April 28, 1964 T. BUDZICH 3,130,674

THROTTLING CONTROL

Filed Jan. 12, 1961 2 Sheets-Sheet 1

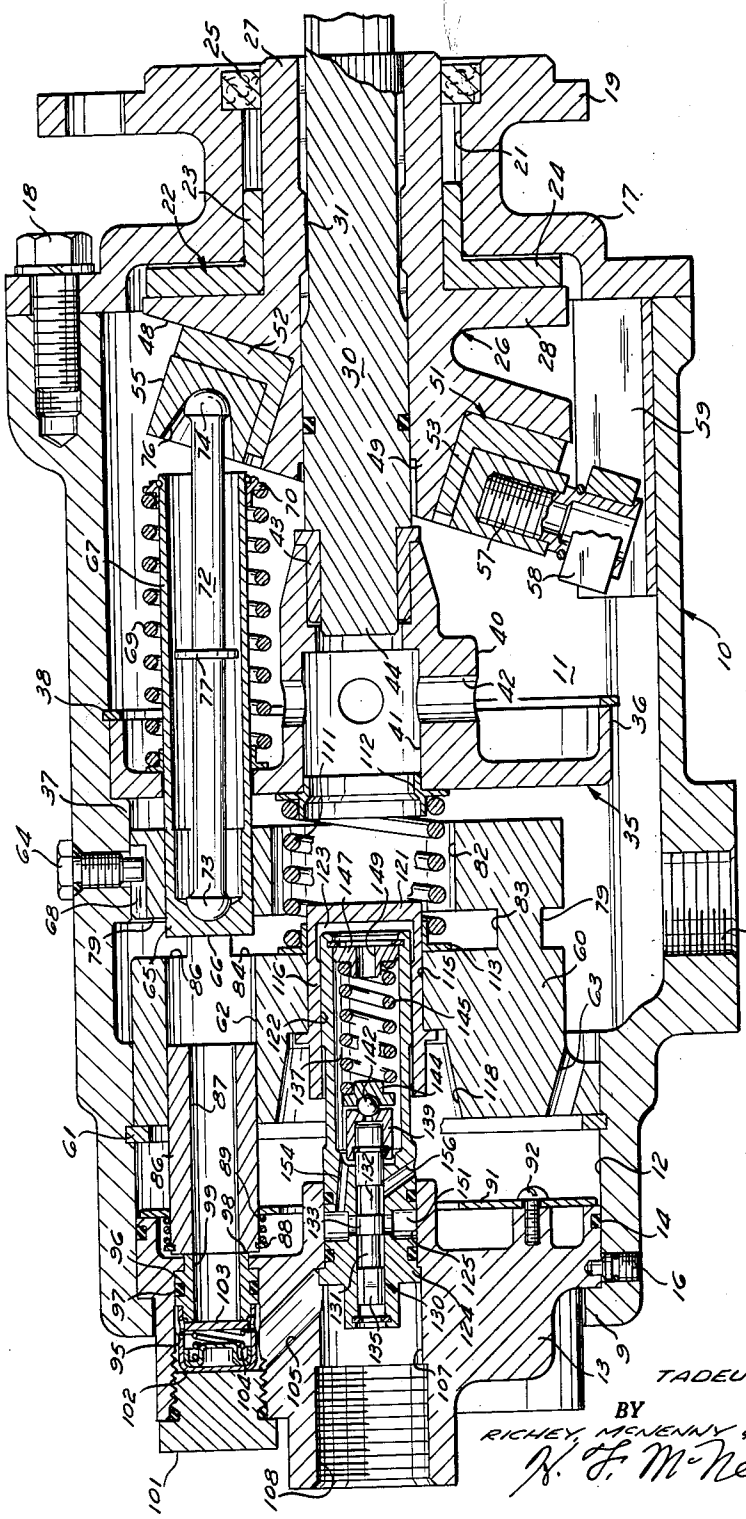


Fig. 1

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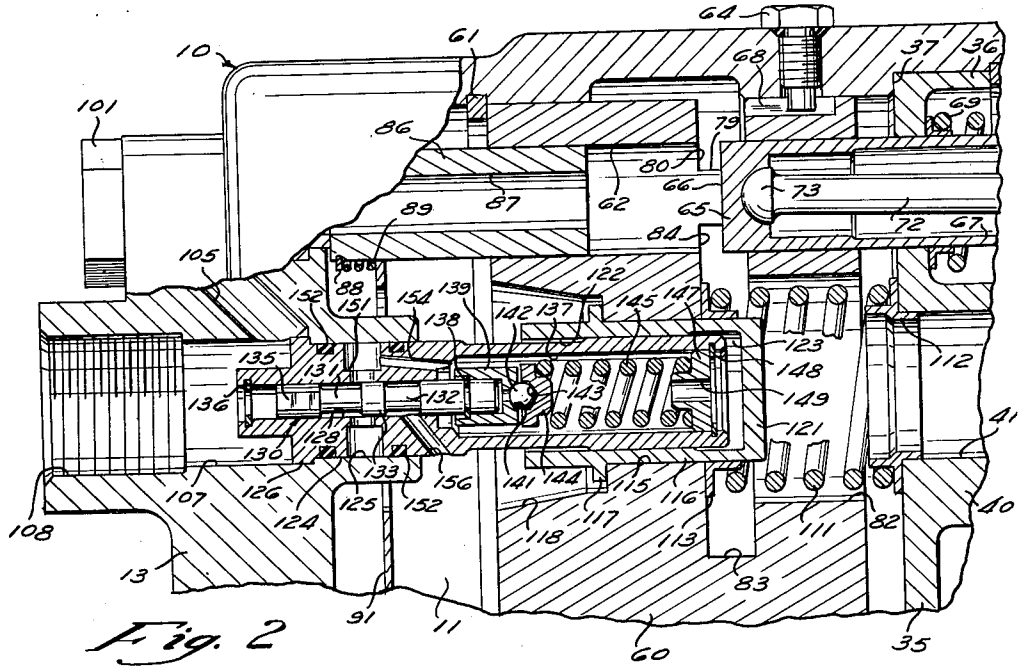


