Nov. 7, 1961 F. B. BURT 3,007,419

POSITIVE DISPLACEMENT PUMP

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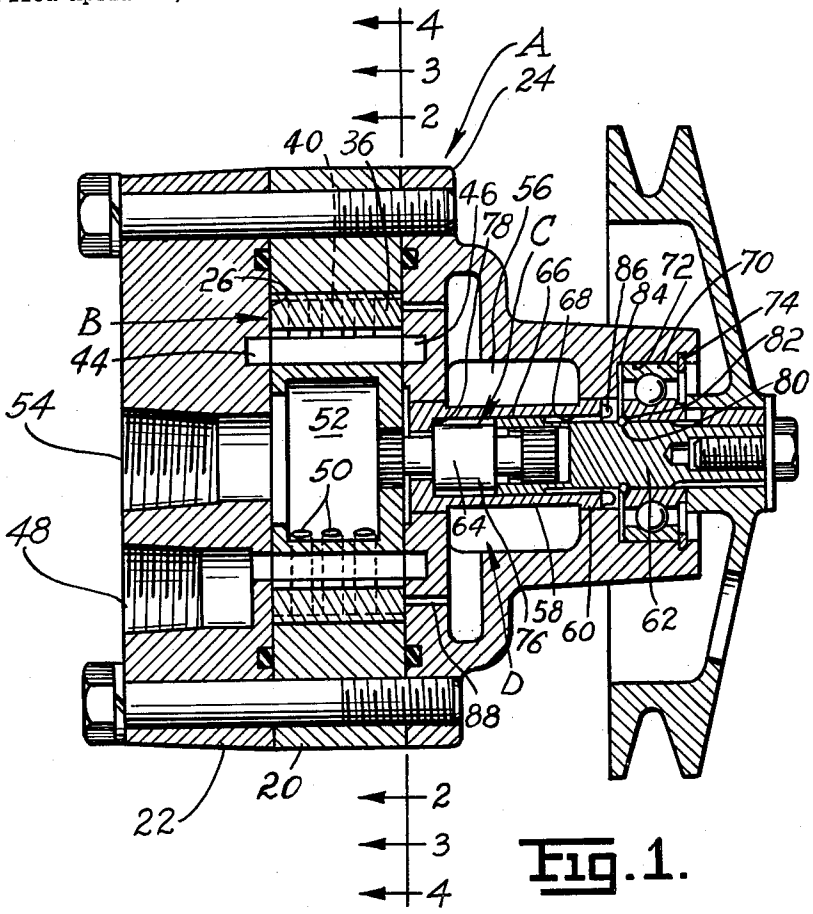


Fig. 1.

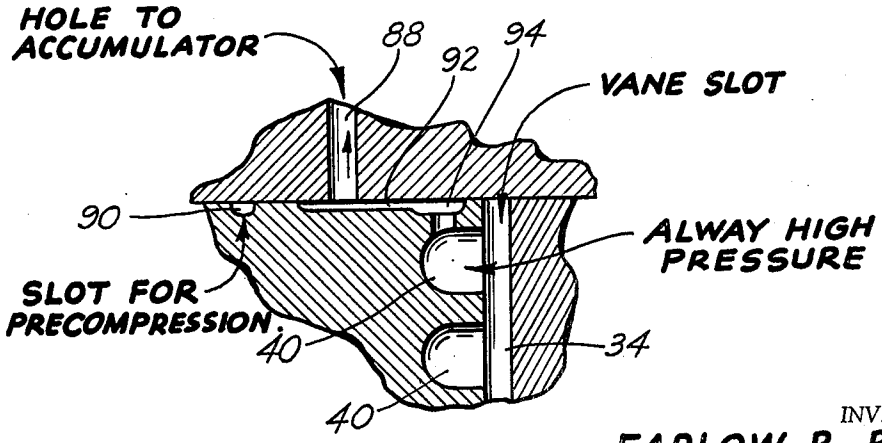


Fig. 5.

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POSITIVE DISPLACEMENT PUMP

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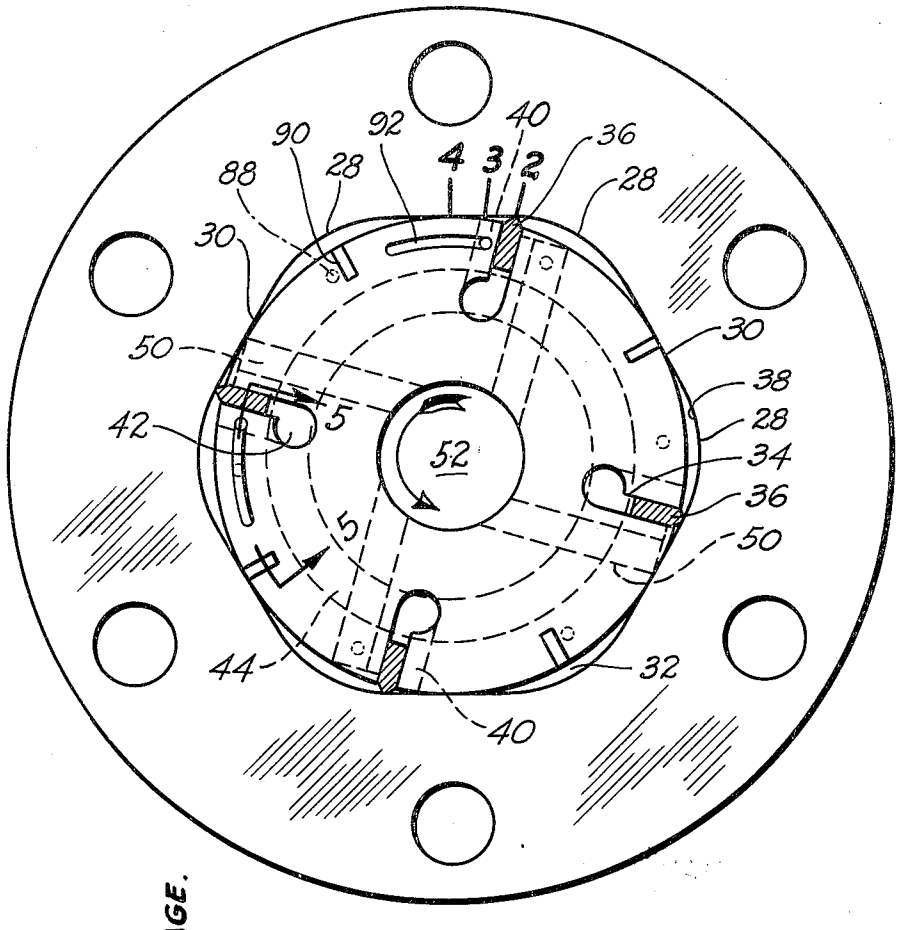


Fig. 2.

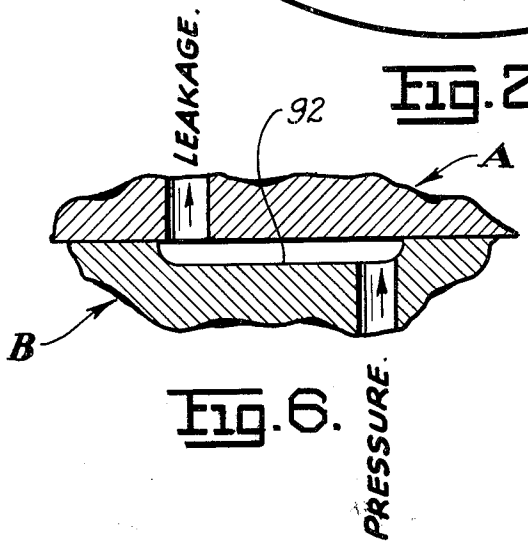


Fig. 6.

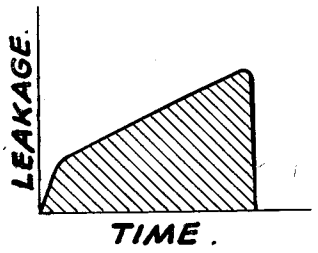


Fig. 7.

