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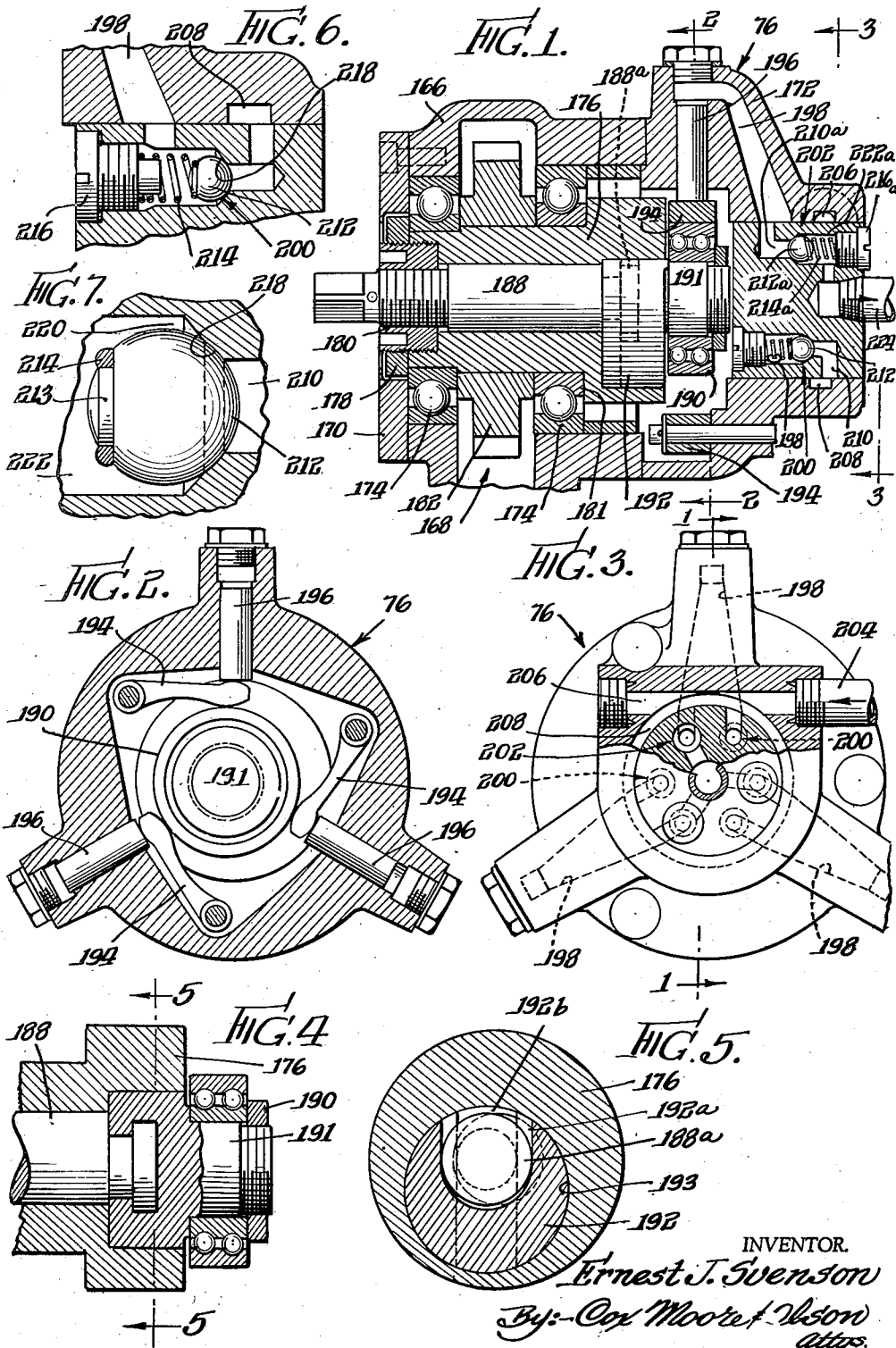
E. J. SVENSON

2,382,452

FLUID PUMPING MECHANISM

Original Filed Aug. 11, 1933

3 Sheets-Sheet 1



Aug. 14, 1945.

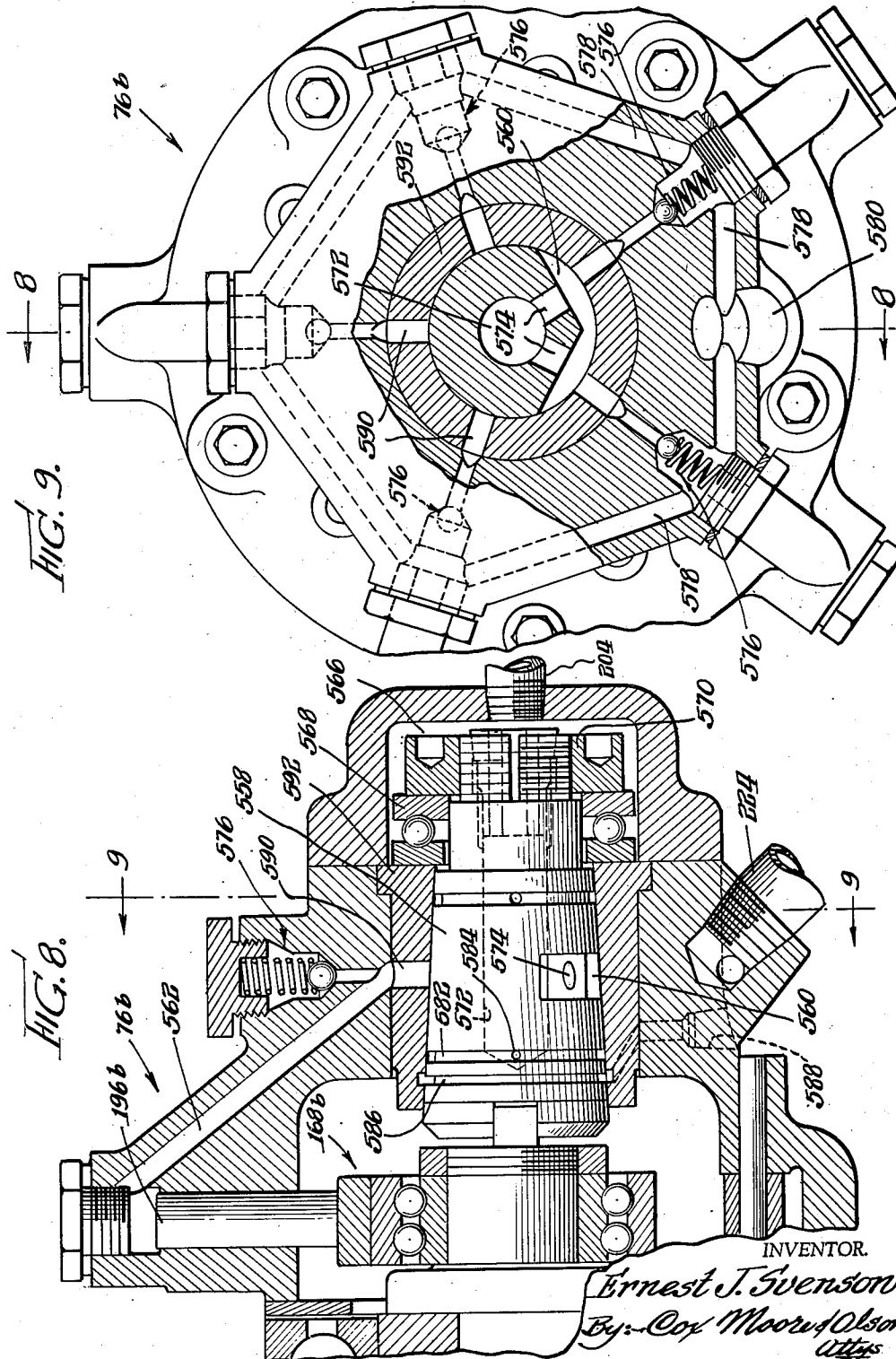
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FLUID PUMPING MECHANISM

