

Dec. 3, 1940.

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2,223,757

PUMP

Filed Aug. 10, 1938

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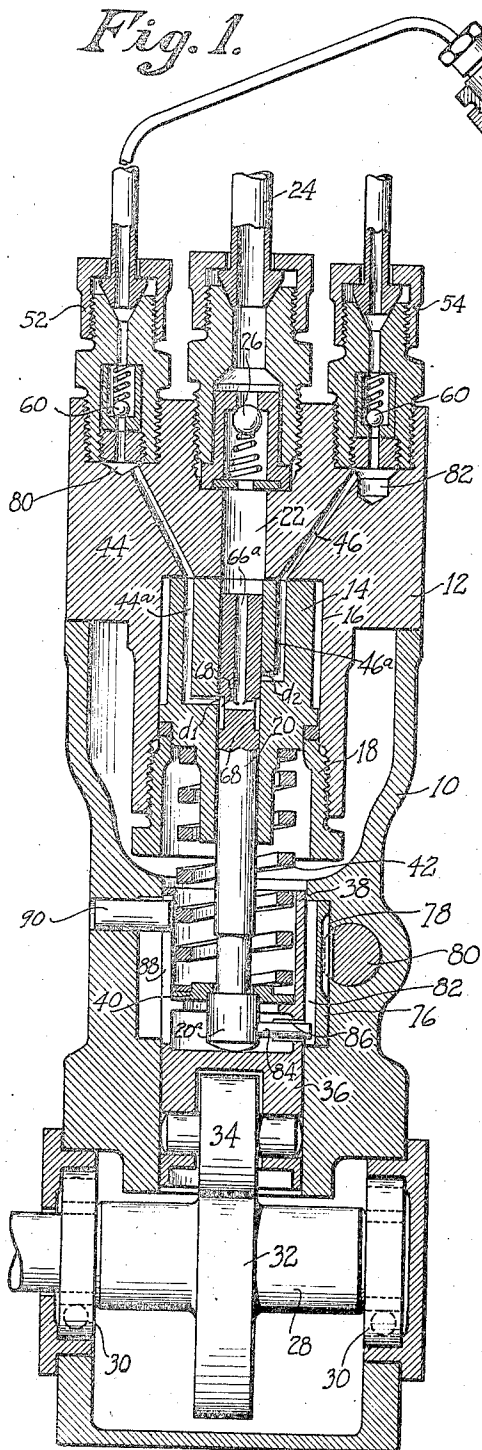
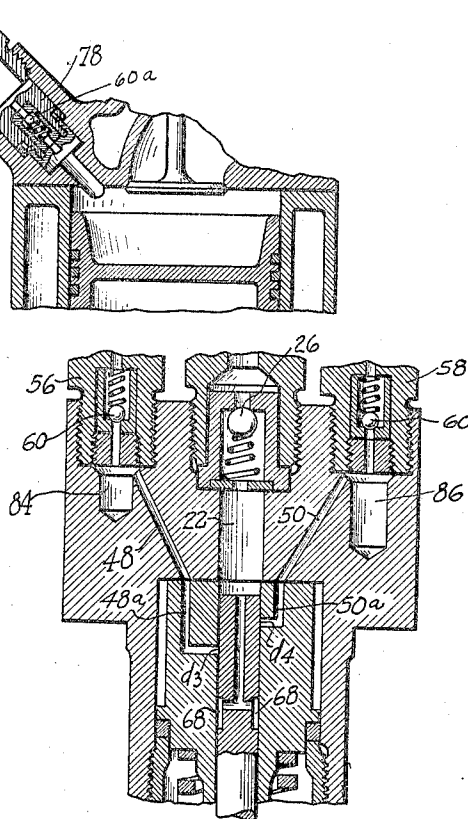
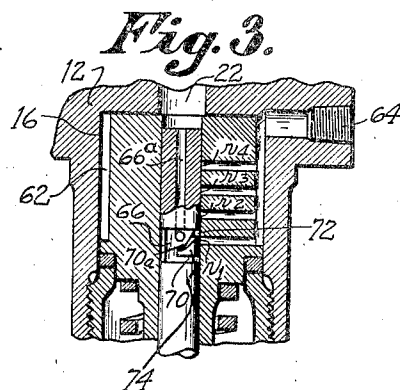


Fig. 2.



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Fig. 4.

