

Dec. 8, 1959

T. BUDZICH

2,915,985

PUMP

Filed June 20, 1957

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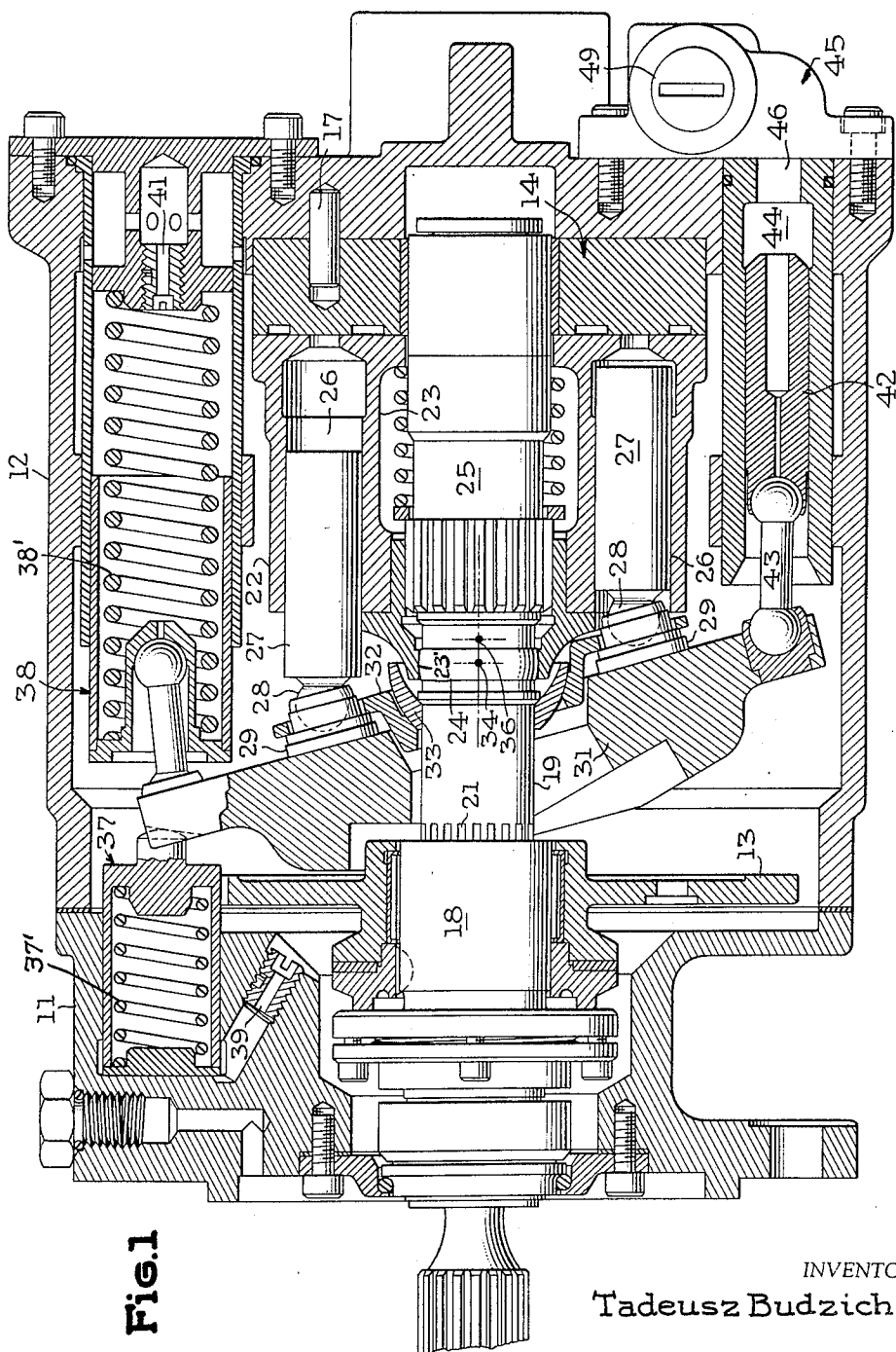


