United States Patent

[11] 3,578,879

[72] Inventors **Emile David Long**
Elmira;
Kenneth Donald Aumick, Pine City, N.Y.

[21] Appl. No. **793,088**
[22] Filed **Jan. 22, 1969**
[45] Patented **May 18, 1971**

[73] Assignee **Gillett Tool Co.**
Buffalo, N.Y.

[54] **SPRING ACTUATED FUEL PUMP FOR FUEL INJECTION SYSTEMS**
19 Claims, 12 Drawing Figs.

[52] **U.S. Cl.** **417/62,** 417/271, 417/471, 417/552
[51] **Int. Cl.** **F04b 23/04,** F04b 27/04
[50] **Field of Search** 103/174, 23, 178, 182, 186, 10, 38; 230/40; 417/62, 271, 471, 552

[56] **References Cited**

UNITED STATES PATENTS

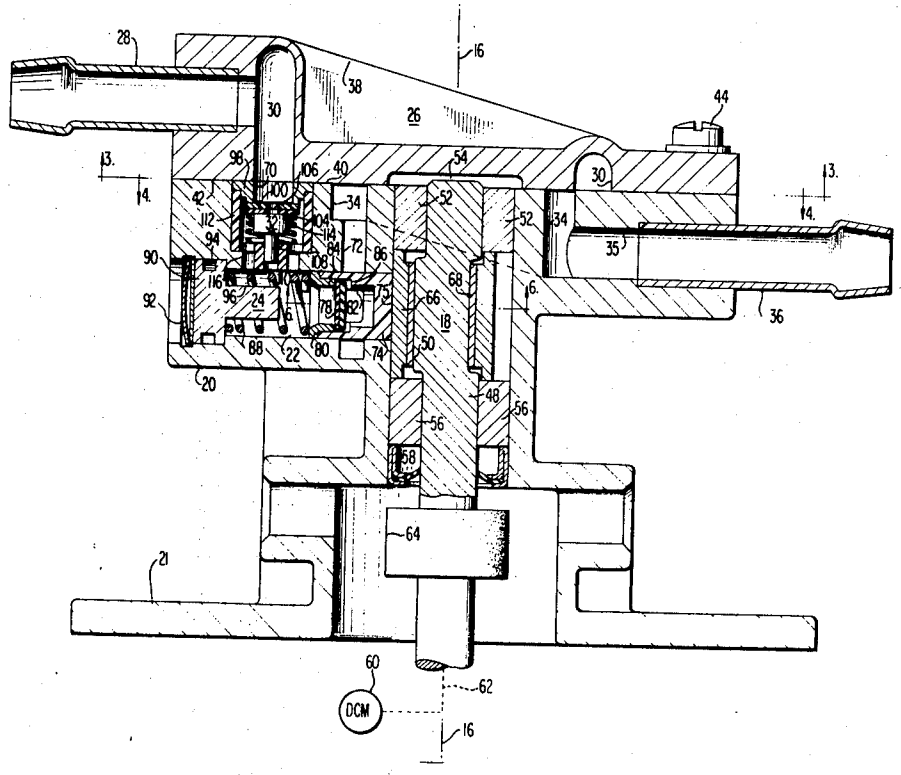
462,666	11/1891	Dykes	103/182
2,018,111	10/1935	Babitch	103/23
2,120,448	6/1938	Twiss	103/186
2,094,171	9/1937	Hoffer	103/23
3,016,843	1/1962	Smith	103/211
3,085,312	4/1963	Evans	308/1(X)
3,118,381	11/1964	Reil	103/174
3,228,587	1/1966	Segebrecht	230/40
3,381,191	5/1968	Bellmer	230/40
2,887,961	5/1959	Hawley	103/84
3,407,746	10/1968	Johnston	103/170

FOREIGN PATENTS

1,290,964	3/1962	France	103/182

Primary Examiner—William L. Freeh
Attorney—Brady, O'Boyle and Gates

ABSTRACT: A motor-driven fuel pump for precision fuel-metering systems of the fuel injection type wherein a plurality of radially disposed reciprocating pistons are actuated during the pumping stroke by a pumping spring after the spring has been cocked by means of a motor-driven eccentric motor coupled to the radially disposed pistons. The plurality of pistons can be staged to provide either series or parallel operation by means of a feed manifold which can provide any desired relationship between inlet and outlet connection depending upon the desired output pressure and/or delivery requirements.



