fuel injection pumps capable of being used with engines having a second number of cylinders.

Another object of the invention is to provide fuel injection pumps for use with engines having one, two, three, four, six, eight or another number of cylinders and to construct such pumps in a way to insure that at least a majority of components useful in one type of pumps can be used, without any changes or with minimal changes, in each other type of pumps.

Briefly stated, one feature of our present invention resides in the provision of a fuel injection pump for supplying fuel to an internal combustion engine having a predetermined (first) number of cylinders, for example, three cylinders. The pump comprises a source of fuel, housing means defining a working chamber, intake channel means adapted to connect the source with the working chamber, discharge channel means adapted to communicate with the working chamber and comprising a first number of separate fuel lines each of which can supply fuel to a separate cylinder of the engine (thus, if the engine comprises three cylinders, the discharge channel means comprises three fuel lines which are machined into or otherwise provided in the housing means), a pumping member movable in the housing means to perform alternating suction and working strokes to respectively draw fuel from the source via intake channel means and into the working chamber and to expel fuel from the working chamber, drive means for imparting to the pumping member, at a rate proportional with the speed of the engine, a series of consecutive movements each of which includes a second number of suction strokes and a second number of working strokes, this second number being higher than and preferably a whole multiple of the first number and the second number of working strokes including a first number of effective working strokes each of which results in expulsion of fuel from the working chamber into the discharge channel means (i.e., into one fuel line), the remainder of the second number of working strokes being ineffective insofar as the expulsion of fuel from the working chamber into the discharge channel means is concerned, and overflow channel means for returning from the working chamber to the source all such fuel which is expelled while the pumping member performs an ineffective working stroke.

In accordance with a presently preferred embodiment of our invention, the pumping member may comprise a rotary piston which is reciprocable in a blind bore of the housing means and each movement of the piston includes a full revolution during which the piston performs an equal number of suction or working strokes, i.e., the number of working strokes during each revolution equals the number of suction strokes during the same revolution and such number is preferably a whole multiple of the number of cylinders in the engine which is associated with the fuel injection pump.

Our novel method can be decsribed as relating to injection of fuel into cylinders of an engine which has a relatively small number of cylinders. Basically, the method comprises the steps of admitting into and expelling from the working chamber of a fuel injection pump (which was built for use in connection with an engine having a relatively large number of cylinders), during each revolution of the engine, metered quantities of fuel at such a frequency that the number of admissions and expulsions exceeds the number of cylinders in the engine having a relatively small number of cylinders, and introducing—during each revolution of the engine—one metered quantity into each cylinder so that at least one metered quantity of fuel expelled from the working chamber during any given revolution of the engine is not admitted into a cylinder. As a rule, the relatively low number of cylinders will be less than four.

The novel features which are considered as characteristic of the invention are set forth in particular in the appended claims. The improved fuel injection pump itself, however, both as to its construction and its mode of operation, together with additional features and advantages thereof, will be best understood upon perusal of the following detailed description of certain specific embodiments with reference to the accompanying drawings, in which:

FIG. 1 is an axial section through a fuel injection pump which is intended for use in a two-cylinder engine but whose pumping member is rotated at a speed corresponding to that of a pumping member in a fuel injection pump for a four-cylinder engine;

FIG. 2 is an enlarged fragmentary section as seen in the direction of arrows from the line II—II of FIG. 1 and illustrates the pumping member of the fuel injection pump in a first angular position;

FIG. 3 is a similar fragmentary section and illustrates the pumping member in a different angular position;

FIG. 4 is an enlarged fragmentary section as seen in the direction of arrows from the line IV—IV of FIG. 1;

FIG. 5 is a section similar to that of FIG. 2 but showing the pumping member and the housing of a fuel injection pump for use in a three-cylinder engine whereby the pumping member is driven at a speed corresponding to that of the pumping member in a pump for a six-cylinder engine;

FIG. 6 is a section similar to that of FIG. 5 but showing the pumping member in an angular position corresponding to that of the pumping member shown in FIG. 3;

FIG. 7 is a section similar to that of FIG. 4 but showing the pumping member and housing of the pump shown in FIGS. 5 and 6;

FIG. 8 is a section similar to that of FIG. 2 but showing the pumping member and the housing of a fuel injection pump for a one-cylinder engine whereby by the pumping member is rotated at a speed corresponding to that of the pumping member in a pump for a four-cylinder engine;