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3 Sheets-Sheet 2



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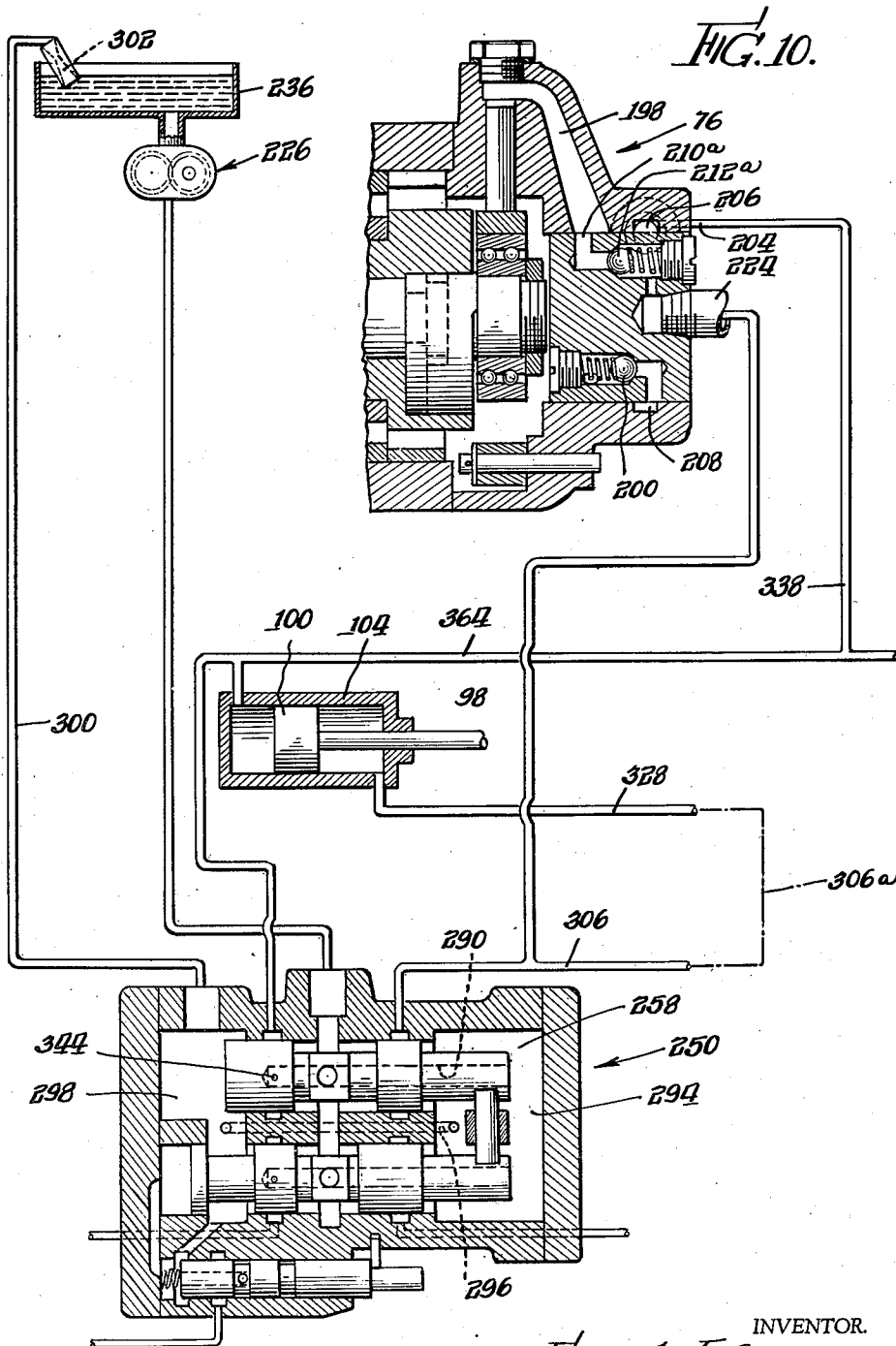
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FLUID PUMPING MECHANISM

Original Filed Aug. 11, 1933

3 Sheets-Sheet 3



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UNITED STATES PATENT OFFICE

2,382,452

FLUID PUMPING MECHANISM

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mesne assignments, to Odin Corporation, Chi-
cago, Ill., a corporation of Illinois

Original application August 11, 1933, Serial No.
684,677. Divided and this application Septem-
ber 16, 1940, Serial No. 356,896

17 Claims. (Cl. 103—174)

This invention relates generally to fluid pump-
ing mechanisms, and more particularly to pump
structures of the variable displacement plunger
type capable of very accurately delivering fluid
at relatively high pressure.