PATENTED MAY 18 1971

3,578,879

SHEET 1 OF 4

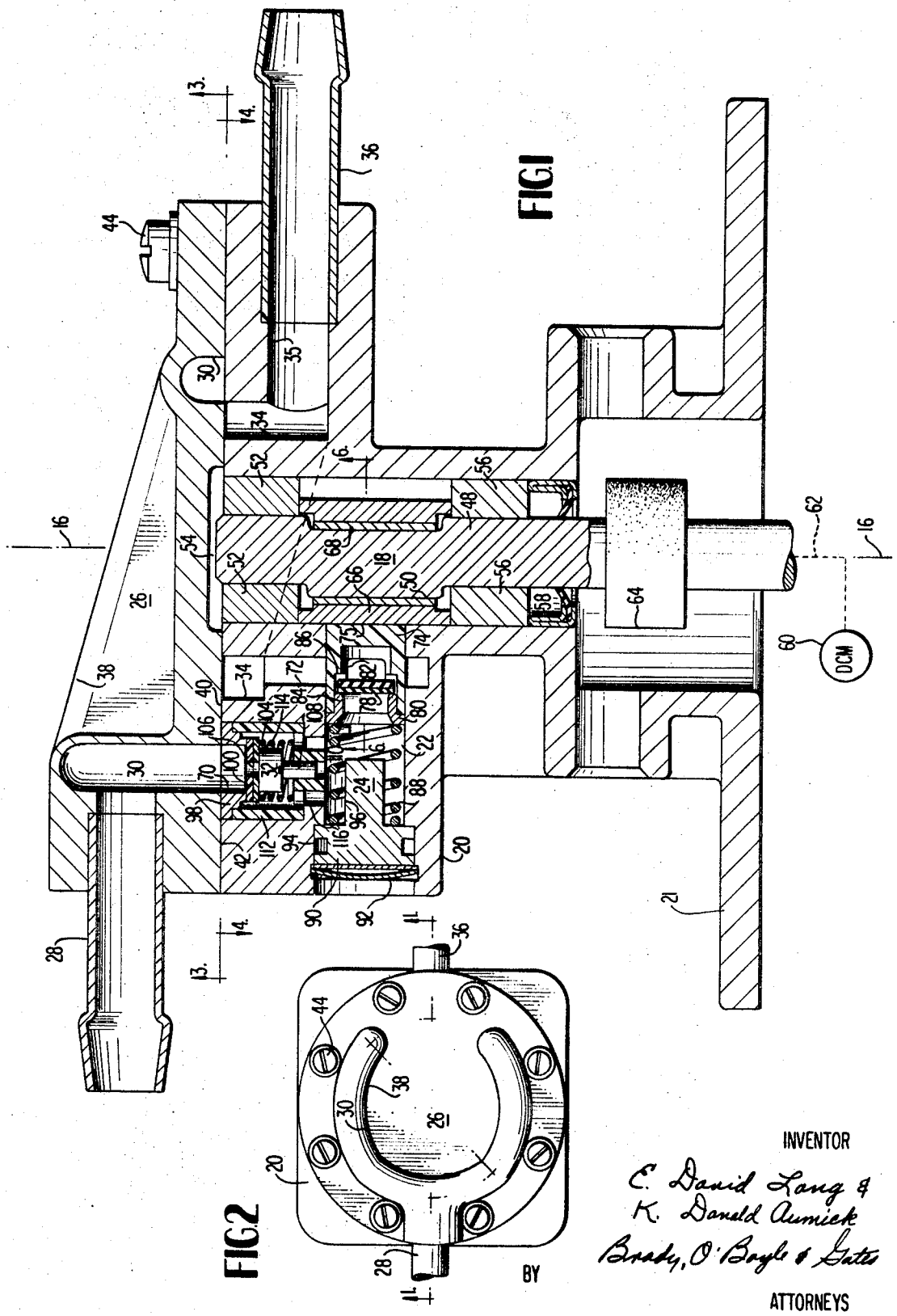


FIG1

FIG2

INVENTOR

E. David Lang &
K. Donald Aumiek

BY

Brady, O'Boyle & Gates

ATTORNEYS

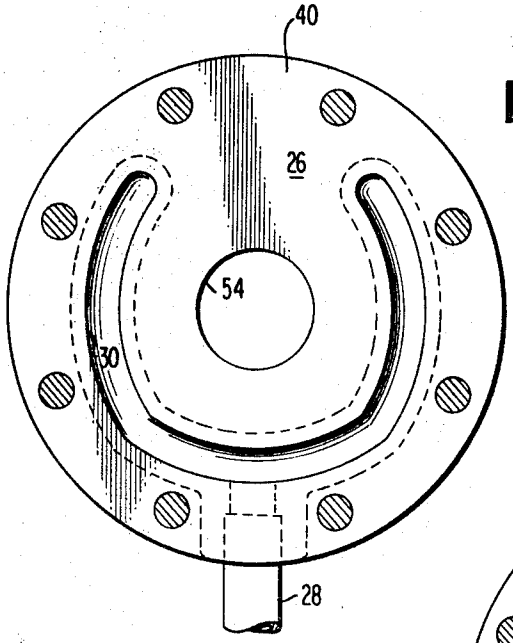


FIG.3

