Sept. 2, 1947. I. E. COFFEY 2,426,965

FUEL PUMP

Filed May 18, 1944 3 Sheets-Sheet 1

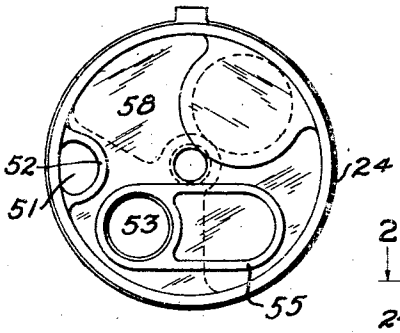


FIG. 3.

FIG. 1.

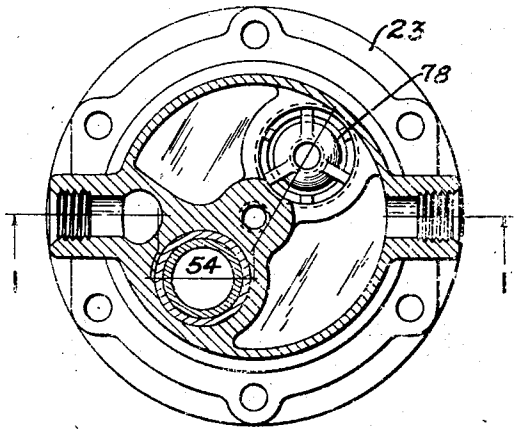


FIG. 2.

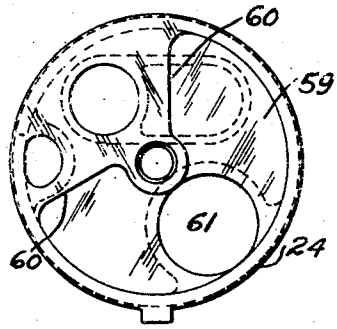


FIG. 4.

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FUEL PUMP

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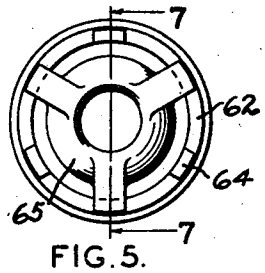


FIG.5.

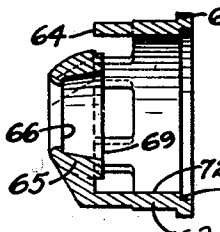


FIG.7.



FIG.8.



FIG.9.

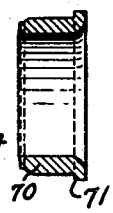


FIG.10.

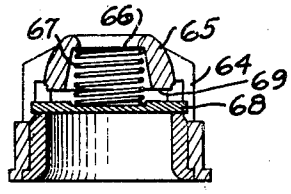


FIG.6.

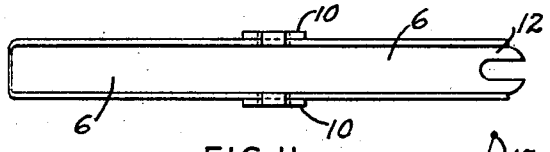


FIG.11.

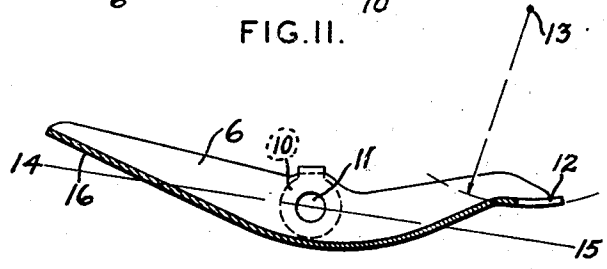


FIG.12.

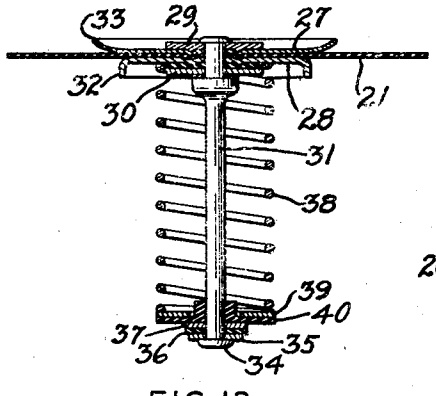


FIG.13.

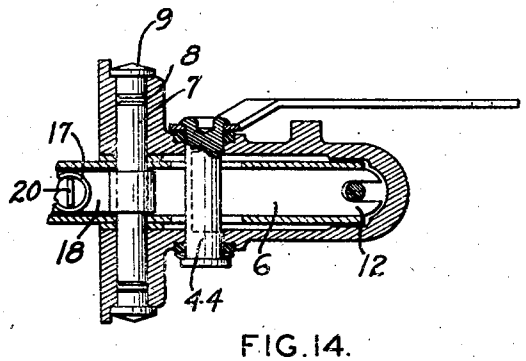


FIG.14.