The present invention is a division of my co-
pending application, Serial No. 684,677, filed
August 11, 1933, now Patent 2,215,257 issued Sep-
tember 17, 1940. The pumping mechanism con-
templated by the present invention relates gen-
erally to the class of pumps disclosed in my is-
sued Patent No. 1,989,117, dated January 29, 1935,
and constitutes an improvement over the struc-
ture disclosed therein. As set forth on page 1
of my aforesaid patent, fluid slippage or leakage
is a problem with which hydraulic engineers have
been confronted. Such fluid slippage or leakage
results in greatly reduced efficiency of operation.
It is, therefore, one of the important objects of
the present invention to provide a pumping mech-
anism which will overcome the aforesaid and
other problems, and to this end the invention
contemplates a novel combination of pumping
elements including automatically operable ball
check valves in cooperative association with fluid
pressure generating elements such as pump
plungers.

More specifically, the invention contemplates
fluid pumping mechanisms, as set forth above,
wherein ball check valves are arranged to func-
tion in timed relation with respect to the move-
ment of pressure generating elements such as
pump plungers.

It is a further object of the present invention
to provide pumping mechanism, as set forth
above, wherein ball valve means is cooperatively
associated with a plurality of shiftable pump
plungers which are designed to be mechanically
shifted in one direction and hydraulically shifted
in the opposite direction.

Another object of the present invention is to
provide pumping mechanism which is particu-
larly adaptable for use in closed hydraulic cir-
cuits wherein the fluid returned in the closed
circuit to the pump may serve to shift the plun-
gers in a given direction and wherein fluid slip-
page or leakage within the pump is eliminated.

The foregoing and numerous other objects and
advantages will be more apparent from the fol-
lowing detailed description, wherein:

Figure 1 is a transverse sectional view of a
variable and relatively slow delivery pumping
mechanism of the type contemplated by the pres-
ent invention, said view being taken substantially
along the line 1—1 of Figure 3;

Figure 2 is a transverse sectional view of said
pump taken substantially along the line 2—2 of
Figure 1;

Figure 3 is an end elevational view of the pump,
shown partly in section, said view being taken
substantially along the line 3—3 of Figure 1;

Figure 4 is a fragmentary detail view of the ec-
centric driving mechanism for the pump struc-
ture shown in Figures 1 to 3 inclusive;

Figure 5 is a fragmentary, transverse, sectional
view of the eccentric driving mechanism taken
substantially along the line 5—5 of Figure 4;

Figure 6 is a detail sectional view of one of the
ball valves disclosing the manner in which said
valve serves to control communication between
the inlet side of the pump and the pump passage-
ways communicating with the plungers or pis-
tons;

Figure 7 is an enlarged detail view of the ball
valve of Figure 6 to more clearly illustrate the
structural features thereof which render it par-
ticularly adaptable for use in closed circuits and
the like as illustrated more particularly in Fig-
ure 10;

Figure 8 is a fragmentary, central, transverse,
sectional view of a modified pumping mechanism
equipped with ball valves, said view being taken
substantially along the line 8—8 of Figure 9;

Figure 9 is a view of the right end of the pump
structure shown in Figure 8, a portion thereof
being broken away along the line 9—9 of Figure
8 to more clearly illustrate the structural ar-
rangement of the valve mechanisms; and

Figure 10 is a circuit diagram illustrating the
manner in which the pumping mechanism of the
present invention may be employed in a closed
circuit.

Referring now to the drawings, wherein like
numerals have been employed to designate simi-
lar parts throughout the various figures, atten-
tion is directed to the fact that, for purposes of
clearness and comparison with the disclosure in
my aforesaid parent application, the numerals
employed herein correspond with the numerals
designating like parts in my aforesaid applica-
tion.

The pump structure shown in the drawings is
designated generally by the numeral 76. This
pumping mechanism includes a central housing
166, which carries a pump driving mechanism des-
ignated generally by the numeral 168, an end plate
170, and an oppositely disposed pump housing 172
serving as an enclosure and support for the plun-
ger and valve elements of the pump. Preloaded
anti-friction bearings 174 provided within the

housing 166 serve as the support for a rotary driving member or sleeve 176. Longitudinal displacement of the sleeve 176 to the right, Figure 1, is prevented by a clamping ring 178 and an adjusting nut or screw 180, while longitudinal displacement of the sleeve 176 to the left is prevented by a flange 181 which engages the right preloaded anti-friction bearing 174.

A sleeve 176 carries a driving gear 182 which is adapted to mesh with another gear (not shown) forming a part of a transmission from a suitable driving means (not shown). A driving member 188 mounted within and rotatably adjustable with respect to the driving sleeve 176 serves as the means for adjusting the eccentricity of a driving ring 190. This ring 190 forms the outer race of a radical ball bearing mounted upon an extended stub shaft portion 191 of a cylindrical adjustment member 192. As best seen in Figure 1, the stub shaft portion 191 is eccentric to the cylindrical adjusting member 192. The member 192 is supported within a cylindrical recess 193 of the member 176 and is eccentric with respect to said 176. It will, therefore, be clear that the adjustment mechanism is such that rotation of the member 192 within the recess 193 will vary the eccentricity of the stub shaft portion 191 with respect to the axis of the sleeve 176. In other words, rotatable adjustment of the cylindrical member 192 causes the stub shaft portion 191 to experience eccentric adjustment with respect to the axis of the sleeve 176 which thereby affects the degree of eccentricity of the driving ring 190.

