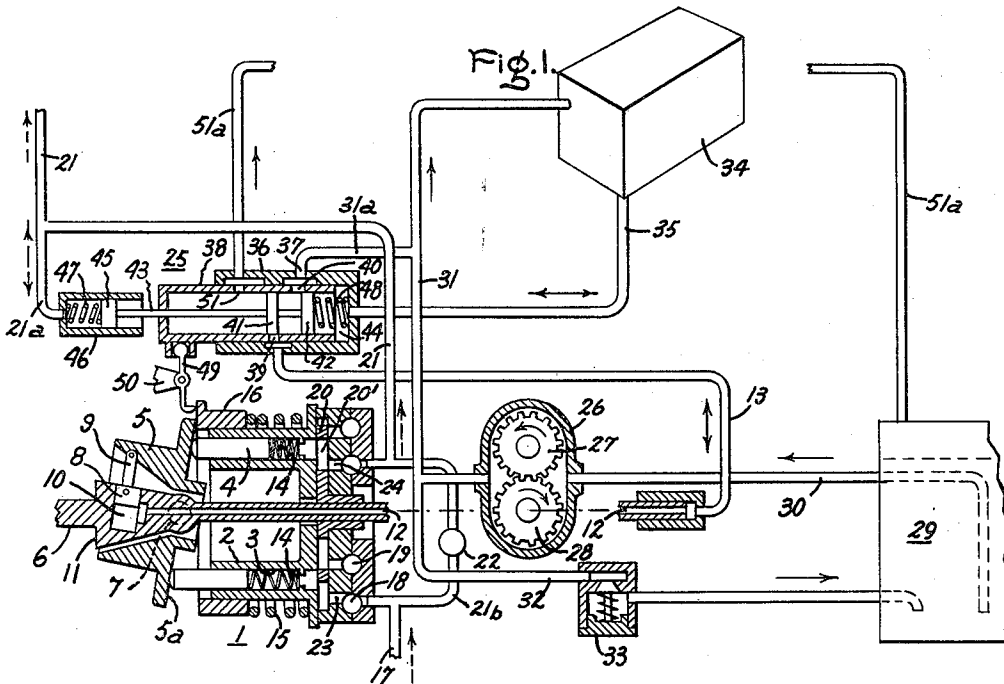
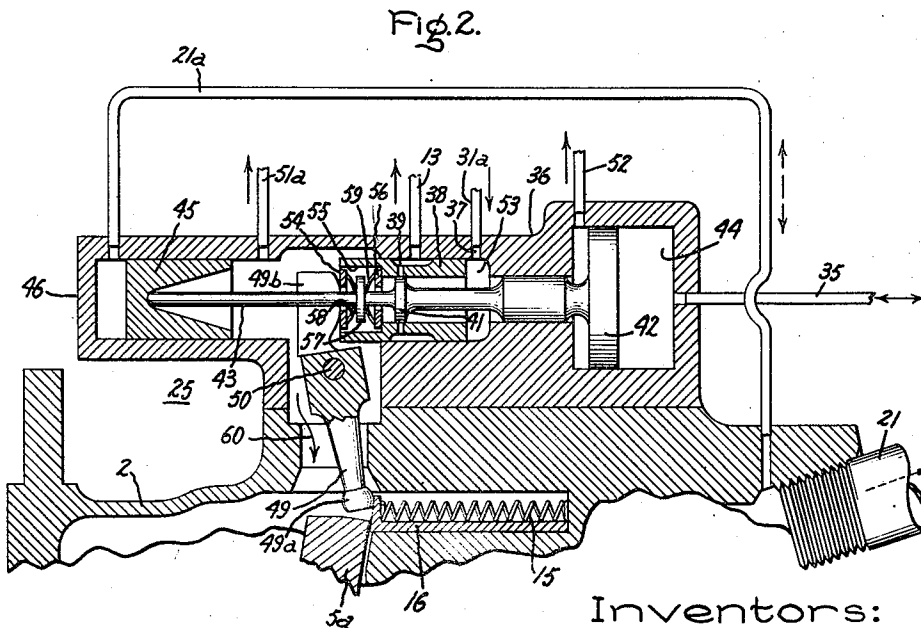


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CONTROL MECHANISM FOR VARIABLE DELIVERY
POSITIVE DISPLACEMENT PUMP
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CONTROL MECHANISM FOR VARIABLE DELIVERY POSITIVE DISPLACEMENT PUMP

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2 Claims. (Cl. 137—620)

This invention relates to multiple piston infinitely variable positive displacement pumps, particularly to means for controlling in a desired manner the displacement and delivery pressure of such a pump. The present case is a continuation-in-part of Serial No. 605,960, filed July 19, 1945, now Patent No. 2,622,393, in the names of Martin A. Edwards, Donald E. Garr, and Hugh M. Ogle.