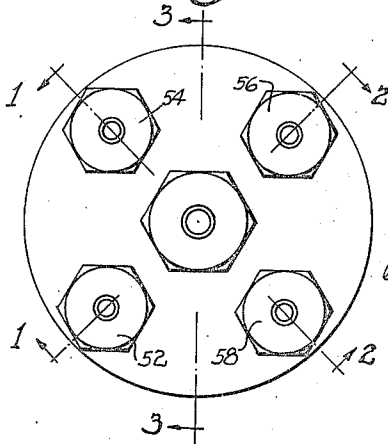


Fig. 5.

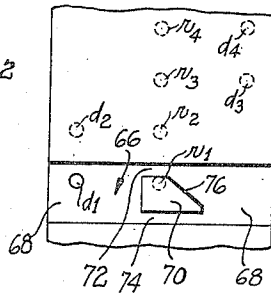


Fig. 6.

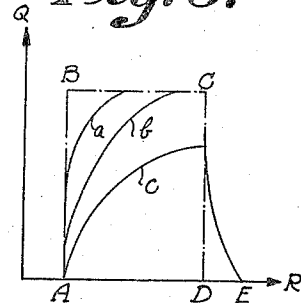


Fig. 7.

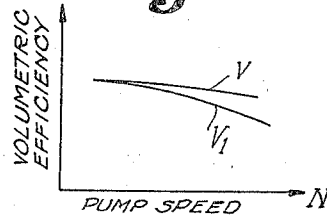


Fig. 8.

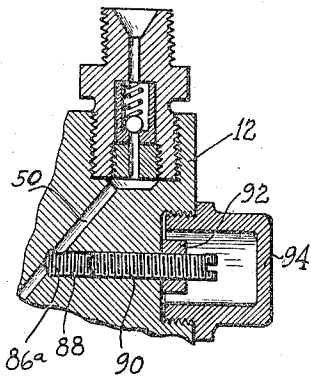


Fig. 10.

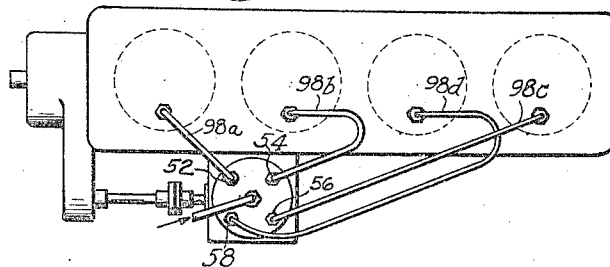


Fig. 11.

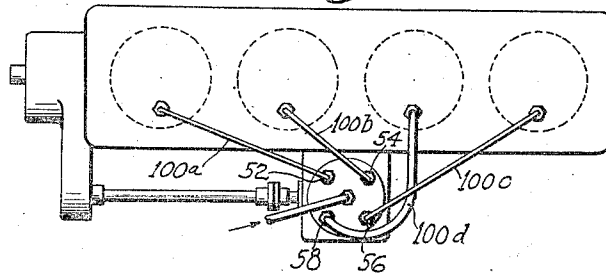
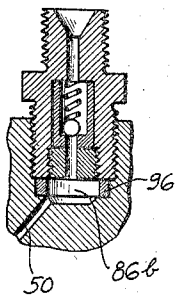


Fig. 9.



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 In Sweden September 28, 1937

9 Claims. (Cl. 123—139)

The present invention relates to plunger pumps and has particular reference to plunger pumps of the type in which liquid is delivered by the pump from a common pump cylinder or pump chamber during one delivery stroke of the plunger to a plurality of places of consumption such for example as the injection nozzles of an internal combustion engine to which liquid fuel is delivered at different times by the pump.

In pumps of the above type a different volume of liquid in the pump chamber is subjected to pressure during each individual delivery period of the pump plunger during the delivery stroke of the plunger, since the liquid delivered during each successive delivery period has to be taken from a single charge of liquid in the pump at the commencement of the delivery stroke, which charge is successively diminished by the deliveries effected during the successive delivery periods.

The general object of the present invention contemplates the provision of improved structure for supplying liquid to a plurality of points of consumption, particularly from a common pump cylinder in which the several quantities of liquid subjected to delivery pressure during different delivery periods, which quantities will for convenience hereinafter be referred to as columns of liquid or liquid columns, are subjected to the influence of compensating factors to insure delivery of liquid to the several points of consumption in quantities which have definitely predetermined relationship. More particularly, the invention aims at the provision of improved structure whereby pumps of the character under consideration may be made to deliver equal quantities of fuel to the several fuel injectors supplied from a single pump cylinder.