Fig. 2

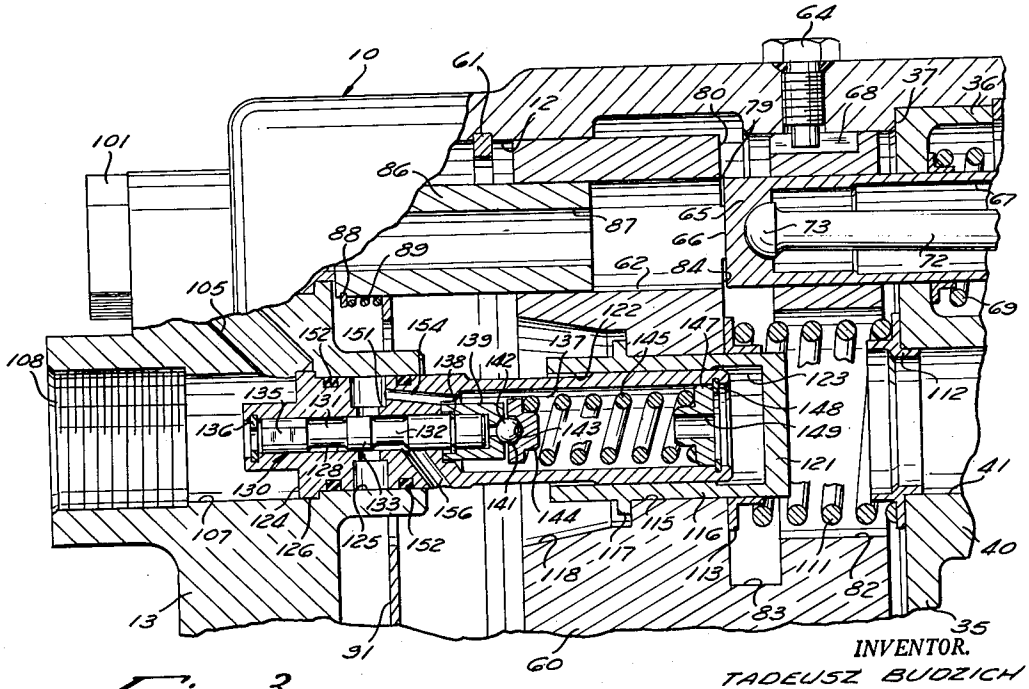


Fig. 3

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THROTTLING CONTROL

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7 Claims. (Cl. 103—37)

This invention relates generally to hydraulic pumps and more particularly to variable volume pumps of the positive displacement, reciprocating piston type.

Pumps of this type are shown in co-pending applications by the same inventor of which this application is a continuation in part, Serial No. 825,005 filed July 6, 1959, Serial No. 847,512 filed October 20, 1959, and Serial No. 17,832 filed March 28, 1960.

The various embodiments of the pump disclosed in the above mentioned co-pending applications generally show a pump housing and a cylinder block which is mounted for axial movement within the housing. The cylinder block is provided with a plurality of cylinders and pistons extending parallel to the longitudinal axis of the housing and cylinder block and in which the pistons are progressively reciprocated by an inclined drive mechanism including a swash or wobble plate. The ends of the cylinders on the other side of the drive mechanism are connected by means of reaction pistons to individual check valves which communicate into a central outlet passage. The cylinder block is shiftable along the longitudinal axis of the pump housing by means of hydraulic pressure and a compression spring arrangement and is controlled by means of a control valve which senses the pressure at the outlet and shifts the cylinder block to vary the volume of the pump and maintain the constant selected output pressure. As shown therein, the individual cylinders are filled through a filling slot which extends circumferentially around the cylinder block and provides a port opening into each of the cylinders in front of the piston. Accordingly, the volume is varied by shifting the cylinder block toward the end of the piston stroke so that the port is closed off at later positions along the length of the stroke of each piston to reduce the effective volume of the piston stroke and hence the volume of output of the pump. This method of control has been known as the "Spill" type of control since the strokes of the pistons are not changed, and the motion of the piston before the loading slot is closed causes the excess fluid above the selected output volume to be spilled out of the cylinders back into the chamber within the pump housing.