Fig. 1

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Fig. 2

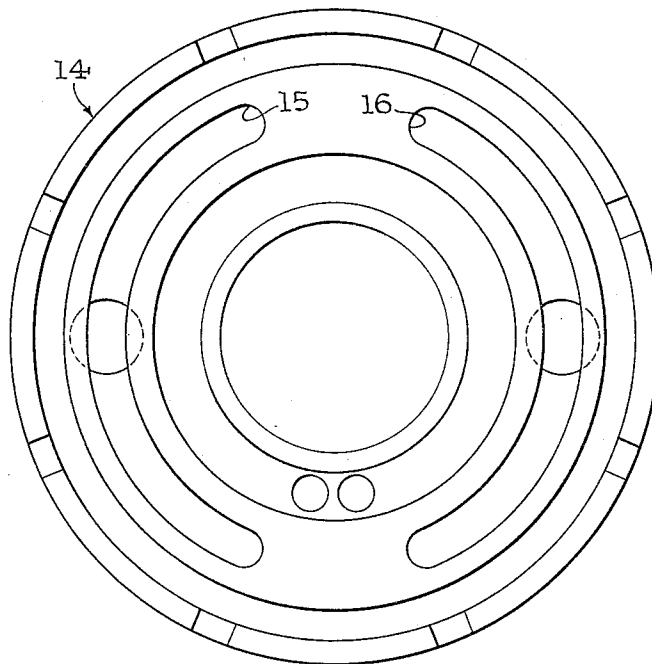
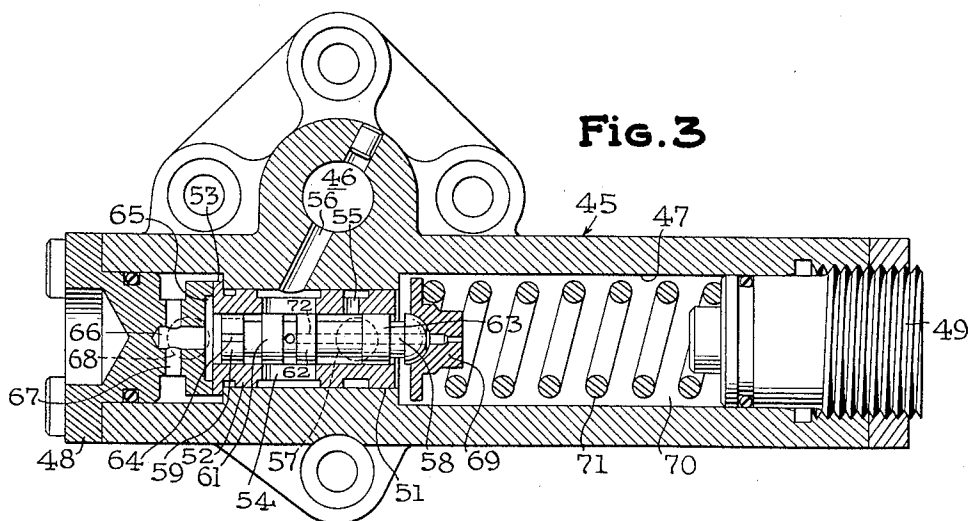