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POSITIVE DISPLACEMENT PUMP

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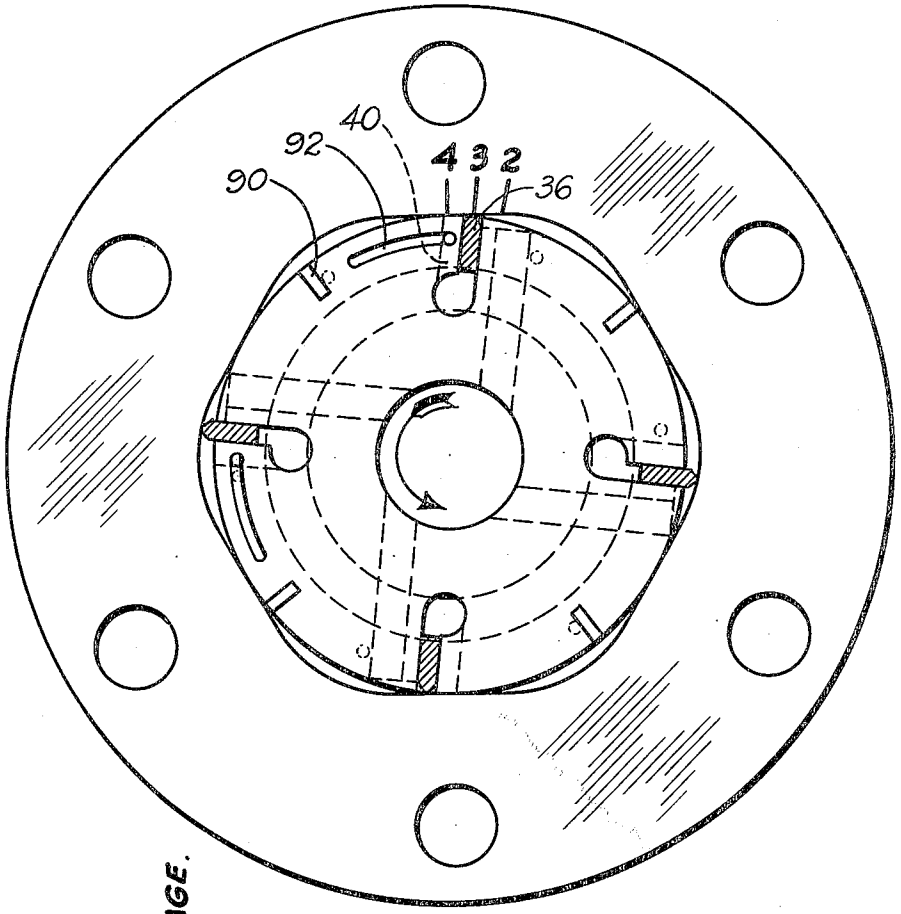


Fig. 3.

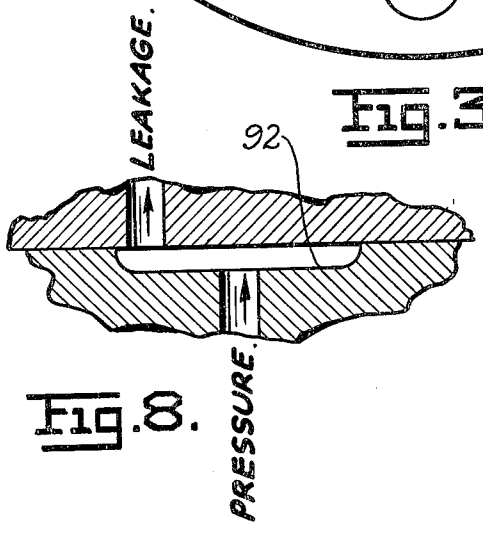


Fig. 8.

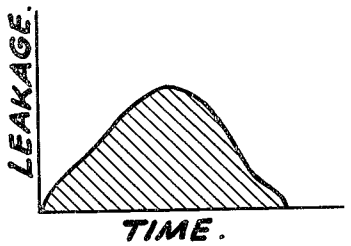


Fig. 9.

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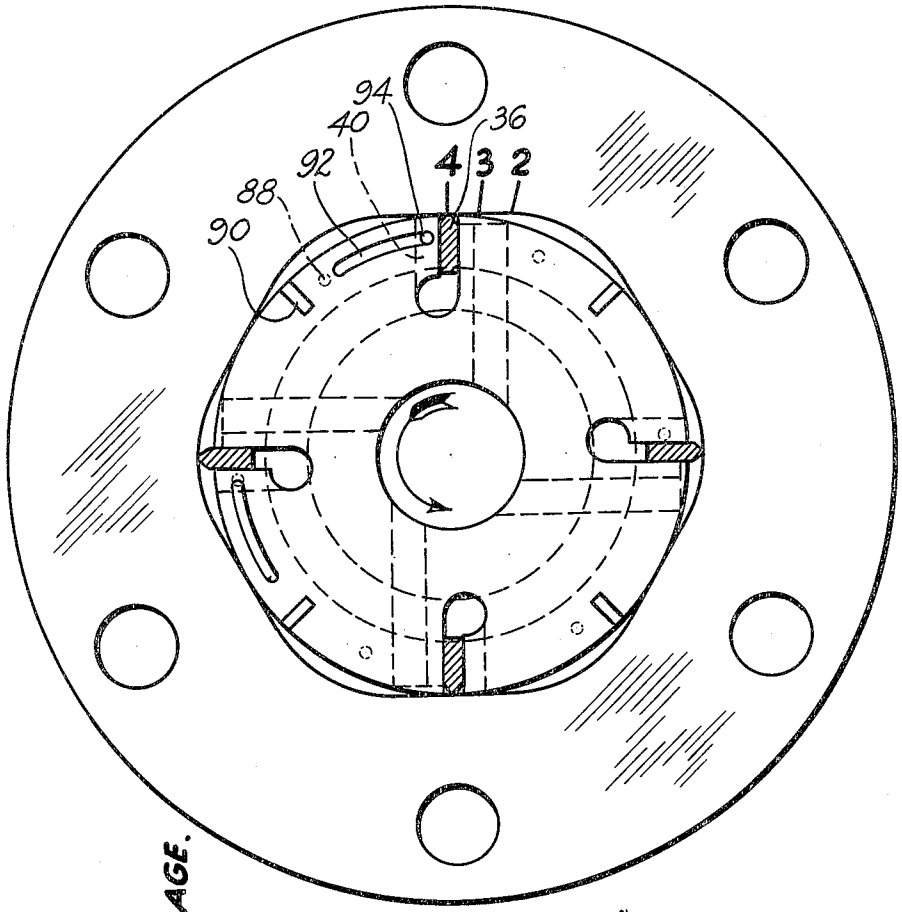


Fig. 4.

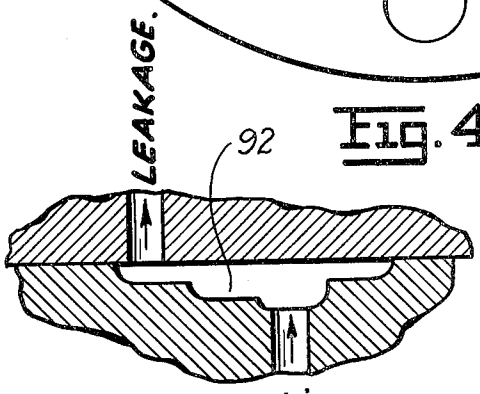


Fig. 10.

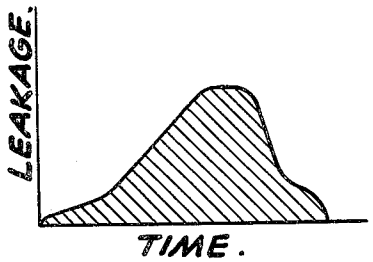


Fig. 11.

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POSITIVE DISPLACEMENT PUMP

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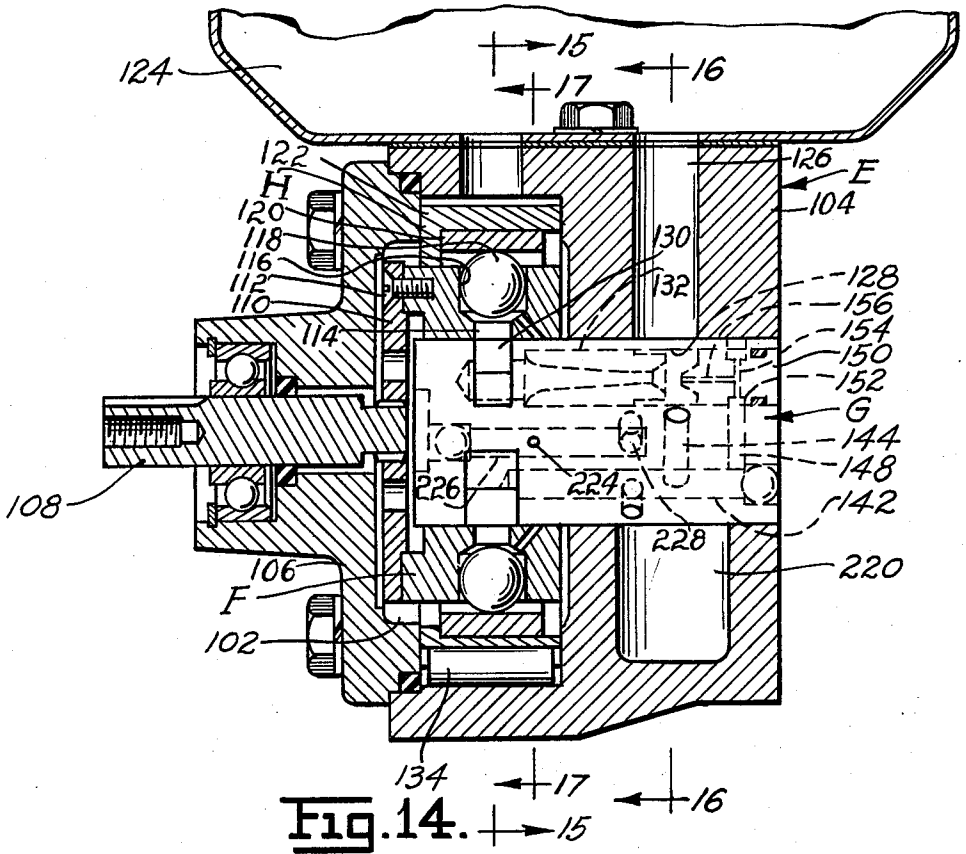


Fig. 14.