Other and more specific objects of the invention and the manner in which such objects may be attained, will best be understood from a consideration of the ensuing portion of this specification in which by way of example the invention will be explained in its application to a single cylinder pump for supplying liquid fuel to the fuel injectors of a four cylinder internal combustion engine.

In the drawings:

Fig. 1 is a section through a pump embodying the invention, the upper portion of the figure being taken on line 1—1 of Fig. 4, and the figure further showing one of the pump outlets connected to a fuel injector of an engine;

Fig. 2 is another section of the upper part of

the pump shown in Fig. 1 taken on the line 2—2 of Fig. 4;

Fig. 3 is a section of a part of the pump shown in Fig. 1 taken on the line 3—3 of Fig. 4;

Fig. 4 is a top plan view of the pump shown in Fig. 1;

Fig. 5 is a developed view of part of the cylinder and plunger structure of Figs. 1 to 3;

Figs. 6 and 7 are diagrams illustrative of certain characteristics of pump operation;

Fig. 8 is a fragmentary section showing a variation of part of the pump structure shown in Figs. 1 and 2;

Fig. 9 is a fragmentary section showing still another variation of the same portion of the pump structure; and

Figs. 10 and 11 are more or less diagrammatic top plan views showing different applications, embodying the invention, of pumps to internal combustion engines.

Referring now more particularly to Figs. 1 to 5, the pump illustrated comprises a casing consisting of a main casing part 10 and a head part 12. A pump cylinder or barrel 14 is fixed in a recess 16 in the head part 12 by means of a threaded nipple 18 and the cylinder has reciprocally and turnably mounted in the bore thereof the pump plunger 20, at the inner end of which is located the pump chamber 22 to which liquid is admitted through the suction line 24 and spring loaded suction valve 26.

In the lower portion of the casing part 10 a cam shaft 28 is mounted in suitable bearings 30. Shaft 28 carries cam 32 which is in contact with roller 34 in the lower part of tappet 36 slidably mounted in a suitable bore in the casing part. The upper portion of the tappet is formed with a cylindrical recess having a circular wall 38. The lower end of the plunger 20 is enlarged to form a foot 20a resting in the bottom of the recess and held in position by a spring retaining washer 40 between which and the pump barrel there is compressed the pump spring 42 which acts to cause the tappet and plunger to follow the reciprocating path of travel determined by the profile of cam 32.

The head part of the pump casing is provided with four delivery passages 44, 46, 48, and 50 (see Figs. 1 and 2) which communicate respectively with the pump outlets 52, 54, 56, and 58, in each case by way of a spring loaded pump delivery valve. Each of the delivery valves is of like construction and all have been designated by reference character 60.

The pump barrel 14 is provided with a series of delivery ports d_1 , d_2 , d_3 , and d_4 which form respectively the inlets for the delivery passages 44—50 and which are placed in communication with the portions of the respective delivery passages located in the head of the pump casing by means of channels constituting in effect extensions of the delivery passage portions in the head and designated respectively as 44a, 46a, 48a, and 50a.

For convenience the pump outlets are uniformly distributed around the pump, as shown in Fig. 4, and for clarity of illustration the delivery ports and portions of the delivery passages adjacent thereto have been shown distributed around the circumference of the pump barrel with uniform angular displacement of 90° between ports, in order to bring the ports into the planes of Figs. 1 and 2. For reasons which will hereinafter be more fully explained, the location of the ports peripherally of the cylinder bore may be somewhat different in practice.

The pump cylinder is further provided with a vertically aligned row of outlet or relief ports r_1 , r_2 , r_3 , and r_4 communicating through suitable transverse bores in the barrel with a common overflow chamber 62 formed between the upper portion of the cylinder barrel and the wall of the recess 16 in the head 12. The overflow recess 16 is in communication with a suitable overflow outlet 64.

Relief ports r_1 — r_4 are located at the same height in the pump barrel as delivery ports d_1 — d_4 , respectively, and further are located in an axial plane different from the axial plane of any of the delivery ports.

