Sept. 6, 1955 F. G. PRESNELL 2,716,945

VARIABLE STROKE ROTARY CYLINDER PUMP

Filed Oct. 17, 1952 2 Sheets-Sheet 1

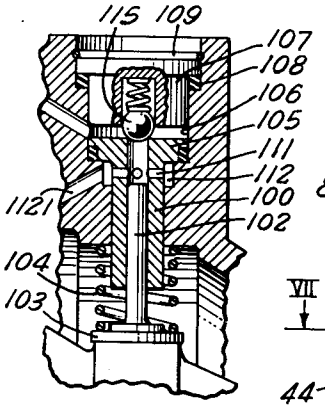


Fig. 4

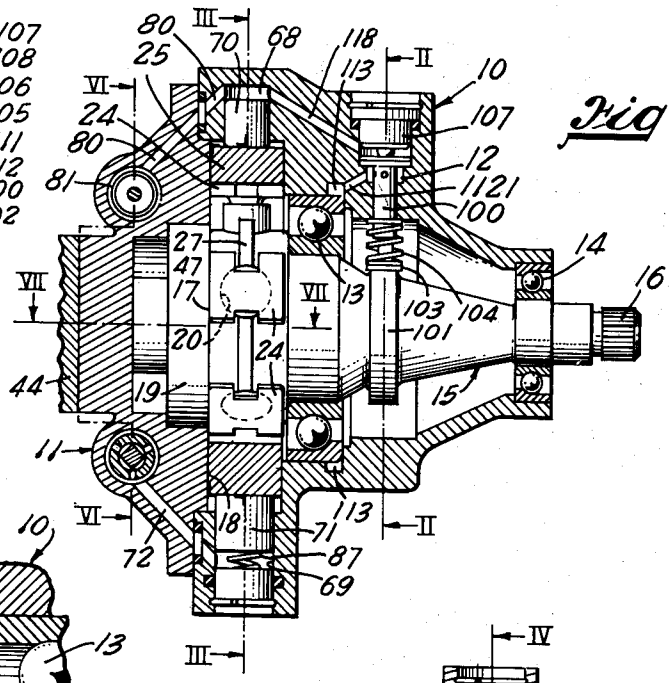


Fig 1

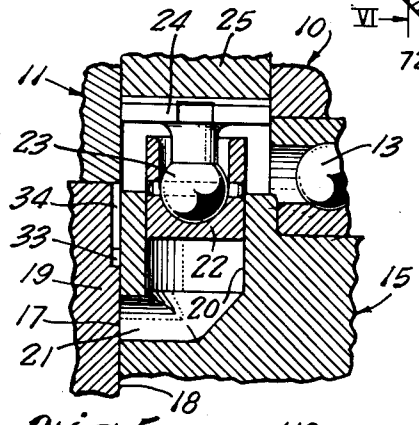


Fig. 5

