

May 12, 1970

J. G. RUSSELL
AXIAL PISTON PUMP

3,512,178

Filed April 24, 1967

2 Sheets-Sheet 1

Fig. 1

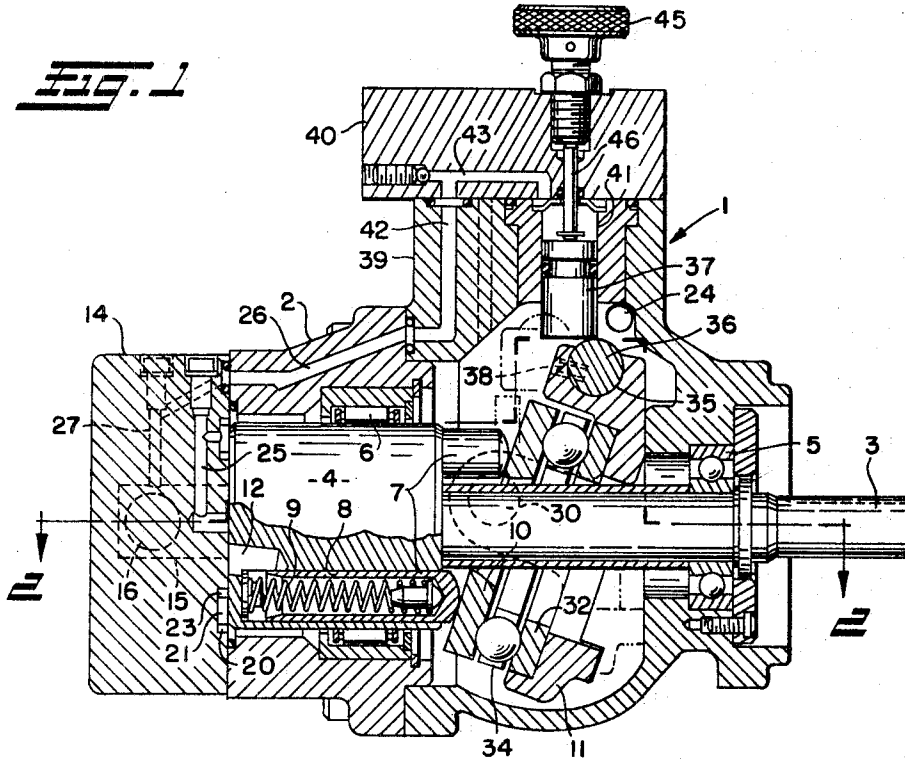


Fig. 2

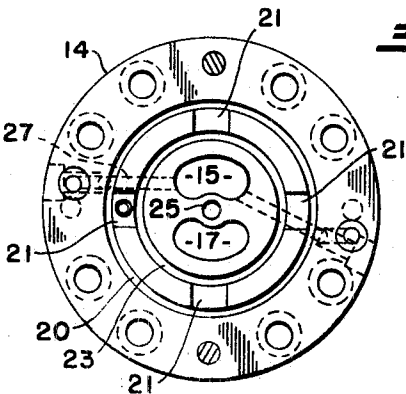
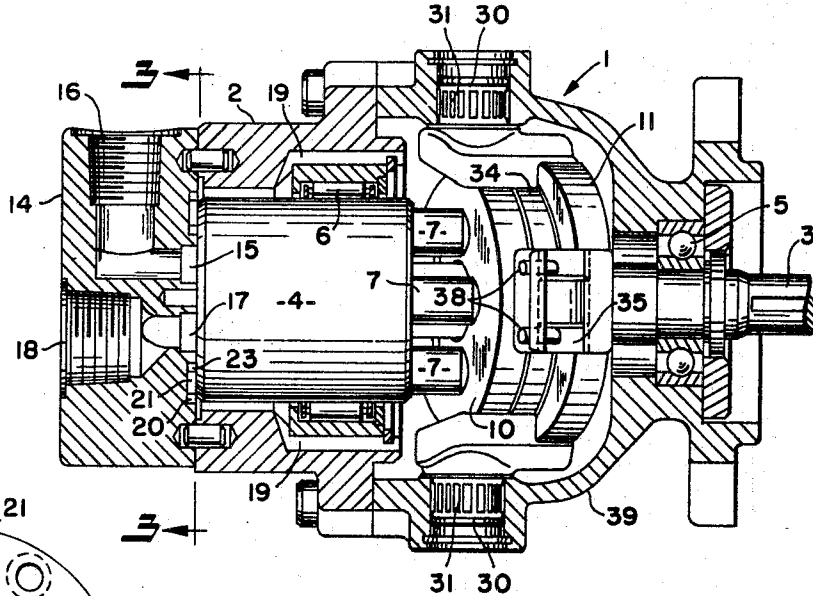