The pump plunger is provided intermediate its ends with a circumferentially extending recess 66 which is in communication with the pump chamber by way of passage 66a in the plunger.

One portion of the recess 66, extending circumferentially of the plunger a sufficient distance to be in line with each of the overflow ports in the pump cylinder, which for convenience will be referred to as cylinder overflow ports, provides a plunger delivery port 68, various sections of which appear in Figs. 1 and 2. As will be observed from Fig. 3, an island-like projection 70, the outer surface of which is in contact with the cylinder bore, extends from the recess 66 to provide two axially spaced plunger overflow ports 72 and 74. The projection 70 provides an inclined control edge 70a for part load control of the quantities of liquid delivered by the pump, different portions of this inclined edge registering with the line of overflow ports in different positions of rotation of the plunger.

To effect the rotational adjustment of the plunger, a sector 76 is mounted for turning movement in the casing around the upper portion of the tappet 36. Sector 76 has external teeth 78 meshing with the teeth of a rack 80 slidably mounted in the pump casing and adapted to be actuated from any desired source of control (not shown). The inner face of sector 76 is provided with a vertical slot 82 which receives the outer end of a pin 84 passing through an arcuate slot 86 in the wall of the tappet and the inner end of which is fixed in the foot 20a of the plunger. The tappet is further provided with a vertical slot 88 in which slides the inner end of a locating pin 90 fixed in the pump casing to prevent turning movement of the tappet. It will be evident from Fig. 1 that move-

ment of the rack 80 along an axis normal to the plane of the paper will result in turning movement of the plunger.

Referring now more particularly to Fig. 5, in which the cylinder and plunger construction, inclusive of ports, is illustrated in developed form, the cylinder relief ports r_1 — r_4 , are shown in their in-line position as in Fig. 3, but the cylinder delivery ports d_1 , d_2 , d_3 , and d_4 are shown peripherally displaced from the positions which have been illustrated for the sake of clarity in Figs. 1 and 2. As diagrammatically shown in Figs. 1 and 2, 90° is the maximum angular displacement between delivery ports and in order to provide sufficient circumferential displacement between cylinder delivery ports to permit the control projection 70 on the plunger to operate properly in different positions of rotation of the plunger, one or more of the delivery ports are in practice advantageously displaced circumferentially from the positions shown in Figs. 1 and 2 to provide a greater peripheral distance between the line of overflow ports and the nearest delivery port. Conveniently, the delivery ports are arranged peripherally as indicated in Fig. 5, with ports d_1 and d_2 in one axial plane to one side of the relief ports and ports d_3 and d_4 in another axial plane to another side of these ports. The delivery ports when so displaced are connected respectively to the uniformly distributed delivery passages in the head by any suitable arrangement and location of the passages in the pump barrel, as will readily be understood by the reader. The operation of the above described pump is as follows, it being assumed that the plunger is in the full load position shown in Fig. 5 and ascending on its delivery stroke with the pump chamber filled with liquid from a preceding suction stroke. As will be apparent from Figs. 1, 3, and 5, the plunger, in the position shown in these figures, has moved to a position in which the plunger delivery port 68 and the cylinder delivery port d_1 are in registry and in which the plunger overflow port 72 has just passed out of registry with the cylinder overflow port r_1 . In this position commencement of delivery through port d_1 is just occurring and will continue until the plunger overflow port 74 comes into communication with the cylinder overflow port r_1 to terminate the period of delivery through port d_1 . By the time this occurs, plunger overflow port 72 is in communication with the cylinder overflow port r_2 to permit continued movement of the plunger in the direction of its delivery stroke and port 68 has come into communication with delivery port d_2 . As soon as the plunger overflow port 72 passes cylinder overflow port r_2 , a second delivery period will commence, which effects delivery through port d_2 . This second delivery period continues until plunger overflow port 74 comes into communication with cylinder overflow port r_2 . This cycle of events is repeated respectively with reference to ports d_3 and r_3 and with respect to ports d_4 and r_4 as the plunger continues to move on its delivery stroke, thus providing a sequence of delivery periods alternating with a sequence of overflow periods.

For part load operation, the plunger is turned so that the inclined control edge 70a is brought into line with the line of cylinder overflow ports and from Fig. 5 it will be apparent that with the plunger in such position, the length of each delivery period as the control projection passes

the respective cylinder overflow ports, will be diminished in length, the extent to which each delivery period is shortened being a function of the extent to which the control edge 70a is moved to the left as viewed in Fig. 5 with respect to the line of cylinder relief ports.