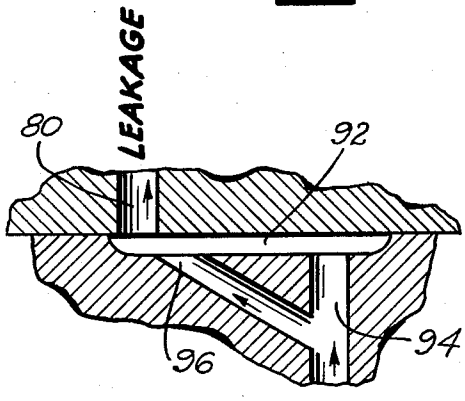


Fig. 12.

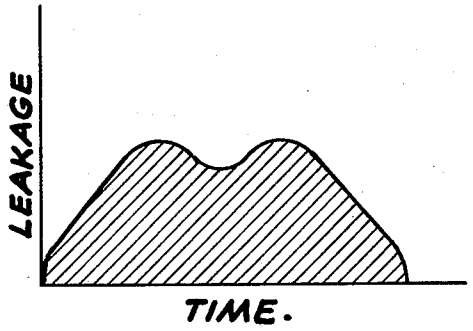


Fig. 13.

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POSITIVE DISPLACEMENT PUMP

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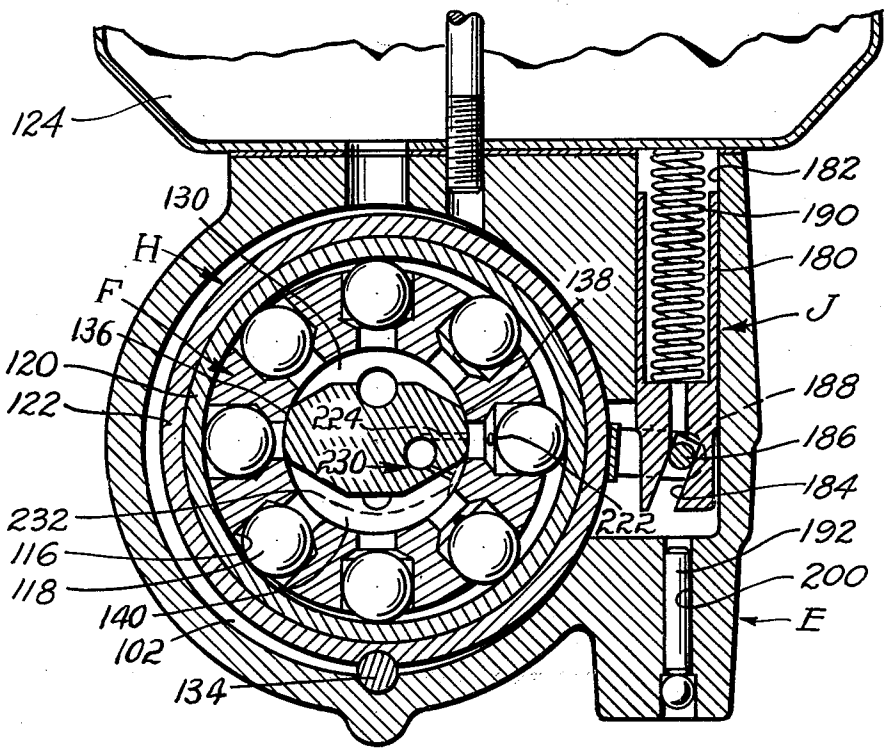


Fig. 15

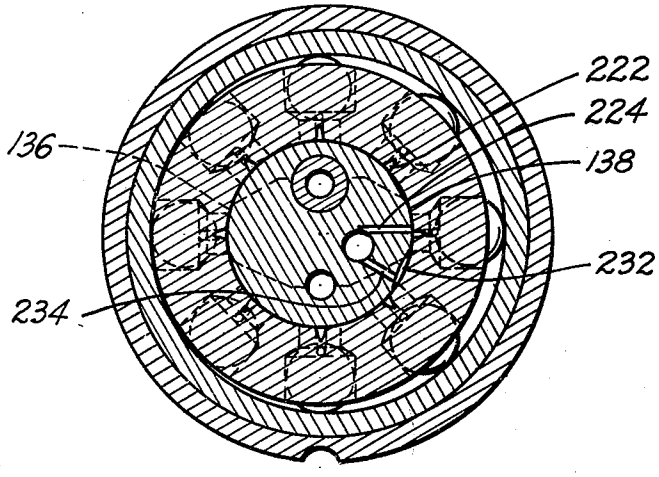


Fig. 17.

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POSITIVE DISPLACEMENT PUMP

Filed April 19, 1957 7 Sheets-Sheet 7

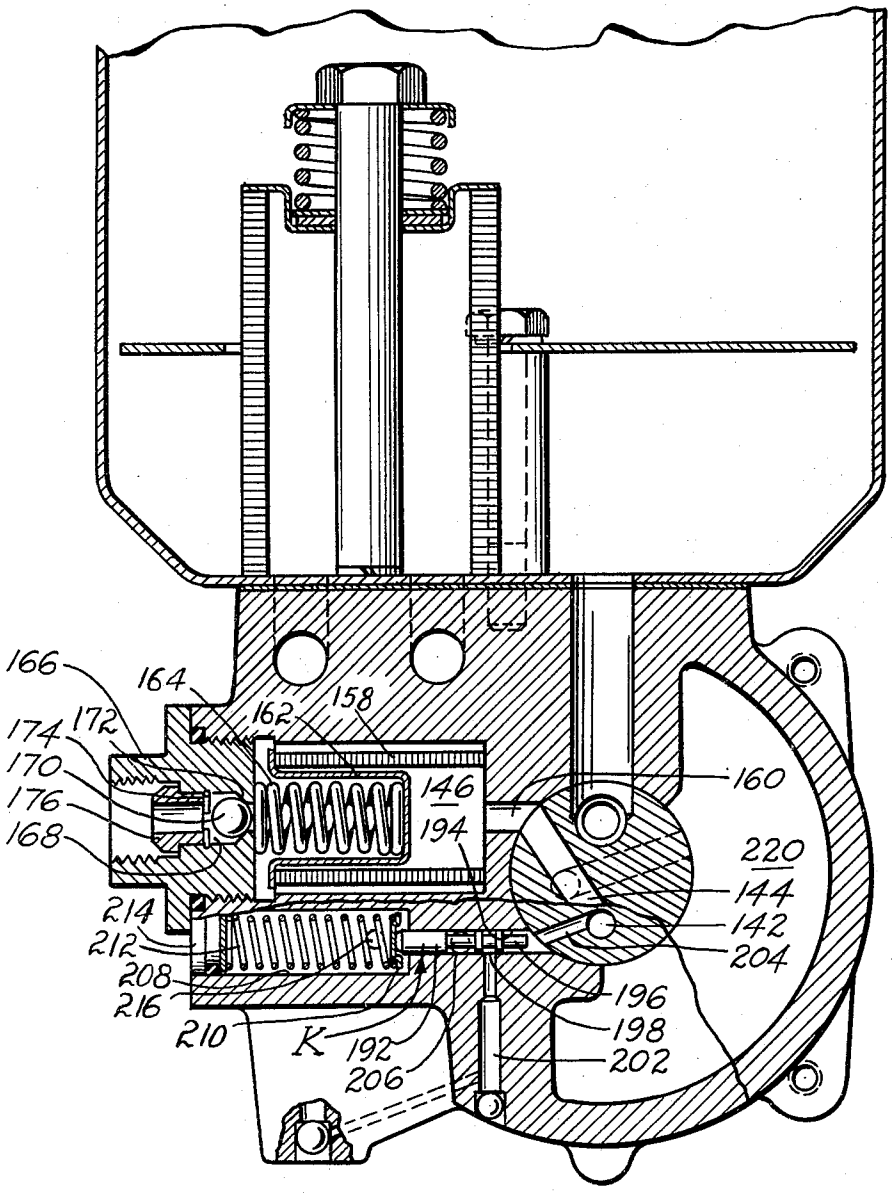


Fig. 16.

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United States Patent Office 3,007,419 Patented Nov. 7, 1961

3,007,419

POSITIVE DISPLACEMENT PUMP

Farlow B. Burt, South Bend, Ind., assignor to The Bendix Corporation, a corporation of Delaware

Filed Apr. 19, 1957, Ser. No. 653,846
11 Claims. (Cl. 103—136)

The present invention relates to positive displacement pumps generally; and more particularly to means for smoothing out pressure pulsations in the discharge of these pumps.

The various positive displacement pumps with which I am familiar (including gear, sliding vane, radial piston, and axial piston pumps) all produce pressure pulsations in their discharge for one of two reasons: (1) these pumps by their very nature may have instantaneous discharge rates which vary throughout the pumping cycle; and (2) where the device is used in high pressure systems, pressure surges will be developed when the pump's individual fluid confining chambers (that have been filled with fluid at suction pressure) are communicated to the pump's discharge because of compressibility of the fluid.

An object of the present invention is the provision of a new and improved positive displacement pump having means which will bleed liquid from the discharge portion of the pump in a manner offsetting the previously referred to pressure surges.