Fig. 3



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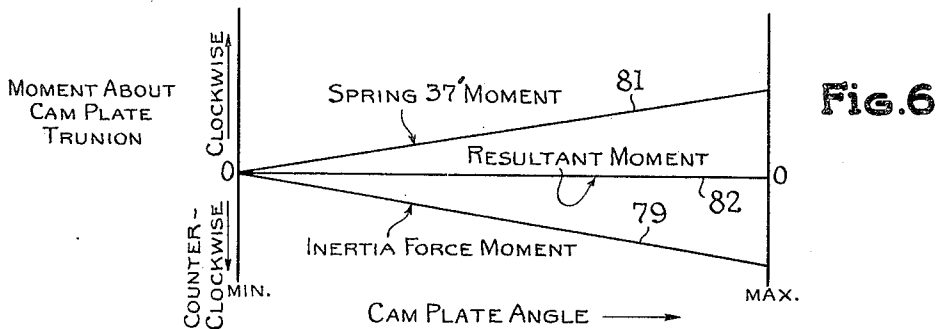
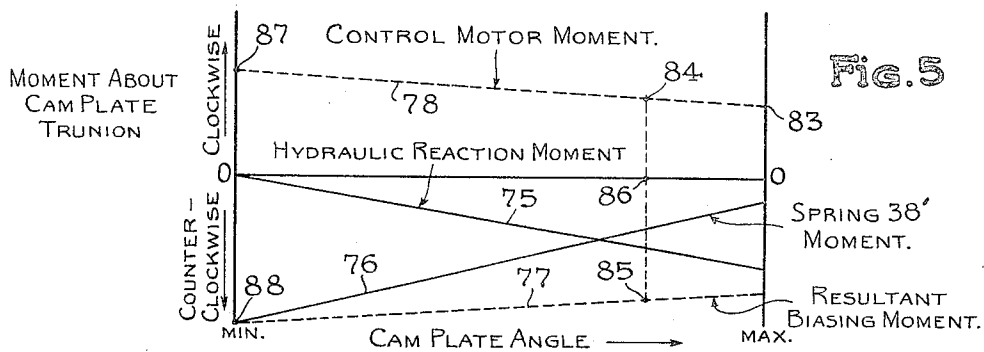
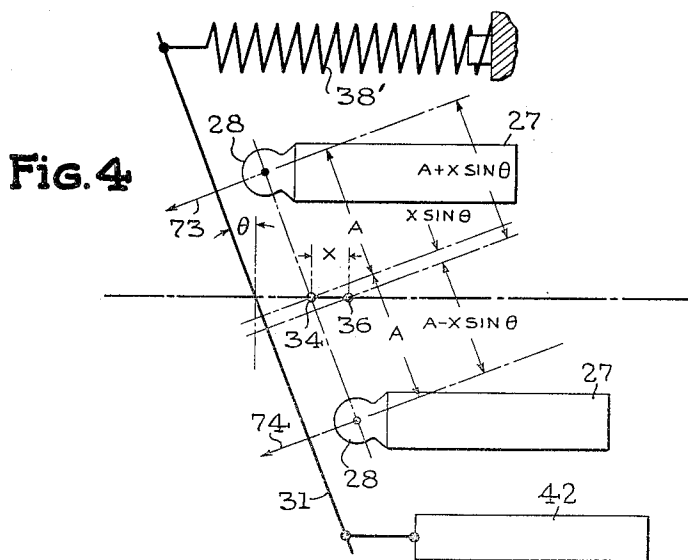
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Application June 20, 1957, Serial No. 666,857

2 Claims. (Cl. 103—162)

This invention relates to fluid pressure pumps and more particularly to variable displacement pumps of the type including a plurality of longitudinally reciprocating pistons and an angularly adjustable cam plate for moving the pistons on their discharge strokes and for governing the lengths of these strokes.

One well known pump of this type comprises a cylinder barrel having a plurality of longitudinally reciprocating pistons, a cam plate mounted for angular adjustment about an axis extending in a direction normal to the axes of the pistons, a spring biasing the cam plate toward its maximum stroke-producing position, a piston motor for moving the cam plate toward its minimum stroke-producing position against the bias of the spring and a control valve for varying the pressure in the motor in accordance with pump discharge pressure. During operation the cam plate remains in its maximum stroke-producing position until discharge pressure reaches a predetermined maximum and then gradually moves toward the minimum stroke-producing position as the discharge pressure continues to increase above this maximum. Since the force exerted by the biasing spring increases as this movement proceeds, it is apparent that the final discharge pressure established by the control system will be greater than the predetermined maximum. For sensitive control action, it is essential that the pressure differential required to move the cam plate between its two limiting stroke-producing positions be as small as practicable.