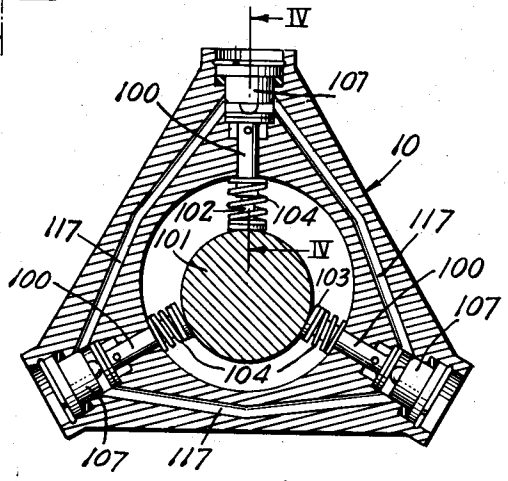


Fig 2

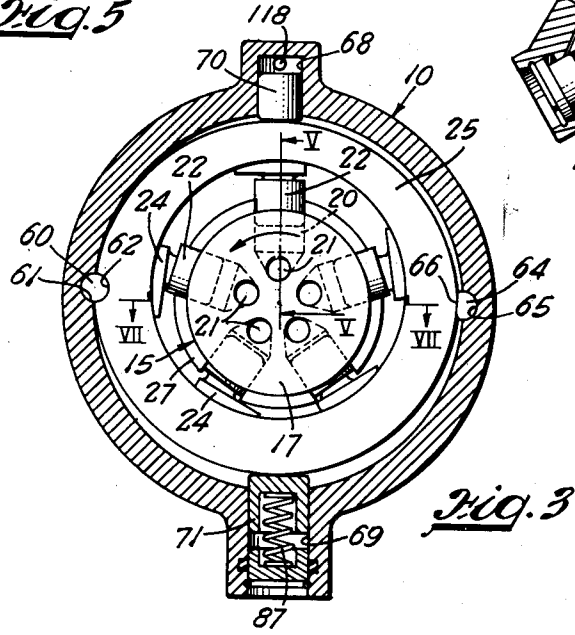


Fig. 3

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VARIABLE STROKE ROTARY CYLINDER PUMP

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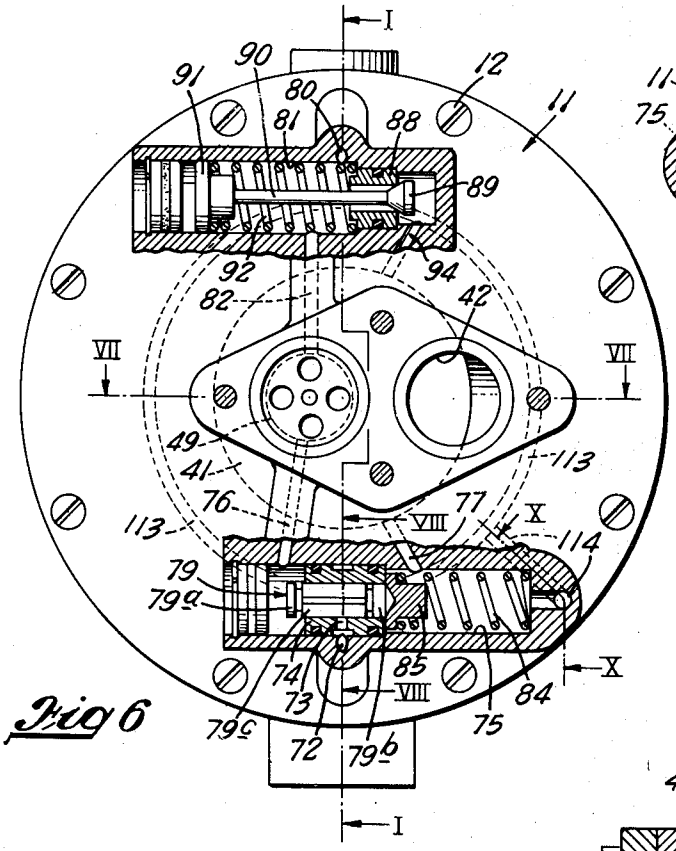