FIG. 9 illustrates the structure of FIG. 8 with the pumping member in an angular position corresponding to that of the pumping member shown in FIG. 3;

FIG. 10 is a section similar to that of FIG. 4 but showing the pumping member and housing of FIGS 8 and 9;

FIG. 11 is a section through the housing and pumping member of a fuel injection pump for a two-cylinder engine wherein the injection of fuel takes place at unequal intervals;

FIG. 12 is a similar section but showing the pumping member of FIG. 11 in a different angular position;

FIG. 13 is a section similar to that of FIG. 4 but showing the parts of FIGS. 11 and 12;

FIG. 14 is an axial section through a further fuel injection pump for use in connection with a one-cylinder engine;

FIG. 15 is an enlarged fragmentary section as seen in the direction of arrows from the line XV—XV of FIG. 14;

FIG. 16 is a similar section but showing the pumping member and housing of a fuel injection pump for use in connection with a two-cylinder engine, the pump being constructed to deliver fuel at regular intervals;

FIG. 17 is a section similar to that of FIG. 16 but showing a modified housing which is utilized in pumps for use in connection with two-cylinder engines receiving fuel at irregular intervals; and

FIG. 18 is a section similar to that of FIG. 15, 16 or 17 but showing the housing of a pump which can be used in connection with a three-cylinder engine.

Referring first to FIG. 1, there is illustrated a fuel injection pump which comprises a housing H and a pumping member here shown as a multistage piston P reciprocable in a composite blind bore **2** of the housing. The bore **2** includes a first or working chamber **6** for a smaller-diameter first stage **1** of the piston, and a second chamber **4** (hereinafter called cylinder chamber) for a larger-diameter second stage **3**. The first stage **1** serves to inject fuel to the cylinders of an engine, not shown, and the second stage **3** serves to control the operation of a pressure-responsive regulating valve **5**. The shaft **3***a* at the outer axial end of the stage **3** forms part of a drive which causes the piston P to rotate and to simultaneously reciprocate with reference to the housing H at a rate depending on the speed of the engine and in a manner well known from the art of fuel injection pumps. For example, the shaft **3***a* may be coupled to a cam having an annular face provided with four equidistant lobes and being driven by a rotary output shaft which receives motion from the engine. Follower rollers which are rotatable about fixed axes track the face of the cam and cause the piston P to move up and down while the piston rotates with the output shaft. The cam and follower assembly **3***b* and the engine E are shown diagrammatically in the lower part of FIG. 1.

The working chamber **6** can receive fuel from a suction space **7** which is formed in the housing H and is connected with a supply conduit **7***a* leading to a suitable source **7***b* of fuel, i.e., a fuel pump which draws fuel from a tank. The working chamber **6** receives fuel from the space **7** when the piston P performs a suction (downward) stroke. Such fuel can flow from the space **7** through two suction ports **8** whose outlets **9** are located in the surface surrounding the bore **2**, four axially extending grooves **10** machined into the periphery of the stage **1**, through an annular groove **11** which is also machined into the periphery of the stage **1** and communicates with the axial grooves **10**, and through a T-shaped bore **12** formed in the stage **1** and connecting the annular groove **11** with the working chamber **6**. The ports **8**, grooves **10**, **11** and bore **12** together constitute a system of intake channels which can convey fuel from the source **7***b* to the working chamber **6** in response to each suction stroke of the piston or pumping member P.

When the piston performs a working (upward) stroke, its stage **1** expels fuel from the working chamber **6** through an axially extending peripheral distributor groove **13** and into one of two radially extending fuel lines **14**. Each of these fuel lines **14** leads to the intake valve of an engine cylinder and contains a ball check valve **15**. The groove **13** and the lines **14** together form a system of discharge channels which can convey fuel from the chamber **6** to the cylinders of the internal combustion engine E when the piston P performs effective working strokes.