The member 188 has a head portion 188a which is eccentric to the axis of the member 188 and registers within a recess 192a in the member 192. A reduced portion which connects the member 188 with the head 188a passes through a transverse recess or slot 192b. Thus when it is desired to vary the eccentricity of the driving ring 190, the nut 180 is first loosened and then rotation is imparted to the member 188 by gripping the outer squared end thereof. Rotation of the member 188 causes rotation of the cylindrical member 192 and this results in varying the eccentricity of the stub shaft portion 191 with respect to the axis of rotation of the sleeve 176 and the member 188. After the required degree of eccentric adjustment has been made, the nut 180 is again tightened. Rotation is imparted to the sleeve 176 and the member 188 by the driving gear 182, and the ring 190 will execute an eccentric motion to a degree which is dependent upon the amount of eccentric adjustment of the stub portion 191 as previously described.

A plurality of fingers 194 are uniformly spaced about the axis of the sleeve 176, and are pivotally supported within the end frame or casing 172 as clearly shown in Figure 2. The free extremities of these fingers are interposed between the driving ring 190 and companion pump pistons or plungers 196. The curvature of the fingers is such as to impart symmetrical acceleration and deceleration of the pistons 196. This feature is fully described in my Patent No. 2,048,524, issued July 21, 1936. Fluid is directed toward and away from the outer ends of the pistons 196 in companion passageways 198, Figures 1 and 3. Each of these passageways 198 communicates with a pair of valves, namely an intake control valve designated generally by the numeral 200 and an outlet control valve 202.

Fluid is directed to the valves 200 from an intake conduit 204, Figure 3, which constantly communicates through a transverse passage 206 with an annular passage or port 208. This annular

passage 208 directs fluid to the intake valves 200 through a passage 210 which is companioned to each valve. Each of the aforesaid valves 200 includes a ball valve member 212. I prefer to employ valve balls comprised of Swedish steel, which has been especially treated to obtain an unusually hard and tough structure. I have found what are well known in the trade as Hultgren Process Brinell balls, to perform very satisfactorily. The balls 212 are normally urged by means of a coiled spring 214, preferably tapered as illustrated in Figure 6, interposed between said balls and a threaded plug 216 into engagement with a valve seat 218. Referring to Figure 7, wherein I have disclosed an enlarged view of the ball and seat therefor, it will be seen that the seat 218 is shaped to conform accurately with a portion of the spherical surface of the valve ball. By having the springs 214 tapered or converging toward the valve ball, as illustrated in Figure 6, the centering of the ball within its seat is materially enhanced.

The ball valve 212 is provided with an annular recess 213 designed to receive the adjacent portion of the coiled spring 214, Figure 7. The portion of the threaded spring 214 which is received by the annular recess 213, is formed with converging walls which are adapted to abut the companion walls or surfaces of the recess. This arrangement positively prevents the ball from being dislodged and maintains the same surface of the ball in contact with the valve seat 218 at all times.

Attention is also directed to the fact that the annular area 220 presented between the outer periphery of the ball 212 and the inner periphery of the valve chamber 222 is less than the cross-sectional area presented by the intake passageway 210. The significance of this construction will be more apparent when a hydraulic circuit for which the pump mechanism 76 is particularly adapted, is described. From the foregoing, it will be apparent that fluid from the conduit or pipe line 204 may pass through the valve 200 and into the passageway 198 companion thereto. The intake of the fluid occurs during the inward stroke of the pistons or plungers 196. During the compression stroke of the plungers 196, fluid passes from the passageways 198 into a passageway 210a which communicates with a discharge valve 202. This valve 202 is similar in structural characteristics to the valve 200 previously described. Each valve 202 includes a ball valve 212a, a chamber 222a, and a threaded plug 216a. Fluid passing through the valve 202 enters a discharge conduit or pipe line 224.

It will be noted that the spring 214a for the valve ball 212a is more powerful than the spring for the intake valve ball 212, whereby to preclude the opening of the ball of the valve ball 212a when the pistons are being charged. In other words, the valve ball 212a will only open when the piston companion thereto is experiencing its compression stroke, and said valve will not open during any other portion of the cycle under normal pressure conditions in the propelling side of the circuit connected therewith.

The pump mechanism just described is designed to deliver fluid to any desired mechanism such as a hydraulic actuator at a predetermined rate. The rate of displacement of the pump can be varied, as previously explained, by merely adjusting the eccentricity of the driving ring 190, which adjustment affects the stroke of the pistons or plungers 196.

For the purpose of explaining one practical application of the feed pump just described, I have

illustrated a circuit diagram in Figure 10. This circuit is a portion of the circuit diagram disclosed in my aforesaid parent application with all parts omitted except those necessary to a clear understanding of the operation of the pump. Assume that one of the pistons 196 is experiencing its outward or compression stroke. Fluid under pressure is thereby forced from the space above the piston 196 through the passages 198 and 210a and past the valve ball 212a into the outlet conduit 224. From the conduit 224, the fluid is conducted into the conduit 306 which is blocked at one end by a control valve designated generally by the numeral 250. A valve member 258 of the control valve mechanism 250 is so positioned as to block one extremity of the conduit 306. The other end of the conduit 306 may be connected through any suitable mechanism designated diagrammatically by a dot and dash line 306a with a conduit 328 which conducts fluid to the right chamber of a hydraulic cylinder 104. This entrance of the fluid into the cylinder 104 causes a piston 100 to move to the left and displace fluid from the left-hand chamber of the cylinder 104 into a conduit 364 and thence through a conduit 338 which communicates with conduit 204. From conduit 204 fluid passes into the inlet passage 206 of the pump and thence into the annular distributing passage 208 as previously described. The passage 208 communicates with all of the inlet ball valves 200. Fluid will not flow through the valve 200 associated with the piston or pistons moving outward because the fluid pressure in the associated passage 198 is greater than the pressure of the fluid which is being returned to the intake side of the pump. However, as the driving ring 190 moves so as to permit the inward movement of one or more of the plungers 196, the pressure in the passages 198 will decrease, thereby permitting fluid to flow from the passage 208 through companion valves 200 of the receding or inwardly moving plungers and will exert a driving force tending to move the plungers inwardly. From the foregoing description it will be understood that when the pump 76 is connected in the above described closed circuit, fluid within the various conduits moves as a unit, one unit being forced under pressure outwardly through the pump passages 198 and the ball valves 202 and thence to the actuator cylinder 104 and the other unit moving from the discharge side of the actuator cylinder 104 into the intake side of the pump through the ball valves 200 and into the outer extremities of the complementary receding pump plungers.