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FUEL PUMP

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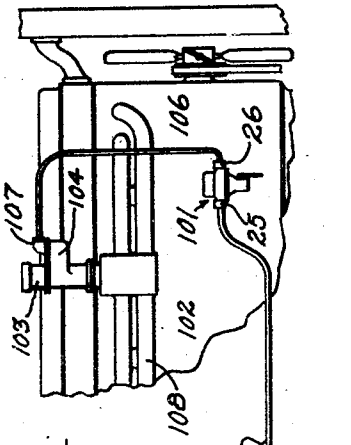


FIG. 18

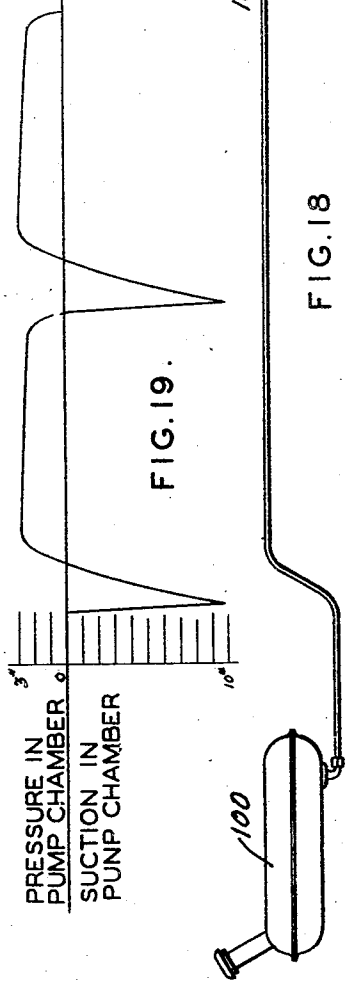


FIG. 19.

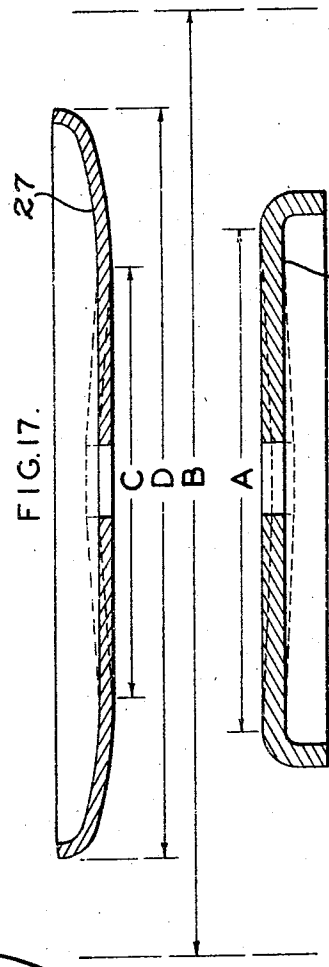


FIG. 17.

FIG. 16.

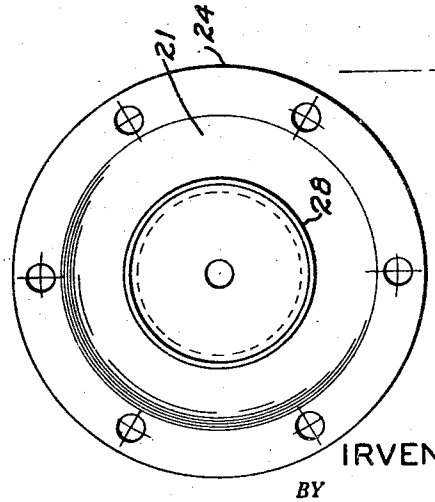


FIG. 15.

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Patented Sept. 2, 1947 2,426,965

UNITED STATES PATENT OFFICE

2,426,965

FUEL PUMP

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Application May 18, 1944, Serial No. 536,162

7 Claims. (Cl. 103—150)

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This invention relates to pumps and particularly to fuel pumps for automotive engines and the like. Pumps previously provided for this purpose have not been entirely satisfactory for a number of reasons, particularly with regard to the handling of boiling fuel.

Another serious trouble with previous designs has been diaphragm failures and short life of the pump, generally.

A very frequent source of trouble in previous fuel pump designs is the wearing of the valves and valve seats so that the pump will not prime itself at cranking speeds, and will not pump boiling fuel fast enough to run the engine.

Automotive fuel pumps are ordinarily operated by the engine cam shaft and must have a capacity sufficient to prime the pump, fill the carburetor bowl, and supply all the fuel required to start a cold engine at cranking speeds which may be as low as 40 R. P. M. Since the pump is ordinarily driven from the cam shaft, the pump is only operated twenty strokes per minute under these conditions. On the other hand, the pump must be capable of continuous operation at engine speeds of at least 4000 R. P. M. or 2000 strokes of the pump. Such high speed operation results in excessive strains on the diaphragms of pumps of previous designs.