A relief bore **16** connects the working chamber **6** with an annular space **17** between the plungers **5***a*, **5***b* of the regulating valve **5**. The annular space **17** can connect the relief bore **16** with a duct **19** communicating with a duct **19***a* which discharges into a compartment **19***b* accommodating a return spring **18**. The spring **18** bears against a head or boss **29** of the valve **5** and urges the boss into sealing engagement with the internal surface **5***c* of the housing H. The compartment **19***b* communicates with the suction space **7** through a return bore **19***c*. When the valve **5** is shifted to the right by fuel pressure in an auxiliary chamber **25** of the housing H and covers a predetermined distance, the relief bore **16** can communicate with the duct **19** and the injection of fuel into one of the fuel lines **14** is terminated in a fully automatic way. The left-hand plunger **5***a* can also permit fuel to flow from the auxiliary chamber **25** into the duct **19***a* and thence into the suction space **7**.

The larger stage **3** can receive fuel through suction ports **20** whose inlets communicate with the suction space **7** and whose outlets **21** communicate with the cylinder chamber **4**. When the piston P performs a working stroke, fuel filling the cylinder chamber **4** can escape through one of several axially extending grooves **22** machined into the periphery of the stage **3**, such fuel then flowing into a connecting bore **23** which contains a ball check valve **24** and discharges into the auxiliary chamber **25**. Thus, the valve **5** can be displaced against the bias of the return spring **18** by the pressure of fuel which is supplied via connecting bore **23**. The spring **18** pushes the valve **5** back in a direction to the left, as viewed in FIG. 1, not only when the piston P performs a suction stroke but also when the piston reaches and dwells at the upper end of its working stroke. The return movement of the valve **5** under the bias of the spring **18** is controlled by a throttle **27**, **28** which regulates the flow of fuel through a throttling passage **27***a* connecting the chambers **4** and **25**. The needle **28** of the throttle extends from the housing H so that it may be adjusted to change the rate of fuel flow through the passage **27***a*.

While the piston P rotates at less than a predetermined speed, the spring **18** invariably returns the boss **29** into abutment with the surface **5***c* before the piston begins the next working stroke. However, if the speed of the piston P exceeds such predetermined speed, the valve **5** has no time to return to its starting position and the auxiliary chamber **25** receives from the connecting bore **23** fuel before the boss **29** returns into abutment with the surface **5***c*. Consequently, a shorter interval of time is needed to shift the valve **5** against the bias of the spring **18** to such an extent that the annular space **17** connects the relief bore **16** with the duct **19**, i.e., the injection of fuel into one of the fuel lines **14** is terminated sooner and the corresponding cylinder of the engine receives less fuel. In other words, fuel admitted by the connecting bore **23** acts not unlike a liquid stop and reduces the amount of fuel which is injected into a cylinder. The reduction in the amounts of injected fuel is proportional with the rotational speed of the piston P, i.e., with the extent to which the speed of the piston exceeds the aforementioned predetermined speed.

In the pump of FIG. 1, the first stage **1** is formed with four equidistant axial grooves **10** (see FIG. 2), and the stage **3** is formed with four equidistant axial grooves **22** (see FIG. 4). Thus, during each of its full revolutions, the piston P will perform four suction strokes and four working strokes. If the pump of FIG. 1 is to supply fuel to the cylinders of a two-cylinder engine E, two of the four working strokes must be ineffective, i.e., the fuel expelled from the working chamber **6** during two of four working strokes performed by the piston P during each of its revolutions must be returned into the suction space **7** or into another space defined by a suitable receptacle connected to or spaced from the housing H.

The pump of FIG. 1 can produce such ineffective working strokes due to the provision of two overflow channels **30** (hereinafter called recesses) which are machined into the periphery of the stage **1** and each of which communicates with one of the axial grooves **10**. FIG. 2 shows that the recesses **30** are located substantially diametrically opposite each other and that each thereof is adjacent to one side of the corresponding groove **10**, as seen in the circumferential direction of the stage **1**. When the piston P performs alternate working strokes, the recesses **30** permit the working chamber **6** to remain in communication with the suction space **7** through the corresponding axial groove **10**, annular groove **11**, bore **12** and suction ports **8**. As shown in FIGS. 2 and 3, the housing H is formed with two ports **8** which are located diametrically opposite each other so that, when the piston P is to perform an ineffective working stroke, each recess **30** communicates with one of the ports **8**.

The operation of the auxiliary pump including the stage **3** is not affected by the recesses **30**.