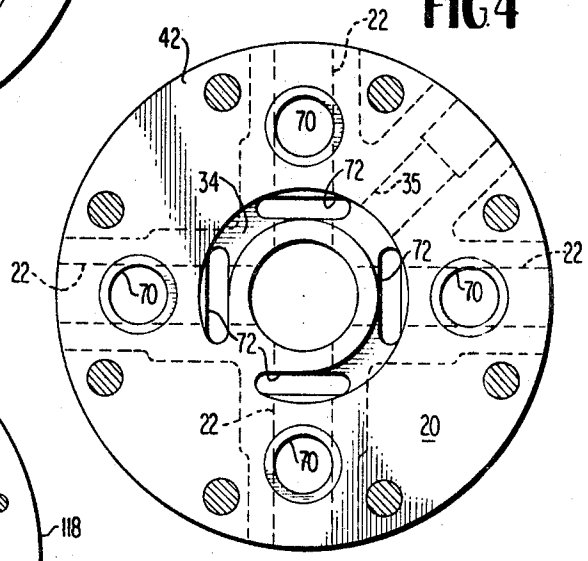


FIG.4

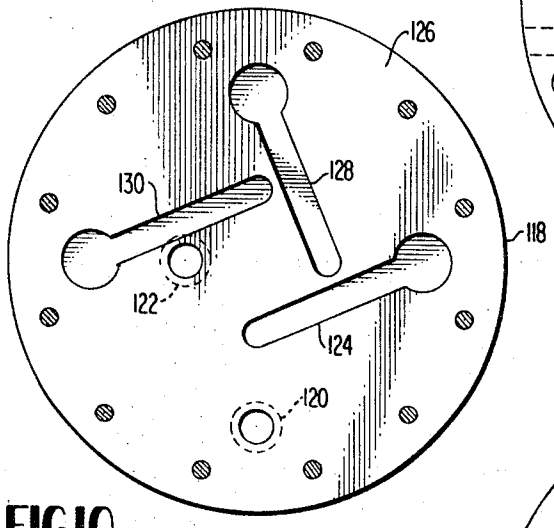


FIG.10

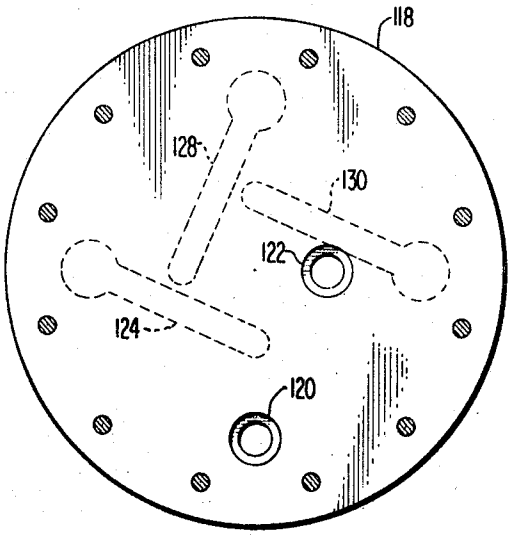


FIG.9

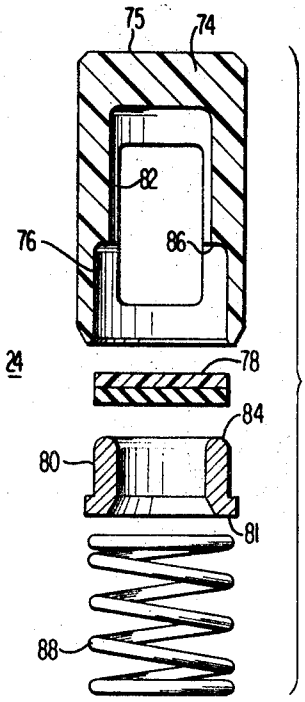
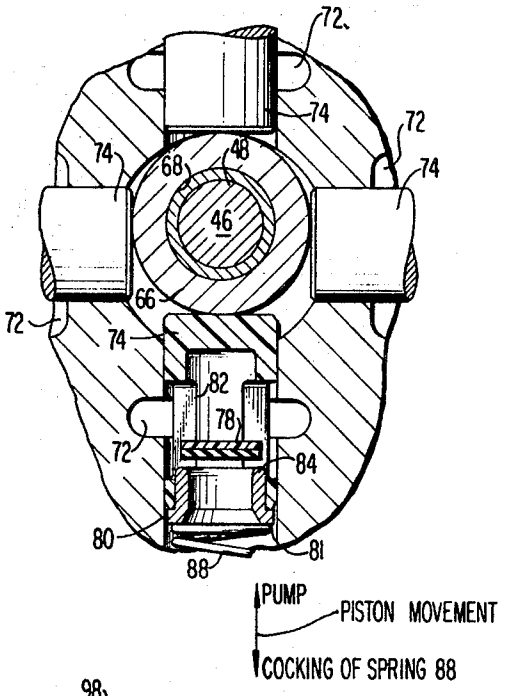