In the circuit disclosed herein, the presence of the piston rod 98 on one side of the piston 100 creates a differential in working area on opposite sides of the piston. Consequently more fluid is discharged from cylinder 104 than is delivered to said cylinder during the movement thereof to the left, Figure 10. Such excess fluid is free to escape through a small radial passage or vent 344 in the valve member 258. From this vent 344 the fluid flows through a longitudinal passage 290, a chamber 294, a longitudinal passage 296, a chamber 298 and then through a conduit 300 and a restricted orifice 302 into a reservoir 236. For a more detailed description of the valve mechanism 250, attention is directed to the explanation given in my aforesaid parent application. It is not essential to a clear understanding of the pump mechanism that the valve mechanism 250 be described in more detail. In instances where no differential exists between

the areas on the opposite sides of the piston and where the actuator cylinder packings are free from leakage, the bleed passage 344 may be done away with. Of course, in instances where equal size piston rods on both sides of the piston 100 are employed and the discharge volume of the cylinder 104 would be theoretically equal to its intake volume, the bleed passage 344 may still be provided to make up for any leakage past the piston, or packings, that might disturb this theoretical equality. It will, of course, be understood that the bleed passage 344 is very small and is intended only to take care of slight volumetric discrepancies between volumetric delivery of the fluid pump to the actuator and volumetric discharge of fluid from the actuator. It is by no means large enough to carry the full pump volume and does not serve as a means to enable the supercharging of the plunger pump 76 by the rapid traverse or gear pump 226 shown in Figure 10. In fact, the construction of the pump 76 is such as to preclude the necessity of the supercharging by reason of the absence of fluid slippage or leakage within the pump.

The ball valves 200 and 202 materially contribute to the efficient functioning of the pump 76 in circuits of the type shown in Figure 10. I have operated such pumps over an extended period of time and have found that the valve balls stand up under the severest operating conditions and that they are adapted for continuous use over an extended period of time without repair or replacement.

Particular attention is directed to the fact that by using the ball valve arrangement in a circuit of the type disclosed herein, the clashing of the balls as they seat themselves is positively precluded. This will be more readily appreciated by referring again to Figures 6 and 7, wherein I have shown the balls 212 seated by the action of the threaded spring 214. Assume now that the pump piston 196, which is companion to the ball valve shown in Figures 6 and 7, begins to experience an inward movement. This enables fluid from the annular port or passage 208 connected with the discharge side of the actuator cylinder 104 to be delivered to the ball 212. This will cause the ball to become unseated and permit fluid to enter the companion passage 198 at a speed which is determined by the speed with which the companion plunger 196 moves inwardly. When the plunger reaches the limit of its intake stroke and is about to move in a reverse outward direction, the flow of fluid past the ball 212 ceases and the pressure on the opposite sides of the ball becomes substantially equalized, thereby enabling the ball to gently seat itself at the instant the pump piston 196 begins its compressive stroke. In other words, there is no clashing or violent seating of the ball 212 but, on the contrary, a very gentle seating thereof. The round seat which corresponds to the spherical surface of the ball also contributes toward the effective functioning of the valve. In this connection I again make reference to the fact that the cross-sectional area of the space 220 indicated in Figure 7, is less than the cross-sectional area in the passageway 210. Thus, a higher speed of fluid takes place within the area 220 than in the passageway 210, thereby enabling the ball 212 to be perfectly centered during the opening thereof. Therefore, as the ball gradually moves into juxtaposition with its seat 218, the speed of the fluid acting within the area 220 serves to maintain the centered relationship of the ball

with respect to its seat, and thereby precludes any engagement or clashing of the ball with the side walls which surround it.

It will also be apparent that by employing the ball valve arrangement just described, fluid pressure acting on the balls cooperates to maintain the balls tightly sealed against their respective seats. This should be clearly distinguished from rotary types of valves and others wherein the pressure of the fluid acting upon the valve does not function to urge the valve against the seat. The importance of the ball valve construction will be more readily apparent when it is understood that in a closed circuit arrangement employed for the purpose of accurately and uniformly propelling a machine tool, the slightest slippage of fluid past a valve in the pump during the compression stroke of the pump plungers will seriously affect the uniformity of fluid flow. Also, such slippage tends to cause a decided increase in fluid temperature. In fact, one of the most serious problems with which hydraulic engineers have heretofore been confronted is that of temperature increase resulting from the slippage or leakage of fluid along the bearing surfaces of rotary valves. In my above described pumping mechanism, the fluid pressure developed during the compressing stroke of the pump plungers will act upon the intake ball valve so as to positively seal the balls against their respective seats. In the ball check type of pump hereinbefore described, the outlet valve 202 will not open until the fluid pressure in the passage 198 companion thereto is greater than the pressure in the conduit 224. Therefore, no back flow from the conduit 224 into the passage 198 can ever take place. This is an important advantage which the ball check type of pump presents over pumps in which the control of fluid depends solely upon mechanically actuated valve members. In instances where mechanically actuated valve members are employed, the timed functioning of the valve is controlled mechanically and not in accordance with or in response to predetermined fluid pressure conditions.

In Figures 8 and 9 I have disclosed a modified plunger pump construction designated generally by the numeral 765. This pump is similar in many respects to the pump 76 previously described, but differs therefrom in providing a tapered rotary valve 558 and a peripheral port 560 which serves to timingly control the flow of fluid toward passages 562. Fluid enters the pump chamber through the conduit 204 which communicates with an end chamber 566. An anti-friction thrust bearing 568 and clamping ring 570 serve to secure the rotary valve 558 against longitudinal movement to the left. Fluid from the chamber 566 passes through an internal longitudinal passage 572 which communicates with the valve port 560 through radial passages 574. As the valve port 560 sweeps through its orbit, it causes fluid to be delivered to the outer ends of the pump pistons 196b during their intake or inward stroke. As the pistons 196b are moved outwardly, fluid under pressure in the passages 562 passes through a ball valve 576 and thence into passageways 578 which communicate with the discharge conduit 224.