Fig. 3

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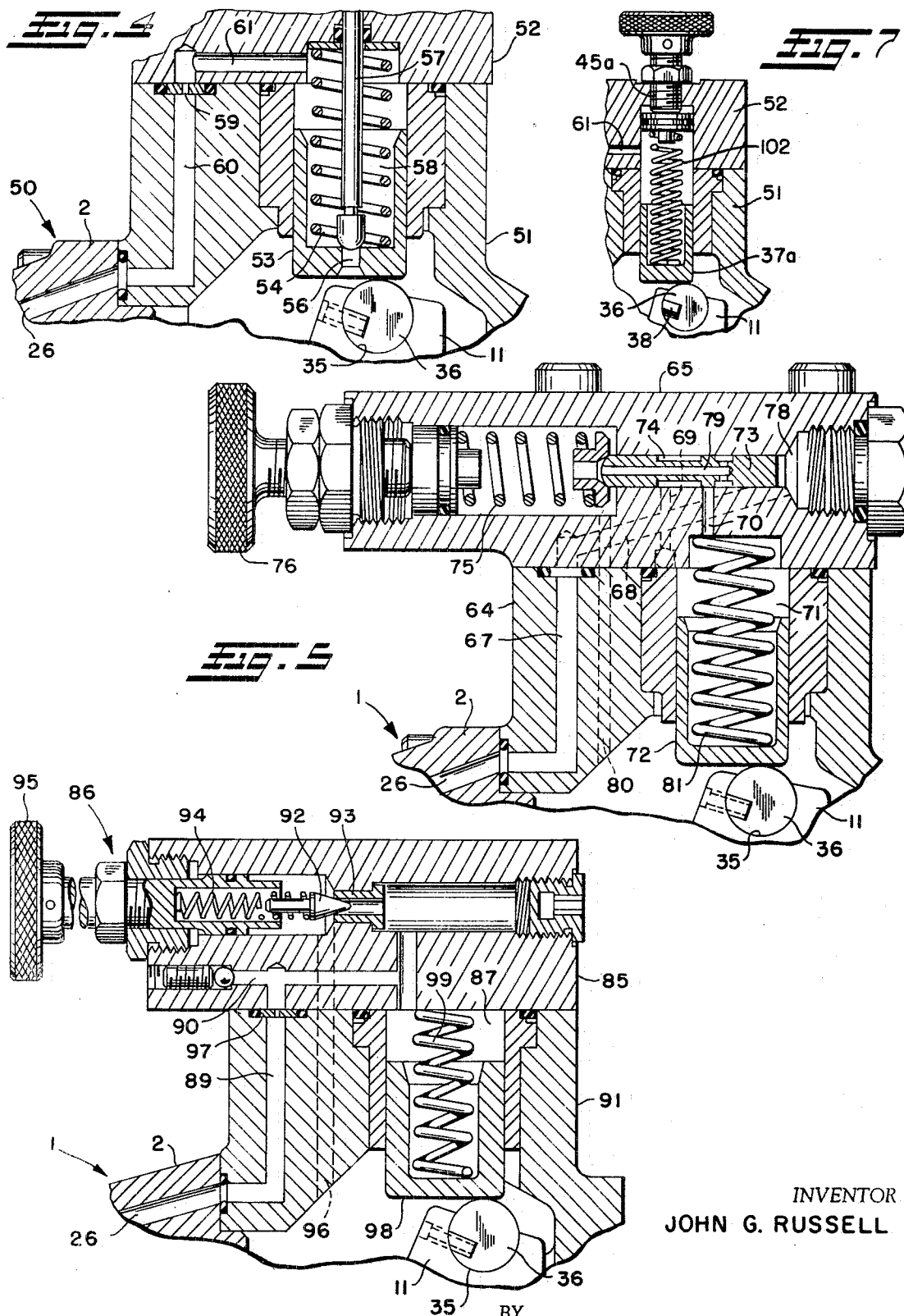
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AXIAL PISTON PUMP