FIG.5

FIG.6



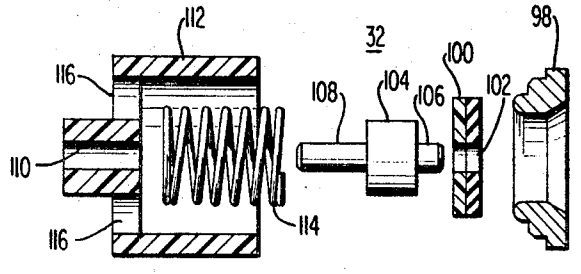


FIG.7

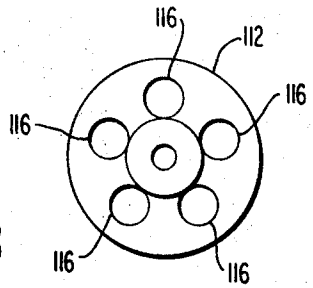


FIG.8

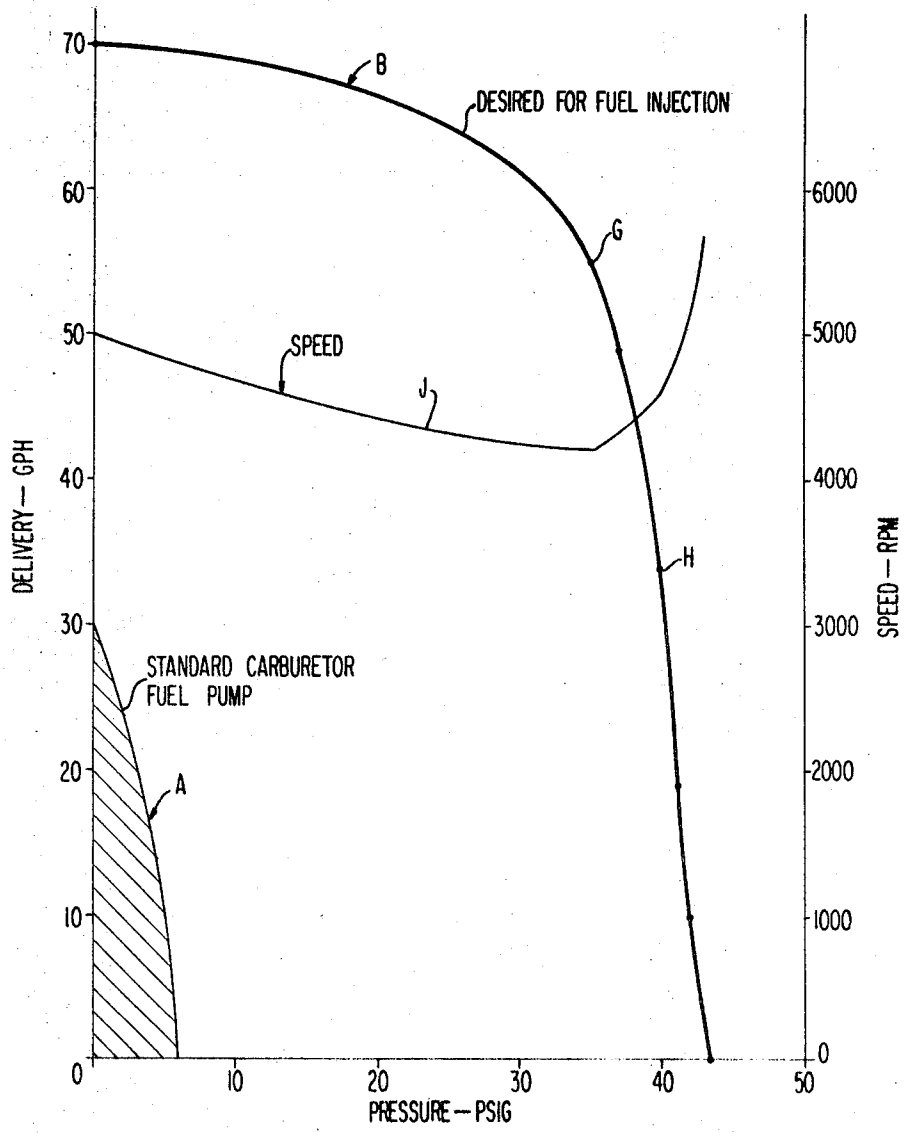
INVENTOR

E. David Lang &
K. Donald Aumick

BY Brady, O'Boyle & Gates

ATTORNEYS

FIG.11



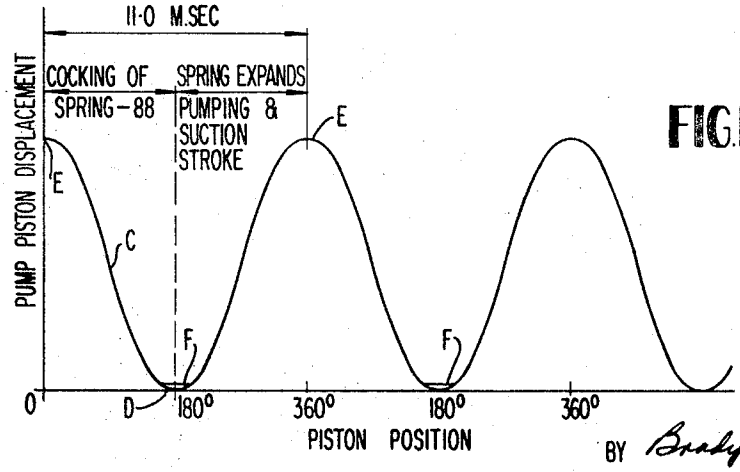


FIG.12

INVENTOR
E. David Lang &
K. Donald Aumiller
BY Brady, O'Boyle & Gates
ATTORNEYS

SPRING ACTUATED FUEL PUMP FOR FUEL INJECTION SYSTEMS

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention is directed to pumps for supplying a fluid such as fuel in a fuel injection system to a predetermined number of cylinders of an internal combustion engine. Fuel pumps for fuel injection systems differ from the standard fuel pump utilized in combination with a fuel carburetor in that the output fuel pressures needed are increased in magnitude by five to 10 times. Moreover, fuel injection systems operating on a noncontinuous flow concept require that the fuel distribution system be adapted to provide fuel at essentially constant pressure regardless of the delivery requirements of the engine. Heretofore such systems have included a multiplicity of pumping concepts wherein a substantial variation in outlet pressure exists over the normal dynamic fuel delivery range of the operating engine. Further, such fuel injection systems require that the pressure of the fuel being held constant be maintained at relatively large values, i.e., around 40 p.s.i. Because the large variation of output pressure as a function of fuel delivery it has been necessary heretofore to provide separate pressure regulator mechanisms so adapted to compensate for the pressure versus delivery characteristics of the associated pump and thereby attempt to maintain a constant line pressure. Such separate pressure regulator means are expensive and are further troubled by a response time problem thereby allowing momentary excursions in the preset fuel pressure characteristics.

In a noncontinuous flow high speed fuel injection system, momentary excursions in output pressure are exceedingly harmful. Further, all pumping mechanisms known to the art heretofore are principally suited and adapted to provide large free delivery at zero or low pressures. Such a characteristic is frequently sought and is desirable in the case of carburetor or fuel transfer-type pumps. However, as the prime fuel supply pump for a fuel injection system, these pumps are not suitable since the great majority of all operation on an automobile takes place under cruising conditions wherein the fuel requirement is usually less than 5 g.p.h. The pumping elements being driven directly by the pump motor are forced to churn the fuel at a rate consistent with its free delivery characteristic. Such churning causes vapor generation within the pump and substantially agitates and weathers the fuel. Further, since such a pump operating under these conditions is essentially at cutoff pressure, the load seen by the pumping motor is at its greatest thereby substantially increasing heat generation as well as wear. Historically such pumps have been totally unsuitable as the prime supply pump for noncontinuous flow fuel injection systems.

2. Description of the Prior Art

In the art of reciprocating piston pumps, a piston is reciprocally mounted in a bore and powered by an external means whereby reciprocal movement of the piston causes valve means to force a fluid to flow unidirectionally from an input port to an output port. In known prior art apparatus this piston may be operated singly or may even be arranged radially around a central axis in combination with a selected number of other pistons and driven by an eccentric located on the central axis which is also perpendicular to the plane of the cylinders. Moreover, the eccentric effects the pumping action by direct action of the eccentric causing reciprocatory motion of the piston, i.e. the pumping stroke is produced by the eccentric as it is forces the piston to move away from the axis of the eccentric.

There is also known to those skilled in the art a fuel pump which is electromagnetically actuated wherein the pumping action is produced by means of a spring which actuates a piston through its pumping stroke. Such teaching is found in U.S. PAT. No. 2,833,221 issued to J. W. Dickey wherein an electromagnet operates to compress or cock a pumping spring which then is adapted to actuate a pumping piston on its discharge stroke. Such a pump, however, comprises a single pumping stage and necessarily includes additional means for overcoming fluctuations of the output pressure. Also it is limited in its ability to provide various types of outputs such as low pressure—high volume or high pressure—low volume depending on the specific requirement. Additionally, such a pump cannot produce a constant pressure vs. variable delivery characteristic required with certain fuel injection systems.