Fig 6

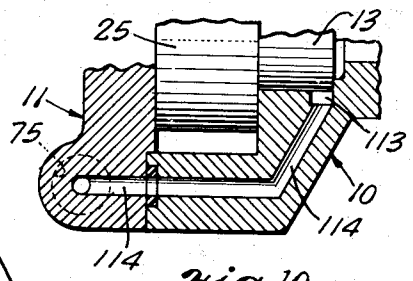


Fig. 10

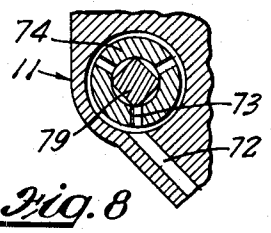


Fig. 8

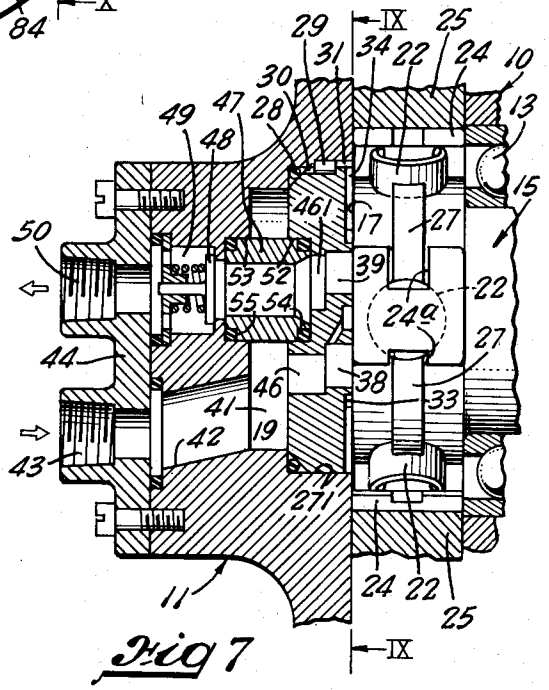


Fig 7

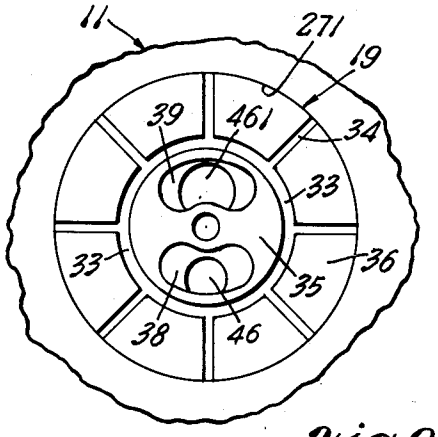


Fig 9

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United States Patent Office

2,716,945

Patented Sept. 6, 1955

2,716,945

VARIABLE STROKE ROTARY CYLINDER PUMP

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Application October 17, 1952, Serial No. 315,269

4 Claims. (Cl. 103—4)

This invention relates to pumps for hydraulic actuating systems and particularly to pumps of the constant pressure variable volume type.

An object of the invention is to provide a constant pressure, variable volume pump in which friction and wear are greatly reduced during periods when there is little or no demand thereon whereby the life of a pump supplying an intermittent demand is extended.

Another object is to provide a simple and effective valving structure for a pump having a multicylinder rotary cylinder block.

Another object is to provide a simple and inexpensive construction for shiftably supporting the piston actuating ring of a rotary radial cylinder pump.

Other more specific objects and features of the invention will appear from the description to follow.

Constant pressure, variable volume rotary radial cylinder pumps are broadly old. In such pumps radial reciprocation of pistons in the cylinders is produced by an encircling ring eccentric to the axis of rotation of the cylinders, and the output is varied to meet the demand by varying the eccentricity of the ring in response to pressure variations. A defect of such a pump is that although the strokes of the pistons are reduced to a low value when no fluid is being drawn from the pump, it is always working against the full pressure for which it is set, to supply leakage losses. Hence frictional losses and wear are not much diminished during no load intervals.

In accordance with the present invention, I increase the efficiency and life of a pump of the type referred to by incorporating in it a small auxiliary constant volume pump of capacity only slightly above the constant leakage loss of the system, and provide for complete unloading of the main pump during idling periods when the auxiliary pump is able to maintain the normal pressure. The slight amount of fluid supplied by the auxiliary pump in excess of leakage losses is exhausted through the usual relief valve of the system without appreciable heating. The complete unloading of the main pump during idling periods substantially eliminates wear thereof and reduces the friction losses. Because of its small volumetric capacity the auxiliary pump can be made very rugged and durable without being expensive.

To further reduce friction losses and wear when the main pump is idling, I provide a special design of valve plate bearing against the cylinder rotor which is urged into sealing relation with the rotor by the pressure in the output passage of the main pump only, this pressure plate not being subject to the pressure in the line, which is maintained by the auxiliary pump. This eliminates all pressure between the sliding valve surfaces when the main pump is not delivering pressure fluid.