Another object of the invention is the provision of a new and improved pump having means of the above described type which comprise valve ports formed in cooperating sliding sealing surfaces of the rotor and an adjacent member such that the timing of the bleeding operation will be accurately and simply maintained with respect to the pump's cycle of operation.

Another object of the invention is the provision of a new and improved positive displacement pump comprising an accumulator reservoir maintained at a pressure approaching pump discharge pressure, means for communicating the accumulator to the pump's individual pressure chambers at a time when they are filled with liquid but valved off from both the suction and discharge portions of the pump to pressurize the individual chambers before they are communicated to discharge (thus partially eliminating pump discharge surges), and valve means operated in phase relation with the pump's rotor to bleed small amounts of liquid out of the pump's discharge in a manner offsetting remaining pulsations and smoothing out the fluid discharged from the pump.

A further object of the invention is the provision of a new and improved sliding vane positive displacement pump of a design having constant theoretical volume displacement per rotor cycle, and in which its rotor and casing have side surfaces which are in sliding sealing engagement and are provided with suitably shaped ports to accomplish the above objects.

A more particular object of the invention is the provision of a new and improved radial piston pump comprising a generally cylindrical pintle and an annular rotor surrounding the pintle, which members are provided with cooperating sliding sealing surfaces in which suitable porting is provided laterally of the cylinders to partially pressurize the individual cylinders prior to the time they are valved to discharge, and to bleed fluid from them when they are opened to discharge in a manner offsetting surges of pressure and thus smoothing out the discharge of the pump.

Further objects of the invention will become apparent to those skilled in the art to which the invention relates from the following description of several preferred embodiments described with reference to the accompanying drawings forming a part of this specification, and in which:

FIGURE 1 is a cross-sectional view of a sliding vane pump embodying principles of the present invention;

FIGURES 2, 3 and 4 are cross-sectional views taken upon the same cross-section line indicated in FIGURE 1, but showing the rotor in different positions;

FIGURE 5 is a fragmentary cross-sectional view taken on the line 5—5 of FIGURE 2;

FIGURE 6 is a fragmentary cross-sectional view similar to FIGURE 5 but showing a slightly different arrangement of porting;

FIGURE 7 is a diagrammatic view of a time-leakage curve which the structure shown in FIGURE 6 provides;

FIGURE 8 is a fragmentary cross-sectional view similar to that shown in FIGURE 5 but showing a different porting arrangement;

FIGURE 9 is a time-leakage curve which the structure shown in FIGURE 8 provides;

FIGURE 10 is a fragmentary cross-sectional view of structure similar to that shown in FIGURE 5 but showing a still further modification of porting;

FIGURE 11 is a time-leakage curve of the structure shown in FIGURE 10;

FIGURE 12 is a still further modification of the porting shown in FIGURE 5;

FIGURE 13 is a time-leakage curve for the structure shown in FIGURE 12;

FIGURE 14 is a cross-sectional view of a radial ball piston pump but showing its pintle in elevation;

FIGURE 15 is a cross-sectional view taken on the line 15—15 of FIGURE 14;

FIGURE 16 is a cross-sectional view taken on the line 16—16 of FIGURE 14; and

FIGURE 17 is a cross-sectional view taken on the line 17—17 of FIGURE 14 and showing the pintle, rotor and cam members only.

Although other embodiments of the invention can be made incorporating the principles set forth and covered by my claims (which embodiments may have their own unique advantages), the preferred embodiment shown in FIGURES 1 through 4 is specifically adapted for use in automotive power steering systems wherein it has particular advantages. As previously indicated, most positive displacement pumps discharge the liquid being pumped in a manner producing pressure pulsations, or surges, either because the pumps do not have a constant rate of discharge or because of the compressibility of the fluid being pumped. The pressure surges so created produce noise and vibrations in the systems to which the pump is connected which noise and vibration is transmitted to the driver's compartment of the vehicle through the steering column and/or frame of the vehicle.

The pump shown in FIGURE 1 generally comprises a casing or body member A formed by means of a generally annular center section 20, and opposite end closure members 22 and 24 to provide an internal pump chamber 26. The opposite end closure members 22 and 24 form generally flat parallel end surfaces for the internal pump chamber 26, which surfaces provide a sliding sealing engagement with respect to a cylindrically shaped rotor B positioned within the pump chamber 26.

The internal surface of the center section 20 of the pump casing A is provided with a plurality of valleys 28 which are separated by means of lobes 30 which sealingly abut the outer surface of the rotor B to provide a plurality of uniformly spaced individual pumping chambers 32 spaced about the rotor. The rotor member B is provided with four uniformly spaced radical slots 34 each of which receives a sliding vane 36 which is biased outwardly into engagement with the internal camming surface 38 of the pump to sweep liquid from the individual pumping chambers 32 during rotation of the rotor member B. Fluid swept from the individual pump-

ing chambers 32 passes through radial discharge slots 40 positioned in the rotor member B just forwardly of each sliding vane 36 to a transverse drilling 42, in the rotor member B at the inner end of each of the radial slots 34. Pump discharge pressure is therefore supplied to the inner end of each of the sliding vanes 36 tending to force the sliding vanes into more firm engagement with the internal camming surface 38 of the pump. Annular discharge grooves 44 and 46 are formed in the end closure members 22 and 24 respectively of the casing for communication with the respective ends of the transverse drillings 42. Discharge flow is therefore free to flow around each of the annular grooves 44 and 46 to distribute pressure equally upon both sides of the rotor and thence to a pump discharge connection 48 communicating with the annular discharge groove 44.

Fluid is added to each of the individual pumping chambers 32, from which fluid has been previously swept by the sliding vanes 36, by means of radial inlet drillings 50 positioned in the rotor member B immediately behind each of the sliding vanes 36. The inner end of the radial drillings 50 open into a centrally located suction chamber 52 in the rotor member B, which chamber receives fluid from the suction connection 54 of the pump in the closure member 22 of the casing A. The cooperating side surfaces of the end closure members 22 and 24, and rotor member B, have a sufficiently close fit to effect a running seal between the discharge grooves 44 and 46 and the inlet chambers of the rotor and casing member. The rotor member B is adapted to be driven by means of a shaft C splined to the center of the rotor member; and the internal camming surfaces 38 are of such a configuration that the volumes swept by each of the sliding vanes 36 during the rotation of the rotor will at all times be constant. The particular contour used for the camming surface 38 is a sinusoidal one; and for a more complete description of the construction and operation of the structure so far described, reference may be had to the Burt et al. application 623,144.

The pump shown in FIGURE 1 is provided with an accumulator reservoir or sink for the purpose of pressurizing the individual pumping chambers at a time when they are isolated from both the suction and discharge passages of the pump—such that when the individual pumping chambers are communicated to the discharge passageways of the pump, the amount of flowback into the pumping chambers due to the compressibility of the liquid will be greatly reduced. The accumulator reservoir or sink D is formed by means of an internal chamber 56 in the end closure member 24 surrounding the drive shaft C of the pump. The accumulator reservoir D is isolated from the shaft's rotating structure by means of a sleeve 58 which is pressed into the opening 60 in the closure member 24 in which the drive shaft C is positioned. The drive shaft C is formed in two sections—an outer section 62 tightly journalled in the end closure member 24, and an inner section 64 adapted to tolerate a small amount of angular movement between the rotor member B and the outer or driven shaft section 62 of the pump. The outer shaft section 62 is journalled at its inner end by a raised portion 66 which slidingly abuts the inner surface 68 of the sleeve 58, and is journalled at its outer end by means of anti-friction means 70 received in a counterbore 72 in the outer end of the axially extending opening 60. The anti-friction means 70 is held in place by the snap ring 74. The inner shaft section 64 is splined at both ends to provide a driving connection with respect to the rotor B, and shaft section 62, respectively; and is retained against axial movement by means of a raised cylindrical section 76 which is positioned between a shoulder 78 in the sleeve 58, and the inner end of the outer shaft section 62. The outer shaft section 62 is retained against removal from the pump by means of a wire snap ring 80 which is seated in a groove 82 in the shaft section 62, and which snap ring abuts the inner end of the anti-friction means 70. An annular seal retaining washer 84 is positioned between the anti-friction means 70 and the inner end of its counterbore 72, and an annular U-packing 86 is positioned between the washer 84 and the outer end of the sleeve 58 to prevent the leakage of oil out around the shaft structure.