SUMMARY

The present invention is directed to an improved piston type fuel pump wherein a pumping piston is reciprocally located in a piston bore and being operable so that the pumping stroke is effected by a pumping spring which is cocked or compressed by means of an eccentric rotor which is driven by an electric motor and wherein the rotor's eccentric surface is adapted to contact one end of the pumping piston. Moreover, a plurality of pumping pistons are located in respective bores radially located in a housing with the rotor being mounted perpendicularly to the plurality of pumping pistons so that the eccentric portion of the rotor is adapted to contact all of the pumping pistons. The housing includes an input feed manifold which is adapted to couple fuel to a respective foot valve which then delivers fuel into the side of each bore intermediate ends thereof whereupon it is pumped through a disc valve disposed in the piston to an output port also located in the side of each bore. Additionally a fluid-collecting ring is also coupled to each output port for feeding the output of the plurality of pumping pistons to a common output. The embodiment also allows for modification of the interconnection of inlet and outlet ports of the pumping pistons by rotation of the input feed manifold with respect to each of the pumping pistons and their respective foot valves or by utilizing an additional distributor plate inserted between the input feed manifold and the housing. Thus, when desirable, the pumping pistons may be staged in any manner desirable.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross-sectional view of the preferred embodiment of the invention taken generally along lines 1–1 of FIG. 2;

FIG. 2 is a plan view of the subject invention illustrating the input feed manifold generally describing a ring type of configuration;

FIG. 3 is a cross-sectional view of the embodiment shown in FIG. 1 taken along the line 3–3, illustrating the underside of the feed manifold shown in FIG. 2;

FIG. 4 is a sectional view of the embodiment shown in FIG. 1 taken along the lines 4–4 and generally illustrating the inlet and outlet ports of four pumping stages arranged at mutually right angles with respect to one another in the housing of the subject embodiment;

FIG. 5 is an exploded cross-sectional view of a pumping piston and valve assembly utilized in combination with the embodiment shown in FIG. 1;

FIG. 6 is a fragmentary cross-sectional view of one piston assembly shown located in the embodiment as taken along lines 6–6 of FIG. 1;

FIG. 7 is an exploded cross-sectional view of a foot valve assembly utilized in combination with the embodiment shown in FIG. 1;

FIG. 8 is a bottom plan view of the foot valve assembly shown in FIG. 7 being illustrative of the openings in the bottom of the foot valve housing;

FIG. 9 is a top plan view of a fluid distributor plate used in connection with the input feed manifold to provide a series connection of the four pumping stages utilized with respect to the subject embodiment shown in FIG. 1;

FIG. 10 is a bottom plan view of the distributor plate shown in FIG. 9 being illustrative of the underside thereof;

FIG. 11 is a graph illustrative of the output characteristic of the subject invention in comparison to a carburetor type fuel pump; and

FIG. 12 is a diagram helpful in understanding the operation of the subject invention.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to FIG. 1, there is shown a motor driven fluid pump of the radial piston type wherein a plurality of pumping pistons are radially arranged around a central transverse axis 16. An eccentric drive shaft or rotor 18 is located along the axis 16 for powering the apparatus. More particularly, FIG. 1 discloses a pump housing 20 including a base member 21 which is adapted to contain four radial piston bores 22, one of which is shown. A reference to FIG. 4 illustrates the four pumping pistons and how they are adapted to be located in quadrature with one another. EAch bore 22 contains a pumping piston and disc check valve assembly 24 which is adapted to pump fluid such as fuel for a fuel injection system. Fuel is coupled to each of the pumping assemblies 24 by means of an input feed mainfold 26 including an input coupling 28 connected to a fuel delivery chamber 30. The feed manifold 26 comprises a raised ringlike structure as shown in FIG. 2 with the delivery chamber 30 having a diminishing cross-sectional area as it proceeds away from the input coupling 28. The chamber 30 will be referred to hereinafter as the "intake ring." Fuel from the intake ring 30 is delivered into the side of each of the piston bores 22 by means of a foot valve assembly 32. Fuel is pumped from the intake ring 30 to an output "collector ring" 34 comprising a circular recess in the upper portion of the pump housing 20 inwardly of the intake ring 30.

While the upper surface 38 of the feed manifold 26 has a sloping contour away from the input connector 28 the lower surface 40 is substantially flat so that it is adapted to be contiguous with the upper surface 42 of the pump housing 20 and is secured thereto by means of screw elements 44.

The drive shaft assembly 18 is located along the central axis 16 of the housing 20 transverse to the pumping piston assembly 24 and comprises a shaft member 48 having an offset region 50 intermediate the ends thereof adjacent the piston assemblies 24. One end of the shaft 48 is secured to the housing 20 by means of the bearing member 52. A small recess 54 is provided in the feed manifold 26 surrounding the end of the shaft 48 so that a small lubricating reservoir may be provided thereat. The opposite end of the shaft 48 is secured to the fuel pump housing by means of the bearing member 56. A seal 58 is provided around the shaft member 48 adjacent the bearing 56 for retaining the bearing 56 and also providing a lubricating reservoir at that end. The shaft member 48 is directly coupled to a DC electric motor 60 and its shaft 62 by means of the rubber coupling 64.

In addition to the shaft member 48 the drive shaft assembly 18 is additionally comprised of a roller member 66 comprised of a hardened steel cylinder mounted around the offset portion 50 of the shaft member 48 and separated therefrom by means of a spacer or liner 68 comprised of low friction material such as Teflon. As the motor 60 turns the eccentric shaft member 48, the eccentric portion 50 rotates and imparts a rolling type motion to the steel roller member 66. The high relative speed imparted by the motor 60 appears essentially between the eccentric portion 50 and the Teflon spacer member 68 however.

Before proceeding to a more detailed description of the assemblies 24 and 32, attention is directed briefly to FIGS. 2 and 3 which disclose cross-sectional views of the embodiment shown in FIG. 1 along lines 3–3 and 4–4, respectively.

FIG. 3 discloses the underside of the feed manifold 26 and the surface 40 and the fuel delivery intake ring 30 which comprises a recess in the body portion of the feed manifold 26. It is shown comprising only approximately three-quarters of a complete continuous ring due to the location of the pumping valves as shown in FIG. 4. Referring now to FIG. 4, there is shown a plan view of the surface 42 of the pump housing 20 and the radial distribution of the bores 22 which are adapted to contain the pumping piston and valve assemblies 24 shown in FIG. 1. Additionally there is disclosed four input ports 70 and four output ports 72 located intermediate the ends of the bores 22. Moreover, each of the output ports 72 couple into the circular collector ring 34 which is then coupled to a single exit channel 35.