Another feature of the invention is a simple and inexpensive arrangement for mounting the piston actuating ring for movement into and out of concentric relation with the rotor.

A full understanding of the invention and the various objects and features thereof may be had from the following detailed description with reference to the drawing, in which:

Fig. 1 is a longitudinal, vertical sectional view of a pump in accordance with the invention, the view being taken along the line I—I in Fig. 6;

Fig. 2 is a cross section taken in the plane II—II of Fig. 1;

Fig. 3 is a cross section taken in the plane III—III of Fig. 1;

Fig. 4 is a detail longitudinal section taken in the plane IV—IV of Fig. 2;

Fig. 5 is a detail longitudinal section taken in the plane V—V of Fig. 3;

Fig. 6 is a cross section taken along the line VI—VI of Fig. 1;

Fig. 7 is a horizontal longitudinal section taken in the plane VII—VII of Fig. 6;

Fig. 8 is a detail vertical longitudinal section taken in the plane VIII—VIII of Fig. 6; and

Fig. 9 is a cross section taken in the plane IX—IX of Fig. 7; and

Fig. 10 is a cross section taken in the plane X—X of Fig. 6.

Referring to Fig. 1, the pump comprises a casing formed in two sections, one consisting of a body **10** and the other constituting a cover **11**, the two being joined together by screws **12** (Fig. 6). The body **10** contains a pair of anti-friction bearings **13** and **14** rotatably supporting a rotor **15**. This rotor extends past the bearing **14** exterior of the casing for driving connection, as by a splined end **16**, to a source of power. The left end face **17** of the rotor is flat and flush with an annular surface **18** of the cover **11** and fits against a valve plate **19**. The largest portion of the rotor **15** immediately to the left of the bearing **13** contains five radial pump cylinders **20**, each of which, as shown in Fig. 5, communicates with a cylinder port **21** terminating in the rotor end face **17**. Each cylinder **20** contains a piston **22** having a socket in its outer end receiving a ball **23** connected to a shoe **24** which bears against the inner surface of a ring **25**. The ring **25** is mounted for movement between a position concentric with respect to the rotor **15**, in which no movement is imparted to the pistons in response to rotation of the rotor, and a position eccentric with respect to the rotor in which the pistons are given in and out pumping motion.

To increase the bearing surface between the pistons and the rotor the latter is provided with an interrupted outwardly extending flange **27** in the mid plane of the pistons. The shoes **24** are slotted at their ends as indicated at **24a** to clear the flange **27** when the pistons and shoes are in their innermost positions.

When the ring **25** is eccentrically positioned with respect to the rotor **15**, each piston is moved outwardly (by centrifugal force) of its cylinder during one half revolution and moved inwardly during the other half revolution. The valve plate **19** connects each port **21** to an inlet passage during the expanding half cycle, i. e. the time when the associated piston is moving outwardly in its cylinder, and connects each port **21** to an outlet passage during the other half of the cycle when the piston is moving inwardly of the cylinder.

Referring to Figs. 7 and 9, the valve plate **19** is circular in cross section and is slidably fitted in a cylindrical recess **271** in the cover **11**. It is sealed with respect to the recess **27** by a resilient packing ring **28** and is restrained against rotation by a key **29** engaging a groove **30** in the periphery of the valve plate and a groove **31** in the wall of the casing cover recess **271**. The front face of the valve plate **19** is divided by a circular groove **33** and by radial grooves **34** extending outwardly there-

from into a central sealing section 35 and surrounding segmental sections 36 which are ground and polished and bear against the end 17 of the rotor 15 which is likewise ground and polished.

The annular row of cylinder ports 21 (Fig. 5) in the face 17 of the rotor are juxtaposed to the central surface 35 of the valve plate 19 and successively register with arcuate inlet and outlet ports 38 and 39 respectively in the central face 35.