The tapered arrangement of the rotary valve 558 precludes slippage of fluid toward the larger end of the valve, namely to the left, Figure 8, and any fluid that may slowly migrate in that direction is received by an annular passage 582 which communicates with the intake passage 572 through a radial opening 584. A second annu-

lar valve passage or port 586 communicates with a passage 588 adapted to be connected with a source of fluid pressure (not shown). For further explanation of this feature of the construction, reference is made to my issued Patent No. 2,078,697, particularly Figures 18 and 25 thereof. The use of fluid pressure at this point serves to insure lubrication and to further counteract any leakage or migration of fluid along the surface of the valve 558 toward the larger end thereof. The valve 558 is driven from a driving mechanism designated generally by the numeral 168b, which is similar in functional and structural characteristics to the mechanism 168 previously described. Other types of driving mechanisms may also be employed, see for example the driving mechanism shown in my issued Patent No. 1,989,118 and my copending application, Serial No. 456,570, filed May 28, 1930, which will issue into Patent No. 2,215,488 on September 24, 1940.

By employing the ball valves 576, I preclude the necessity of using a second port in the valve member 558 similar to the port 586. I thus prevent the rotary valve 558 from experiencing fluid pressure which might have a tendency to cause the valve to bind or wear along the peripheral surfaces positioned at each side of the peripheral valve port 560. While the fluid pressure from the passageways 562 does exert itself through the radial passages 580, provided in the bushing 582, which supports the rotary valve 558, this pressure is exerted against a relatively small area of the valve and hence merely serves to counteract the pressure of fluid acting within the peripheral port 560. This construction maintains the valve 558 in a balanced or floating condition within the bearing or bushing 582. That is to say, the pressure of fluid acting on one side of the valve is substantially counterbalanced by the pressure of the fluid acting on the opposite side, and thus maintains the valve in a state of rotative balance.

From the foregoing it will be apparent that the present invention contemplates a novel and practical pumping structure which is particularly designed for very accurate volumetric delivery of fluid at relatively high pressures. There are environments which require very small volumetric fluid displacement at relatively high pressures. As pointed out in my issued Patent No. 2,140,565, beginning page 9, left column 1 line 65, when a combined body of fluid is subjected to relatively high pressures, the fluid tends to experience a molecular change or compression. In my aforesaid patent I recommend the use of a circuit shown in Figure 24 of said patent when a plunger pump of the type described herein is employed for extremely small fluid displacement under relatively high pressures. Under such conditions the ball check valves disclosed herein function very efficiently and in fact are superior to conventional rotary valves with which I am familiar. As previously pointed out, the valve ball on the high pressure or discharge side of the pump will not open unless the pressure of the fluid between the ball and the pump plunger is greater than the pressure of the fluid positioned beyond said ball. This materially contributes to the efficient functioning of the pump under conditions of relatively low displacement and high pressure. I also prefer to construct the housing material associated with the valve balls of a uniform alloy steel treated so that it will wear in conformity with the surface of the ball, although not as hard as the ball. This enables balls to be used over an extended period of time without

requiring replacement. In fact, the longer the ball is used the more perfect becomes the seat in the material associated with the ball. It should also be understood that care must be exercised in properly freezing or otherwise fitting within the pump housing 172 the core or member which carries the ball check valves 200 and 202. There must be a very tight fit between these parts in order to positively prevent fluid leakage along the peripheral surface of this core. The pivoted pump fingers function in cooperation with the ball check valves to increase the efficiency of operation thereof. Not only do these fingers insure symmetry in acceleration and deceleration of the pump plungers, but they also insure the proper dwell or period of inactivity at each end of the plunger stroke and this insures the proper opening and closing of the ball check valves.

The invention is obviously not limited to the specific structural details disclosed herein, but is capable of other modifications and changes without departing from the spirit and scope of the appended claims.

The invention is hereby claimed as follows:

1. In combination with a source of liquid under pressure, a pump including a plurality of reciprocable plungers, a housing therefor, a driving mechanism for shifting said plungers during their compression stroke only and limiting the intake stroke thereof, valve means including pressure actuated inlet valve members and pressure actuated outlet valve members preset for operation at different pressures to open and close in timed relation under the pressure control of said plungers, said inlet and outlet valve members being arranged in pairs, each pair companion to one plunger, a common passageway for directing fluid through one member of each pair of valve members to the plungers, and a common passageway for directing fluid through the other member of each pair of valve members away from said plungers.

2. In combination with a source of liquid under pressure, a variable displacement pump comprising a fixed casing having an intake passage communicating with the source of liquid and a discharge passage, a plurality of cylinders in said casing communicating with each of the passages, a piston reciprocable in each of said cylinders, means for varying the stroke of movement of the pistons within the cylinders comprising means imparting an exhaust stroke only to said pistons and limiting the intake stroke thereof at all displacements, a pressure actuated valve controlling the communication between the intake passage and each cylinder opening to permit an influx of the liquid from the passage to the cylinder for imparting an intake stroke to the piston and closing to prevent discharge from the cylinder, and an independent pressure actuated valve controlling communication between the discharge passage and each cylinder opening upon an exhaust stroke of the piston and closing upon an intake stroke to prevent return flow into the cylinder.

3. In combination with a source of liquid under pressure, a pump comprising a housing having an intake passage communicating with the source of liquid and a discharge passage, a plurality of cylinders in said housing communicating with each of the passages, a plunger reciprocable in each of said cylinders, driving mechanism for imparting an exhaust stroke only to said plungers and limiting the intake stroke thereof, a first pressure actuated valve controlling the communi-

cation between the intake passage and each cylinder opening to permit the influx of the liquid from the passage to the cylinder for imparting an intake stroke to the plunger and closing to permit discharge of liquid from the cylinder, and a second pressure actuated valve controlling communication between the discharge passage and each cylinder opening upon an exhaust stroke of the plunger and closing upon an intake stroke of the plunger to prevent return of flow of liquid into the cylinder, said first pressure actuated valve being constructed to shift to its open position when the liquid pressure differential thereacross exceeds a predetermined amount, and the second pressure actuated valve being constructed to open when the liquid pressure differential thereacross exceeds a greater predetermined amount.