The distribution of the ports **8** and **20**, axial grooves **10** and **22**, and recesses **30** in the pump of FIG. 1 is such that the stage **1** can inject fuel at regular intervals. The common plane of the grooves **10** which are not connected with recesses **30** makes an angle of 90 degrees with the common plane of the remaining two grooves **10**. The grooves **22** and suction ports **20** are respectively equidistant from each other, and the ports **8** communicate with the bore **2** at points which are located diametrically opposite each other. The housing H is formed with only two fuel lines **14** which communicate with the bore **2** at points located diametrically opposite each other. FIG. 1 merely shows one of the lines **14**.

FIG. 2 illustrates the stage **1** in an angular position while the piston P performs an effective working stroke. The grooves **10** and recesses **30** cannot communicate with the suction ports **8**, and the groove **13** connects the working chamber **6** with one of the fuel lines **14**. Therefore, and since the stage **1** moves upwardly, as viewed in FIG. 1, the volume of the chamber **6** is reduced and a measured amount of fuel is compelled to overcome the bias of the spring in the check valve **15** and to flow on to the intake valve of the corresponding cylinder of the engine E.

FIG. 3 illustrates the stage **1** in an angular position when the piston P performs an ineffective working stroke. The two suction ports **8** communicate with the recesses **30** and with the corresponding axial grooves **10** so that, as the piston P moves upwardly, the stage **1** expels fuel from the working chamber **6** via bore **12**, annular groove **11**, the two grooves **10**, recesses **30**, suction ports **8**, and back into the space **7**. In FIG. 3, the stage 1 is displaced through 90 degrees with with reference to the position shown in FIG. 2. While the two grooves **10** communicate with the suction ports **8** via recesses **30**, the groove **13** does not communicate with either of the two fuel lines **14**.

The auxiliary pump operates in a manner as shown in FIG. 4. Thus, the grooves **22** of the stage **3** are sealed from the suction ports **20** whenever the piston P performs a working stroke so that the stage **3** expels fuel into the connecting bore **23** and hence into the auxiliary chamber **25**. During each suction stroke, the grooves **22** connect the cylinder chamber **4** with the suction ports **20**, and and one of the grooves **22** connects the chamber **4** with the connecting bore **23** during each working stroke. The bore **23**, chamber **25**, duct **19***a*, compartment **19***b* and bore **19***c* together form a composite passage which connects the chamber **4** with the space **7** and wherein the flow of fuel is regulated by the valve **5**.

The pump of FIGS. 1 to 4 may be used, with minor modifications, to supply fuel to the cylinders of a four-cylinder internal combustion engine. The rotational speed of the piston P can remain the same which is important because the aforementioned cam-and-follower assembly

3b in the drive for the piston P can remain unchanged. The piston will perform four suction strokes and four working strokes in response to each full revolution, regardless of whether it is used in a fuel injection pump for two-cylinder engines E or four-cylinder engines. The housing of both types of pumps will be substantially the same, i.e., all that is necessary to use the housing H of FIG. 1 in a pump for a four-cylinder engine is to provide this housing with four equidistant fuel lines **14** and with four check valves **15**. The auxiliary pump including the stage **3** remains unchanged, and the same holds true for the regulating valve **5**. For use in a pump for four-cylinder engines, the stage **1** of the piston P will be without recesses **30**. The throttling passage **27***a*, the throttle **27**, **28** and the number of working and suction strokes per revolution of the piston will remain unchanged.

Known fuel injection pumps which are used for engines having a large number of cylinders and which comprise a liquid stop operating on the samme principle as the liquid stop in the pump of FIG. 1 are provided with throttles which, as a rule, have a relatively large cross-sectional area. This reduces the likelihood of clogging by impurities which are present in the fuel and allows for more precise regulation of the injected amounts of fuel. It is to be borne in mind that a relatively small adjustment of the throttle can bring about a substantial change in the amounts of injected fuel if the passage which is throttled is of relatively small cross sectional area. In other words, and if the cross-sectional area of the passage which is controlled by a throttle is rather large, minor adjustments in the rate of flow of fuel through such a relatively large passage will result in minor changes in the amounts of injected fuel which is highly desirable for the aforediscussed reason, i.e., that the amounts of injected fuel can be regulated with a higher degree of precision.