The present invention differs from the "Spill" type in that the effective output volume of the pump is varied by regulating the amount of fluid within the pump housing which enters each cylinder during the retracted portion of the piston stroke. The cylinder block has a circumferential filling slot which when the pump is adjusted for maximum volume is completely opened at the rearward reversal of the piston to allow the portion of the cylinder forward of the piston to be filled with fluid. As the piston moves forward it seals off the port and forces the fluid ahead of the piston out through the check valve into the outlet passage. To reduce the effective output volume of the pump, the filling slot or port is shifted with the cylinder block toward the rearward position of the pistons thereby decreasing the effective port area. Since the flow of fluid into the cylinder through this port is a function of the effective area of the port and the pressure differential between the pressure within the cylinder and the pressure within the pump housing, a reduction of the effective port area lessens the amount of fluid which can enter the cylinder. As a result, the cylinder is not completely filled when the pump is adjusted to an output

volume less than maximum. If the cylinder block is shifted so that the port is in the full rearward position where it is never uncovered by the end of the piston, no opportunity presents itself for the cylinder to be filled with fluid and the effective output volume of the pump is reduced to zero.

An advantage of this arrangement is that the distance through which the cylinder block must be shifted by the control mechanism is greatly reduced over that required for the spill type of pump control. The spill type of pump control requires the cylinder block to be shifted through a linear distance substantially equal to the full length of the piston stroke to vary the effective output volume of the pump from maximum volume down to zero volume. On the other hand, using the throttling type control of the present invention, it is necessary to shift the cylinder block only through a distance substantially equal to the linear width or length of the filling port slot and this distance may be equal to 25% of the piston stroke or less. By reducing the distance through which the cylinder block is shifted, the pump is able to respond more quickly to changes in demand at the output of the pump, since the cylinder block moves faster over the shorter distance. Furthermore, the resulting arrangement allows the pump to be more compact in size since the shortened length of stroke of the cylinder block allows the pump housing to be shortened by at least 75% of the stroke length over that necessary if the spill type control were employed.

Another advantage of the throttling type control of the present invention is that the pump consumes less power at or near the zero flow condition. When a pump is operated using the spill type control at or near zero flow, the cylinder is completely filled with fluid on the rearward stroke of the piston and this fluid is then spilled outwardly through the loading port during the forward stroke. Thus the pistons are required to exert a force to push the fluid within the cylinder out through the loading port and this force required creates friction on the bearings, which together with the churning of the oil being pumped into and out of the cylinders, requires additional power which is transformed into heat within the pump. Thus not only is less power required during these conditions with the present invention, but the pump runs considerably cooler and eliminates the occasional necessity of providing a cooling system for the fluid passing through the pump.

Another advantage of the throttling control of the present invention is an increase in the volumetric efficiency of the pump. In making a multiple piston pump, a certain clearance tolerance is necessary between the piston and cylinder walls to prevent possible seizing and to eliminate excessively high sliding friction. This clearance provides a leakage path for the fluid under pressure around the sides of the piston and out through the loading port. In a pump employing a spill type control, this passageway is not closed off until the effective length of the stroke begins, and when the pump is operating at a relatively low volume of flow condition, the piston must pass through a considerable portion of its effective stroke before the distance between the piston head and the loading slot is sufficiently great to prevent substantial leakage past the piston. According to the present invention, the piston always passes the loading slot to close it off at the beginning of its stroke, so that the length of the leakage path past the piston is always relatively long even under less than maximum volume conditions.

Another advantage of the present invention is that the pump operates at a relatively low noise level. In the spill type control, shock waves are produced both by the fluid being spilled out of the cylinder and by the piston

which during the intermediate volume condition closes off the loading slot during a relatively high velocity portion of its stroke. However, a pump employing throttling type control never spills fluid out into the pump chamber to create a shock wave. When the pump of the present invention is operated at less than maximum volume, the cylinder will not be completely filled as the piston moves forward to close off the filling slot and any compressive shocks between the head of the piston and the fluid within the cylinder occur only after the cylinder has been completely sealed.

Additional features and advantages of the pump of the present invention will readily become apparent upon an examination of the drawings and following detailed description of the preferred embodiment of the invention.

In the drawings:

FIG. 1 is a longitudinal cross-sectional view of a pump constructed in accordance with the present invention;

FIG. 2 is an enlarged fragmentary cross-sectional view showing the cylinder block and pump control with a piston at the rearward end of its stroke when the cylinder block is shifted for maximum output volume; and

FIG. 3 is an enlarged fragmentary cross-sectional view similar to that of FIG. 2, but showing the cylinder block shifted toward the minimum output volume position.