The cover 11 of the casing defines immediately back of the valve plate 19 (Fig. 7) a circular recess 41 in communication with an inlet passage 42 which is communicated with an inlet connection 43 formed in a plate 44 secured to the left end of the cover 11. The valve plate 19 contains an inlet passage 46 which extends thereto and connects the arcuate inlet port 38 with the chamber 41. It will be observed therefore that the major portion of the rear surface of the valve plate 19 is exposed to the inlet pressure in the chamber 41.

The outlet port 39 in the face of the valve plate 19 is communicated, by an outlet passage 461 extending through the plate, with the rear face of the plate and thence through a tubular member 47 and past a check valve 48 to a main outlet passage 49 in the casing cover which is in communication with an outlet connection 50 in the plate 44. The check valve 48 serves to prevent output pressure being applied to the rotor and valve plate when the piston actuating ring 25 is adjusted into concentric position with respect to the rotor, and pressure is being maintained in the main outlet passage 49 by an auxiliary pump to be described later.

The connection comprising the tubular member 47 connecting the valve plate outlet passage 461 with the main outlet passage 49 serves the additional purpose of producing a pressure force acting to hold the valve plate in sealing engagement with the rotor when the pump is in operation. The front end of the tubular member 47 fits in a cylindrical recess 52 in the valve plate 19, and in a cylindrical recess 53 in the cover 11 and is fitted for longitudinal sliding movement in recesses 52 and 53 but is fluid sealed with respect thereto by a pair of sealing rings 54 and 55. Pressure fluid within the tubular member 47 exerts a pressure force urging the valve plate 19 against the rotor in opposition to the force in the outlet port 39 in the face of the plate, which tends to separate the plate from the rotor. Thus, referring to Fig. 9, it will be observed that the area of arcuate port 39 is greater than the area of passage 461, and the pressure force tending to separate the plate 19 from the rotor is proportional to the difference between the areas of port 39 and passage 461. On the other hand, pressure fluid within passage 461 penetrates between the end of member 47 and the bottom of recess 52 and between the sealing ring 54 and the bottom of recess 52 to produce a pressure force urging the plate 19 against the rotor, which force is proportional to the difference between the areas of passage 461 and the recess 52. To insure that the valve plate will be held in sealing engagement with the rotor, the diameter of the tubular member 47 and recess 52 is such that the area of the recess 52 in the back of the valve plate is greater than the area of the outlet port 39 in the front face of the plate. Furthermore the axis of the cylindrical surface 52 is so located with respect to the position of the arcuate port 39 that its center of force is in alignment with the center of force developed against the port 39.

When the pump is operating, its suction may reduce the pressure in the chamber 41 substantially below atmospheric, and it is to prevent suction reaching the front face of the valve plate that the sealing ring 28 is provided. In the particular design shown in Fig. 1, the interior of the housing is not sealed, the device being intended to be positioned in direct communication with a supply tank so that atmospheric pressure prevails generally within the housing except in suction passages and parts that are connected thereto. If the sealing ring 28 were not provided, suction in the inlet passages and chambers 42 and 41 might draw air from the casing into the chamber 41 which would then be delivered to the outlet passage with the pressure fluid. Plate 19 is urged into light contact with the rotor, even when no pressure exists within tube 47, by the elasticity of packing rings 28, 54 and 55.

Referring to Fig. 3, the ring 25 is mounted for movement into and out of concentric relation with the rotor by a pin 60 which is engaged by juxtaposed substantially semi-cylindrical grooves 61 and 62 in the body 10 and the ring 25 respectively. The ring is held in engagement with the pin 60 by a diametrically opposite pin 64 which is mounted in a substantially semi-cylindrical groove 65 in the body 10 and has a face 66 engaging a matching face on the ring 25. The engaging faces on the pin 64 and the ring 66 are curved with their centers of curvature located at the pin 60. The arrangement described permits the necessary limited swinging movement of the ring 25 about the pin 60 as a center between a position in which the ring is concentric with the rotor and a position in which it is substantially eccentric thereto. Fig. 3 shows ring 25 in position of maximum eccentricity in which the greatest pumping effect is produced.