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3 Claims

ABSTRACT OF THE DISCLOSURE

Axial piston pump (or motor) wherein resultant thrust load of pistons acting on cam plate tends to move cam plate toward zero displacement position, and wherein a delivery pressure actuated control piston acting on cam plate with increasing moment arm tends to move cam plate toward maximum displacement position. For manual adjustment of displacement fluid pressure on control piston assists in movement of control piston to increase displacement. For automatic pressure compensation, control piston moment on cam plate exceeds resultant piston moment whereby upon reduction of fluid pressure in control cylinder (as by venting), the then dominating resultant piston moment moves cam plate toward zero displacement position.

The present invention relates generally as indicated to an axial piston pump (or motor) and more especially to a variable displacement axial piston pump in which the cam plate engaged by the axial pistons may be shifted manually or automatically to various tilted positions with respect to the rotational axis of the pump. More particularly, the present invention relates to a pump of the character indicated which may be readily converted from manual adjustment of capacity to automatic adjustment as by pressure compensation.

It is a principal object of this invention to provide an axial piston pump in which the cam plate pivot is offset from the rotational axis of the pump so that the resultant of the axial thrust loads of the pistons on said cam plate tends to swing the cam plate toward zero displacement position.

It is another object of this invention to provide a pump of the character indicated in which the swinging of the cam plate toward zero displacement position by the resultant thrust load of the pump pistons is partially balanced by a delivery pressure-actuated control piston which acts on the cam plate with increasing moment arm about the cam plate pivot so that manual shifting of said control piston to swing said cam plate toward maximum displacement position may be accomplished with relatively small loads on the adjustment mechanism which acts on the control piston.

It is another object of this invention to provide a pump of the character indicated which is pressure compensated and in which the swinging of the cam plate toward zero displacement position is overbalanced by a delivery pressure actuated control piston which acts on the cam plate with increasing moment arm about the cam plate pivot to tend to swing the cam plate to maximum displacement position, said control piston having pressure sensing means associated therewith effective at a desired maximum delivery pressure of the pump to decrease the pressure acting on said control piston to a value such that the then dominating piston thrust moment on said cam plate swings said cam plate toward zero displacement position.

Other objects and advantages of the present invention will become apparent as the following description proceeds.

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To the accomplishment of the foregoing and related ends, the invention, then, comprises the features hereinafter fully described and particularly pointed out in the claims, the following description and the annexed drawings setting forth in detail certain illustrative embodiments of the invention, these being indicative, however, of but a few of the various ways in which the principle of the invention may be employed.

In said annexed drawings:

FIG. 1 is a central, vertical cross-section view through the rotational axis of a variable displacement pump embodying manual adjustment of the displacement;

FIG. 2 is a cross-section view taken substantially along the line 2—2, FIG. 1;

FIG. 3 is an elevation view along line 3—3, FIG. 2 of the port adapter which cooperates with the end of the cylinder block to provide intake and discharge valving;

FIG. 4 is a fragmentary cross-section view of the FIG. 1 pump in which is embodied a displacement adjusting device adapted for remote control; and

FIGS. 5, 6, and 7 are fragmentary cross-section views of the FIG. 1 pump in which are embodied two different forms of pressure compensating controls for automatically decreasing pump capacity whenever the delivery pressure exceeds a predetermined value (FIGS. 5 and 6), or for decreasing pressure at increased flow (FIG. 7).

Referring now more particularly to the drawings, and first to FIGS. 1 to 3, the pump 1 therein shown comprises a housing 2 in which the drive shaft 3 and cylinder block 4 are journaled as on bearings 5 and 6, the cylinder block 4 having a circular array of axial pistons 7 (herein five in number, for example) reciprocable in cylinder bores 8 therein and actuated by springs 9 toward the right as viewed in FIG. 1 against the inclined face of the bearing ring 10 of the cam plate 11.