The specific pump construction hereinbefore described by way of example constitutes per se no part of the invention herein claimed, and forms the claimed subject matter of my co-pending application Serial No. 224,211, filed August 10, 1938.

Referring now more particularly to Fig. 6, the ordinates of the diagram shown in this figure represent the quantity (Q) of liquid delivered during a delivery or injection period and the abscissa represent rotation of the pump cam shaft (R) in degrees. If the several outlets of the pump are assumed to be connected to different injectors, one of which is indicated at 78 in Fig. 1 and which is shown as provided with a spring loaded delivery valve 60a, and if the speed of rotation of the cam shaft is constant, the quantities of fuel forced from any one nozzle per unit of time, (for example, per second) are equal to each other and are proportional to the pressure (P) in the fuel delivery passages and lines. Under this assumed condition the ordinates of the diagram indicate pressure as well as quantity. Theoretically, the pressure in the pump chamber and in the delivery line during a delivery period can rise along the line A—B, the delivery valve 60a (Fig. 1) then opening to effect delivery along the line B—C through the nozzle of the injector. At point C the pump is unloaded. In actual practice, however, injection does not take place in the above outlined theoretical manner. The speed of the pump cam shaft is dependent upon the speed of the engine and the speed of the plunger is dependent during the delivery period on the cam shaft speed. In the following analysis it has been assumed, for the sake of simplicity, that during a delivery period there is no change of speed so that the delivery stroke for each delivery period is carried out at approximately constant speed corresponding to the prevailing speed of rotation of the pump cam shaft. The form of the curve representing fuel delivery depends in part on the degree of compressibility of the injected fuel. From examination of injection occurring through an injection nozzle, the quantity (Q) of fuel injected, as a function of the pressure drop, is determined by the following equation:

$$Q = \mu F \sqrt{2gp \frac{10}{\gamma}} \quad (1)$$

where μ indicates the injection nozzle outflow coefficient;

F the cross-section of the nozzle orifice or sum of the cross sections of all nozzle orifices;

g acceleration due to gravity;

p the pressure drop across the nozzle orifice, that is the injection pressure minus the cylinder pressure; and

γ the specific gravity of the fuel.

The following relationship exists between the pressure rise in the fuel column ahead of the pump plunger, and the time:

$$A v dt = (B - A v) a dp + C \sqrt{p} dt \quad (2)$$

where A denotes the cross-section of the pump plunger;

B the volume of fuel supported by the pump plunger at the beginning of a delivery period;

$$C = \mu F \sqrt{2g \frac{10}{\gamma}}$$

v is the speed of the pump plunger; and
a the compressibility of the fuel.

Differential Equation (2) shows that after commencement of the application of pressure by the pump plunger, there occurs a volumetric alteration in the column of fuel subjected to pressure by the pump plunger, which column includes the liquid under pressure in the pump and in the delivery line leading to the injector. This alteration in volume is partly due to compression of the liquid and partly due to the passage of fuel from the injection nozzle. The course of the variation in pressure is thus dependent on the volume of the fuel column and on the quantity of fuel injected, the latter being proportional to the square root of the pressure. Under no circumstances, therefore, does the pressure rise perpendicularly in correspondence with the line A—B, but rises gradually, for example in accordance with curves such as those shown at a, b, and c on the diagram. Curve a represents the character of pressure rise occurring in pumps having smaller volume than that of pumps in which the pressure rise is in accordance with curve c. Such being the case it is evident that curve b represents the character of pressure rise in a pump of intermediate volume.

At the point C a pump overflow or relief port is opened to unload the pump and terminate the delivery period. Until the delivery valve 60a closes, fuel flows through the pump overflow passage of the pump. Theoretically, the pressure should fall along the line C—D of Fig. 6 but in practice there is a considerable lag or retardation of the pressure drop. In practice the pressure drop on release follows a curve such as indicated by C—E on the diagram. When the delivery valve 60a is closed, the portion of the fuel column between said valve and the nozzle orifice or orifices expands through the latter into the combustion chamber of the engine. The delivery valve may in known manner be disposed in the body of the injector as herein illustrated only a slight distance ahead of the nozzle. Only in the case of fuel injection devices without delivery valves, that is, if the structure were as indicated in Fig. 1 but without a delivery valve, does the pressure drop substantially in accordance with the theoretical line C—D.