Referring now to the drawings in greater detail, it will be seen that the pump includes a pump housing 10 enclosing a chamber 11. At the one end, pump housing 10 is formed with an internal cylindrical end portion 12 within which is fitted an outlet housing 13. Outlet housing 13 is fitted against an inwardly projecting flange 9 on pump housing 10 and held in place by means of set screws indicated at 16. The periphery of outlet housing 13 is provided with an O-ring seal 14 to prevent leakage around the joint between the outlet housing and the pump housing. An inlet opening 15 is formed in the side of pump housing 10 adjacent its mid-point for admission of fluid into chamber 11.

The other end of pump housing 10 is closed off by a drive housing 17 secured in place by means of cap screws 18. Drive housing 17 may have a flange 19 for mounting the pump in position on machinery, an electric motor or the like. Drive housing 17 is provided with an axial bore 21 within which is fitted a bearing member 22. Bearing member 22 includes an axial portion 23 extending along bore 21, and a radial portion 24 extending outwardly along the wall of the drive housing. A drive member 26 is rotatably journaled within axial portion 23 of bearing 22, and has a shank portion 27 extending axially outward to make sealing contact with an oil seal 25 in the outer end of axial bore 21. Drive member 26 has a radially extending flange portion 28 making bearing contact with radial portion 24 of bearing member 22 to absorb the thrust exerted thereon. A drive shaft 30 extends axially through the drive member 26 and is secured thereto by a spline connection indicated at 31. The drive shaft 30 projects axially outward from the drive housing 17 and is secured to a suitable motor or drive mechanism for driving the pump.

A spaced distance inwardly from the drive housing 17, a transverse web member 35 is mounted within chamber 11 in pump housing 10 and is provided with an axially extending peripheral flange 36 adapted to fit against the walls of pump housing 10. At its one end, peripheral flange 36 abuts against an inwardly extending annular shoulder 37 formed on the pump housing, and is secured in position against axial movement by a snap ring indicated at 38. Web member 35 has an axial extension 40 projecting toward the drive housing 17. Extension 40 is provided with an axial bore 41 and openings 42 which allow free circulation of fluid within the chamber 11 from one side of the web member 35 to the other. At its outer end, extension portion 40 is provided with a pilot bearing insert 43 which receives the end portion 44 of drive shaft 30.

The drive member 26 is provided with an inclined face 48 and a hub portion 49 which extends perpendicular to the inclined face 48. A bearing member 51 is mounted on drive member 26 to have a radial flange portion 52 extending outwardly along the inclined face 48, and an axial sleeve portion 53 which extends along the hub portion 49. Bearing 51 serves to journal a wobble plate 55 which is supported on the radial and axial portions 52 and 53, respectively. At its lower end, wobble plate 55 is provided with a projecting stud 57 on which is rotatably secured a guide block 58. Guide block 58 is adapted to slide longitudinally within a channel 59 formed in the pump housing 10 to prevent rotation of the wobble plate 55 as it is oscillated by rotation of the drive member 26.

A cylinder block 60 is slidably journaled within the cylindrical end portion 12 of pump housing 10 for axial movement back and forth therein. The range of movement of the cylinder block is limited in the one direction by the web member 35 and in the other direction by a snap ring 61 fitted within an annular groove in the cylindrical end portion 12. The cylinder block is restrained from rotating within the pump housing by a set screw 64 in the housing wall which has a dog point engaging a longitudinal groove or guide slot 68 on the outer periphery of the cylinder block. Cylinder block 60 is provided with a plurality of axially extending cylinder bores 62 which are spaced equidistantly about the cylinder block at points equidistant from the axis of the drive shaft 30. Only one cylinder bore has been shown for purposes of clarity, but it is understood that the pump employs a plurality of cylinders similar to the one shown in the drawings. Cylinder block 60 is also provided with several passages 63 extending therethrough from one side to the other so that fluid can circulate within the chamber 11 on both sides of the cylinder block.

A piston or plunger 65 is fitted within one end of each of the cylinder bores 62 and has a head portion 66 on the end away from drive member 26. Piston 65 has a tubular skirt portion 67 extending toward the wobble plate 55. A compression spring 69 is fitted around the tubular skirt portion 67 and abuts at the one end against the web member 35. At the other end, compression spring 69 abuts against a spring retainer 70 secured to the end of the tubular skirt portion 67. Thus compression spring 69 normally biases the piston 65 toward the wobble plate 55. A piston rod 72 is positioned within the tubular skirt portion 67 and has a ball end 73 which seats against the head portion 66 on the piston. At its other end, piston rod 72 has another ball end 74 which is seated within a cup-like recess 76 in the face of wobble plate 55. At its mid-point, the piston rod 72 is provided with a radially extending flange 77 to maintain the piston rod 72 in approximate axial alignment within tubular skirt 67 if either end of the piston rod should become unseated from its seat against the piston head 66 or the wobble plate 55.

The cylinder block 60 has an annular groove or outer filling slot 79 extending around its periphery and opening into each of the cylinder bores 62 to admit fluid therein. An axial bore 82 extends partially through the cylinder block 60 from the end adjacent web member 35 and is provided with an enlarged annular groove or inner filling slot 83 which opens into the cylinder bores 62 at their inner sides. Forward edge 80 of filling slot 79 and the forward edge 84 of the inner filling slot 83 lie in the same plane, and the spacing of the edges 80 and 84 from the piston head 66 determines the effective port area of the two filling slots 79 and 83.