The most obvious way to reduce this differential is to employ a biasing spring having a very low rate, i.e., a spring in which the force required to produce unit deflection is small. However, this solution is not as satisfactory as it might seem because, since the size of a spring varies inversely with its rate, the pump housing would become larger and heavier as the rate decreased. In environments where size and weight are critical, this change could not be tolerated. Furthermore, in some cases it is imperative that the cam plate move rapidly from its minimum to its maximum stroke-producing position upon a decrease in discharge pressure. Inasmuch as the speed with which the cam plate moves depends on its mass and on the accelerating force applied to it, the spring force acting on the cam plate must be large. A low rate spring satisfying this requirement would tend to be unstable because of the extreme length necessary to provide this force and rate combination. As a result, the spring would be apt to buckle laterally and consequently the reliability of the pump would be seriously impaired.

The object of this invention is to provide a displacement controlling mechanism of the type mentioned which affords sensitive control action without incurring the size, weight and reliability penalties mentioned. Basically the invention comprises a cam plate which is supported for angular adjustment about an axis so located that the forces transmitted to the cam plate by the pistons produce a moment about said axis urging the cam plate toward

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its maximum stroke-producing position. The piston force moment increases in a substantially linear manner with movement of the cam plate toward this stroke-producing position and serves to supplement the moment of the biasing spring. As a result, the effective rate of the biasing spring is reduced without changing its physical size.

In the preferred form of the invention, only the moment due to the hydraulic reaction force component of the total piston force is used to augment the biasing spring. The moment due to the other component, namely the inertia force of the piston itself, is balanced by a second spring which produces a moment urging the cam plate toward its minimum stroke-producing position and which varies in the same way as the inertia force moment. Since the magnitude of the inertia force depends on the speed of the pump, it is desirable to eliminate this component so that the location of the cam plate support axis will be independent of pump speed. In this way, the pump can be operated at different speeds without the need for relocating a major portion of the pumping mechanism.

The preferred embodiment of the invention will now be described in relation to the accompanying drawings, in which:

Fig. 1 is an axial section of a variable displacement pump incorporating the invention.

Fig. 2 is a view of the face of the valve plate showing the porting arrangement.

Fig. 3 is a sectional view of the control valve showing the valve plunger in its lap position.

Fig. 4 is a schematic diagram illustrating the unbalanced hydraulic reaction force moment.

Fig. 5 is a graphical representation of the control moments acting on the cam plate.

Fig. 6 is a graphical representation of the inertia force moment and its balancing moment.

Referring to Fig. 1, the pump employing the invention comprises a housing having separable sections 11 and 12 which are connected by suitable mating flanges and bolts (not shown) and which, when assembled, locate and rigidly hold an intermediate wall 13. A valve plate 14, containing inlet and discharge ports 15 and 16 (see Fig. 2), is freely seated within a bore formed in the housing section 12 and is constrained against rotation by pin 17. A two-part drive shaft, having telescoping sections 18 and 19 joined by splines 21, is journaled in wall 13 and valve plate 14 for supporting and driving rotary cylinder barrel 22. The cylinder barrel is formed with an axial bore 23 having a portion 23' which is in great circle engagement with spherical enlargement 24 carried by the shaft for providing a universal and axially slidable support for the cylinder barrel 22. A torque tube 25, formed with external and internal splines at its respective opposite end which mate with splines formed on the cylinder barrel 22 and the shaft section 19, respectively, connects these members in driving relation. This feature is fully described and claimed in applicant's copending application Serial No. 656,574, filed May 2, 1957. The cylinder barrel 22 contains a circumferential series of longitudinal cylinder bores 26 which extend through the barrel and are arranged to sequentially register with inlet and discharge ports 15 and 16 as the cylinder barrel rotates.