Each cylinder bore 8 has a passage 12 through the left end of the cylinder block 4 which is in sliding sealed engagement with the end face of the port adapter 14, the latter being of well known form including an arcuate discharge passage 15 leading to the discharge port 16 and an arcuate intake passage 17 leading to the intake port 18. Thus, as the cylinder block is rotated in the housing 2, the pistons 7 which are being moved to the right by their springs 9 as viewed in FIG. 1 will draw fluid into the respective bores 8 through the intake port 18 and passage 17; and the pistons 8 which are being moved to the left by engagement with the cam plate 11 will pump fluid out of their respective cylinder bores 8 through the discharge passage 15 and discharge port 16. The interior of the pump housing 2 is lubricated by leakage flow and maintained at intake pressure by way of the passages 19 which lead from circular groove 20, radial grooves 21, and circular groove 23 to the drain passage 24. The space between arcuate passages 15 and 17 is maintained at low pressure by the passage 25 which leads to one radial groove 21 as shown. On the other hand, the passage 26 in the housing 2 is communicated by way of passage 27 with the discharge passage 15 for a purpose which will hereinafter be described in detail.

The cam plate 11 aforesaid (also well known in the art as a swash plate), is herein shown as being of ring-like form having diametrically opposite trunnions 30 which are journaled in the bearings 31 in the pump housing 2 and has seated therein one bearing ring 32 of a thrust bearing 34 of which the other bearing ring 10 is disposed for engagement by the ends of the pistons 7. As best shown in FIG. 1, the axis of the trunnions 30 is offset with respect to the rotational axis of the cylinder block 4 so that in considering the difference in moment arms of the points of contact of the pistons 7 with the cam plate 11, the resultant thrust on the

cam plate 11 by the pistons 7 in the discharge zone will always tend to swing said cam plate 11 in a counterclockwise direction as viewed in FIG. 1, toward a position of less or zero displacement. By reason of the disposition of the axis of the cam plate trunnions 30 as aforesaid, even when the cam plate 11 is in zero displacement position with the face of ring 10 perpendicular to the rotational axis of the pump 1, there will yet be a resultant piston thrust force due to springs 9, resisting clockwise rotation of the cam plate 11. For any given discharge pressure, the counterclockwise moment increases as the cam plate 11 swings toward zero displacement position.

The upper side of the cam plate 11 has a cylindrical recess 35 which is parallel to the trunnions 30 and in which is disposed a cylindrical roller 36 preferably formed with a slight flat (not shown) to engage the bottom plane end of a control piston 37, the roller 36 being held in centered position in the recess 35 as by means of the pins 38 overlying the ends of said roller 36.

Bolted or otherwise attached to the housing 2 and to each other, are housing parts 39 and 40 which provide a cylinder 41 in which the control piston 37 is movable, and passages 42 and 43 communicating the cylinder 41 with the aforesaid high pressure passage 6. It is to be noted that the control piston 37 operates along a line perpendicular to the rotational axis of the pump 1 and that the moment arm thereof producing clockwise turning moment on cam plate 11 rapidly increases as the cam plate 11 is moved from the zero displacement position to a tilted position, the maximum displacement position being shown in FIG. 1.

The rate of increase of the clockwise moment is made to correspond with the rate of increase of the counterclockwise moment. The area of the control piston 37 is such that the delivery pressure acting downwardly thereon is slightly underbalanced with reference to the resultant counterclockwise moment of the pump pistons 7 acting on the cam plate 11. Therefore, the cam plate 11 position, which determines pump delivery, is easily varied by a light external force.