A reaction piston 86 is slidably journaled within the other end of each of the cylinder bores 62 to make sealing contact therein to prevent leakage of fluid between the reaction piston and the cylinder bore. Each of the reaction pistons 86 has an axial bore 87 extending therethrough, and at the end away from the piston 65 the reaction piston 86 is fitted with a snap ring 88 which

forms one abutment for a compression spring 89 fitted around the end of the reaction piston. The other end of compression spring 89 abuts against a retainer plate 91 secured to the inner face of the outlet housing 13 by suitable screws 92.

The outlet housing 13 is fitted with a bore or chamber 95 opposite each of the reaction pistons 86 to receive a port member 96. Port member 96 is provided with an O-ring seal 97 to make sealing contact with the bore 95 and has a projecting end 98 to make sealing contact with the end of the adjacent reaction piston 86. A bore 99 extends through port member 96 to connect with the axial bore 87 in the reaction piston 86. The opposite end of the bore or chamber 95 is closed off by a suitable plug 101, and the remainder of bore 95 contains a cage 102 which encloses a spring 104 biasing a check valve plate 103 against the end of the port member 96. Thus the check valve plate 103 serves to permit fluid within the reaction piston 86 and port member 96 to pass outwardly, but prevents fluid from entering the port member 96 from the outlet side. A passage 105 extends from each of the bores 95 radially inward to connect to an axially located outlet passage bore 107 in the outlet housing 13. Outlet bore 107 is provided with a suitable threaded portion 108 for connection to the outlet line.

A cylinder block spring 111 is positioned within the axial bore 82 and abuts at one end against a suitable thrust washer 112 fitted within the axial bore 41 on web member 35. At its other end, spring 111 abuts against another thrust washer 113 positioned on the cylinder block 60. Cylinder block 60 is provided with a reduced bore 115 to receive a cup-like cylinder member 116 which is prevented from moving toward the web member 35 by a radially extending flange 117 which abuts against the bottom of a counterbore 118 on the other end of the cylinder block 60. The cylinder member 116 includes an end wall 121 closing off a bore 122 therein which defines a chamber 123. A control valve housing 124 extends into bore 122 to make a sliding sealing fit therein. The other end of control valve housing 124 fits within a reduced bore 125 in outlet housing 13 coaxial with outlet bore 107. A radially extending flange 126 within the outlet bore 107 retains the control valve housing 124 in place.

The control valve housing 124 has an axial bore 128 extending therethrough to slidably receive a valve spool 130. The valve spool 130 has annular reduced portions 131 and 132 adjacent its mid-point separated by a land portion 133 therebetween. The outer end of valve spool 130 adjacent outlet bore 107 is provided with several longitudinal flats 135 thereabout to permit fluid to pass inwardly along these flats from outlet bore 107 to reach the first reduced portion 131. A snap ring 136 is fitted within the outer end of axial bore 128 to retain the valve spool 130 therein. The other end of valve spool 130 projects into a chamber 137 within control valve housing 124 and is provided with a snap ring 138 to make bearing contact with a cap member 139 which fits over the end of the valve spool.

The outer end of cap member 139 has a valve seat 141 to receive a bearing ball 142 fitted within a socket 143 on a spring abutment member 144. Spring abutment member 144 receives one end of a helical compression control spring 145 extending axially within the chamber 137. The other end of control spring 145 abuts against a plug 147 retained within chamber 137 by a snap ring 148. A bore 149 passes through plug 147 to provide communication between the chamber 137 within control valve housing 124 and the chamber 123 in the cylinder member 116.

An annular port 151 extends around the control valve housing 124 within the reduced bore 125 to communicate with the portion of axial bore 128 adjacent the land portion 133 on valve spool 130. O-ring seals 152 are provided on the outer periphery of control valve housing

124 on each side of the annular port 151 for sealing purposes. A passage 154 connects the annular port 151 to chamber 137, and a drain passage 156 extends through control valve housing 124 from a point adjacent the other reduced portion 132 on valve spool 130 and communicates with the chamber 11 within the pump housing.

Turning now to the operation of the pump, it will be understood that the chamber 11 within the pump housing 10 is normally completely filled with hydraulic fluid. When the pump is stationary, the cylinder block spring 111 biases the cylinder block 60 into abutting contact with the snap ring 61 so that the filling slots 79 and 83 provide the maximum port area for the cylinders 62. At no point within the pump is any fluid maintained under pressure. When the pump is started, the drive shaft 30 will rotate the drive member 26 to oscillate the wobble plate 55 and cause the pistons 65 to reciprocate within their respective cylinder bores 62. At each forward stroke of the piston, the fluid trapped within the cylinder bore 62 and within the reaction piston 86 is forced outwardly through port member 96 past the check valve plate 103 to be discharged through the outlet bore 107. The control valve mechanism will be in the position shown in FIG. 3, so that the control spring 145 will bias the valve spool 130 toward the left and connect the chambers 123 and 137 to the drain passage 156.