When operating at 2,000 cycles per minute, the inlet and outlet valves of the pump have to be opened and closed with great rapidity, and this is especially hard on the inlet valve and seat, because its operation of opening and closing must be accomplished in a very small portion of the cycle, when the pump is operating at normal speed and capacity.

Another difficulty with previous pumps is that they have, in themselves, tended to set up vapor lock or boiling of the fuel by reason of the excessive suction developed in the fuel at the intake port of the pump and in the line leading back to the tank. In high speed operation of the pump, the fuel column leading to the inlet valve must be started and stopped with great rapidity, the only means preventing the separation or boiling of the fuel being the atmospheric pressure in the fuel tank or wherever it can act on the fuel.

An object of my invention is to produce a new and improved fuel pump capable of lifting and handling low boiling fuels at high temperatures.

Another object of my invention is to produce a new and improved fuel pump for internal combustion engines capable of quickly priming itself and filling the carburetor bowl under low speed engine cranking conditions.

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Another object of my invention is to produce a fuel pump for internal combustion engines in which the fuel in the intake conduit is refrigerrated by vaporization of fuel in the pumping chamber.

It is a further object of my invention to produce a diaphragm type fuel pump in which the strains on the diaphragm are minimized by providing for the acceleration and deceleration of only short columns of fuel.

It is a further object of my invention to produce a new and improved fuel pump of the reciprocating type in which the maximum capacity of the pump is limited by a calibrated restriction at or near the intake fuel of the pump.

It is another object of my invention to produce a new and improved fuel pump of the reciprocating type in which the fuel inlet column or pipe is interrupted by an air chamber near the inlet valve of the pump and in which the inlet passage between the air chamber and the inlet valve has sufficient capacity to contain at least as much fuel as can be discharged at one stroke of the pump during normal and high speed operation.

It is a further object of my invention to provide a fuel pump of the above described character, in which the displacement of the pumping element at low speed operation is much greater than the capacity of the inlet reservoir between the air chamber and the inlet valve.

Another object of my invention is to produce a fuel pump for automotive engines in which the life of the pump may be definitely expected to exceed the life of any engine or vehicle to which it is likely to be applied.

Another object of my invention is to produce a new and improved valve capable of maintaining high sealing qualities after long use under high speed operating conditions.

Another object of my invention is to provide a new and improved operating lever for high speed operation.

In order to eliminate the faults of previous designs, and to produce a new and improved pump capable of accomplishing the above described and other objects, I have invented the pump described and shown in the following specification and accompanying drawings, referring to which:

Figure 1 is a sectional elevation of a pump according to my invention.

Figure 2 is a sectional plan view taken along the irregular sectional line 2—2 of Figure 1.

Figure 3 is a detailed plan view of the main air dome member.

Figure 4 is an inverted plan view of the main air dome member.

Figure 5 is a plan view of a valve cage.

Figure 6 is a sectional elevation of a valve assembly.

Figure 7 is a sectional view taken along the line 7—7 of Figure 5.

Figure 8 shows the valve spring.

Figure 9 is a sectional view of the valve.

Figure 10 is a sectional view of the valve seat member.

Figure 11 is a plan view of the operating lever or hammer.

Figure 12 is a sectional elevation of the hammer.

Figure 13 is a sectional elevation of the diaphragm assembly.

Figure 14 is a sectional plan view taken along the line 14—14 of Figure 1, parts being broken away.

Figure 15 shows the diaphragm and lower supporting washer.

Figure 16 is a sectional view of the lower supporting washer.

Figure 17 is a sectional view of the upper supporting washer.

Figure 18 is a diagram showing the approximate position of the main fuel tank with respect to the engine and fuel pump in an automotive installation.

Figure 19 is a diagram showing the approximate pressure and suction curve existing in the pumping chamber.

Referring first to Figure 18, the reference numeral 100 shows a main fuel tank which is normally mounted at a lower level than the pump which is generally indicated by the reference numeral 101. The pump is mounted on and driven by an internal combustion engine 102 having the usual carburetor 103 and float chamber 104, the details of which are not shown but are well understood by those skilled in the art. A fuel conduit 105 of comparatively small diameter and great length leads from the fuel tank to the pump inlet connection 25, and an outlet conduit 106 also of small diameter and considerable length is connected to the fuel pump outlet at 26 and to the carburetor float chamber at 107. It may be noted that the engine is provided with an exhaust manifold 108 which frequently becomes red hot and is likely to cause boiling of the fuel as it passes through the conduit 105 to the pump inlet. It may also be noted that while boiling occurs in the pump chamber and in the outlet, it is not of such great importance as boiling in the inlet. This is because any boiling which occurs posterior to the inlet valve creates its own discharge pressure, while boiling in the inlet conduit prevents induction of fuel into the pump chamber and causes vapor lock.