4. In combination with a source of fluid under pressure, a pump including a housing having an intake passage communicating with the source of liquid and a discharge passage, a plurality of cylinders in said housing communicating with each of the passages, a plunger reciprocable in each of said cylinders, means imparting an exhaust stroke only to said plungers and limiting the intake stroke thereof, pressure actuated means including a valve member controlling the communication between the intake passage and each cylinder opening to permit an influx of the fluid from the passage to the cylinder for imparting an intake stroke to the plunger, said valve means including a spring actuating said valve member to closed position automatically during a reversal of movement of the plunger, and independent pressure actuated valve means including a valve member controlling communication between the discharge passage and each cylinder opening upon an exhaust stroke of the plunger, said last mentioned valve means including a spring actuating said last mentioned valve member to closed position during reversal of movement of the plunger, the spring of the last mentioned valve means being stronger than the spring of the first mentioned valve means.

5. In combination with a source of liquid under pressure, a variable speed, variable displacement pump comprising a casing having an intake passage communicating with the source of liquid and a discharge passage, a cylinder in said casing communicating with each of the passages, a piston reciprocable in said cylinder, means for varying the path of travel of the piston within the cylinder comprising means imparting an exhaust stroke only to said piston and limiting the intake stroke thereof at all displacements and speeds, a pressure actuated inlet valve means including a shiftable valve member controlling the communication between the intake passage and the cylinder opening to permit an influx of the liquid from the passage to the cylinder for imparting an intake stroke to the piston, said inlet valve means including a spring actuating said valve member to closed position to prevent discharge from the cylinder, the force of said spring being such in relation to the pressure of the source as to permit the liquid from the source to maintain the inlet valve member in open position during the entire intake stroke of the plunger at all speeds and displacements, and spring controlled pressure actuated outlet valve means controlling communication between the discharge passage and the cylinder opening upon an exhaust stroke of the piston and closing upon an intake stroke to prevent return flow into the cylinder,

said inlet valve being preset to open at a lower pressure differential than the outlet valve.

6. In combination with a source of liquid under pressure, a variable displacement pump comprising a casing having an intake passage communicating with the source of liquid and a discharge passage, a cylinder in said casing communicating with each of the passages, a piston reciprocable in said cylinder, means for varying the path of travel of the piston within the cylinder comprising means imparting an exhaust stroke only to said piston and limiting the intake stroke thereof at all displacements, an inlet valve controlling the communication between the intake passage and the cylinder opening to permit an influx of the liquid from the passage to the cylinder for imparting an intake stroke to the piston and closing to prevent discharge from the cylinder, and an independent pressure actuated outlet valve controlling communication between the discharge passage and the cylinder opening upon an exhaust stroke of the piston and closing upon an intake stroke to prevent return flow into the cylinder, said inlet and outlet valves being preset to open at different predetermined operating pressures.

7. A variable displacement pump for use in a closed fluid circuit comprising a casing having an intake passage and a discharge passage, a plurality of cylinders in said casing communicating with each of the passages to receive liquid at a predetermined pressure and to discharge liquid from said cylinders at a higher pressure, a piston reciprocable in each of said cylinders, means for varying the path of travel of the pistons within the cylinders comprising means imparting an exhaust stroke only to each of said pistons and limiting the intake stroke of each piston at all displacements, pressure actuated spring controlled inlet valve means controlling the communication between the intake passage and each cylinder opening to permit an influx of the liquid from the passage to the cylinder at said predetermined pressure for imparting an intake stroke to the piston and closing upon completion of the intake stroke to prevent discharge from the cylinder, and a second pressure actuated outlet valve means including a shiftable valve member controlling communication between the discharge passage and each cylinder, said second pressure actuated outlet valve means including a spring urging said valve member to closed position and constructed to permit movement of the valve member to open position only at a pressure above the pressure of the liquid in the discharge passage and to close upon completion of the exhaust stroke of the piston to prevent return flow of liquid into the cylinder during the intake stroke, said outlet valve being present to open at a higher differential operating pressure than said inlet valve.

8. In combination with a source of liquid under pressure, a variable displacement pump comprising a casing having an intake passage communicating with the source of liquid and a discharge passage, a plurality of radially disposed cylinders in said casing communicating with each of the passages, a piston reciprocable in each of said cylinders, a driving mechanism for imparting an exhaust stroke only to said pistons and limiting the intake stroke thereof, said driving mechanism including an eccentric driving member adjustable as to eccentricity to vary the displacement of said pistons, independently supported followers interposed between said eccentric driving mem-

ber and said pistons, said driving member and followers being constructed to operate said piston in definite timed relation for all displacements, pressure actuated inlet valves controlling the communication between the intake passage and the cylinders opening in said timed relation for intervals determined by the displacement of the pistons to permit an influx of liquid from the passage to the cylinders for imparting intake strokes to the piston and closing in said timed relation to prevent discharge from the cylinders, and independent pressure actuated outlet valves controlling communication between the discharge passage and the cylinders opening in said timed relation for intervals determined by the displacement of the pistons in their exhaust strokes to discharge liquid from the cylinders and closing in said timed relation upon the completion of the exhaust strokes to prevent return flow into the cylinders during the intake strokes, said outlet valves being preset to open with a higher differential pressure than the inlet valves.

9. In combination with a source of liquid under pressure, a pump including a housing having intake and discharge passages and a plurality of plunger accommodating bores, a reciprocable plunger associated with each bore, driving mechanism for shifting said plungers in one direction and limiting their movement in the opposite direction, said plungers being adapted to be shifted in said opposite direction by liquid pressure, pressure actuated inlet valves operable at a given pressure for controlling the flow of liquid to each bore from said source through said intake passage, and independent pressure actuated outlet valves operable only at a pressure greater than said given pressure for controlling the flow of fluid from each bore to said discharge passage in timed relation with the flow of fluid to each bore from said intake passage.