The individual pumping chambers 32 are pressurized from the accumulator reservoir D by means of drilled passageways 88 (one for each valley 28 of the pump) extending between the accumulator chamber 56 and the internal pump chamber 26. The drilled passageways 88 open into the portion of the side walls of the chamber 26 which are sliding sealingly abutted by one side surface of the rotor member B. A plurality of radially extending grooves 90 (one for each vane) is positioned in this sliding sealing side surface of the rotor B in a manner adapted to communicate the reservoir D with the individual pumping chambers 32 when the grooves 90 are in registry with the drilled passageways 88. The radial extending grooves 90 are so positioned relative to the suction and discharge portion of the rotor so as to be in registry with the drilled passageways 88 at a time when the individual pumping chambers are isolated from both the suction and discharge passageways of the rotor. The drilled passageways 88 and radially extending grooves 90 will preferably be so constructed and arranged that the edge of the groove 90 will start to communicate with the passageway 88 just after the time that the suction groove 50, which has previously filled the pumping chamber, has moved past the sealing edge of the forwardly positioned land 30 of the pumping chamber to be pressurized (such a position is shown in the upper left hand corner of FIGURE 2). In this rotor position, the sliding vane 36, which is about to sweep the pumping chamber involved, will assume a position as shown in FIGURE 2 and as indicated in each of the FIGURES 2, 3, and 4 by the numeral 2. Continued rotation of the rotor B causes the groove 90 to move over the drilled passageways 88 thereby permitting flow from the accumulator into the pumping chambers being pressurized. Just prior to the time that the leading edge of the discharge groove 40 of the trailing vane about to sweep the pumping chamber registers with the sealing edge of the trailing lobe 30, the radially extending grooves 90 will move out of registry with the drilled passageways 88 (this position is shown in FIGURE 3 and is indicated by the numeral 3 in each of the FIGURES 2, 3, and 4). It will therefore be seen that pressurizing of the individual pumping chambers must take place while the rotor is moving between the positions 2 and 3, and that the drilled passageways 88 and grooves 90 will be so proportioned as to use as much of this rotor movement as is feasible.

According to further pinciples of the invention, a side stream of fluid is bled out of the pump discharge in amounts and at time intervals which will offset the pressure surges created in the pump discharge when the individual pumping chambers 32 are communicated therewith. Inasmuch as the accumulator reservoir D will not always be maintained at a pressure equaling the pressure discharge of the pump, a slight amount of backflow due to compressibility of the fluid will be experienced when the individual pumping chambers 32 are first communicated to the discharge of the pump. This will produce a periodic and rhythmical pressure pulsation or surging in the pump discharge, which pulsations will have a frequency generally proportional to the number of pumping chambers and the speed of rotation of its rotor. The precise configuration of these pressure pulsations or waves will vary depending upon the type of positive displacement pump involved (gear, sliding vane, radial or axial piston), and will vary to some degree depending upon the formation of the pump discharge chambers and/or the system to which the pump is connected. In all cases the time-rate pattern or configuration of the pulsations can be

accurately determined by means of an oscilloscope or other suitable laboratory means.

The means provided in the pump shown in FIGURE 1 for bleeding fluid out of the pump discharge in amounts offsetting these pressure surges, comprises a plurality of ports or grooves **92** in the sliding sealing surface of the rotor B positioned in such manner as to register with the drilled passageways **88** subsequent to the time that the individual pumping chambers **32** are first communicated with the associated discharge slot **40** of the sliding vane **36** which forces the liquid out of the pumping chamber involved. Pressure flowback into the pumping chambers **32** creates the valleys or low points in the pump pressure discharge curve during the time that the rotor is moved from the position shown in FIGURE 3 to the position shown in FIGURE 4 (and as indicated by the numerals **3** and **4** of each of the FIGURES 2, 3 and 4). The start of the discharge stroke for the pumping chamber involved occurs at a time when the trailing sliding vane **36** is positioned over the center of the trailing lobe **30**; and continues while the sliding vane **36** moves through the pumping chamber to the center of the lobe **30** on the other or downstream side of the pumping chamber involved. The rising portion of the pump pressure discharge curve will occur generally over the interval in which the sliding vane sweeps the individual pumping chamber.

The porting or grooves **92** are brought into communication with the drilled passageways **88** at some time interval after the appropriate sliding vane **36** starts to sweep the individual pumping chamber involved. This is shown in the upper left-hand corner of FIGURE 4 wherein the groove **92** is out of registry with the passageway **88**. Shortly thereafter the leading edge of the groove **92** is brought into registry with the passageway **88** to start the process of bleeding liquid out of the pump discharge.

The bleed ports or grooves **92** are connected to the pump discharge system by means of a passageway or port **94** communicating its trailing edge with the radial discharge slots **40**. The radial discharge passageway **94** and the bleed port or grooves **92** are so shaped as to vary the amount of fluid bled out of the pump discharge in a manner offsetting the pressure pulsations. The instantaneous rate of fluid flow through the groove **92** and port **94** will be controlled by the cross-sectional area of the groove **92** in combination with the length of groove intermediate the passageway **88** and the inlet port **94**; and the maximum amount or peak value of the flow experienced when the inlet port **94** is in registry with the passageway **88** will be controlled by the sizes of either one or both of the passageways **88** and **94**. As a general rule a groove **92** of constant predetermined cross-section will give a uniformally increasing rate of leakage as the rotor is rotated; and the slope of the time-leakage curve can be changed by changing the cross-sectional area of the groove **92**. Where the change in the cross-sectional area of the groove **92** is abrupt, sharp changes in the slope of the time-leakage curve will be experienced; and the over-all length of the time-leakage curve will be controlled by the over-all length of the groove **92**.

FIGURES 6 through 13 show several forms of leakage groove, and the type of time-leakage curve which they provide (the even number figures showing the construction of the groove, and the next succeeding odd number figure showing the time-leakage curve which it provides). The several grooves shown utilize the principles above set forth, and show the manner in which the principles can be combined to provide time-leakage curves of practically any configuration. These time-leakage curves, of course, should be tailored to provide a time-leakage curve shape which generally corresponds to the shape of the pressure pulsations of the pump in which the leakage grooves are to be used; such that the instantaneous rate of leakage so provided is proportional to the instantaneous pressure intensity of its impulses or surges.

FIGURE 6 shows a bleed port configuration having a substantially constant uniform cross-section for the full length of the groove, and a pressure inlet port adjacent its trailing edge. This type of groove when passing over a suitable outlet port produces a time-leakage curve having a substantially constant or uniform slope—indicating that the rate of discharge for the bleed port builds up at a substantially constant rate, and thereafter drops off sharply when the trailing edge of the groove moves past the cooperating outlet port.

FIGURE 8 shows a bleed port configuration in which the pressure inlet passageway or opening is positioned in approximately the center of the groove. With such a configuration, the leakage builds up at a generally uniform rate until the pressure inlet passage is in register with the outlet port; and thereafter, decreases at a substantially uniform rate until the trailing edge of the groove passes the outlet port.

FIGURE 10 shows a stepped bleed groove configuration in which the configuration of the cross-sectional groove is changed in a series of steps to provide a changing rate of discharge. As the leading edge of the discharge groove **92** opens into registry with the outlet port, a generally rapid build up of flow is initiated until the outlet port is completely opened. Thereafter, the leakage rate increases at a substantially constant rate until the deeper portion of the groove moves into registry with the outlet port. Thereafter the discharge flow increases at a faster rate than was provided by the first stepped portion of the groove until the pressure inlet port begins to move into registry with the outlet port. The discharge rate gradually changes to a maximum rate, which continues for a short time, until the trailing edge of the port begins to move into registry with the outlet port. The flow rate continues to gradually decrease until the trailing constant cross-sectional area of the groove is in registry with the outlet port, and thereafter the discharge flow decreases at a substantially constant rate until the trailing edge of the groove begins to choke off the flow out through the discharge port.

FIGURE 12 shows a bleed port having two pressure inlet openings. The time-leakage curve for such a configuration of bleed port will have two peaks spaced apart in accordance with the spacing of the groove's inlet openings. The bleed groove shown in FIGURE 2 is provided with a substantially constant uniform cross-section through its entire length to give rising and falling portions of the curve having substantially the same slope or rate of change. The rate of discharge builds up to a peak at about the time that the first inlet passage of the groove **92** is in registry with the outlet port; and there after decreases at approximately the same rate of change until the discharge port is approximately centered between the two inlet ports **94** and **96**. When the discharge port **88** is intermediate the inlets **94** and **96**, flow of course can proceed from both of the inlet ports through the outlet port to create a minimum flow when the outlet port **88** is approximately centered between the two inlets. Thereafter the flow increases again as the inlet port **94** moves closer to the outlet port **88** until the inlet port **94** is in registry with the outlet port, at which time a peak is again reached. Thereafter the flow will decrease again until the trailing edge of the groove **92** moves past the outlet port **88**. Peaks of different intensity could be provided by regulating the sizes of the branch passageways **94** and **96**; and other minor fluctions in the curve can be provided by varying the depth or width or both of the various sections of the groove **92**. Tapered leading and trailing edges of the groove **92** will likewise change or smoothen out the leading and trailing edges of the time-leakage curve.

As a general comment, the principles set forth above can be used to provide a shape of time-leakage curve which generally corresponds to the shape of the pressure pulsations as determined by an oscilloscope or other means. Thereafter cut and try means can be used to vary the shape and cross-section of the groove at appro-

priate points, and/or the size of the inlet and outlet openings may be changed to match the time-leakage curve of the pump whose discharge is to be smoothened out.