The reference numeral 1 shows a main casting having a flange 2, by means of which it is bolted on to the frame of the engine as at 3. It will be understood that while the pump is primarily designed for automotive engines, it is capable of other uses and can be attached to any support 3 so as to be properly positioned with respect to a cam 4 carried by a rotating shaft 5 which preferably turns in a clockwise direction, so that the sweep of the cam will be inward with respect to the hammer or operating lever 6, which is pivotally mounted on the body member 1 by means of the floating pivot shaft 7.

This shaft is mounted in the bore 8, the ends of which are closed by sealing plugs or rivets 9 which permit substantial end play of the shaft.

The hammer 6 is formed of a curved and channel-shaped sheet metal member having bent-over lugs 10 having a central opening 11 which is accurately finished to line up with corresponding holes in the sides of the hammer to form a bearing for the shaft 7. One end of the hammer rides on the cam, as indicated in Figure 1, and the other is provided with a fork or hammer head 12 which is slightly curved on a radius about the point 13, as indicated in Figure 12.

It will be noted that the floor of the channel is substantially curved about the bearing 11 in such a manner that a diameter of the bearing, produced as indicated at 14 and 15, passes on the same side of the cam contact portion 16 and the hammer head 12, so that a component of the force on the cam necessary to produce the hammer blow is taken in tension and not in pure bending strain. This feature of construction is of importance, because the high speed of operation and the violence of the hammer blow when the hammer encounters the resistance of the diaphragm would otherwise cause crystallization and breakage of the lever.

The lever is yieldably urged in an anti-clockwise direction by the spring 17, which is seated against the flange 2, and a seating member 18 which is mounted on the shaft 7 and provided with a seating fork 19 and an upturned locating member 20, as shown in Figure 1.

The pump diaphragm 21 is formed of synthetic rubber or the like formed on a cloth base, and is held in place between the flange 22 on the main casting and a corresponding flange 23 on the valve body 24, which is provided with inlet and outlet connections 25 and 26, respectively. The diaphragm is clamped in position between the flanges, which are held together by any suitable means, such as bolts or rivets (not shown). The center portion of the pump diaphragm is rigidly held in between upper and lower diaphragm washers 27 and 28, respectively, which are, in turn, clamped between washers 29 and 30 riveted on to the shaft 31, as indicated in Figure 13.

The shape and curvature of the washers 27 and 28 are important and form a substantial part of my invention. The lower washer 28 has a slightly convex or conical central portion which is pressed flat in the assembly. The diameter of this washer preferably bears a relation to the working diameter of the diaphragm in the order of six to ten. The flat portion of the lower washer is surrounded by a curved flange 32 which has a sectional radius of approximately ⅛ inch.

The upper washer has a central conical portion approximately the same diameter as the oppositely disposed surface of the lower washer. This surface is surrounded by a flange 33 which is curved on a decreasing radius in such a manner that its cross section is a section of a parabola or ellipse. The full diameter of the washer 27 bears a relation to the working diameter of the diaphragm of approximately 8½ to 10, and the short radius of the ellipse which roughly corresponds to the curvature of the flange 33 is approximately one fifth of the full operating stroke of the pump.

The shaft 31 is provided with a head 34 against which is seated a metal washer 35. A cushion washer 36 of pliable, but only slightly flexible, material is positioned next to the metal washer, and a third washer 37 is made of hard plastic

material on a cloth base and with highly polished surfaces. A spring 38 having sufficient strength to move the diaphragm in one direction is mounted against the washer 28 and against a metallic washer 39 which rests on a sealing washer 40 formed of synthetic rubber or other oil proof sealing material and slidably or snugly fitted to the shaft 31.

The sealing washer 40 is mounted in the seat 41 in the main casting. When mounted as shown in Figure 1, the strength of the spring 38 acting against the diaphragm 21 has sufficient force to produce the pressure in the pumping chamber 42 which the pump is desired to maintain.

An auxiliary operating lever 43 rigidly mounted on a transverse rock shaft 44 having a cam surface 45, as indicated in Figure 1, is provided. The function of the member 43 is to permit the operator to manually rock the hammer 6 to prime the pump or carburetor when the engine is not in operation.