In a fuel injection pump which must supply metered quantities of fuel to a six-cylinder engine, the revolving piston performs a large number of working strokes during each of its revolutions (i.e., six working strokes), whereas a pump which is built specifically for use with three-cylinder engines comprises a piston which performs only three working strokes. The required cross-sectional area of the throttled passage is inversely proportional to the length of time intervals between successive working strokes, i.e., a pump for a six-cylinder engine can have a throttling passage whose cross-sectional area is twice the cross-sectional area of the throttling passage in a conventional pump for use with three-cylinder engines. This holds true for all types of fuel injection pumps including those which inject fuel into a manifold and wherein the metering operation is controlled by a suction throttle.

It will be seen that a pump which is constructed in accordance with our present invention can be provided with a more sensitive and more accurate liquid stop because the number of working strokes performed by the revolving piston per each revolution is much higher than in presently known fuel injection pumps for engines having a relatively small number of cylinders. This is due to the fact that, in using a fuel injection pump in connection with engines having one, two or three cylinders, the number of working strokes (including effective and ineffective working strokes) per full revolution of the piston is a whole multiple of the number of cylinders. The throttling passage **27** has a cross-sectional area which suffices to prevent clogging by impurities and which allows for highly precise selection of the exact moment when, at a given r.p.m. of the engine E, the regulating valve **5** allows for flow of fuel from the relief bore **16** into the duct **19** and back to the source, e.g., to the suction space **7**.

FIGS. 5, 6 and 7 illustrate certain details of a fuel injection pump which is used in connection with a three-cylinder engine. The stage **1** of the piston is formed with six equidistant axial grooves **10** and with three equidistant overflow channels or recesses **30** each of which communicates with one of the grooves **10**, i.e., grooves

10 communicating with recesses **30** alternate with the remaining three grooves **10**. The stage **3** has six equidistant axial grooves **22** and the housing of this pump is provided with six equidistant suction ports **20**. During each of its revolutions, the piston which includes the stages **1** and **3** of FIGS. 5 to 7 performs six suction strokes and six working strokes. The housing of this pump has three equidistant fuel lines **14**, not shown in FIGS. 5 to 7. FIG. 5 shows the stage **1** in an angular position when the piston performs an effective working stroke. The grooves **10** and recesses **30** cannot communicate with the suction ports **8** and the groove **13** (see FIG. 1) communicates with one of the three fuel lines **14**. In FIG. 6, the stage **1** is shown in an angular position when the piston performs an ineffective working stroke. Each of the three recesses **30** communicates with one of the three suction ports **8** so that the stage **1** can expel fuel from the working chamber **8** (see FIG. 1) back into the space **7**.

A pump which is used in connection with a six-cylinder engine is very similar to the pump of FIGS. 5 to 7. All that is necessary is to replace the piston with one whose stage **1** has six equidistant grooves **10** but is without recesses **30**. The housing of the pump for a six-cylinder engine may have a total of six equidistant suction ports **8**. Otherwise, the pump for a six-cylinder engine operates in the same way as described in connection with FIG. 1.

Referring now to FIGS. 8 to 10, there are shown certain details of a pump which is used in connection with a one-cylinder engine. The piston which includes the stages **1** and **3** respectively shown in FIGS. 8–9 and 10 performs four suction strokes and four working strokes during each of its full revolutions; however, three of the working strokes are ineffective because the stage **1** is formed with four axial grooves **10** and with three overflow channels or recesses **30**. The housing of this pump has a single port **8** which communicates with a different recess **30** during three of each four successive working strokes. Furthermore, the housing has a single fuel line **14**, not shown in FIGS. 8 to 10. The stage **3** (see FIG. 10) is constructed in the same way as described in connection with FIG. 4. In FIG. 8, the stage **1** is shown in an angular position when the piston performs an effective working stroke because the grooves **10** and recesses **30** cannot communicate with the single port **8** and the piston is assumed to expel fuel from the chamber **6** (see FIG. 1) through the groove **13** and into the single fuel line **14**. In FIG. 9, the stage **1** assumes an angular position in which the piston performs an ineffective working stroke because one of the recesses **30** communicates with the port **8**.

The advantages of the fuel injection pump for a one-cylinder engine are the same as those described in connection with FIGS. 1 to 4. The free cross-sectional area of the throttle **27**, **28** which is used in this pump is four times the size of the cross-sectional area of the throttle in a fuel injection pump which is built especially for a one-cylinder engine.