Pistons 27, carrying spherical heads 28 and universally attached shoes 29, are mounted in cylinder bores 26 for reciprocation by cam plate 31 and nutating plate 32. Nutating plate 32 is seated on a collar 33 having a spherical outer surface which engages a similarly shaped recess formed in the nutating plate. The center of this spherical surface as well as the center of spherical enlargement 24 is located at the point of intersection 34 of the axis of the drive shaft and the plane of the centers of spherical piston

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heads 28. The cam plate is supported by yokes and trunnions (not shown) for angular adjustment about an axis extending in a direction normal to the axis of the drive shaft and intersecting that axis at the point 36. It should be noted that this point 36 is longitudinally displaced from the point 34 in a direction away from the cam plate.

As shown in Fig. 1, the upper end of the cam plate 31 is connected with two spring-biased plungers 37 and 38 which, under the action of springs 37' and 38' respectively, urge the cam plate toward minimum and maximum stroke-producing positions. The chambers behind the plungers are connected with the interior of the pump housing by restricted passages 39 and 41 which function to damp the motion of cam plate 31.

The lower end of this cam plate is universally connected with motor piston 42 by connecting rod 43. The motor piston contains a longitudinal passage which transmits fluid from motor working chamber 44 to the spherical end of connecting rod 43 for lubricating the universal joint. The working chamber 44 communicates with control valve 45 via passage 46.

Referring to Fig. 3, the control valve 45 comprises a housing having a through central bore 47 and plugs 48 and 49 for closing and sealing the opposite ends of the bore. Intermediate its ends, the bore 47 contains a reduced diameter portion 51 into which is pressed the valve sleeve 52. A flange 53, formed on the sleeve, is clamped between the annular shoulder formed at the left end of the reduced diameter portion 51 and the end of plug 48 for holding the valve sleeve in position. Two ports 54 and 55 are located in valve sleeve 52; the first communicating with motor chamber 44 via passages 56 and 46 and the second communicating with the interior of the pump housing via passage 57. Slidable within the valve sleeve is a plunger 58 having four spaced annular lands 59, 61, 62 and 63. The land 59 contains longitudinal slots 64 for connecting the space 65 between plug 48 and valve sleeve 52 with the recess between lands 59 and 61. The space 65 communicates with the discharge port 16 of the pump via longitudinal bore 66 and radial passages 67 formed in plug 48 and port 68 located in the valve housing. The right end of the plunger 58 carries a hemispherical head for universally supporting spring seat 69. A spring 71, reacting between plug 49 and seat 69, urges the plunger toward plug 48. Restricted passage 72, formed in the plunger 58, connects spring chamber 70 with the motor port 54 in the valve sleeve.

When the cylinder barrel 22 rotates, cam plate 31 moves each piston 27 on its discharge stroke against the force of the pressure fluid in cylinder bore 26 acting on the piston. For convenience, this force will be termed the hydraulic reaction force and it equals the pressure in cylinder bore 26 multiplied by the area of the piston and divided by the cosine of the angle of inclination of the cam plate. Referring to Fig. 4, the hydraulic reaction forces exerted on the cam plate by two of the pistons are indicated at 73 and 74 and the angle of inclination of the cam plate is designated θ and is measured between the plane of the face of the cam plate and the vertical. These forces act in directions normal to the surface of the cam plate and pass through the centers of spherical piston heads 28. In the following description it is assumed that the horizontal projections of the pistons 27 on a plane extending in a direction normal to the cam plate trunnion axis 36 form a symmetrical pattern about the axis of the drive shaft. Actually this pattern varies continuously between symmetrical and asymmetrical arrangements as the barrel rotates but since for most pumps of this type the frequency of this variation is about 1,000 cycles per second, the assumption is believed to afford a reasonable and practical basis for describing the invention.