10. A pump adapted for use in a closed fluid circuit wherein fluid under pressure is delivered to an actuator and fluid discharged from said actuator is returned under pressure to the intake side of the pump, including a housing having a plurality of cylinders, a reciprocable plunger in each cylinder, a rotary eccentric driving mechanism for imparting an exhaust stroke only to said plungers, fluid pressure imparting an intake stroke to said plungers, means for limiting the intake stroke of said plungers, valve means for controlling the flow of fluid toward and away from said plungers including a pressure actuated inlet ball valve for each plunger, and a pressure actuated outlet ball valve for each plunger and operable to open position only at a pressure higher than the pressure required to open the inlet ball valve, fluid inlet channel means communicating with all of said inlet ball valves, and outlet channel means communicating with all of said outlet ball valves.

11. In a pump structure, a housing, a plurality of cylinders formed in said housing, a piston reciprocable in each cylinder, means for imparting progressively increasing and decreasing speeds of movement to the pistons during the reciprocation thereof, a chamber in the housing, means for varying the strokes of the pistons within the cylinders, and pressure actuated valve means for each of said cylinders comprising a passage forming a reduced extension of said chamber, said passage and chamber being in fluid communication with the cylinder, a hardened metal valve ball disposed in the chamber, a valve seat formed at the juncture between said cham-

ber and said passage, and a spring for urging the ball against said seat whereby to close said passage, the ball being surrounded by an annular space within the chamber which is less in cross-sectional area than the cross-sectional area of the passage, whereby to insure a greater fluid velocity through said annular space than through said passage to effect the centering of the valve ball with respect to its seat.

12. In a pump structure, a housing, a plurality of cylinders formed in said housing, a piston reciprocable in each cylinder, means for imparting progressively increasing and decreasing speeds of movement to the pistons during the reciprocation thereof, means for varying the strokes of the pistons within the cylinders, and pressure actuated valve means for each of said cylinders comprising a chamber in the housing, a passage forming a reduced extension of said chamber, said passage and chamber being in fluid communication with the cylinder, a hardened metal valve ball disposed in the chamber, a valve seat formed at the juncture between said chamber and said passage, a spring for urging the ball against said seat whereby to close said passage, means for holding the spring in predetermined position, and means interlocking a portion of the spring with the ball whereby to hold the ball in predetermined position with respect to its seat during its opening and closing movements, so as to insure the continuous presentation of the same portion of the ball into engagement with its seat.

13. In a pump structure, a housing, a plurality of cylinders formed in said housing, a piston reciprocable in each cylinder, means for imparting progressively increasing and decreasing speeds of movement to the pistons during the reciprocation thereof, means for varying the strokes of the pistons within the cylinders, and pressure actuated valve means for each of said cylinders comprising a chamber in the housing, a passage forming a reduced extension of said chamber, said passage and chamber being in fluid communication with the cylinder, a hardened metal valve ball disposed in the chamber, a valve seat formed at the juncture between said chamber and said passage, a spring for urging the ball against said seat whereby to close said passage, means for holding the spring in predetermined position, and means comprising an annular recess formed on the valve ball interlocking a portion of the spring with the ball whereby to hold the ball in predetermined position with respect to its seat during its opening and closing movements, so as to insure the continuous presentation of the same portion of the ball into engagement with its seat.

14. In a pump structure, a housing, a plurality of cylinders formed in said housing, a piston reciprocable in each cylinder, means for imparting progressively increasing and decreasing speeds of movement to the pistons during the reciprocation thereof, means for varying the strokes of the pistons within the cylinders, and pressure actuated valve means for each of said cylinders comprising a chamber in the housing, a passage forming a reduced extension of said chamber, said passage and chamber being in fluid communication with the cylinder, a hardened metal valve ball disposed in the chamber, a valve seat formed at the juncture between said chamber and said passage, a spring for urging the ball against said seat whereby to close said passage, means holding the spring in predetermined position, means interlocking a portion of the

spring with the valve ball whereby to insure the presentation of the same portion of the valve ball against its seat during operation of the ball, and the ball being surrounded by an annular space within the chamber which is less in cross-sectional area than the cross-sectional area of the passage, whereby to effect a fluid centering of the valve ball in respect to its seat.

15. In a pump structure, a housing, a plurality of cylinders formed in said housing, a piston reciprocable in each cylinder, means for imparting progressively increasing and decreasing speeds of movement to the pistons during the reciprocation thereof, means for varying the strokes of the pistons within the cylinders, and pressure actuated valve means for each of said cylinders comprising a chamber in the housing, a passage forming a reduced extension of said chamber, said passage and chamber being in fluid communication with the cylinder, a hardened metal valve ball disposed in the chamber, a valve seat formed at the juncture between said chamber and said passage, said valve seat having a spherical surface for cooperative engagement with the ball, and said seat being of less hardness than the ball, and a spring for urging the ball against said seat whereby to close said passage.

16. In a pump structure, a housing, a plurality of cylinders formed in said housing, a piston reciprocable in each cylinder, means for imparting progressively increasing and decreasing speeds of movement to the pistons during the reciprocation thereof, means for varying the strokes of the pistons within the cylinders, and pressure actuated valve means for each of said cylinders comprising a chamber in the housing, a passage forming a reduced extension of said chamber, said passage and chamber being in fluid communication with the cylinder, a hardened metal valve ball disposed in the chamber, a spherical valve seat formed at the juncture between said chamber and said passage, said valve seat being of less hardness than the valve ball, and a spring for urging the ball against said seat whereby to close said passage, the ball being surrounded by an annular space within the chamber which is less in cross-sectional area than the cross-sectional area of the passage, whereby to effect the fluid centering of the valve ball with respect to its seat.

17. In a pump structure, a housing, a plurality of cylinders formed in said housing, a piston reciprocable in each cylinder, means for imparting progressively increasing and decreasing speeds of movement to the pistons during the reciprocation thereof, means for varying the strokes of the pistons within the cylinders, and pressure actuated valve means for each of said cylinders comprising a chamber in the housing, a passage forming a reduced extension of said chamber, said passage and chamber being in fluid communication with the cylinder, a hardened metal valve ball disposed in the chamber, a spherical valve seat formed at the juncture between said chamber and said passage, said seat being of less hardness than the valve ball, a spring for urging the ball against said seat whereby to close said passage, means for holding the spring in position, and means interlocking a portion of the spring with the valve ball whereby to insure the presentation of the same ball surface against said seat during operation of the valve.