The ring 25 is automatically shifted between concentric and eccentric positions in response to the pressure in the main outlet passage 49 (Fig. 7) of the pump. To this end, the body 10 (Fig. 3) is provided with two diametrically opposite radially extending cylinders 68 and 69 respectively which are disposed substantially 90° displaced on opposite sides of the ring-supporting pin 60. Cylinder 68 contains a piston 70 bearing against the ring 25 for urging it into concentric relation with respect to the rotor, and cylinder 69 contains a piston 71 bearing against the ring 25 for urging it into eccentric relation with respect to the rotor. Referring to Fig. 1, the outer end of cylinder 68 is connected by communicating ducts 80 in the body 10 and the cover 11 with a relief valve cylinder 81. This cylinder is connected (Fig. 6) by a duct 82 to the main outlet passage 49. Therefore the output pressure of the pump is at all times applied to cylinder 68 and urges piston 70 to move the ring 25 into concentric (non-pumping) relation with the rotor.

The cylinder 69 (Figs. 1 and 3) is of larger diameter than the cylinder 68 so that when outlet pressure is applied thereto the piston 71 is able to overcome the piston 70 and shift the ring 25 into eccentric position. However, outlet pressure is not delivered directly to the cylinder 69 but is delivered thereto under the control of a pressure-responsive valve. Thus cylinder 69 is connected by a passage 72 (Fig. 1) in the body 10 and cover 11 to a port 73 (Figs. 6 and 8) in a sleeve 74 positioned in a cylinder 75 in the cover 11. The left end of the cylinder is communicated by a duct 76 with the main outlet passage 49 in the cover. The right end of the cylinder 75 is communicated by a duct 77 with the inlet chamber 41. The sleeve 74 contains a shuttle valve element 79 having cylindrical end sections 79*a* and 79*b* adapted to seal with the sleeve 74, and an intermediate polygonal guide section 79*c*. The valve element 79 is urged into a leftmost position as shown in Fig. 6 by a helical compression spring 84 acting through a pusher member 85. In this leftmost position of the valve element 79, the cylindrical section 79*b* on the right end thereof isolates the ports 73 from the inlet duct 77 but connects them through the left end of the cylinder 75 and the duct 76 with the main outlet passage of the pump so that pump output pressure is delivered to the cylinder 69 (Fig. 3). This condition exists whenever the outlet pressure is below the predetermined maximum value. As the outlet pressure increases, the force developed thereby on the valve element 79 (Fig. 6) overcomes the force of the spring 84 and moves the valve element to the right into position in which the cylindrical section 79*a* of the valve element seals with the

sleeve 74, blocking application of outlet pressure to the port 73. At the same time, the other cylindrical end section 79*b* of the valve element is moved clear of the sleeve 74 so that the ports 73 are connected to the inlet chamber 41.

Therefore when the pump output or line pressure reaches a desired value the pressure in cylinder 69 (Fig. 1) is reduced to a low value, permitting the piston 70 to restore the ring 25 to concentric relation with the rotor and completely stop the pumping action.

To insure that the ring 25 will be in eccentric position when the pump is started and no pressure exists therein, a light helical compression spring 87 is provided back of the piston 71 to force it inwardly. The force of this spring 87 is inconsequential compared to the pressure forces that exist during normal operation of the pump.

The relief valve cylinder 81 (Fig. 6) has been previously referred to. This cylinder 81 contains a sleeve 88 the right end of which constitutes a valve seat for a poppet 89 which is mounted on the right end of a stem 90, the left end of which is conected to a dash pot piston 91. A helical compression spring 92 compressed between the sleeve 88 and the piston 91 urges the poppet 89 against its seat. The right end of the cylinder 81 is connected by a duct 94 with the main return chamber 41 so that the output pressure existent in the cylinder 81 on the left side of the sleeve 88 tends to open the poppet 89 and will do so when its value increases sufficiently to overcome the spring 92. The relief valve is set to open at a pressure appreciably higher than that required to actuate the control valve element 79 which deactivates the main pump. The reason for this is that a small auxiliary pump next to be described maintains the pressure in the outlet passage during periods of no demand when the main pump is deactivated.