As previously mentioned, in FIG. 1 the cam plate 11 has been adjusted to maximum displacement position and if it be desired to decrease the pump displacement, the adjusting screw 45 threaded in the housing part 40 is turned to move it upward, whereupon the counterclockwise rotation of the cam plate 11 due to the dominating counterclockwise moment of the piston thrust, will raise the control piston 37 and the rod 46 interposed between said control piston 37 and the adjusting screw 45, thus to decrease the pump displacement. Similarly, if the cam plate 11 is at the zero displacement position or at some intermediate displacement position, the displacement of the pump 1 may be increased simply by screwing down the adjusting screw 45 which, through the rod 46, will push the control piston 37 downwardly to increase the tilt of the cam plate 11 as desired. Because the counterclockwise moment of the cam plate 11 is partially counterbalanced by discharge pressure acting on the control piston 37, the displacement of the pump 1 can easily be varied just by turning the adjusting screw 45 in one direction or the other by hand.

Referring now to FIG. 4, the pump 50 may be the same as the pump 1 in FIGS. 1 to 3 except for the displacement adjusting mechanism. In FIG. 4 there is provided housing parts 51 and 52 similar to those in FIG. 1, except that the control piston 53 is biased by spring 54 and has an opening 56 therein which is controlled by a remotely controlled valve rod 57. In this case, the area of the control piston 53 is such that the high pressure fluid acting thereon together with spring 54 will overbalance the counterclockwise moment of the thrust load of the pistons 7 on the cam plate 11.

The valve rod 57 may be actuated by any suitable means, such as horsepower sensing means, not shown. The valve rod 57 is pressure balanced, i.e., the diameter of the seat 56 is the same as the diameter of the seal with housing part 52 and when the rod 57 is moved upwardly the pressure in the control chamber 58 is vented to intake via the interior of the housing 2 at a faster rate than fluid can be replenished into the chamber through the orifice disk 59 in the high pressure passage 60-61. Accordingly, with the pressure thus decreased in the control chamber 58, the axial thrust of the pistons 7 on the cam plate 11 exerts a then dominant counterclockwise moment on the cam plate 11, whereby the control piston 53 is forced upwardly against the biasing spring 54 and the decreased pressure therebehind. As soon as the control piston 53 is raised to engage its seat 56 with the valve rod 57 the pressure in the control chamber 58 builds up to halt further upward movement of the control piston 53, and tends to move the control piston 53 downwardly to shift the cam plate 11 toward its maximum displacement position. However, as soon as the control piston 53 moves downwardly a slight amount again to vent the chamber 58 faster than it can be replenished through the orifice disk 59, the pressure will decrease and the cam plate 11 again will tend to move to a position of lower pump displacement.

Accordingly, when the valve rod 57 is in any preset position, the displacement and outlet pressure will be maintained at a preselected value by alternate opening and closing of the control chamber 58. When it is desired to increase the displacement of the pump 50, the valve rod 57 will be shifted downwardly to maintain the chamber 58 in closed condition whereby the cam plate 11 will be turned clockwise toward maximum displacement position by the then dominant clockwise moment of the high pressure in control chamber 58. It is to be understood that various forms of servo control mechanisms (not shown) may be associated with the valve rod 57 to maintain predetermined output conditions of the pump 50.

Referring now to FIG. 5, the housing parts 64 and 65 again provide passages 67, 68, 69, and 70 which lead from high pressure passage 26 to a control cylinder or chamber 71 in which there is disposed a spring-biased control piston 72, the area of the control piston 72 being greater than that of the control piston 37 in FIG. 1 so that there will be an over-balancing clockwise moment on the cam plate 11 holding the same in its maximum displacement position. The housing part 65 has a spring-biased slide valve 73 therein which normally allows passage of fluid into the control chamber 71 via the passages 67 to 70 and the groove 74 of the slide valve 73. The slide valve 73 is biased to its normally open position by the spring 75, the compression of which can be adjusted to any desired value of the adjusting screw 76. The high pressure passage 68 also leads to a chamber 78 to which the end of the slide valve 73 is exposed whereby, if the delivery pressure of the pump exceeds a predetermined maximum, this pressure acting on the end of the slide valve 73 will shift the latter toward the left to close communication of the high pressure with the control chamber 71 and to open the control chamber 71 to drain pressure through the passages 79 in the slide valve and through the passage 80 of the mating housing parts 64 and 65 which lead to the interior of the pump housing 2.