The compressibility of the fuel affects the efficiency of the pump. The volumetric efficiency of the pump is determined by the ratio of the areas ACE:ABCD. For this reason pumps having delivery valves therefore have a higher volumetric efficiency than pumps without delivery valves.

Differences in the volumes of liquid may also occur, inter alia, for reasons of pump design. Thus, multi-cylinder pumps always show small deviations in the volumetric capacities of the different fuel chambers and passages. The diagram of Fig. 7 illustrates the different volumetric efficiencies V and V₁ of two cylinders of a multi-cylinder fuel pump at different pump speeds N. This fact is one of very great practical importance, particularly when the pump is applied to an internal combustion engine of the injection type, since if the different cylinders of the same engine have injected therein different amounts

of fuel, the desired steady and even running of the engine, with equal power output from each of the cylinders at any given load, will be materially and adversely affected.

5 In the case of a pump in which a series of delivery periods are effected from a single cylinder during the course of one delivery stroke of the plunger, it is evident that the fuel column in the pump chamber becomes progressively smaller for each such delivery period. Consequently, if 10 the desired equality of delivery during the several delivery periods is to be obtained, equalization must be provided in order that the plunger may have approximately the same volume of fuel in front of it during each delivery period.

15 In accordance with the present invention this may be provided for in different ways, some of which will now be described by way of example.

Referring now to Figs. 1 and 2, it will be observed that the delivery passage 44 through which liquid is delivered during the first delivery period when the volume of liquid in chamber 22 is the greatest, discharges into a relatively very small chamber 80 ahead of the appropriate delivery valve 60. Delivery passage 46 through which the next delivery period is effected, at a time when the volume of liquid in the pump chamber is less than when delivery is effected through passage 44, discharges to a chamber 82 the capacity of which is larger than that of chamber 80 and which is likewise situated ahead of the appropriate delivery valve. In similar manner passages 48 and 50 (Fig. 2), through which delivery is effected in sequence during periods in which the volume of liquid in the pump chamber is in each case smaller than during the preceding delivery period, discharge respectively to chambers 84 and 86, chamber 84 being of larger capacity than chamber 82 and chamber 86 being of larger capacity than chamber 84. These chambers may, for convenience, be designated as compensating or equalizing chambers. In the construction illustrated in Figs. 1 and 2, the equalizing chambers are conveniently formed, as shown, by drilling to different predetermined distances, the bottom portion of the recesses in the head of the pump which receive the different delivery valve assemblies.

Another arrangement for providing equalizing chambers of different capacity is illustrated in Fig. 8 in which each delivery passage, passage 50 being shown by way of example in the figure, is intersected by a bore 88 closed by a threaded plug 90 which may be adjusted to secure the desired capacity of the compensating chamber 86a formed at the inner end of the bore. The position of plug 90 is advantageously fixed by a lock nut 92 and the exposed end of the plug protected by a suitable cover or cap 94.

60 In Fig. 9 another arrangement is illustrated in which the compensating chamber 86b is formed between the bottom of the delivery valve assembly and the bottom of the recess which receives it by means of a spacing ring 96 the width of which will evidently determine the volume of the compensating chamber. It will be readily appreciated that the desired compensating chambers of different volume may readily be obtained with this arrangement by utilizing spacing rings of different width located at the bottoms of cavities of like form in the pump head.

To a certain extent, volumetric equalization of the fuel columns to different injectors may be effected by providing fuel lines leading from the pump unit to the several injectors, which lines

are of unequal volumetric capacity. Such unequal volumetric capacity may be effected by provision of lines of unequal length and equal internal diameter or by utilizing lines having different internal diameters, or both.

As illustrated in Fig. 10, the delivery lines 98a, 98b, 98c, and 98d are of equal diameter and unequal length, the arrangement being such that the shortest line 98a is attached to pump outlet 52 through which delivery is effected when the pump chamber has maximum volume, the remaining lines of progressively increasing length being connected to the pump outlets in the order of progressively decreasing pump chamber volume. The cylinders to which these lines are connected are indicated in Fig. 10 as the cylinders of a four cylinder engine with a conventional 1-2-4-3 firing order.