The operation of the pump shown in FIGURE 1 causes fluid to flow through the pump inlet port or suction connection **54** to the suction chamber **52** of the rotor B; whereupon, flow proceeds from the inlet drillings **50** to the individual pumping chambers **32** as the inlet drillings **50** are rotated past the pumping chambers. Each of the pumping chambers **32** are of course separated by lobes **30** which are adapted to sliding sealingly engage the outer surface of the rotor B in such manner as to valve off each of the pumping chambers for an interval of time after an inlet groove **50** has been moved past one of the chamber's separating lobes **30** and prior to the time that the following discharge slot **40** is moved into registry with the other separating lobe **30** for the chamber. During this portion of the time that the individual pumping chambers are valved off from both the suction and discharge portions of the pump, the individual pumping chamber is pressurized by means of fluid from the accumulator chamber **56** by means of the passageway **88** and the radially extending groove **90** which are positioned in the sliding sealing side surfaces of the internal pump chamber, and rotor, respectively. During this interval, the groove **90** is brought into registry with the passageway **88**—permitting flow from the accumulator **56** to raise the pressure in the isolated pump chamber from its original suction pressure to a pressure approximately that at which the accumulator is maintained. Immediately thereafter, the discharge slot **40** is brought into communication with the pumping chamber to connect the pumping chamber with the discharge portion of the pump. The trailing sliding vane **36** thereafter sweeps the fluid out of the pumping chamber through the discharge slot **40**, transverse drilling **42**, annular grooves **44** and **46**, to the discharge connection **48** of the pump.

The pressurized of the individual pump chambers prior to their being valved to discharge appreciably helps to reduce the amount, or intensity of the pressure pulsations produced in the pump discharge. Inasmuch as the pressure in the accumulator cannot be maintained exactly at pump discharge pressure, some pressure pulsations are still produced when the individual pumping chambers are connected to discharge. According to the principles of the present invention suitable bleed passages **92** are utilized to bleed off fluid from the discharge of the pump in amounts, and at time intervals which will offset the remaining pulsations in the pump discharge. Shortly after the sliding vane starts to force fluid out of a filled pumping chamber **32**, the bleed groove **92** is moved in registry with the drilled passageway **88** communicating with the reservoir D. This causes fluid to flow out of the discharge of the pump through the inlet port **94**, groove **92** to the drilling **88**, thereby replacing fluid previously withdrawn from the accumulator D to maintain the reservoir at its generally predetermined pressure. The size and shape of the grooves **92** and **94** are of course varied as previously explained to provide a time-leakage curve which will generally match the pressure pulsation curve of the pump so as to offset the pulsations and smoothen out the pressure of the fluid discharged from the pump. Inasmuch as the pump shown in the drawing is hydraulically balanced such that the cycle of operation of opposite vanes is identical, only two pressure bleed grooves **92** need be used to smoothen out the pump operation.

While the preferred embodiment uses the pressure bleeding process to replenish the accumulator chamber, this need not be done; and if preferred, the bleed stream can be returned directly to the suction of the pump or even to an external source.

FIGURES 14, 15, 16 and 17 show the invention as embodied in a radial ball piston pump which generally comprises a body member E having an internal chamber **102** therein in which an annular rotor F is journalled about an axially extending pintle G which projects into the internal chamber **102** from one end wall of the pump. The opposite end wall **106** of the pump is made in the form of a removable cover member suitably bolted in place; and the annular rotor F is adapted to be rotated about the pintle by means of a drive shaft **108** journalled in the cover member **106**. The inner end of the drive shaft **108** is splined to a drive plate **110** positioned over the adjacent end of the pintle, and the outer edges of which are fastened to the annular rotor F by means of a plurality of machine screws **112** only one of which is shown.

The annular rotor member F is provided with a plurality of radially extending openings **114** therethrough the radially outer ends of which are accurately counterbored to form cylinders **116** in which individual ball pistons **118** are positioned. The ball pistons **118** are retained within the cylinder by means of an annular camming member H which extends around the outer surfaces of the rotor and on which the balls **118** are adapted to roll. The annular camming member H shown in the drawing utilizes a race **120** of a commercially obtainable anti-friction bearing, pressed into a support member **122** which is suitably guided and supported for eccentric movement with respect to the rotor F.

The pump shown in the drawing is adapted to be supplied with oil from a reservoir **124** which is bolted directly to the top surface of the pump. Oil from the reservoir passes through a vertical opening **126** in the body member E to a longitudinally extending drilling **128** in the axially extending pintle G. The top surface of the pintle G directly beneath the annular rotor F is notched out as at **130** to provide inlet communication between the inner end of the cylinders **116** and the inner end of the longitudinally extending drilling **128**; and a venturi section **132** is pressed into the longitudinally extending drilling **128** between the inlet passage **126** and the notch **130** supplying the rotor. The particular embodiment shown in the drawing utilizes a pressurized suction wherein the fluid from the inlet passage **126** is forced into the throat of the venturi section **132** to the cylinders **116** by means of a pressure impinging stream presently to be described.

The radial piston pump shown in the drawings is adapted to be driven clockwise as seen in FIGURE 15. The annular camming member H is supported for eccentric movement with respect to the annular rotor F by means of an abutment pin **134** recessed into the lower end of both the body member E and the support member **122** in such manner as to limit all but a rocking movement of the camming member H with respect to the rotor F. Maximum displacement of the pump will be provided when the camming member H is in the position shown in FIGURE 15 of the drawing. With the camming member H in the position shown, the ball pistons **118** will be in their innermost positions with respect to the cooperating cylinders **116** when the inner end of the cylinders **116** are moved out of engagement with the land portion **136** of the pintle into communication with the inlet groove **130** on the upper surface of the pintle. The positioning of the camming member H is such that centrifugal force moves the ball pistons **118** radially outwardly in their cooperating cylinders **116** as the balls roll around the race **120** to a position approximately 180° from the start of the inlet stroke. As the ball pistons approach their outward limit of travel, the inner end of the cylinders **116** move out of communication with the inlet groove **130** to a position wherein a land **138** valves off or completely isolates the cylinders **116** from both suction and discharge.

Continued rotation of the rotor during the second half of each revolution causes the ball pistons **118** to roll around the lower half of the race **120**, thereby causing the pistons **118** to be moved inwardly to their most

inwardly or starting position. Just after the time that the ball pistons **118** start to move inwardly, the inner ends of the cylinders **116** move off the land **138** into communication with the discharge groove **140** in the lower surface of the pintle G. The discharge groove **140** is quite similar to the inlet groove **130** (extending over a similar arc of the pintle); but is separated from the inlet groove **130** by the land portions **136** and **138**. Fluid forced into the discharge groove **140** by the inward movement of the ball pistons **118** passes through a longitudinal discharge drilling **142** in the pintle G to a transverse drilling **144** leading to a discharge chamber **146** in the body member E. Some discharge fluid is used for the pressurizing of the pump inlet stream by means of a transverse drilling **148** communicating the longitudinal discharge drilling **142** with a nozzle **150** in the inlet drilling **128**. The passage **148** opens into an annular groove **152** in the nozzle **150**; and a transverse drilling **154** between opposite sides of the recess or groove **152** communicates with a small longitudinal drilling **156** which directs the high pressure stream into the throat of the venturi section **132**. Fluid being discharged from the pump passes through an annular filter **158** held into engagement with the bottom end of the discharge chamber **146** surrounding its inlet **160** by means of a retainer **162** and a coil spring **164**. The coil spring **164** is in turn held in place by a threaded outlet fitting **166** screwed into the outer end of the discharge chamber **146**. The fitting **166** is provided with a centrally located discharge opening **168** therethrough containing a check valve **170** and cooperating valve seat **172** for the prevention of return flow through the pump. A bent wire **174** is inserted between the check valve **170** and a tube fitting **176** in the opening **168** to prevent the ball from restricting flow out of the discharge connection.

The rate of discharge of the pump shown in FIGURES 14 through 17 can be varied by shifting the camming member H with respect to the rotor F in a manner changing the eccentricity therebetween to vary the stroke of the individual pistons **118**. Shifting of the camming member D is accomplished by means of a slide structure J comprising a cylindrically shaped slide **180** positioned in a bore **182** extending at substantially right angles to the displacement changing movement of the camming member H. The lower end of the cylindrically shaped slide **180** is provided with a milled slot **184** angularly disposed with respect to the direction of movement of the slide **180**—which slot receives a pin **186** fastened between the outer ends of a bifurcated or U-shaped bracket **188** which is brazed to the support member **122** of the camming member H. The slide **180** is biased downwardly by a coil spring **190** suitably positioned between the slide **180** and the bottom of the reservoir **124** which closes off the upper end of the bore **182**. The camming member H is therefore normally urged into a position providing maximum displacement for the pump by the coil spring **190**; and the displacement of the pump is decreased by means of a position **192** adapted to be biased against the lower end of the slide **180** to cause the slide to be moved upwardly. Upward movement of the slide of course causes the camming member H to be moved toward its concentric position thereby decreasing the stroke of the ball pistons **118**.

The piston **192** is actuated by means of pressure from a control valve K which receives its pressure from the discharge of the pump. The control valve K comprises a bore **194** in the body member E in which a spool valve **196** is positioned in a manner permitting its lands to straddle a control port **198** in the side walls of the bore **194**. The control piston **192** is positioned in the upper end of a bore **200** positioned beneath the slide structure J; and the lower end of the bore **200** is communicated to the control port **198** through suitable passageways **202** in the body member E. The inner end of the control valve bore **194** is communicated with the discharge drilling **142** in the pump pintle C by a drilling **204**; and the outer end of the bore **194** is communicated with the internal chamber **102** of the pump by means of an opening **206** in the body member E. The outer end of the spool valve **196** projects into a spring chamber **208** where it is abutted by a spring retaining plate **210** which is biased inwardly by a coil spring **212**. The outer end of the spring chamber **208** is closed off by a suitable sealing member **214** held in place by the threaded outlet fitting **166**; and the spring chamber **208** is also vented to the internal pump chamber **102** by means of a drilling **216** in the body member. For a better understanding of the construction and operation of radial piston pump so far described, reference may be had to the Farlow B. Burt application Serial Number 649,370, filed March 29, 1957.