The auxiliary pump is shown in Figs. 1, 2 and 4. It comprises three radially extending small pump barrels 100 mounted in bores provided therefor in the body 10 surrounding an eccentric surface 101 on the pump rotor 15. Each barrel 100 is supplied with a plunger 102 having a shoe 103 on its lower end urged against the eccentric surface 101 by a helical compression spring 104. Each pump barrel 100 has an enlarged head 105 at its upper end fitting in a counterbore 106 in the casing, the outer end of which counterbore is closed by a plug 107, the plug being sealed by a sealing ring 108 and held in position by a retaining ring 109. Each pump barrel 100 has a plurality of radially extending inlet ports 111 communicating with a small counterbore 112 in the body 10 which counterbore is communicated by passages 1121 with an annular passage 113 defined by a groove in the body 10 and the outer race of the bearing 13. This annular passage 113 is in turn communicated by passage 114 (Fig. 6 and Fig. 10) extending through the casing 10 and the cover 11 to the right end of the cylinder 75 as shown in Fig. 6. As previously described, the right end of cylinder 75 is in turn connected to the main inlet passage by the duct 77.

Referring again to Fig. 4 on each inward stroke of a plunger 102 it uncovers the inlet ports 111 and draws in fluid from the main inlet passage through the path just described. On each outward movement of a plunger 102, it first covers the inlet ports 111 to seal them off and then forces the fluid in the barrel past an outlet check valve 115 into the counterbore 106 below the closure cap 107. All three counterbores are interconnected by ducts 117 as shown in Fig. 2, and the uppermost counterbore, as shown in Fig. 1, is connected by a duct 118 to the cylinder 68 which as previously described is at all times connected to the main outlet passage through the relief valve cylinder 81.

The dimensions of the auxiliary pump are so chosen that its output is very small compared to the maximum output of the main pump. The volumetric output of the auxiliary pump is designed to be slightly in excess

of the leakage losses in the system supplied by the pump so that the auxiliary pump will maintain maximum pressure in the system when there is no demand thereon, and a slight excess of fluid will be constantly exhausted through the relief valve. Whenever appreciable fluid is withdrawn from the pump the pressure drops sufficiently to permit the spring 84 (Fig. 6) to shift the valve element 79 to the left, in which position pressure fluid is supplied to the cylinder 69 (Fig. 1) to shift the ring 25 into eccentric relation with respect to the rotor thereby setting the main pump into action.

As previously indicated, an important feature of the invention is that when the demand on the pump is less than that supplied by the auxiliary pump the main pump is completely unloaded and the friction reduced to a very low value. Thus when the ring 25 is restored to neutral position the pump ceases to pump any fluid whatsoever and the check valve 48 (Fig. 7) prevents any return to the main pump of the pressure fluid delivered by the auxiliary pump. Any residual pressure in the main pump is quickly dissipated by the unavoidable leakage, so that the shoes 24 bear against the ring 25 only with the centrifugal force induced in the pistons and shoes by their rotation. To insure unloading of the main pump, ring 25 may be arranged to be shiftable slightly below the concentric, neutral, position. At the same time, the elimination of pressure within the tubular member 47 (Fig. 7) eliminates the pressure force urging the pressure plate 19 against the rotor end, and there is substantially no force between those elements. The overall result is that wear is substantially eliminated and the frictional load of the pump is greatly reduced when no demand is made thereon.

Although for the purpose of explaining the invention, a particular embodiment thereof has been shown and described, obvious modifications will occur to a person skilled in the art, and I do not desire to be limited to the exact details shown and described.