As evident from FIG. 5, when the control chamber 71 is thus vented by movement of the slide valve 73 toward the left, the axial thrust of the pistons 7 on the cam plate 11 will rotate the cam plate 11 in a counterclockwise direction toward a position of decreased displacement and then as the delivery pressure decreases, the spring 75 will bias the slide valve 73 to reopen the connection between the high pressure passages 67 to 70 and the control chamber 71, whereby the control piston 72 with the assistance of its biasing spring 81, will again force the cam

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plate 11 to rotate in a clockwise direction to its position of maximum displacement at which it will remain unless the pressure again increases sufficiently to actuate the slide valve 73 as aforesaid.

In FIG. 6 instead of employing a slide valve 73 as in FIG. 5, the housing part 85 is provided with an adjustable relief valve 86. When the pressure in the control chamber 87 (through passages 89 and 90 in the respective lower and upper housing parts 91 and 85 which communicate with high pressure passage 26) increases to a value sufficient to force the relief valve member 92 out of engagement with its seat 93, as determined by the adjustment of the spring 94 by the adjusting screw 95, the control chamber 87 will again be vented to intake pressure through the passage 96 leading into the interior of the pump housing 2 faster than fluid can be replenished into the control chamber 87 via the orifice disk 97. In this way there is a reduction of pressure in the control chamber 87, whereupon the cam plate 11 can again move in a counter-clockwise direction to move control piston 98 upward to decrease pump displacement. If the delivery pressure then drops to desired value, the spring 94 will bias the relief valve member 92 to seated position, whereupon the pressure will again build up in the control chamber 87, whereupon the control piston 98 with assistance of its biasing spring 99, will urge the cam plate 11 to move in a clockwise direction to its maximum displacement position, as shown in FIG. 6.

From the foregoing it can be seen that the present invention provides a simplified form of variable displacement axial piston pump 1 in the FIGS. 1 to 3 embodiment of which displacement is manually adjusted through a fluid actuated control piston 37 which produces a turning moment on the cam plate 11 to partially balance the opposite turning moment due to the resultant of the axial thrust loads of the pistons 7 on the cam plate 11; in the FIG. 4 embodiment of which the displacement of the pump 1 is varied by the axial positioning of the valve rod 57 as by suitable remote control means, said valve rod 57 cooperating with a control piston 53 that produces on the cam plate 11 a turning moment that exceeds that due to the resultant of the thrust loads of the pistons 7 on the cam plate 11; and in FIGS. 5 and 6 of which there is automatic pressure compensation either by an adjustable spring-biased slide valve 73 (FIG. 5) or by an adjustable relief valve 86 which has a spring-biased relief valve member 92 responsive to control chamber 87 pressure (FIG. 6).

Furthermore, in all embodiments of the invention herein disclosed, the fluid pressure actuated control piston 37, 53, 72, or 98 moves along a line which is perpendicular to the rotational axis of the pump and exerts a progressively increasing turning moment on the cam plate 11 which corresponds to the progressive increasing counter-moment due to axial thrust of the pump pistons 7 against the cam plate 11.

In FIGS. 1 to 3 if the hydraulic moments are balanced then the manual adjusting means 45 and 46 need only overcome the cam plate 11 moment due to the piston return springs 9. However, in the other forms of the invention disclosed in FIGS. 5 and 6, the hydraulic moment of the control piston 72, or 98 exceeds the hydraulic moment of the resultant of the axial thrust of the pistons 7 so that the cam plate 11 will always be urged toward maximum displacement position when less than predetermined maximum pressure is delivered by the pump 1 but when such predetermined maximum pressure is exceeded the control chamber pressure will be decreased so that the hydraulic moment due to the pistons 7 will prevail to tend to move the cam plate 11 toward zero displacement position. In FIG. 4 the hydraulic moment of the control piston 53 likewise exceeds the hydraulic moment of the axial thrust of the pistons 7 so that the cam plate will be urged toward maximum displacement position when valve rod 57 restricts passage

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56 to maintain a predetermined pressure on the top side of the control piston. When the restriction is such that the pressure is less than the predetermined amount, the hydraulic moment due to the pistons 7 will prevail to move the cam plate 11 toward zero displacement position. In each of the forms of the invention disclosed in FIGS. 4, 5, and 6, it is preferred that the turning moment exerted on the cam plate 11 by the respective spring biased control pistons 53, 72, and 98 exceed the opposite turning moment exerted on the cam plate 11 by the resultant of the bias of the return springs 9 on the pistons 7 so that the pump will not be stopped in zero displacement position. In other words, the cam plate 11 will be in tilted position (preferably maximum displacement position) whenever the pump is started so as to immediately displace fluid for pressurizing the respective control chambers 58, 71, and 87.