Means similar to that described for the first described embodiment is also provided for the pump shown in FIGURES 14 through 17. Pressure from an accumulator **220** is communicated to each of the cylinders **116** intermediate the time that they are valved off from the suction chambers of the pump and the time that they are connected to the discharge passageways of the pump. Control of the pressurizing of the individual pumping cylinders **116** is controlled in this instance by means of valve porting similar to that used in the previous embodiment. In the present instance the control porting is positioned in the sliding sealing cylindrical surfaces of the pintle and rotor, respectively, which surfaces rotatably support the rotor member F. Angularly drilled passageways **222** (one for each of the cylinders **116**) are provided in the rotor member F for the purpose of communicating each of the cylinder chambers **116** with the portion of the inner cylindrical surface of the rotor positioned to one side of the plane in which the cylinders are located. A pressure fill port **224** is provided in the pintle G adjacent the land portion **138** for registry with each of the angularly drilled passageways **224** as they revolve therepast. Pressure is supplied the fill port **224** from the accumulator **220** through a longitudinal drilling **226** in the pintle G, which intersects the drilling forming the pressure fill port **224** at one end, and which is communicated to the accumulator **220** at its other end by means of a transverse drilling **228**.

Fluid pressure from the accumulator **220** is used to pressurize each of the cylinders **116** when the ball pistons **118** therein have reached their outermost limits of travel and during the time that each cylinder is valved off from both the inlet groove **130** and the discharge groove **140** by the land **136**. Fluid pressure from the accumulator **220** is bled to each cylinder at this instance in a controlled amount by means of the drilling which forms the pressure fill port **224**. Each of the angularly drilled passageways **222** in the rotor F are brought into registry with the fill port **224** at a time when the trailing edge of the cylinder openings **114** have moved out of communication with the inlet groove **130** of the pintle. The sizes of the passageways **222** and fill port **224** are such that the passageways in the fill port will move out of registry with each other just prior to the instant that a cylinder, which has been pressurized moves into communication with the discharge groove **140** of the pintle. By this expedient each of the cylinders **116** are successively pressurized from the accumulator, and are immediately thereafter valved to discharge before any appreciable leak down of pressure in the individual cylinders takes place.

The present embodiment is also provided with means similar to that of the previous embodiment for bleeding fluid from its discharge at times and in amounts which will offset its discharge pulsations to smoothen out the fluid delivered by the pump. In the present embodiment, this is accomplished by a second set of porting in the cylindrical surfaces of the pintle and rotor member located in the plane containing the fill port **224**, such that the angularly drilled passageways **222** may also be used for the bleed down operation. In the embodiment shown

in the drawing, the bleed down porting **230** located in the pintle is adapted to register with the angularly drilled passageway **222** of a cylinder **116** immediately following the time that the cylinder is first valved to discharge. The precise angular position at which the registry of the passageway **222** and bleed down porting **230** begins to take place will of course be determined by the pressure pulsation curve of the pump as determined by an oscilloscope, or other suitable means. Inasmuch as three other cylinders **116** are also communicating with the discharge groove **140** during this same period of time, the bleed porting **230** could probably be advanced 45°, 90°, or 135° to cooperate with anyone of these other cylinders **116** which are valved to discharge during the same period of time.

The bleed down porting **230** is provided with a bleed groove or recess **232** in the sliding sealing surface of the pintle—which groove is constructed in accordance with the principles previously described with reference to the FIGURES 5 through 13. The outlet port **234** for the porting **230** is formed by a transverse drilling communicating the groove **232** and the longitudinal drilling **226**, in the pintle, which leads to the accumulator **220**.

Operation of the pump should be readily discernible by those skilled in the art from the above description reciting the cooperation between the various elements. Suffice it to say that hydraulic fluid from the reservoir **124** passes through the inlet passageway **126** to the venturi section **132** where the impingement of a high pressure stream through the longitudinal drilling **156** into the throat of the venturi section produces a positive pressure in the inlet groove **130** of the pintle C. Rotation of the rotor B successively communicates the cylinders **116** with the inlet groove **130** during the portion of the rotor cycle wherein the ball pistons **118**, which are in rolling contact with the camming member H, move radially outwardly in their cylinders. Outward movement of the ball pistons **118** causes a quantity of fluid to be added to each cylinder while the cylinders are communicated to the inlet groove **130**; and at approximately the time that the ball pistons **118** have reached their outward limit of travel, the inner openings **114** of the cylinder slide over the land portion **138** of the pintle to isolate the cylinders from communication with both the inlet and outlet systems of the pump. Shortly after the cylinders **116** become valved off from the inlet groove **130** and prior to the time that the cylinders are communicated with the discharge groove **140** of the pintle, each cylinder is pressurized with fluid from the accumulator **124** by the rotation of the cylinder's angularly drilled passageway **222** into communication with the pressure fill port **224** of the pintle. Each cylinder is rapidly brought up to accumulator pressure and immediately thereafter each cylinder is successively valved off from the accumulator and then communicated with the discharge groove **140** of the pintle. Continued rotation of the rotor with respect to the camming member H causes the ball pistons **118** to move inwardly in their cylinders **116** discharging the fluid into the discharge groove **142** through passageways **142** and **144** to the discharge chamber of the pump. At the same time a small side stream is continuously supplied to the suction pressurizing nozzle **156** through the transverse drilling **148**; and a second side stream of high pressure fluid is intermittently supplied to the accumulator **220** through the bleed down porting **230**. The bleed down porting **230** is so positioned, constructed and arranged as to bleed pressure out of the discharge of the pump in amounts and at times which will offset the pressure surges which are created when the individual cylinders **116** are connected to discharge—thereby smoothening out the flow of fluid discharged from the pump to alleviate the noise and vibration produced by pressure pulsations. For a more complete understanding of the manner in which this is accomplished reference may again be had to the explanation of FIGURES 5 through 13.

The pump shown in FIGURES 14 through 17 is a variable displacement pump which can control the amount of fluid delivered by the pump by changing the eccentricity of its camming member H with respect to its rotor F. The pump is designed to control the eccentricity of the camming member in a manner adapted to provide substantially constant discharge pressure. Pump discharge pressure is supplied to the inner end of the spool valve **196** of the control valve K—which pressure is opposed by the spring **212**. When this pressure exceeds the setting of the spring **212**, movement of the spool valve **196** permits pressure to be communicated with the control port **198**, whereupon control pressure is exerted against the bottom end of the control piston **192**. Upward movement of the piston **192** forces the slide J upwardly—producing a shifting of the pin **196** in a direction causing the camming member H to move concentrically with respect to the rotor member F. The stroke of the ball pistons **118** is therefore reduced to cause a reduction in the displacement of the pump per rotor cycle, until such time as the quantity of fluid delivered by the pump equals the demand of the system to which the pump is connected at the set pressure as determined by the spring **212**.

Although the invention has been described in considerable detail, I do not wish to be limited to the particular constructions shown and described; and it is my intention to cover hereby all novel adaptations, modifications, and arrangements thereof which come within the practice of those skilled in the art to which the invention relates.

I claim:

1. In a positive displacement pump: a stationary member having inlet and discharge passages and an accumulator reservoir therein, a rotatable member having sliding sealing engagement with said stationary member, at least one fluid chamber in one of said members, means for successively displacing liquid from said chamber and for causing a new supply of liquid to enter said chamber in timed relation to the movement of said rotatable member, said chamber being valved to said outlet passages when liquid is being displaced from said chamber and being valved to said inlet passages when liquid is caused to enter said chamber, and said chamber being isolated from the inlet and outlet passages intermediate the times they are valved to the inlet and outlet passages; said pump producing pressure surges in said discharge passages when said chamber is discharged thereto; first and second ports in the sliding sealing surface of one of said members, said sliding sealing surface of said one of said members having a recess therein communicating with said second port, a third port in the other of said members for successive registry with said first port and said recess, said first and third ports being constructed and arranged to communicate said chamber with said accumulator reservoir during the time that the chamber is isolated from the inlet and discharge passages, and said recess and said third port being constructed and arranged to conduct flow out of the discharge passages in a manner offsetting said pressure surges to smoothen out the fluid discharged from said pump.