Considering now the piston and valve assembly 24 located in the bore 22, FIG. 5 shows an exploded view of the assembly while FIG. 1 discloses the assembly in situ. The valve assembly 24 is comprised of a hollow cylindrical piston 74 which is closed at one end but includes an annular recess 76 at the other end into which a circular disc valve member 78 is adapted to fit, being retained therein by means of a hollow circular valve seat 80 having a distal end portion 84 which acts as the seat. The valve seat 80 is press fitted into the piston 74. The piston 74 additionally contains a rectangular opening 82 in each side thereof. When the disc valve 78 is mounted inside of the piston 74 and held in place by the valve seat 80 the valve 78 comprises a freely movable valve movable between the raised portion 84 of the valve seat 80 and the shoulder 86 inside of the piston 74. Movement of the disc valve 78 between these two positions operate to pass or block the flow of fuel from the region surrounding the valve seat 80 to the inner portion of the piston 74 and rectangular opening 82 which is adjacent the output port 72. Pumping action is provided by means of a pumping spring 88 positioned against the assembly 24 in combination with the drive shaft assembly 46 including the steel roller member 66 in a manner to be described hereinafter.

Reference to FIG. 6 illustrates the manner in which four piston valve assemblies 24 such as disclosed in FIG. 5 are arranged in quadrature with the piston valves 74 being in contact with the roller member 66. Additionally, the disc valve 78 is shown in its open position away from the surface 84. In this position fluid such as fuel surrounding the pumping spring 88 is adapted to flow around the flapper valve 78 into the interior of the piston 74 and into the output port 72 by means of the rectangular opening 82. At this point it might be stated that the operation of the piston valve assembly is such that the rotation of the drive shaft assembly 46 causes compression or cocking of the pumping spring 88 as well as lifting the flapper valve 78 as shown in FIG. 6. As the rotor portion 50 turns away from the piston 74 the pumping spring 88 produces a pumping stroke by expanding against the outer edge 81 of the valve seat 80 causing the flapper valve 78 to seat on the surface 84 at which time liquid or fuel retained interiorally of the piston 74 is forced out of the rectangular opening 82 into the output port 72 and then into the collector ring 34 by movement of the piston 74 toward the roller member 66.

This can further be demonstrated by reference to FIG. 1 wherein the piston valve assembly such as shown in FIGS. 5 and 6 is located in its respective bore 22, being held in place by means of a retaining plug 90 and a spring clip 92. Additionally, an O-ring 94 is provided for obtaining a liquid seal. It is interesting to note that the retaining plug additionally includes a protrusion 96 which is adapted to reduce the volume surrounding the pumping spring 88 to assist in priming of the piston valve assembly 24.

Delivery of the fluid or fuel to the piston pumping assembly 24 is produced by the foot valve assembly 32 coupled into the side of the bore 22 intermediate the ends thereof and more particularly at the location of the pumping spring 88. The volume surrounding the pumping spring 88 interior of the bore 22 comprises the input chamber of the piston assembly 24 and is adapted to be coupled to the intake ring 30 by means of the foot valve assembly 32 which has its input port 70 coupled to the intake ring 30. The details of the foot valve assembly 32 are shown in an exploded view at FIG. 7 and is comprised of a valve seat 98 and a circular valve member 100 including a mating hole 102 for engagement with a valve stem 104 by means of the projection 106. The valve stem 104 additionally includes a guide pin 108 which is adapted to mate with an elongated guide 110 in the foot valve housing 112. A biasing spring 114 is located interiorally of the foot valve housing 112 and is adapted to bias the valve member 100 against the

valve seat 98. The foot valve housing 112 additionally includes five holes 116 in the lower end thereof adjacent the extended recess 112 so that fuel may pass into the bore 22.

Considering the overall operation of the embodiment shown in FIG. 1, the electric motor 60, which is preferably a DC motor, causes rotation of the shaft 48 to effect a rolling-type movement of the roller member 66 against the piston valve 74 urging it toward the retaining plug 90 and thereby cocking or compressing the pumping spring 88. Although the eccentric shaft 48 is turning at a rapid rate, little or no turning of the roller member 66 occurs with reference to the piston face 75. A given point on the circumference of the roller 66 tends to stay relatively stationary with reference to a piston face 75. At this time the disc or flapper valve 78 moves away from the valve seat 80 against the shoulder 86 of the piston 74. This action causes an increment of fuel trapped in the bore 22 of the piston 74 to be transferred to the back or inner side of the flapper valve 78 with the quantity being transferred determined solely by length of stroke of the piston 74 during the cocking stroke. The fuel flows around the disc valve 78 into the interior portion of the piston 74. Next as the eccentric portion 50 of the drive shaft 48 rotates away from the piston 74 the pumping spring 88 moves the piston 74 away from the foot valve assembly 82 at which time the disc valve 78 seats on the surface 84 of the valve seat 80 causing fuel captured inside of the piston 74 to be urged into the exit port 72 from the opening 82 and from there into the collecting ring recess 34. At the same time a sucking action is created against the normally closed spring biased foot valve 32 causing it to open and allowing fuel to enter the bore 22 from the intake ring 30.

The present invention moreover is adapted to provide a variable volume substantially constant pressure output over a predetermined operating range. This is illustrated with reference to FIGS. 11 and 12. FIG. 11 is illustrative of the output characteristic of the present invention as compared against a standard carburetor fuel pump. Curve A is indicative of the fact that the output of such a standard carburetor fuel pump is not only not constant for its delivery capability but is also not desirable for operation in connection with a fuel injection system. The output characteristic of the present invention however is illustrated by Curve B and indicates that its output capability is in the region of 70 gallons per hour for unobstructed output flow (zero output pressure) but as the output is loaded for example, with a plurality of fuel injectors, a transition region G occurs and the output pressure remains relatively constant.

This characteristic can further be explained by referring to the illustrative diagram of FIG. 12 wherein a sinusoidal curve C is representative of the travel of the piston 74 in the bore 22 between the limits D and E which are dictated by the maximum throw or offset of the eccentric portion 50 of the shaft 48 FIG. D. The zero or 360° point E would be the farthest projection of the piston 74 towards the center axis 16 of the housing 20. This position defines the end of the pumping stroke. The movement of the piston 74 back into the bore 22 by means of the roller 66 on the other hand effects cocking of the pumping spring 88. The 180° degree position is defined as that point where the face 75 of the piston 74 coincides with the edge of the bore 22 which is common to the surface of the roller 66.