As pressure builds up within the outlet bore 107, the pressure exerts a force on the valve spool 130 tending to force it toward the right to compress the control spring 145. The control spring 145 is preloaded so that it will not begin to compress and allow movement of the valve spool 130 until the outlet pressure approaches the regulated level. When the outlet pressure reaches the level determined by the amount of preload applied to the control spring 145, the valve spool 130 moves toward the right and assumes the position as shown in FIG. 2.

With the spool in this position, fluid entering the axial bore 128 flows past the flats 135 to reach the first reduced portion 131. Since the valve spool 130 has been shifted so that the land 133 now only partially covers the annular port 151, the fluid flows from reduced portion 131 outwardly through annular port 151, through passage 154 into chamber 137. From here, the fluid will pass through the bore 149 in plug 147 to enter the chamber 123 in the cylinder member 116. The pressure within the cylinder member 116 exerts a force upon the end wall 121 tending to move the cylinder member 116 and hence cylinder block 60 toward the right to compress the cylinder block spring 111. As this is done, the edges 80 and 84 of the filling slots 79 and 83 are moved toward the head portion 66 of piston 65 as it is in the retracted position. Thus, motion of the cylinder block in this direction decreases the effective area of the ports formed by the filling slots and throttles down and decreases the amount of fluid which can enter the cylinder bores through these ports during the reversal of the piston. Therefore, when the piston moves through its forward or pumping stroke, the interior of the cylinder bore 62 and the reaction piston 86 will not be completely filled with fluid, and a smaller volume of fluid will be discharged through the port member 96 into the outlet bore 107. Since a smaller volume of fluid is introduced into the outlet bore on each stroke of the piston, the effective volume of the pump is decreased and the fluid pressure in the outlet bore will thereby be reduced.

As the piston is retracted, a partial vacuum will exist within the cylinder bore, and when the filling slot post is uncovered, the fluid within the pump housing chamber 11 will flow inward to fill the cylinder bore to an extent determined by the amount of throttling of fluid flow through the ports as determined by the effective port area. Since the fluid flow into the cylinder bore through the port is throttled and restricted by the reduced effective port area, the volume of flow becomes a function of the area and pressure differential as well as time, and the rate of flow is

independent of time. Therefore, under these conditions the output volume of the pump will be substantially independent of the rotary speed of the wobble plate and the rate of reciprocation of the pistons since the period of time during which the inlet port is uncovered is expressed as a percentage of the piston stroke and is independent of the number of reciprocations per minute of the piston. It will be understood that the compression spring 89 biasing the reaction piston 86 into contact with the port member 96 is sufficiently strong that the external fluid pressure will not be able to shift the reaction piston away from the port member to allow fluid to enter the cylinder even with the highest vacuum in the cylinder when the ports remain closed so that there is no output from the pump.

When the pressure within the outlet bore 107 drops below the preselected pressure determined by the amount of preload on the control spring 145, the biasing force of spring 145 will be greater than the force exerted by the pressure in outlet bore 107 upon the valve spool 130. Therefore, the valve spool 130 will be shifted to the left and assume the position shown in FIG. 3. With the valve spool 130 in this position, the land portion 133 is shifted to the left of the annular port 151 which is connected to the other reduced annular portion 132 on the valve spool. Under these conditions, the fluid within the chambers 123 and 137 will exhaust through the passage 154 into the annular port 151, and from there flow past the other reduced annular portion 132 to the drain passage 156 and hence into pump housing chamber 11. As the fluid is drained from the chambers 123 and 137, the cylinder block spring 111 will shift the cylinder block 60 in a leftward direction to increase the effective port area of the filling slots 79 and 83. This decreases the throttling effect of the ports and allows more fluid to enter the cylinder bores 62 to increase the volume of fluid pumped into the outlet bore 107 at each stroke of the piston. After the pressure within the outlet bore 107 has been increased to the regulated value, the valve spool 130 will be shifted back to the intermediate position where the land portion 133 blocks off the annular port 151 to prevent any flow of fluid to or from the chambers 123 and 137 and maintain the cylinder block 60 in that regulated position.

It will therefore be seen that responsive to an increase or decrease in the pressure within the outlet bore 107, the cylinder block 60 will be shifted either to the right or to the left to decrease or increase, respectively, the effective port area of filling slots 79 and 83. Since the cylinder block 60 need move only through the short distance equal to the effective port length of the filling slots, that is, the distance between the edges 80 and 84 and the head 66 of the piston as shown in FIG. 1, the cylinder block will be able to move quite rapidly even though it may have considerable mass. This allows the pump to respond quite rapidly to sudden transient changes in rate of output flow and hence in the pressure within the outlet bore 107.