A main air dome casting 46 is mounted on a diaphragm 47 against a seat in the side of the valve body 24. A cover member 48 containing a portion of the inlet air dome space 49 is held in position by the screw 50 which is threaded into the valve body to clamp the cover and the air dome in place. The inlet passage 51 extends into the valve body and up into the air dome in which it is surrounded by a baffle 52 which extends to a point substantially higher than the passage 53 leading to the inlet valve 54. The passage 53 is provided with an upstanding flange 55 over which is fitted the strainer 56, the latter being held in position by the boss 57 which extends inwardly from the top of the cover 48. The passage 53 forms an inlet charge chamber for supplying fuel to the fuel inlet, the latter being controlled by means of the inlet valve 54 provided at its lower end adjacent the pumping chamber. The inlet charge chamber has a volume approximately as great as the predetermined maximum displacement of the movable wall 21.

The inlet air dome 58 contains substantial volume which effectively includes the volume of the chamber 49 and is more than double the displacement of the movable wall of the pumping chamber. I have discovered that the relative volumes of the air domes and pump stroke have a definite bearing on the life of the diaphragm as well as capacity of the pump, and since these relative volumes form an important part of my invention, the following examples are given for the assistance of those skilled in the art in making and using the invention. If a particular automobile engine requires 3 gallons per hour for cold starting when cranked at 40 R. P. M. or 20 pump strokes per minute and a maximum possible capacity of 29 gallons per hour at 4,000 R. P. M. or 2,000 pump strokes, I have found that satisfactory results can be obtained with the following approximate dimensions and proportions:

Working diameter of pump diaphragm inches	2.47
Maximum stroke of diaphragm do	.25
Diaphragm displacement (full stroke) c. c.	10
Maximum stroke of diaphragm during high speed operation (2,000 R. P. M.) while pumping liquid fuel only inches	.03
Discharge spring and exhaust restriction calibration to give diaphragm displacement during high speed operation while pumping liquid fuel only c. c.	1

Intake air dome volume c. c.	62
Outlet air dome volume c. c.	15

The outlet air dome is closed against the admission of fuel by means of the diaphragm 47 and the air volume in this dome is approximately one-third of the volume of the air inlet dome 58. In order to give this diaphragm substantial working capacity, I form a substantial part of the air dome volume in a low chamber 59 which extends more than halfway around the base of the air dome casting and terminates in walls 60. In order to prevent unrestricted movement of the diaphragm beyond the distance which it can be flexed with safety, I keep the ceiling of the chamber 59 low, so as to stop the movement of the diaphragm beyond a predetermined limit, and form the remainder of the volume in a chamber 61 which extends upwardly, as shown in Figure 1, but which does not have sufficient diameter to permit undue strain on the diaphragm 47. It will be understood that the greatest strain on the diaphragm 47 does not result from the action of the spring 38, but from the vapor pressure which builds up as a result of heat when the engine is standing idle, just after a run in which the engine has been thoroughly heated up.

The construction of the valves has an important bearing on the life and operation of the pump. The design of the inlet valve according to my invention is more critical than the design of the outlet valve, because it has to operate at a much higher rate of speed, and also because it is required to perform a restricting function which will be described later. For convenience, I have made the exhaust valve identical with the inlet valve.

The valve comprises a cage 62 having a shoulder 63 which can be seated in corresponding bores formed in the die cast valve body. Struts 64 extend upwardly from the body member, and some or all of these struts are integrally connected with an annular spring seat and stop member 65, as shown in Figures 5 to 7. A shoulder 66 is formed in the member 65 to receive the spring 67 which normally holds the valve 68 in closed position. The stop portion 69 must be accurately and smoothly finished parallel to the valve 68 and also positioned in such a manner as to definitely limit the opening movement of the valve to a position substantially less than that which would be required to fully clear the passageway through the valve. The valve cage 62 is preferably formed of die cast material.

The valve seat member 70 has a shoulder 71 and is press-fitted into the bore 72 of the valve cage and firmly pushed up against the shoulder 73. A seat portion 74 of slightly conical shape, as indicated in Figure 10, is formed at the end of the valve seat member, and this is carefully lapped to a substantial line contact against the valve 68. This valve is preferably formed of Bakelite or other hard plastic material on a cloth base which is very light so as to move with very little inertia and strike the seat 74 and stop 69 with very little kinetic energy.

Disk valves of the general type shown herein have been previously used, but they have not been satisfactory in regard to wearing quality, as well as for other reasons. To the ordinary observer, a disk valve made of Bakelite and lightly seated by a small spring against a metal seat seems to present minimum possibilities for wear. There appears to be no rubbing between the valve and the seat. Even if rapidly operated, the small in-

ertia of the valve should not apparently do any damage to the seat or to the valve stop.

In actual practice, previous valves have been subject to rapid wear and have actually worn out the seats and cages without any apparent reason. I have discovered that because of this, rapid wear lies in a valve cage and spring which causes or permits the valve to assume a slightly angular position with respect to the seat or stop during its opening or closing movement. If this occurs during the movement of the valve from one position to the other, it does not strike dead and flat, but strikes with a gyrating movement, the same as when a coin is dropped on a table. During this movement, there is a definite rubbing between the valve and seat which is probably the cause of the surprising amount of wear which has occurred in previous valves of this type. By carefully centering the disk with respect to the valve seat and the spring with respect to the valve and making the seats and stops absolutely parallel when new, I have substantially eliminated this cause of wear on the valves.