Assuming that the present invention is loaded by means of a fuel injection system, fuel will fill the collector ring 42 and the output port 72 as well as the interior portion of the piston 74. As the pressure on the output side of the disc valve 78 builds up, a transition region G (FIG. 11) occurs for example, in the region of 35 p.s.i.g. whereupon the travel of the pumping piston 74 becomes very small in comparison to its total possible displacement. This region is illustrated by the region F of curve C in FIG. 12. Such operation is analogous to a spring-loaded piston riding in a bore of infinite length containing fuel with the spring being of infinite length located on the opposite side of the piston from the fuel. If a fuel outlet is placed at the end of the infinitely long bore and is coupled to a load, the load offered by the spring will be constant regardless of the piston position. The pressure in the fuel column can be determined very simply by dividing the available spring force by the area of the piston. It becomes obvious that fuel existing between the piston and the outlet is free to move only on demand made by the load at which time the piston will move to displace only that volume of fuel required. On this basis, motion is slaved entirely to the demand relationship. Spring force being held constant implies a constant outlet pressure regardless of demand. This is an ideal concept however, the present invention is a practical embodiment of this concept in that the pumping spring 88 although having a finite length is being maintained in compressed state at essentially constant length due to the very high cocking rate as determined by the speed of the DC motor 60. The cocking action of the spring 88 at the same time allows the fuel reservoir to be replenished with the amount of replenishment based solely upon the fuel required during the preceding cycle of the motor shaft 62 and consequently the rotor 48. Throughout, however, it should be observed that the fuel pressure is exactly equal to that value established by the spring 88 and the pumping motion of the plunger or piston 74 is accordingly determined solely by the output demand and not by the motor 60.

Also it is significant to note that whereas in prior apparatus the pumping is produced by the actuating motor usually having some kind of an eccentric mechanism coupled thereto, the present invention derives its pumping action by means of a pumping spring 88 which is designed not merely for biasing purposes but in actuality performs the word and significantly reduces the load on the DC motor 60 when the point of transition G occurs. At the point of transition the work load which is the product of the output pressure and delivery reduces as the output delivery decreases; however, the pressure remains essentially constant due to the pumping spring characteristic. The decrease in work load causes the DC motor 60 to speed up which is characteristic of such a motor. This is evidenced by curve J of FIG. 11. The speed up of the motor 60 acts to further vertically straighten the substantially constant pressure portion H of curve B, FIG. 11.

The embodiment of the present invention shown and described with reference to FIGS. 1 through 8 discloses a feed manifold configuration 26 wherein each of the four pumping stages are commonly fed by the intake ring 30 and produce a common output by means of the collector ring 34 and the output channel 35. The four pumping stages then may be said to be operating in parallel. It is also within the scope of the present invention that the four pumping sections or stages may be operated other than is presently illustrated. For example, by rotating the feed manifold 26 one-eighth of a revolution one of the pumping sections may be rendered inoperative. Also the present invention contemplates a distributor plate such as shown in FIGS. 9 and 10 being placed between the undersurface 40 of the feed manifold 26 and the upper surface 42 the pump housing for providing a configuration where the four stages for example are cascaded, i.e. operated in series fashion.

Directing attention now to FIGS. 9 and 10, there is disclosed the upper side and lower side, respectively, of a series distributor plate 118 and includes an input opening 120 and an output opening 122. The plate is adapted to feed the input fuel to a single pumping section only by means of the input opening 120 which is adapted to then allow fuel to flow therethrough from the intake ring 30. The fuel pumped by the first stage is then coupled to an adjacent stage by means of an intermediate channel 124 fabricated into the undersurface 126 of the distributor plate. The intermediate channel 124 does not extend through the width of the distributor plate 118 but is merely of sufficient depth to provide unrestricted flow of fuel from the output of the first pumping stage to the input of the second. A third and a fourth intermediate channel 128 and 130, respectively, series couples the first and second stages to the third and fourth stages whereupon the output opening 122 feeds into the collector ring 34 adjacent the output channel 35.

By use of other configurations of intermediate channels in a distributor plate such as shown with respect to the distributor plate **118** other combinations of the pumping sections can be achieved such as, for example, a series-parallel combination wherein pairs of pumping sections are operated in series with a single input and output.

What has been shown and described therefore is an improved fluid pump which is particularly suitable for being utilized as a constant pressure fuel pump for a fuel injection system for internal combustion engines. The arrangement and operation of the pumping sections being operated by a pumping spring having been cocked by an eccentric rotor driven by a DC electric motor provides an improvement in constant output pressure over the operating range heretofore unobtainable. Finally, the present invention discloses the manner in which the various stages can be suitably intercoupled by selective manifold means whereby the pump may be modified to suit the requirements of a particular application.

Although the present invention has been described with a certain degree of particularity, it should be understood that the present disclosure has been made by way of example only and that numerous changes in the detail of the apparatus in the combination or arrangement of elements may be resorted to without departing from the scope and spirit of the invention.

We claim:

1. A piston-type pump adapted to supply fluid on demand under substantially constant pressure, comprising in combination:

a pump housing including a powered rotatable drive shaft means located along the central axis of the housing, said drive shaft means having an offset portion intermediate the ends thereof;

at least one piston bore located in said housing transverse to said central axis and extending radially therefrom;

a pumping piston assembly including a reciprocally movable piston located in said bore and having an open end portion, a face portion, a generally hollow inner portion, and a generally cylindrical side portion, said face portion being adapted to cooperate with said offset portion of said drive shaft means so that said piston is moved thereby away from said central axis of said housing during a suction stroke of operation, a freely movable circular disc check valve member mounted in said hollow inner portion being adapted to travel a predetermined distance axially therein between an open and a closed position, a generally hollow cylindrical valve seat fitted into said open end portion of said piston and having a distal end projecting inwardly of said inner portion, said distal end being adapted to restrain the travel of said disc check valve member in one direction and defining said closed position thereby, said piston additionally having a shoulder located in said hollow inner portion opposing said distal end of said valve seat and being adapted to restrain the travel of said disc check valve member in an opposite direction and defining said open position thereby, at least one opening in said side portion, said opening having a dimension axially along said side portion at least equal to the travel of said disc check valve member and in registration therewith, whereby fluid passes through said generally hollow valve seat and around said disc check valve member into said hollow inner portion when said disc check valve member is in said open position but passes only through said at least one opening out of said generally hollow inner portion when said disc check valve member is in its closed position seated against said cylindrical valve seat during a pumping stroke of operation;