Since the piston head 66 closes off the filling slots as it moves past their edges 80 and 84 at the very beginning of its stroke before the piston has built up appreciable velocity, substantially no shock wave will be produced by the piston as it moves past this port and the noise of the pump will therefore be greatly decreased. Furthermore, since the filling slots are always closed near the beginning of the stroke, the only time that fluid will be forced out of the cylinder 62 is when the pump is operating at maximum volume and the cylinder 62 is completely filled with fluid at the beginning of each stroke. However, under this condition only a small amount of fluid will be forced out during the initial portion of the piston stroke before the filling slots are completely closed. When the pump is operating at less than maximum volume, the cylinder 62 will not be completely filled, and therefore no fluid will be forced outward through the ports as the piston moves forward. Furthermore, since the filling slots are covered during the initial portion of the piston stroke regardless of the output volume of the pump, the piston head 66 will at all times be well past the filling slot edges 80 and 84 during the terminal portion of the stroke when maximum pressure is developed in the cylinder bore, whether the effective volume of the pump is at a maximum or at a minimum. Therefore, a piston head 66 will effectively seal the cylinder bore 62 and allow a minimum of leakage between the piston and the cylinder bore during the pumping stroke.

Although the pump shown and described herein employs an automatic self-regulating control for shifting the cylinder block to vary the output volume, it is also contemplated that other automatic control means or even a manual control means can be employed to shift the cylinder block to vary the effective output volume of the pump. It is also appreciated that various modifications and rearrangements of the pump structure may be made and that the invention can be applied to pumps of a different structure without departing from the scope of the invention as defined in the following claims.

What is claimed is:

1. A pump comprising a pump housing providing a fluid chamber therein, an inlet to said fluid chamber, an outlet on said pump housing, a cylinder block mounted for slidable movement within said fluid chamber along a longitudinal axis, a plurality of cylinder bores in said cylinder block parallel to said longitudinal axis, a piston within each of said cylinder bores, drive means in said pump housing to progressively reciprocate said pistons in said cylinder bores between forward and retracted positions, means connecting each of said cylinder bores to said outlet, an inlet port for each cylinder bore in said cylinder block, said inlet port having forward and rearward axially spaced edges, said cylinder block defining a predetermined maximum displacement position when the head of the piston is intermediate said port edges when said piston in the retracted position, control means to shift said cylinder block with respect to said pump housing and said drive means in the direction to shift the forward edge of said inlet port toward the retracted piston position to decrease the inlet port area exposed when the piston is in the retracted position to throttle fluid flow into said cylinder bore and decrease the effective output volume of the pump, and stop means on said pump housing to prevent movement of said cylinder block in the opposite direction from said maximum displacement position.

2. A pump comprising a pump housing providing a fluid chamber therein, an inlet to said fluid chamber, an outlet on said pump housing, a cylinder block mounted for slidable movement within said fluid chamber along an longitudinal axis, a plurality of cylinder bores in said cylinder block parallel to said longitudinal axis, a piston within each of said cylinder bores, drive means in said pump housing to progressively reciprocate said pistons in said cylinder bores between forward and retracted positions, means connecting each of said cylinder bores to said outlet, an inlet port for each cylinder bore in said cylinder block, said inlet port having forward and rearward axially spaced edges, said cylinder block defining a predetermined maximum displacement position when the head of the piston is intermediate said port edges when said piston is in the retracted position, control means operable responsive to the fluid pressure in said outlet port to shift said cylinder block with respect to said pump housing and said drive means in the direction to shift the forward edge of said inlet port toward the retracted piston position to decrease the inlet port area exposed when the piston is in the retracted position to throttle fluid flow into said cylinder bore and thereby decrease the effective output volume of the pump, and stop means on said pump housing to prevent movement of said cylinder block in the opposite direction from said maximum displacement position.

3. A pump comprising a pump housing providing a fluid chamber therein, an inlet to said fluid chamber, an outlet on said pump housing, a cylinder block mounted for slidable movement within said fluid chamber along an longitudinal axis, a plurality of cylinder bores in said cylinder block parallel to said longitudinal axis, a piston within each of said cylinder bores, drive means in said pump housing for progressively reciprocating said pistons in said cylinder bores between forward and retracted positions, means connecting each of said cylinder bores to said outlet, an inlet port for each cylinder bore in said cylinder block, each of said inlet ports having forward and rearward axially spaced edges, said cylinder block defining a predetermined maximum displacement position when the head of the pistons are intermediate said port edges with the piston in the retracted position, an expansible chamber fluid motor operatively connected between said cylinder block and said pump housing to shift said cylinder block with respect to said pump housing and said drive means in the direction to move the forward edges of said inlet ports toward the retracted positions of the pistons to decrease the port area exposed when the pistons are in the retracted position to throttle fluid flow into said cylinder bores and thereby decrease the effective output volume of the pump, stop means on said pump housing to prevent movement of said cylinder block in the opposite direction from said maximum displacement position, spring means biasing said cylinder block in opposition to said fluid motor, and control valve means operable to control the flow of fluid to and from said fluid motor.