2. In a rotary positive displacement pump of the sliding vane type: a body member having an internal chamber therein whose side surfaces are formed by surfaces of revolution positioned about an axis extending through said surfaces, a rotor member in said internal chamber, said rotor having side surfaces which are in sliding sealing engagement with said side surfaces of said body member, said members also having generally axially extending surfaces one of which is a surface of revolution and the other of which forms lobes and valleys which abut the generally axially extending surface of revolution to provide individual fluid confining chambers, a plurality of radial slots in the member having the generally axially extending surface of revolution, a sliding vane in each slot for sliding engagement with the lobes and val-

leys of the other member, said member having the generally axially extending surface of revolution being provided with a discharge passage positioned forwardly of each vane for receiving fluid forced out of said valleys by said vane and having inlet passages behind each vane for adding fluid to said fluid chambers, said pump producing surges in discharge pressure as the fluid confining chambers are communicated to said discharge passages, a sink maintained at a lower pressure than said discharge port, said sink being sufficiently isolated from said discharge port so that pressure fluctuations in said sink do not appreciably affect the pressure in said discharge port, and first and second ports in the adjacent sliding sealing side surfaces of said members for bleeding fluid out of said discharge passages to said sink, said ports being so positioned as to be brought into registry with each other when the fluid is being forced out of said discharge passages by said vanes, and being shaped and sized to produce a flow through the ports in an amount generally proportional to said surges to smoothen out the pressure level in said discharge passages.

3. In a rotary positive displacement pump of the sliding vane type: a body member having an internal chamber therein whose side surfaces are formed by surfaces of revolution positioned about an axis extending through said surfaces, a rotor member in said internal chamber, said rotor having side surfaces which are in sliding sealing engagement with said side surfaces of said body member, said members also having generally axially extending surfaces one of which is a surface of revolution and the other of which forms lobes and valleys which abut the generally axially extending surface of revolution to provide individual fluid confining chambers, a plurality of radial slots in the member having the generally axially extending surface of revolution, a sliding vane in each slot for sliding engagement with the lobes and valleys of the other member, said member having the generally axially extending surface of revolution being provided with a discharge passage positioned forwardly of each vane for receiving fluid forced out of said valleys by said vane and having inlet passages behind each vane for adding fluid to said fluid chambers, said pump producing surges in discharge pressure as the fluid confining chambers are communicated to said discharge passages the pressure of which varies according to a generally predetermined pattern, said body member also having an accumulator reservoir therein which is sufficiently isolated from said discharge passages at all times so that pressure fluctuations in said reservoir produce substantially no effect in said discharge passages, a first port in said body member communicating said reservoir with one side surface of said pump chamber, a second port positioned in the adjacent side surface of said rotor in such manner as to periodically register with said first port during rotation of said rotor, said ports being constructed and arranged to pressurize one of said fluid chambers after it is filled with fluid and before it is communicated with said discharge passage, and a third port positioned in said adjacent side surface of said rotor in such manner as to periodically register with said first port, said third port being sized and shaped to conduct flow out of said one of said fluid chambers when it is communicating with said discharge passage at a rate and amount generally proportional to said pattern of said pressure surge to smoothen out the pressure of the fluid discharged from said discharge passages.

4. In a radial piston pump: a housing having an internal pump chamber therein, a pintle in said pump chamber, a generally annularly shaped rotor positioned in sliding sealing engagement around said pintle, said rotor having at least one cylinder chamber extending radially therethrough between its surface adjacent said pintle and its radially outer surface, a camming member eccentrically positionable around said rotor, a piston in each cylinder chamber adapted to abut said camming member in a manner causing each piston to move in and out with respect to its cylinder chamber, said pintle having suitable inlet passages for communication with each cylinder chamber as its piston moves radially outwardly and suitable discharge passages for communication with each cylinder chamber as its piston moves radially inwardly, said pump producing surges in discharge pressure as each cylinder chamber is communicated to said discharge passages the intensity of which varies according to a predetermined pattern, a sink maintained at a lower pressure than said discharge port, said sink being sufficiently isolated from said discharge port so that pressure fluctuations in said sink do not appreciably affect the pressure in said discharge port, a first port in said sliding sealing surface of said pintle and communicating with said sink, a second port in said sliding sealing surface of said rotor and communicating with one of said cylinder chambers, said ports being so positioned as to be brought in registry with each other when the fluid is being forced out of said cylinder chamber, and being so shaped and sized as to produce a flow through the ports in an amount generally proportional to said pattern of said surges to smoothen out the fluid discharged from said discharge passages.

5. In a radial piston pump: a housing having an internal pump chamber therein, a pintle in said pump chamber, a rotor positioned in sliding sealing engagement around said pintle, said rotor having at least one cylinder chamber extending radially therethrough between its surface adjacent said pintle and its radially outer surface, a camming member eccentrically positionable around said rotor, a piston in each cylinder chamber adapted to abut said camming member in a manner causing each piston to move in and out with respect to its cylinder chamber, said pintle having suitable inlet passages for communication with each cylinder chamber as its piston moves radially outwardly and suitable discharge passages for communication with each cylinder chamber as its piston moves radially inwardly, said pump producing surges in discharged pressure as each cylinder chamber is communicated to said discharge passages the intensity of which varies according to a generally predetermined pattern, said body member also having an accumulator reservoir therein which is sufficiently isolated from said discharge passages at all times so that pressure fluctuations in said reservoir produce substantially no effect in said discharge passages, a first port in the sliding sealing surface of said pintle communicating with said accumulator reservoir, a second port in the adjacent sliding sealing surface of said rotor for periodic registry with said first port, said second port communicating with one of said cylinder chambers, said ports being constructed and arranged to pressurize said one of said cylinder chambers after it is filled with fluid and before it is communicated with said discharge passages, and a third port in said sliding sealing surface of the pintle for periodic registry with said second port, said third port being sized, positioned and shaped to conduct flow out of said one of said fluid chambers when it is communicating with said discharge passages at a rate and amount generally proportional to said pattern of said pressure surges to smoothen out the fluid discharged from said discharge passages.

6. For use with a positive displacement hydromechanical device having flow passages which are normally held at pressures above a predetermined level but in which there are rhythmic pressure pulsations; a pair of sliding sealing surfaces which are slid over each other in a prescribed path and in timed relation to said rhythmic pressure pulsations; a first port located in one of said pair of sliding sealing surfaces and which communicates with said passages during said pressure pulsations; a second port located in the other of said pair of sliding sealing surfaces and which communicates to a region of pressure which is below said predetermined level; one of said sliding sealing surfaces having a groove which communicates said first and second ports as said ports slide past each

other, and said groove having a cross-section, length and position relative to said ports to provide a bleed flow out of said flow passages which substantially offsets said rhythmic pressure pulsations.

7. For use with a positive displacement hydromechanical device having flow passages which are normally held at pressures above a predetermined level but in which there are rhythmic pressure pulsations; a rotary valve comprising a pair of sliding sealing surfaces which are slid over each other about an axis of revolution in timed relation to said rhythmic pressure pulsations; a first port located in one of said pair of sliding sealing surfaces and which communicates with said passages during said pressure pulsations; a second port located in the other of said pair of sliding sealing surfaces and which communicates to a region of pressure which is below said predetermined level; one of said sliding sealing surfaces having a groove which communicates said first and second ports as said ports slide past each other, and said groove having a cross-section, length and position relative to said ports to provide a bleed flow out of said flow passages which substantially offsets said rhythmic pressure pulsations.

8. In a positive displacement hydromechanical device having a casing member which houses a positive displacement rotor member therein that rotates about an axis of revolution and that has side surfaces that are in sliding sealing contact with said casing, said casing further including flow passages which are normally held at pressures above a predetermined level but in which there are rhythmic pressure pulsations throughout the rotary cycle of said rotor: a first port located in said sliding sealing surface of said rotor; a second port located in said sliding sealing surface of said casing; one of said ports communicating with said flow passages during said pressure pulsations, and the other of said ports communicating to a region of pressure which is below said predetermined level; and one of said sliding sealing surfaces having a flow restricting recess which communicates said first and second ports as said ports slide past each other, said recess being so proportioned and positioned that the length of said recess between said ports and through which fluid flow takes place changes during the rotation of said rotor to vary the flow rate out of said flow passages in a manner generally offsetting said rhythmic pressure pulsations.

9. A positive displacement pump as set forth in claim 8, and wherein said recess extends to either side of said port in said one of said surfaces, whereby the flow through said ports gradually builds up to a maximum and thereafter gradually diminishes as said ports move past each other.

10. A positive displacement pump as set forth in claim 8, and wherein said recess has changing cross sections throughout its length, whereby the flow through said ports builds up at a changing rate.

11. In a positive displacement hydromechanical device having a casing member which houses a positive displacement rotor member therein that rotates about an axis of revolution and that has side surfaces that are in sliding sealing contact with said casing, said casing further including flow passages which are normally held at pressures above a predetermined level but in which there are rhythmic pressure pulsations throughout the rotary cycle of said rotor: first porting comprising first and second spaced apart openings in the sliding sealing surface of one of said rotor and casing members, a recess in said surface of said one member connecting said first and second openings, second porting in the sliding sealing surface of the other of said members positioned to periodically register and communicate with said recess, one of said portings communicating with said flow passages during said pressure pulsations, and the other of said portings communicating to a region of pressure which is below said predetermined level, said recess being so positioned that said second porting opens into changing positions of said recess as the second porting and recess move past each other to vary the flow rate out of said flow passages in a manner generally offsetting said rhythmic pressure pulsations.

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UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,007,419 November 7, 1961

Farlow B. Burt

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 2, line 67, for "radical" read -- radial --; column 7, line 39, for "pressurized" read -- pressurizing --; column 9, line 59, for "position" read -- piston --.

Signed and sealed this 29th day of May 1962.

(SEAL)
Attest:

ERNEST W. SWIDER
Attesting Officer

DAVID L. LADD
Commissioner of Patents