I claim:

1. A variable volume pump assembly for supplying fluid to a system having a small continuous demand and a relatively large intermittent demand comprising: means defining inlet and outlet passages; a large capacity variable volume main pump unit having a suction port and a discharge port and having a control element movable to vary its pumping action from zero to maximum; means connecting said suction port to said inlet passage; a check valve for passing fluid from said discharge port to said outlet passage and preventing reverse flow; means responsive to pressure in said outlet passage for actuating said control element to reduced said pumping action to zero in response to a predetermined pressure and to incite said pumping action at a lesser pressure; a small constant volume auxiliary pump unit having volumetric capacity exceeding but of the same order of magnitude as said continuous demand and having a suction port connected to said inlet passage and a discharge port connected to said outlet passage; and common drive means for simultaneously driving both said pump units; whereby said predetermined pressure is maintained by said auxiliary unit and said variable volume unit is deactivated and depressurized when the demand on said assembly is within the capacity of said constant volume unit, and the variable volume unit is activated and pressurized only during periods when the demand exceeds the capacity of the constant volume unit and the outlet pressure drops to said lesser pressure.

2. An assembly according to claim 1 in which said first pump unit comprises a cylinder rotor having cylinders therein, pistons reciprocable in said cylinders in response to rotation of said rotor, and means for varying the strokes of the pistons in response to movement of said control element; said rotor having an annular face containing ports extending to said respective cylinders; a valve plate contacting said annular face in sealing re-

lation and having inlet and outlet ports cooperating with said cylinder ports in said rotor face; and means responsive to pressure fluid in said discharge port of said main pump urging said valve plate and said rotor face together whereby the friction between said face and plate varies with the output pressure developed by said first pump unit and deactivation of said unit substantially eliminates such friction.

3. A rotary pump comprising a casing containing inlet and outlet passages; a rotor containing cylinders and means for rotating it in said casing; pistons in said cylinders and means for reciprocating them in response to rotation of said rotor; said rotor having an annular face concentric with its axis of rotation and an annular row of cylinder ports in said face each communicating with a separate cylinder; a valve plate having a front face adapted to seal against said rotor face and having spaced inlet and outlet ports in the path of said cylinder ports, and having inlet and outlet passages extending from said ports through said plate; means supporting said plate against radial and rotary movement and for axial movement toward and away from said rotor face; said plate having a rear face defining a cylindrical sealing surface normal to said front face and surrounding the orifice of said outlet passage in said rear face; a tubular member having at one end a cylindrical surface complementary to and in sliding sealing relation with said cylindrical surface of said plate for conducting fluid from said outlet passage in the plate; means communicating the other end of said tubular member with said outlet passage in said casing; passage means in said casing communicating the rear end of said inlet passage in said plate with said inlet passage in said casing; said cylindrical sealing surface on said plate being of larger area than said outlet port in the front face of said plate whereby pressure fluid in said outlet port and said tubular member produces a resultant force urging said plate into sealing contact with said rotor face.

4. A rotary pump according to claim 3 in which said cylinders extend radially inwardly from the periphery

of said rotor and said means for reciprocating said pistons comprises: a ring encircling said rotor in the plane of said pistons and actuating means between each piston and said ring for limiting outward movement of said pistons, said casing defining an approximately cylindrical inner wall of larger diameter than and surrounding said ring; means for supporting said ring from said casing for movement between concentric and eccentric positions with respect to said rotor comprising a pair of juxtaposed cylindrically curved axially extending grooves in the outer surface of said ring and said inner wall of said casing respectively and a cylindrical pin fitting said grooves and supporting the ring for pivotal movement with respect to said casing; a second cylindrically curved groove in said inner wall of said casing approximately diametrically opposite said first mentioned grooves, and a second pin in said second groove and bearing against said ring for maintaining said ring and casing in contact with said first pin, said second pin having a cylindrical surface portion of the same curvature as and fitting against said second groove and having another cylindrical surface portion bearing against said ring, said other cylindrical surface portion and the adjacent surface of said ring having their centers of curvature at said first mentioned pin; and means for rocking said ring about said first pin between concentric and eccentric positions with respect to said rotor.

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