4. A pump comprising a pump housing providing a fluid chamber therein, an inlet to said fluid chamber, an outlet on said pump housing, a cylinder block mounted for slidable movement within said fluid chamber along an longitudinal axis, a plurality of cylinder bores in said cylinder block parallel to said longitudinal axis, a piston within each of said cylinder bores, drive means in said pump housing to progressively reciprocate said pistons in said cylinder bores between forward and retracted positions, means connecting each of said cylinder bores to said outlet, inlet port means on said cylinder block opening into each of said cylinder bores, said inlet port means including an annular groove on the outer periphery of said cylinder block, said annular groove defining forward and rearward edges of said inlet port at said cylinder bores, said cylinder block defining a predetermined maximum displacement position when the head of the piston is intermediate said port edges when the piston is in the retracted position, control means to shift said cylinder block with respect to said pump housing and said drive means in the direction to shift the forward edge of the inlet port toward the retracted position of the piston to decrease the port area exposed when the piston is in the retracted position to throttle fluid flow into said cylinder bore and thereby decrease the effective output volume of the pump, and stop means on said pump housing to prevent movement of said cylinder block in the opposite direction from said maximum displacement position.

5. A pump comprising a pump housing providing a fluid chamber therein, an inlet to said fluid chamber, an outlet on said pump housing, a cylinder block mounted for slidable movement within said fluid chamber along a longitudinal axis, a plurality of cylinder bores in said cylinder block, parallel to said longitudinal axis, a piston within each of said cylinder bores, drive means in said pump housing to progressively reciprocate said pistons in said cylinder bores between forward and retracted positions, means connecting each of said cylinder bores to said outlet, inlet port means including an annular groove on the outer periphery of said cylinder block opening into said cylinder bores, said annular groove defining forward and rearward axially spaced edges of the inlet port at said cylinder bores, said cylinder block defining a predetermined maximum displacement position when the head of the piston is intermediate said port edges when the piston is in the retracted position, an expansible chamber fluid motor operatively connected between said cylinder block and said pump housing and adapted to shift said cylinder block with respect to said pump housing and said drive means in the direction to shift the forward edge of the said inlet port toward the retracted position of said piston to decrease the port area exposed when the piston is in the retracted position to throttle fluid flow into said cylinder bores and decrease the effective operative volume of the pump, stop means on said pump housing to prevent movement of said cylinder block in the opposite direction from said maximum displacement position, spring means biasing said cylinder block in opposition to said fluid motor, and control valve means operable to control the flow of fluid to and from said fluid motor.

6. A pump comprising a pump housing providing a fluid chamber therein, an inlet to said fluid chamber, an outlet on said pump housing, a cylinder block mounted for slidable movement within said fluid chamber along a longitudinal axis, a plurality of cylinder bores in said cylinder block parallel to said longitudinal axis, a piston within each of said cylinder bores, drive means in said pump housing to progressively reciprocate said pistons in said cylinder bores between forward and retracted postions, means connecting each of said cylinder bores to said outlet, inlet port means for said cylinder bores, said inlet port means including annular groove on the periphery of the cylinder block opening into said cylinder bores, an axial bore in said cylinder block, said inlet port means also including an annular groove in said cylinder bores, said outer and inner annular groove having coplanar forward and rearward axially spaced edges, said cylinder block defining a predetermined maximum displacement position when the head of piston is intermediate said port edges when said piston is in the retracted position, control means to shift said cylinder block with respect to said pump housing and said drive means in the direction to shift the forward edges of said annular grooves toward the retracted position of the piston to decrease the port area exposed when the piston is in the retracted position to throttle fluid flow into the cylinder bore and thereby decrease the effective output volume of the pump, and stop means on said pump housing to prevent movement of said cylinder block in the opposite direction from said maximum displacement position.

7. A pump comprising a pump housing providing a fluid chamber therein, an inlet to said fluid chamber, an outlet on said pump housing, a cylinder block mounted for slidable movement within said fluid chamber along a longitudinal axis, a plurality of cylinder bores in said cylinder block parallel to said longitudinal axis, a piston within each of said cylinder bores, drive means in said pump housing to progressively reciprocate said pistons in said cylinder bores between forward and retracted positions, means connecting each of said cylinder bores to said outlet, inlet port means for said cylinder bores including an annular groove on the outer periphery of said cylinder block opening into said cylinder bores, an axial bore in said cylinder block, said inlet port means also including an annular groove in said cylinder block extending radially outward from said axial bore into said cylinder bores, said outer and inner annular grooves having coplanar forward and rearward axially spaced port edges, said cylinder block defining a predetermined maximum displacement position when the head of the piston is intermediate said port edges when the piston is in the retracted position, an expansible chamber fluid motor operatively connected between said pump housing and said cylinder block to shift said cylinder block with respect to pump housing and said drive means in the direction to shift said forward edges

of said annular grooves toward the retracted position of the pistons to decrease the port area exposed when the piston is in the retracted position to throttle fluid flow into the cylinder bore and decrease the effective output volume of the pump, stop means on said pump housing to prevent movement of said cylinder block in the opposite direction from said maximum displacement position, spring means biasing said cylinder block in opposition to said fluid motor, and control valve means

operable to control the flow of fluid to and from said fluid motor.

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