In Fig. 11 another arrangement is illustrated in which lines 100a, 100b, 100c, and 100d of different size are connected to the pump outlets in a relation such that the volumes of the respective lines are progressively greater for delivery periods during which the pump chamber volumes are progressively smaller. The engine shown in Fig. 11, like that shown in Fig. 10, is one having the conventional 1-2-4-3 firing order.

From the foregoing description it will be apparent that the principles of the invention may be embodied in various different structural forms and the invention is accordingly to be understood as embracing all forms of construction falling within the scope of the appended claims when they are construed as broadly as is consistent with the state of the prior art.

What is claimed is:

1. A plunger pump including a pump cylinder having a series of delivery outlets and separate conduits for conducting liquid under pressure from the pump chamber through different outlets to different places of consumption during different delivery periods, different ones of said conduits having different volumetric capacities for the liquid under pressure therein during the respective delivery periods, and the volumetric capacity of a conduit through which liquid is delivered later in the delivery stroke of the plunger being greater than the volumetric capacity of a conduit to which liquid is delivered earlier in the delivery stroke of the plunger.

2. A plunger pump including a cylinder, a plurality of pump outlets and separate conduits placing different outlets in communication with the pump chamber at different times during the delivery stroke of the plunger, said conduits having different volumetric capacities and the relation to each other of the capacities of the different conduits being the inverse of the relation of the volumetric capacity of the pump chamber at the different times when delivery is effected through different conduits.

3. A plunger pump including a cylinder, a plurality of delivery passages, different passages being in communication with the pump chamber at different times during the delivery stroke of the plunger, and at least certain of said passages having an equalizing chamber associated therewith for establishing liquid columns of substantially equal volume subjected to delivery pressure when the different passages are in communication with the pump chamber.

4. A plunger pump including a cylinder, a plurality of delivery passages, different passages being in communication with the pump chamber at different times during the delivery stroke of

the plunger, at least certain of said passages having an equalizing chamber associated therewith, and means adjustable to vary the volumetric capacities of said equalizing chambers.

5. A plunger pump including a cylinder, a plurality of delivery passages, different passages being in communication with the pump chamber at different times during the delivery stroke of the plunger, at least certain of said passages having an equalizing chamber associated therewith, said equalizing chambers being formed between different pump parts removably secured together and spacing elements between said parts for determining the volumes of the different chambers formed between them.

6. A plunger pump including a cylinder, a plurality of pump outlets, a plurality of delivery passages, different passages serving to separately connect the pump chamber to different outlets in different positions of the plunger on its delivery stroke, a pump delivery valve for each passage, and at least certain of said passages having an equalizing chamber associated therewith between the pump cylinder and the associated delivery valve, different chambers having different volumes to compensate for the different volumes of the pump chamber at the different times in the delivery stroke of the plunger when the different passages are brought into communication with the pump chamber.

7. In a fuel injection system for internal combustion engines, the combination with a fuel pump of the plunger type and a plurality of fuel injectors for supplying fuel at different times to different engine cylinders, of a plurality of fuel delivery lines separately connecting different pump outlets to different injectors, said delivery lines having different volumetric capacities proportioned relative to each other and to the in-

ternal volumetric capacity of the pump at different delivery periods to compensate for different pump chamber volumes during different delivery periods.

8. In a fuel injection system for internal combustion engines, the combination with a plunger pump including a pump cylinder and a plurality of pump outlets, different outlets communicating with the pump cylinder at different times during the delivery stroke of the plunger, and a plurality of fuel injectors each provided with a delivery valve determining the fuel pressure required to initiate injection, of a plurality of fuel lines each connecting a different pump outlet with a different injector, the value of the volumetric capacity of each of said fuel lines being such that when added to the volumetric capacity of the delivery passage in the pump with which it communicates and the volume of the pump chamber when communication between it and the pump chamber is established, the volume of the liquid column subjected to delivery pressure is substantially equal to the volume of the equivalent liquid column subjected to delivery pressure for delivery through any other of said lines.

9. A plunger pump for supplying liquid under pressure from a common pump chamber to different places of consumption at different times during the discharge stroke of the pump plunger through separate delivery lines, and compensating means associated with the pump for compensating for the difference between the volumes in said pump chamber at the commencement of each of the different delivery periods to establish substantially equal volumetric relation between the quantities of liquid subjected to delivery pressure at the commencement of each delivery period.

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