A feature of the invention is that the stop **69** is made circular, so that any wear on the valve is distributed on an annular surface. In this way, I avoid throwing the valve slightly out of parallelism with the seat, which would otherwise result from a slight turning of the valve.

Another important feature of the invention is the limitation of the movement of the valves. The movement of the valves between the seat and stop occurs with considerable violence, and I have discovered that by snubbing or limiting this movement, the life of the valves and diaphragm can be substantially increased. In order to hold the end of the spring **67** in fixed position with respect to the valve to prevent wear, I counterbore the valve, as indicated at **75**. This counterbore may be made of such a diameter as to snugly fit the end of the spring, but good results may also be obtained by making the shoulders of the counterbore with a radius or chamfer so as to centralize the spring with respect to the valve.

I have found that the movement of the valve between the seat and stop tends to occur with great violence, and that the valve should be made as light as possible, so that it will strike the seat and stop with minimum inertia. In order to keep the valve flat and straight, it must have a thickness of at least $^{3}/_{64}$'' at the rim, but by counterboring it, as shown, the diameter of the center can be reduced to approximately $\frac{1}{32}$'' or even less. The material used in making the valve should be as light as possible, and I have found that a hard plastic on a cloth base, such as Celeron, Formica, Micarta, or Spauldite, may be used with satisfactory results.

In designing the valve, I have discovered that it is important for a number of reasons to limit the opening movement, and I prefer to make the space between the valve seat and stop approximately $\frac{1}{16}$ of the diameter of the passageway controlled by the valve. The valve will still operate satisfactorily up to an opening movement of $\frac{1}{8}$ of the passage diameter after which further wear occurs very rapidly, resulting in the ultimate destruction of the valve. When the valve is new, I allow it to open only approximately $\frac{1}{16}$ of the passage diameter. Wear of the valve seat and stop eventually increases the opening to about $\frac{1}{8}$ of the passage diameter during the normal life of the pump, which is greater than the life of any automobile engine to which it is likely to be applied.

In case the wear should be excessive after extreme length of use, it is desirable to provide some means to prevent further wear on the stop. I accomplish this function by making the spring **67** of a sufficient number of turns so that it will go solid when the valve opening reaches approximately $\frac{1}{8}$ of the passage diameter, so that the opening of the valve cannot have become so great as to result in destruction of the valve due to the inertia which it would acquire in opening a substantial distance. It will be understood that the valve is opened by the flow of liquid and that the first part of the liquid flow is comparatively slow and gentle, so that by snubbing the valve movement during this part of the cycle, it is not subjected to the greater acceleration which it would acquire during the latter part of the impulse of fuel flow.

One very important function of the inlet valve is to permit free flow of fuel or vapor into the pump chamber during cranking of the engine, but to definitely restrict the flow of fuel into the pump chamber during the intake stroke, so as to produce a short period of low pressure in the pump chamber to lower the boiling point of the fuel, to cause some boiling in the pump chamber under incipient boiling conditions, whereby the lowered temperature resulting from partial vaporization of fuel in the pump chamber will be transmitted to the walls of the valve body to the fuel entering at the connection **25**.

It will be noted that the fuel is led directly into this connection, so as to intimately contact and be cooled by the metal of the valve body before it passes into the dome **58** where it is comparatively insulated from temperature changes.

In operation, the lever **6** is held in the position shown in Figure 1 by means of the spring **17**, so that one end rests on the cam **4**, which preferably rotates in a clockwise direction. Turning the cam rocks the lever on the pivot shaft **7** which floats in the bearings **8** and **11**. Assuming there is no pressure in the passage outlet, the spring **38** moves the diaphragm **21** to the position shown in Figure 1 until it is withdrawn by contact of the hammer **12** with the hammer pad or shock absorber **35—36—37**.

The first part of the intake stroke of the diaphragm occurs rather slowly, due to the changing movement of the cam, and during this part of the movement, the intake valve **54** is opened to the full extent permitted by the stop **69**. As the cam continues in its movement, the diaphragm **21** moves more rapidly, and the valve is firmly held in contact with the stop. This somewhat restricts the movement of the liquid during the central part of the stroke, and produces a sharp pressure drop in the pump chamber, as shown in the diagram in Figure 19. The diaphragm or pump chamber **42** is eventually filled due to the dwell at the end of the stroke, and also due to the fact that this chamber is not emptied at each stroke except under very low speed operating conditions. In other words, the cycle of fuel flow lags slightly behind the cycle of diaphragm operation and some vapor will appear in the pump chamber during incipient boiling conditions. During the next cycle of cam movement, which is somewhat less than 180°, if the rotation of the cam is clockwise, the hammer **12** is out of contact with the hammer pad (except during some cranking or vapor locking conditions), and the spring **38** returns the diaphragm **21** to discharge the fuel and any vapor through the exhaust valve **78**. This valve func-