As shown in Fig. 4, the forces 73 and 74 act at equal moment arms (A) about the point of intersection 34 of the

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plane of the centers of the spherical piston heads 28 and the shaft axis. If the cam plate trunnion axis were located at this point, the moments due to the hydraulic reaction forces acting on the pistons located above the shaft axis would counterbalance the moments of the forces acting on those pistons located below this axis. However, in accordance with this invention, the trunnion axis is displaced from the point 34 a distance (X) and located at the point 36 and therefore, the moment arm of the force 73 is $(A+X \sin \theta)$ while the moment arm of the force 74 is $(A-X \sin \theta)$. As a result of this inequality, a net counterclockwise moment acts on the cam plate and urges it toward its maximum stroke-producing position. This net moment varies with the sine of the cam plate angle θ and for small angles the variation is substantially linear. It will be apparent that the magnitude of the total hydraulic reaction force moment acting on the cam plate 31 equals the magnitude of this net moment multiplied by the number of pairs of pistons in communication with discharge port 16.

Referring now to Fig. 5, the variation of the hydraulic reaction force moment with cam plate angle is represented by curve 75 and the variation of the moment of spring 38' with cam plate angle is represented by curve 76. Since both of these curves are straight lines their sum, the resultant curve 77, is also a straight line. An inspection of these curves will reveal that the slope of curve 77 is less than that of curve 76 and therefore, in effect, the rate of spring 38' has been reduced without altering its physical size. The curve 78, which is a mirror image of the resultant curve 77, represents the moment which must be exerted by motor piston 42 in order to move the cam plate 31 toward its minimum stroke-producing position against the biasing moment represented by curve 77. The slopes of curves 77 and 78, which are the same, depend on the distance (X) in Fig. 4, and by varying this distance it is possible to make these slopes as small as desired. As a practical matter, the slopes should not be made too flat because tolerance accumulations during assembly may reduce them to zero or even reverse their sense. If this happens, the pump control would become unstable. However, using this method, slopes equivalent to a pressure differential of about 30 p.s.i. become practical at discharge pressures of 3000 p.s.i.

From curve 76 it can be seen that spring 38' exerts a preload moment on the cam plate 31 when the latter is in its maximum stroke-producing position. It should be noted that this preload is not an essential feature of the invention but rather is used to produce faster movement of the cam plate and therefore more rapid response.

In addition to the hydraulic reaction force, each piston also exerts another force on the cam plate which, for convenience, will be termed the inertia force of the piston. This force equals the mass of the piston multiplied by its longitudinal acceleration. Referring to Fig. 1, it will be seen that all of the pistons 27 located above the shaft axis are either moving to the right at the start of their discharge strokes or are moving to the left at the end of their suction strokes. Thus, the pistons are subjected to accelerating forces supplied by cam plate 31 for increasing the velocity of the pistons in the first group and for reducing the velocity of the pistons in the second group. The accelerating forces acting on both groups of pistons act to the right with the result that the cam plate experiences an equal and opposite reaction which tends to tilt it in a counterclockwise direction about trunnion axis 36. The pistons 27 located below the shaft axis are either moving to the left at the start of their suction strokes or are moving to the right at the end of their discharge strokes. Therefore, these pistons are also subjected to accelerating forces but in this case, the forces act to the left and are supplied by nutating plate 32. However, the reaction to these forces is provided by cam plate 31 through the shoes of those pistons 27 located above the shaft axis and consequently, these pistons also urge the

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cam plate in a counterclockwise direction. Those pistons lying on the shaft axis will exert no inertia force on the cam plate because they will be in the midstroke position where their acceleration is zero.

The inertia force moment varies with the sine of the cam plate angle θ and for small angles the variation is linear. Curve 79 of Fig. 6 graphically illustrates this relationship. In order to balance this moment, the spring 37' is designed to afford a moment curve 81 which is a mirror image of the inertia force moment curve 79. The resultant moment curve 82 lies on the zero moment line and illustrates the cancellation of the inertia force moment at all angles of the cam plate.