FIGS. 11, 12 and 13 again show certain parts of a fuel injection pump for a two-cylinder engine. This pump is very similar to that of FIGS. 1 to 4 with the exception that the stage **1** of the piston has a single overflow channel or recess **30** and that the annular distance between the ports **8** is only 90 degrees. Thus, in making a full revolution, the piston will perform two consecutive ineffective working strokes followed by two effective working strokes. Stated in another way, the angular distance between a first position in which the piston performs a first effective working stroke and a second position in which the piston performs the next effective working stroke is 270 degrees, and the angular distance between the second and first positions (as seen in the direction in which the piston rotates) is only 90 degrees. When the piston rotates, the recess **30** first sweeps one of the ports **8** during a first working stroke to make such stroke ineffective and thereupon sweeps the second port **8** during the next working stroke, again to render such second working stroke ineffective. The two ineffective working strokes are followed by two effective working strokes. Of course, the angular distance between the two fuel lines of the housing H shown in FIGS. 11 to 13 must be 90 degrees.

The auxiliary pump including the stage **3** (see FIG. 13) is constructed and operates in the same way as described in connection with FIG. 4.

Common to all of the fuel injection pumps shown in FIGS. 1 through 13 is the feature that their pistons P perform ineffective working strokes when the working chamber **6** for the first stage **1** is free to communicate with the suction space **7** through such overflow channel or channels which, during the suction strokes of the piston, serve a part of intake channels for admission of fuel to the chamber **6**. In other words, overflow channels have portions which are common to intake channels and such common portions are provided in part in the periphery of the piston.

FIG. 14 shows a modified fuel injection pump which differentiates from the previously described pumps in that fuel which escapes or is expelled from the working chamber **6** when the piston P is to perform an ineffective working stroke flows through the distributor groove **13** of the first stage **1**, i.e., along the same route as such fuel which is actually injected into a cylinder. In other words, the overflow channel means and the discharge channel means of the pump shown in FIG. 14 have a common portion **13** which is machined into the periphery of the stage **1**. This is achieved by forming the housing H with three overflow bores **31** which are located at the level of the fuel line **14** and are distributed circumferentially of the stage **1** in a manner as shown in FIG. 15. The pump of FIG. 14 is intended for use in connection with a one-cylinder engine and its housing H has a single fuel line **14**. Since the piston P is assumed to perform four suction strokes and four working strokes during each of its full revolutions, the housing H must be provided with three equidistant overflow bores **31** and with a single fuel line **14** having a ball check valve **15**. Three of the four working strokes are ineffective because, during each such ineffective working stroke, the axially extending distributor groove **13** of the stage **1** communicates with one of the overflow bores **31** so that fuel expelled from the working chamber **6** can flow back into the suction space **7**. All remaining details of the fuel injection pump shown in FIGS. 14 and 15 are the same as those of the pump which was described in connection with FIG. 1.

FIG. 15 shows the stage **1** of the modified pump in an angular position when the piston P performs a suction stroke. The groove **13** is out of communication with the fuel line **14** or with the overflow bores **31** and the bore **12** (see FIG. 14) communicates with the suction space **7** via ports **8**, annular groove **11**, and axial grooves **10**.

FIG. 16 shows a portion of a different pump which is similar to the pump of FIGS. 14 and 15 with the exception that its housing H comprises two fuel lines **14** alternating with two overflow bores **31**. This pump is intended for use in connection with a two-cylinder engine and each second working stroke of the piston P is ineffective. The fuel lines **14** and the overflow bores **31** are respectively located diametrically opposite each other and each overflow bore **31** is located exactly midway between the fuel lines **14**. During each of its revolutions, the piston P of the pump shown in FIG. 16 performs four working strokes (including two effective working strokes which alternate with two ineffective working strokes) and four suction strokes.

FIG. 17 shows a portion of a further fuel injection pump wherein the housing H comprises the same number of overflow bores **31** and fuel lines **14** as in FIG. 16. However, the angular distance between the overflow bores **31** is only 90 degrees and the angular distance between the fuel lines **14** is also only 90 degrees. Therefore, this pump will operate in the same way as described in connection with FIGS. 11 and 12, i.e., two effective working strokes (when the distributor groove **13** communicates

with one of the fuel lines **14**) will be followed by two ineffective working strokes when the distributor groove **13** communicates with one of the overflow bores **31**. This pump is used in connection with a two-cylinder engine and its piston P performs a total of four working strokes and four suction strokes during each full revolution of its drive shaft *3a*. The remaining parts of this pump are constructed in the same way as shown in FIG. 14.