a pumping spring located in said bore on the opposite side of said piston assembly from said drive shaft means and having one end thereof in contact with the proximal end of said cylindrical valve seat and being operable to effect said pumping stroke;

means located in said bore adjacent the opposite end of said pumping spring for providing a rigid base for said pumping spring and additionally closing one end of said bore;

fluid input means coupled to said bore in the region of the location of said pumping spring for delivering fluid to said piston assembly; and

fluid output means including an output port extending across said bore in the region of the location of said piston side portion and adapted to be in registry with said at least one opening, said drive shaft means being operable to compress and cock said pumping spring whereupon fluid in said bore moves said freely movable disc check valve member against said shoulder and flows into said hollow inner portion, said pumping spring subsequently effecting a relatively short movement of said piston towards said central axis of said housing on output demand whereupon the fluid in said hollow inner portion urges said disc check valve against said valve seat which then forces the fluid captured in said hollow inner portion of said piston into said output port.

2. The invention as defined by claim **1** wherein said at least one opening comprises a generally rectangular opening wherein said axial dimension is substantially greater than the travel of said circular disc check valve member.

3. The invention as defined by claim **1** wherein said distal end of said valve seat has a generally tapered shape for minimizing the contact surface between itself and said disc check valve member.

4. The invention as defined by claim **3** wherein said tapered shape comprises a generally rounded surface.

5. The invention as defined by claim **1** wherein said fluid input means comprises normally closed valve means, but being responsive to the movement of said piston assembly towards said central axis of said housing to open and deliver fluid into said at least one piston bore.

6. The invention as defined by claim **5** wherein said normally closed valve means comprises a spring-biased foot valve.

7. The invention as defined by claim **1** wherein said fluid input means additionally comprises an input feed manifold including a distribution chamber for delivering fluid to said at least one piston bore.

8. The invention as defined by claim **1** wherein said output port is located substantially parallel to said central axis of said pump housing and wherein said output means additionally includes a substantially circular recessed collector ring adjacent said output port and coupled thereto, and having a center substantially along said central axis of said pump housing.

9. A piston-type pump adapted to supply a fluid such as fuel to an internal combustion engine on engine demand and under substantially constant pressure, comprising in combination:

A pump housing including a powered rotatable drive shaft means located along the central axis of the housing, said drive shaft means additionally having an offset portion intermediate the ends thereof;

a plurality of piston bores located in said housing transverse to said central axis and extending radially therefrom;

a pumping piston assembly including a reciprocally movable piston located in each of said plurality of bores adapted to cooperate with said drive shaft means, said piston having a substantially cylindrical outer surface and a substantially hollow inner portion including an abutment therein, an unbiased circular disc check valve mounted in said inner portion being adapted to travel a predetermined linear distance axially between an opened and closed position, a generally hollow cylindrical valve seat inserted into one end of said piston and having an end extending inwardly of said inner portion, said end having a tapering projection to contact said disc check valve and restrain the travel of said check valve in one direction and defining the closed position thereby, said piston additionally including at least one relatively large opening between said cylindrical outer surface and said inner portion of said piston and extending a distance equal to at least half

of the length of said piston whereby fluid may pass through said cylindrical valve seat into said inner portion of said piston when said disc check valve is in said opened position whereupon it is in contact with said abutment, and then passes through said relatively large opening from said inner portion when said check valve is in said closed position;

a pumping spring located in each of said plurality of bores on the opposite side of said respective piston from said drive shaft means and having one end thereof in contact with said respective cylindrical valve seat;

means located in each of said plurality of piston bores adjacent the opposite end of the respective pumping spring and additionally closing one end of the respective bore;

respective fluid input means coupled into the side of each of said plurality of piston bores substantially transverse to the axis of said pumping spring adjacent said cylindrical valve seat; and

respective fluid output means including an output port extending across each of said plurality of piston bores in the region of the location of said relatively large opening and adapted to come into registry therewith.

10. The invention as defined by claim **9** wherein said respective fluid input means comprises respective second valve means normally biased in a closed position but becoming operative to open when said check valve is in said closed position; and

a unitary feed manifold means attached to said housing adjacent each respective second valve means and includes a delivery chamber for selectively feeding input fluid to each said second valve means.

11. The invention as defined by claim **10** and additionally including an apertured plate located between said unitary feed manifold means and said housing for selectively directing fluid flow between said respective fluid input means to said respective output means.

12. The invention as defined by claim **10** wherein said unitary feed manifold means includes a curvalinear delivery chamber located substantially transverse to said central axis of said housing.

13. The invention as defined by claim **9** wherein said plurality of piston bores comprises four piston bores.

14. The invention as defined by claim **13** wherein said four piston bores are located mutually perpendicular to one another.

15. The invention as defined by claim **9** wherein said respective output means additionally includes a substantially circular collector ring coupled to each said output port, being located adjacent said input means and wherein said input means includes feed manifold means being attached to said housing and having a closed surface portion facing said substantially circular collector ring.

16. The invention as defined by claim **15** wherein said plate comprises a single input aperture and a single output aperture and a plurality of recesses located in one side thereof next to said respective input and output means to selectively interconnect said plurality of piston bores.

17. The invention as defined by claim **9** wherein said offset portion of said drive shaft means additionally includes a roller member coaxially fitted over said offset portion and a spacer member located between said offset portion and said roller member for providing slidable contact therebetween thereby allowing said roller member to revolve freely about the axis of said offset portion.

18. The invention as defined by claim **17** wherein said roller member is comprised of metal and has a substantially cylindrical configuration and wherein said spacer member is comprised of a low friction material and also has a substantially cylindrical configuration.

19. The invention as defined by claim **18** wherein said low friction material comprises Teflon.