The magnitude of the inertia force moment varies not only with the angle of inclination of cam plate 31 but also with the rotary speed of cylinder barrel 22. The curve 79, therefore, represents only one of a family of inertia force moment curves in which the slope of each member curve is governed by the rotary speed of the pump. Since the inertia force moment is cancelled by spring 37' and does not affect the location of the cam plate trunnion axis 36 or the slope of control motor moment curve 78, it is only necessary to change the rate of spring 37' when operation at a different speed is desired. It will be apparent that this moment could be used in conjunction with the hydraulic reaction force moment to reduce further the rate of spring 38, but that such use would limit the versatility of the pump because of the difficulty of relocating the cam plate trunnion axis each time the operating speed is changed.

When the pump is operating, discharge fluid will be transmitted to the space 65 of control valve 45 via port 68, radial passages 67 and longitudinal passage 66. This fluid flows through the longitudinal slots 64 formed in land 59 and reacts against the left face of the land 61. When the pump is first started, the right face of this land, as well as motor chamber 44 and spring chamber 70, is subjected to the low pressure prevailing inside the pump housing via passage 57, the recess between lands 62 and 63 and port 54. As discharge pressure increases, the pressure differential across land 61 moves plunger 58 to the right against the bias of spring 71. When the plunger 58 reaches the lap position, shown in Fig. 3, land 62 will interrupt communication between port 54 and passage 57. Further movement of the plunger 58 causes land 61 to uncover port 54 establishing communication between working chamber 44 and valve space 65. Since spring chamber 70 is connected to the port 54 by restricted passage 72, the ensuing pressure increase in working chamber 44 will also occur in the spring chamber 70. When the sum of the fluid pressure and spring forces acting on the right end of plunger 58 is sufficient to overcome the force of the discharge pressure acting on land 61, the plunger 58 will move to its lap position as shown in Fig. 3. As a result of this operation, the pressure established in working chamber 44 will equal the difference between discharge pressure and that pressure necessary to move the plunger 58 to the right from its lap position against the force of spring 71.

The fluid pressure thus established in motor chamber 44 causes piston 42 to exert a clockwise moment on cam plate 31, but until the magnitude of this moment increases above the point 83, in Fig. 5, the cam plate will remain in its maximum stroke-producing position. Should the discharge pressure now increase to a value such that the pressure in working chamber 44 produces a moment 84, as shown in Fig. 5, the cam plate will move in a clockwise direction to a position in which the resultant moment, represented by point 85 on curve 77, is just balanced by the control motor moment 84. At this time, the angle of inclination of the cam plate will correspond to the value represented by point 86. Further increases in discharge pressure will effect similar reductions of the angle of inclination of the cam plate. When the control motor moment reaches the point 87, the maximum resultant

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moment 88 will be balanced and cam plate 31 will be in its minimum stroke-producing position.

If the pump discharge pressure should now decrease, the valve plunger 58 would move to the left connecting working chamber 44 with the pump housing, across land 62. The pressure in chamber 44 would then decrease until the force of the discharge pressure on land 61 could again move plunger 58 to its lap position against the sum of the forces of spring 71 and the fluid pressure in chamber 70. The decrease in pressure in working chamber 44 would reduce the moment exerted by piston 42 and consequently the biasing moment, represented by curve 77 in Fig. 5, would move the cam plate toward its maximum stroke-producing position until the control motor moment 78 was again balanced by the resultant biasing moment 77.

As stated previously, the slopes of curves 77 and 78 are so selected that a pressure change of 30 p.s.i. in motor chamber 44 will move the cam plate 31 from its maximum to its minimum stroke-producing position. It has also been stated that in some cases, it is imperative that the cam plate 31 move rapidly from its minimum stroke-producing position to its maximum stroke-producing position upon a decrease in discharge pressure. In a typical pump designed for a maximum discharge pressure of 3,030 p.s.i., these two conditions are satisfied when the moments represented by points 83 and 87 of Fig. 5 correspond respectively to working chamber pressures of 1,500 and 1,530 p.s.i. This means that the valve spring 71 would first permit communication between valve space 65 and working chamber 44 when the discharge pressure reached 1,500 p.s.i. and that as this pressure rose to 3,000 p.s.i., the pressure in working chamber 44 would progressively increase to 1,500 p.s.i. The next 30 p.s.i. increase in discharge pressure would, under the conditions stated above, cause the motor piston 42 to move the cam plate toward its minimum stroke-producing position in accordance with curve 78. It should now be obvious that although the absolute magnitude of the moment 87 must be large so that a large accelerating moment 88 can be used, this fact does not affect the discharge pressure differential corresponding to the difference between the moment 87 and the moment 83. In other words, the present invention affords a specified degree of sensitivity regardless of the absolute values of the moments acting on the cam plate.