tions in substantially the same way as the intake valve 54, although its movement against the stop does not tend to occur with such great violence, the spring 38 having limited force while the movement of the cam and hammer is substantially positive. The valve also serves as a restriction on the amount of fuel discharged, so as to limit the gallons per hour which the diaphragm can be called on to pump, but without placing any corresponding limit on the amount of vapor which can be pumped to get rid of a vapor locking condition. The pump diaphragm is accordingly available to pump at least ten times as much volume of vapor as it will of liquid fuel. Of course, this would result in excessive strain and wear of the diaphragm and operating mechanism, except for the fact that vapor lock occurs only occasionally, and vapor does not present great resistance to the movement of the diaphragm.

During the exhaust stroke, the suction in the dome 53 refills the dome with fuel from the inlet 25, or at least brings in enough fuel to supply the next intake stroke of the pump. The exhaust stroke of the pump also displaces the diaphragm 47 against the top of the low chamber 59 and against the air in the chamber 61, so that the discharge of fuel from the exhaust of the pump can be continuous throughout the cycle. It will be understood that in a pump of this character, both the inlet and exhaust pipe lines are long and of small diameter, and the fuel has comparatively great inertia, so that without continuity of flow at high speeds, the diaphragm and operating mechanism would be subjected to severe stresses.

One of the most important features of the invention is the construction of the inlet 53 and air dome 53 in such a manner that the column of fuel which must be started in motion at the beginning of the intake stroke is of large diameter and very short, but of volume corresponding to a full charge for normal and high speed operation. Modern motor fuels have comparatively high vapor pressure, and the application of any substantial suction by the diaphragm has a great tendency to cause the fuel to separate or vapor lock. This would not be so important, except for the fact that when the fuel is separated, the vapor collects in bubbles which do not disappear as quickly as they form.

By the construction herein shown and described, the column of fuel present in the dome above the inlet valve is not much more than one inch long. Also, it has greater diameter at the surface than in the passage. By this means, the power required to accelerate the fuel column and the internal suction developed within the column, itself, are held to an absolute minimum.

The device according to this invention departs substantially from prior practice in the restriction at the inlet valve and at the exhaust valve. The inlet restriction definitely limits or reduces the internal suction which can be placed on the fuel at the inlet side of the valve. The amount of suction on the fuel in the pump chamber on the other side of the restriction is increased, but the formation of vapor in the pumping chamber, if it occurs, does not appear to give any serious trouble, but on the contrary, its formation has a cooling action which is helpful in preventing the development of vapor lock. The diaphragm is capable of discharging the vapor at each stroke and it is not allowed to accumulate. This is because of the selective action of the restricted valve in passing vapor more freely than liquid fuel. Any vapor which forms in the inlet air dome 53 may accumulate, so that the elasticity of the large volume of accumulated vapor would become a serious factor, if not withdrawn by a full stroke of the diaphragm, but the vapor forming in the dome and chamber is mostly disposed of at the end of the stroke and cannot accumulate to do serious damage. Since both inlet and exhaust valves have a very selective action in passing a greater volume of vapor than of fuel, it is practically impossible to vapor lock the pump, and yet the diaphragm is not subjected to severe stresses or required to operate through more than a small fraction of its stroke during normal or high speed operation.

I claim:

1. In a fuel pump for an internal combustion engine, a pump chamber, a movable wall in said pump chamber, inlet and outlet passages for the pump chamber, inlet and outlet valves controlling said inlet and outlet passages, a fuel inlet charge chamber for maintaining directly adjacent the inlet side of said inlet valve a charge of fuel greater than the maximum displacement of said movable wall during high speed operation, stop means for limiting the opening movement of said valve to restrict the flow of fuel from said fuel charge to said pump chamber to less than the rate of displacement of said movable wall, whereby a pressure drop of comparatively great amplitude as compared with the simultaneous pressure drop in the inlet passage and charge chamber anterior to the inlet valve will be produced in said chamber to enhance the vaporization of the fuel therein for lowering the temperature at the inlet.

2. In a reciprocating pump for high speed operation, a pumping chamber having a movable wall forming a pumping element, yieldable means tending to move said movable wall in one direction, positive means for moving the movable wall in the other direction, a fuel inlet opening for said pumping chamber, an inlet charge chamber for said fuel inlet, a fuel outlet passage and a valve therefor, an air chamber forming an air dome and connected with a fuel supply passage and enclosing the inlet charge chamber, a valve at the end of the inlet charge chamber adjacent the pumping chamber, said inlet charge chamber being constructed and arranged to hold approximately the full amount of fuel which the pumping chamber is calibrated to receive at one stroke of said movable wall during high speed operation, said air chamber having a volume substantially greater than that of said inlet charge chamber, whereby the acceleration of fuel from said inlet charge chamber into the pump chamber may be accomplished without accelerating the fuel in said supply passage to a corresponding extent and the load on said pump is proportionately reduced.