FIG. 18 illustrates a portion of a further pump wherein the housing H comprises three equidistant fuel lines **14** alternating with three equidistant overflow bores **31**. This pump is used in connection with a three-cylinder engine but can be readily modified to be used in connection with a six-cylinder engine. This can be done by replacing the housing H with a housing having no overflow bores **31** but provided with six equidistant fuel lines **14**. The angular distance between the adjoining fuel lines **14** and overflow bores **31** shown in FIG. 18 is 60 degrees. When the piston P completes a full revolution, it has performed a total of six working strokes (including three effective strokes and three ineffective working strokes) and six suction strokes.

The advantages of the pumps shown in FIGS. 14 and 18 are the same as those which were described in connection with FIG. 1. Thus, a pump which is used in connection with a three-cylinder or two-cylinder engine can be rapidly converted into a pump for use with a six-cylinder or four-cylinder engine. Also, a pump in use for a one-cylinder engine can be converted for use in connection with a four-cylinder engine.

Without further analysis, the foregoing will so fully reveal the gist of the present invention that others, can, by applying current knowledge, readily adapt it for various applications without omitting features which fairly constitute essential characteristics of the generic and specific aspects of our contribution to the art and, therefore, such adaptaions should and are intended to be comprehended within the meaning and range of equivalence of the following claims.

What is claimed as new and desired to be protected by Letters Patent is:

1. In a fuel injection pump for supplying fuel to an internal combustion engine having a predetermined number of cylinders, in combination: a source of fuel; housing means defining a working chamber; passage means extending from said source of fuel to said working chamber; discharge channel means adapted to communicate with said working chamber; a piston mounted in said working chamber rotatable about its axis and axially reciprocable to perform alternate suction and working strokes to respectively draw fuel from said passage means into said chamber and to expel fuel from said chamber into said discharge channel means; a plurality of angularly displaced axial grooves formed in the periphery of said piston and equal in number to a multiple of said predetermined number and respectively adapted to establish communication between said chamber and said passage means during each suction stroke of said piston; a plurality of recesses connected respectively to some of said grooves and each extending from the respective groove in a direction opposite to the direction of rotation of said piston for such a distance as to establish also during the respective working stroke communication between said chamber and said passage means so as to render the pump ineffective during the respective working stroke; and means for rotating at a speed proportional to the speed of the engine said piston about its axis and to reciprocate said piston during each revolution to a number of suction and working strokes which is equal to the number of axial grooves formed in the periphery of said piston.

2. A structure as set forth in claim **1**, wherein said discharge channel means comprises a substantially axially extending distributor groove provided in the periphery of said piston and communicating with said working chamber, and fuel line means provided in said housing means and positioned to communicate with said groove when said piston performs an effective working stroke, the number of said fuel line means being equal to said first number and each such fuel line means being arranged to feed fuel to a separate cylinder of the engine, said passage means comprising bore means provided in said housing means to connect said recess with said source when said piston performs an ineffective working stroke, the number of said bore means being equal to the difference between said second and first numbers.

3. A structure as set forth in claim **1**, wherein said piston is a multistage piston having a first stage which is arranged to draw fuel into and to expel fuel from said working chamber.

4. A structure as set forth in claim **3**, wherein said piston comprises a second stage operating in a second chamber of said housing means, said second chamber being connected with said source and further comprising regulating valve means responsive to pressure of fuel expelled from said second chamber by said second stage to provide a liquid stop which reduces the quantity of fuel entering said discharge channel means when the rotational speed of said piston exceeds a predetermined value.

5. A structure as set forth in claim **4**, wherein said regulating valve means comprises a slide valve movable in an auxiliary chamber provided in said housing means, said housing means further having a connecting bore for conveying fuel from said second chamber to said auxiliary chamber and a throttling passage connecting said last named chambers.

6. A structure as set forth in claim **5**, further comprising means for regulating the flow of fuel through said throttling passage.

7. A structure as set forth in claim **4**, wherein said housing means is provided with a relief bore connected to said working chamber and said valve means is arranged to connect said relief passage with said source to thus terminate the outflow of fuel into said discharge channel means in the course of an effective working stroke when said second stage expels a predetermined amount of fuel from said second chamber.

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