In addition to permitting the use of a spring 38' exerting a large accelerating moment, the high pressure in working chamber 44 also affords another advantage. In the controlling range of the pump, the working chamber pressure continuously varies between 1,500 and 1,530 p.s.i. and, therefore, the pressure differentials between this chamber and space 65 of valve 45 and between this chamber and the interior of the pump housing are substantially equal. This means that when valve plunger 58 moves equal distances to either the right or to the left of its lap position, as shown in Fig. 3, the cam plate will move at equal rates in either the stroke-decreasing or the stroke-increasing directions, respectively, because the fluid flowing into or out of working chamber 44 will always be under a net pressure head of about 1,500 p.s.i. This feature also contributes to the sensitivity of the control mechanism.

As stated previously, the drawings and description relate only to a preferred embodiment of the invention. Since many changes can be made in this embodiment without departing from the inventive idea, the following claims should provide the sole measure of the scope of the invention.

What is claimed is:

1. In a variable displacement pump of the type including a rotary cylinder barrel, a plurality of longitudinally reciprocating pistons mounted in the cylinder barrel and having spherical surfaces at one of their ends, the centers of these surfaces lying in a common plane,

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and a cam plate associated with the pistons at their spherical ends for moving them on their discharge strokes and for governing the length of these strokes, the improvement which comprises means supporting the cam plate for angular adjustment about an axis extending in a direction normal to the axis of rotation of the cylinder barrel and intersecting that axis at a point displaced in the direction of the cylinder barrel from the intersection of the cylinder barrel rotational axis and said common plane, whereby the piston forces produce a moment about the adjustment axis which urges the cam plate toward its maximum stroke-producing position and which increases in magnitude as the cam plate moves toward that position; spring means connected with the cam plate for producing a moment about the adjustment axis which urges the cam plate toward its maximum stroke-producing position, the maximum magnitude of this moment being greater than the maximum magnitude of the piston force moment; and motor means connected with the cam plate for moving it toward its minimum stroke-producing position against the bias of the piston force moment and the moment of the spring means.

2. In a variable displacement pump of the type including a rotary cylinder barrel, a plurality of longitudinally reciprocating pistons mounted in the cylinder barrel and having spherical surfaces at one of their ends, the centers of these surfaces lying in a common plane, and a cam plate associated with the pistons at their spherical ends for moving them on their discharge strokes, the improvement which comprises means supporting the cam plate for angular adjustment about an axis extending in a direction normal to the axis of rotation of the cylinder

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barrel and intersecting that axis at a point longitudinally displaced in the direction of the cylinder barrel from the intersection of the cylinder barrel rotational axis and said common plane, whereby the hydraulic reaction forces and the inertia forces of the pistons produce moments about the adjustment axis which urge the cam plate toward its maximum stroke-producing position and which increase in magnitude as the cam plate moves toward that position; first spring means connected with the cam plate for producing a moment about the adjustment axis urging the cam plate toward its maximum stroke-producing position, the maximum magnitude of this moment being greater than the maximum magnitude of the hydraulic reaction force moment; second spring means connected with the cam plate for producing a moment about the adjustment axis which urges the cam plate toward its minimum stroke-producing position, the magnitude of this moment being equal to the magnitude of the inertia force moment at all positions of the cam plate; and motor means connected with the cam plate for moving it toward its minimum stroke-producing position in opposition to the hydraulic reaction moment and the moment of the first spring means.

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