3. In a fuel pump, means forming an expansible chamber, a movable diaphragm forming a wall of said chamber, yieldable means tending to move said diaphragm in one direction, positive means for moving the diaphragm in the other direction, inlet and outlet passages for said chamber, inlet and outlet valves controlling said passages, seats for said valves, and stop means for limiting the opening movement of said valves to not more than one-sixth of the inside diameter of their seat whereby the rotary canting movement of said valves due to move-

ment of the fluid passing therethrough is limited and abrasive action between the valves and their seats is prevented.

4. In a fuel pump for pumping liquids which are subject to boiling, a fuel inlet passageway, an inlet chamber connected to receive fuel from said inlet passageway, a pumping chamber having a movable wall capable of a predetermined maximum displacement, a fuel inlet valve for said pumping chamber, a fuel inlet charge chamber directly adjacent and anterior to said inlet valve, said fuel inlet charge chamber having a volume approximately as great as the predetermined maximum displacement of said movable wall, an air dome formed in the upper part of said inlet chamber above the level of the inlet of said fuel inlet charge chamber, said air dome having a volume more than double the said maximum displacement of said movable wall, and an outlet valve for said pumping chamber, an outlet air dome directly adjacent said outlet valve, an outlet diaphragm for said outlet air dome, the volume of air in said outlet air dome being not more than one-third of the volume of the inlet air dome.

5. In a fuel pump for pumping liquids which are subject to boiling, a fuel inlet passageway, an inlet chamber connected to receive fuel from said inlet passageway, a pumping chamber having a movable wall capable of a predetermined maximum displacement, a fuel inlet valve for said pumping chamber, a fuel inlet charge chamber directly adjacent and anterior to said inlet valve, said fuel inlet charge chamber having a volume approximately as great as the predetermined maximum displacement of said movable wall, an air dome formed in the upper part of said inlet chamber above the level of the inlet of said fuel inlet charge chamber, said air dome having a volume more than double the said maximum displacement of said movable wall, and an outlet valve for said pumping chamber.

6. In a fuel pump for pumping liquids which are subject to boiling, a fuel inlet passage, an inlet chamber connected to receive fuel from said inlet passageway, a pumping chamber having a movable wall capable of a predetermined maximum displacement and forming a pumping element, yielding means for moving the movable wall in one direction, positive means for moving the movable wall in the other direction, a fuel inlet opening for said pumping chamber, an inlet charge chamber for said fuel inlet, a valve at the end of the inlet charge chamber adjacent the pumping member, said inlet charge chamber being directly adjacent and anterior to said inlet valve and having a volume approximately as great as the predetermined maximum displacement of said movable wall, an air dome formed in the upper part of said inlet chamber above the level of the inlet of said fuel inlet charge chamber, said air dome having a volume more than double the maximum displacement of said movable wall and substantially greater than the volume of the inlet charge chamber whereby upon

operation of said pumping element direct acceleration of fuel from said inlet charge chamber into the pumping chamber may be accomplished without accelerating the fuel in said supply passage to a corresponding extent and the load on said pumping element is proportionately reduced, and an outlet valve for said pumping chamber.

7. In a fuel pump for pumping liquids which are subject to boiling, a fuel inlet passage, an inlet chamber connected to receive fuel from said inlet passageway, a pumping chamber having a movable wall capable of a predetermined maximum displacement and forming a pumping element, yielding means for moving the movable wall in one direction, positive means for moving the movable wall in the other direction, a fuel inlet opening for said pumping chamber, an inlet charge chamber for said fuel inlet, a valve at the end of the inlet charge chamber adjacent the pumping member, said fuel charge chamber being directly adjacent and anterior to said inlet valve, said inlet charge chamber having a volume approximately as great as the predetermined maximum displacement of said movable wall, an air dome formed in the upper part of said inlet chamber above the level of the inlet of said fuel inlet charge chamber, said air dome having a volume more than double the maximum displacement of said movable wall and substantially greater than the volume of the inlet charge chamber whereby upon operation of said pumping element direct acceleration of fuel from said inlet charge chamber into the pumping chamber may be accomplished without accelerating the fuel in said supply passage to a corresponding extent and the load on said pumping element is proportionately reduced, an outlet valve for said pumping chamber, an outlet air dome directly adjacent said outlet valve, and an outlet diaphragm for said outlet air dome, the volume of air in said outlet air dome being approximately one-third of the volume of the inlet air dome.

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