In the alternate form of FIG. 7, control piston 37a is smaller in diameter than piston 53 so that pump outlet pressure on the top side of the piston is insufficient of itself to move cam plate 11 toward maximum displacement against the hydraulic moment of pistons 7 but such pressure is augmented by spring 102 for accomplishing this. Spring 102 may be adjusted by screws 45a. This construction provides another means of automatic pressure compensation that results in a less pressure at large flows than the forms of FIGS. 5 and 6, which is sometimes desirable.

It is to be noted that the housing parts 39 (FIGS. 1 and 2), 51 (FIGS. 4 and 7), 64 (FIG. 5), and 91 (FIG. 6), may be of identical construction except for the tubular members in which the respective control pistons 37, and 37a, 53, 72, and 98 are reciprocally mounted.

I therefore particularly point out and distinctly claim as my invention.

1. A pump comprising a housing having an intake port and a discharge port; a cylinder block journaled in said housing and having a circular array of cylinder bores disposed in generally parallel relation to the rotational axis of said cylinder block; pistons reciprocable in the respective cylinder bores; a cam plate adjacent one end of said cylinder block pivotally mounted in said housing about an axis transverse to the rotational axis of said cylinder block; said cam plate being engaged by said pistons to cause reciprocation of the latter during rotation of said cylinder block; valve means in said housing serially communicating said cylinder bores with said intake and discharge ports as said pistons are thus reciprocated in said cylinder block; the resultant of the thrust of said pistons against said cam plate producing an increasing turning moment resisting movement of said cam plate to increased displacement position; said cam plate having a cylindrical recess in its outer periphery whose axis is parallel to the axis of said cam plate; a cylindrical roller disposed in said recess; and a control piston in said housing actuated by fluid pressure in said discharge port and acting on said cylindrical roller in a direction producing an increasing turning moment on said cam plate tending to move it to increased displacement position.

2. The pump of claim 1 further comprising pin means in said cam plate overlying the ends of said roller for retaining said roller in centered position in said recess.

3. A pump comprising a housing having an intake port and a discharge port; a cylinder block journaled in said housing and having a circular array of cylinder bores disposed in generally parallel relation to the rotational axis of said cylinder block; pistons reciprocable in the respective cylinder bores; a cam plate adjacent one end of said cylinder block pivotally mounted in said housing about an axis transverse to and offset from the rotational axis of said cylinder block; said cam plate being engaged by said pistons to cause reciprocation of the latter during rotation of said cylinder block; valve

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means in said housing serially communicating said cylinder bores with said intake and discharge ports as said pistons are thus reciprocated in said cylinder block; the offset of said cam plate being such that the resultant of the thrust of said pistons against said cam plate produces an increasing turning moment resisting movement of said cam plate from zero to maximum displacement position; control piston means in said housing actuated by fluid pressure in said discharge port and acting on said cam plate at a zone for producing a corresponding increasing turning moment on said cam plate tending to move it from zero to maximum displacement position thus to maintain approximately the same turning moment differential at various positions of said cam plate; the turning moment of said cam plate due to thrust of said pistons thereagainst being less than that due to thrust of said control piston thereagainst; and means for decreasing the fluid pressure acting on said control piston relative to the fluid pressure acting on said pistons to a value such that said cam plate is turned toward zero displacement position, said means comprising a valve which operates at a predetermined fluid pressure to decrease the fluid pressure acting on said control piston relative to

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the fluid pressure acting on said pistons to permit turning of said cam plate toward zero displacement position, said valve comprising a spring-biased slide valve which is actuated by predetermined fluid pressure to isolate said control piston from said discharge port and to permit escape of the fluid acting on said control piston.

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