An object of the invention is to provide improved control mechanism for a pump of the type described in which a fluid pressure control signal is supplied to a control device which automatically regulates the pump discharge pressure.

Another object is to provide a device for automatically controlling the discharge pressure of a variable displacement pump, so as to maintain a predetermined ratio of discharge pressure to a control pressure signal supplied to the device.

Still another object is to provide a pump control mechanism of the type described which may be readily designed to maintain either a constant ratio or a variable ratio between the pressure control signal and the discharge pressure, as the delivery varies.

A further object is to provide pump control means of the type described which will effectively produce stable operation over a wide range of flow rates, discharge pressures, and rotational speed.

Other objects and advantages will be apparent from the following description taken in connection with the accompanying drawing, in which Fig. 1 is a diagrammatic representation of a multiple piston infinitely variable displacement pump having control mechanism in accordance with the invention; and Fig. 2 is a diagrammatic representation of a modification of our control device.

Referring now to Fig. 1, our invention is shown as applied to a pump 1 having a housing 2 defining a plurality of circumferentially spaced cylinders 3 each containing a piston 4 arranged to reciprocate in a direction parallel to the axis of the pump. The outer ends of the pistons are engaged by a wobble member or "swash-plate" 5, which is supported on a rotatable shaft 6 by means of a transverse pivot or trunnion indicated in dotted lines at 7. The angular relation between the member 5 and the shaft 6 is controlled by a servo-motor including a piston 8 having a link 9 pivotally connected to the wobble member 5 and slidably disposed in a pressure chamber 10 formed in an enlarged portion 11 of the shaft 6. Shaft 6 has a longitudinal bore 12 connecting the pressure chamber 10 to a pipe 13. As will be seen more clearly hereinafter, liquid is supplied through conduit 13 to the chamber 10 in order to actuate piston 8 and position member 5 in accordance with the pump displacement required to produce the desired delivery pressure. Suitable means, for instance a coil spring 14 located in each cylinder 3, is provided for biasing the respective pistons 4 into engagement with the thrust face 5a of the wobble member.

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As will be understood by those familiar with pumps of this type, the stroke of the pistons is produced by the angularity of the rotating wobble member 5, the stroke being zero when the thrust surface 5a is normal to the axis of shaft 6 and increasing as the angularity increases. Wobble member 5 is biased to the neutral or zero delivery position by a suitable spring 15 surrounding the pump housing 2 and engaging an axially slidable ring member 16. As will be apparent from Fig. 1, ring 16 engages the outer periphery of the wobble member 5. It will be obvious that the single spring 15 may be replaced by a plurality of small coil springs circumferentially spaced around the housing 2, as in Fig. 2.

The liquid being pumped is supplied from a reservoir (not shown) through conduit 17 to an annular suction channel 18 formed in the housing 2, and a second annular passage 19 communicates with the discharge conduit 21. A by-pass conduit 21b, containing a suitable pressure relief valve 22, may be arranged as in Fig. 1 to prevent the discharge pressure from rising above a preselected value. Associated with each cylinder 3, is a port 23 adapted to establish communication with the inlet passage 18 and a second port 24 providing communication with the discharge passage 19. Interposed between the cylinders 3 and the passages 18, 19, is a valve plate 20 adapted to rotate with shaft 6 and having a plurality of circumferentially spaced openings 20' which establish communication between the cylinders and the passages 18, 19, in a particular sequence as the shaft 6 and wobble member 5 rotate. In Fig. 1, the upper cylinder is shown discharging to passage 19, while the lower cylinder is in communication with the port 23 and inlet passage 18.

Further details of construction of the pump 1 are believed unnecessary, as they do not form a part of our invention. It is necessary only to note that the infinitely variable pump 1 has a fluid pressure servo-motor 8, 10 which is energized to cause the displacement to vary, while a certain member (i. e. the ring 16) is positioned in accordance with the displacement at any moment. It will be obvious that our invention, as described below, may be applied to many different forms of variable delivery positive displacement pump having the two above-mentioned components.

Our invention resides particularly in the device 25 which controls the displacement of the pump 1 by regulating the supply of an operating liquid to the servo-motor 8, 10. A suitable control liquid, for instance a hydraulic oil, is supplied by an auxiliary pump 26, represented in Fig. 1 as being of the well-known gear type having a pair of intermeshing gears 27, 28 one of which may be mounted on the shaft 6 so as to be driven thereby. Auxiliary pump 26 draws liquid from a reservoir 29 through a suction conduit 30 and discharges at a suitable pressure to conduit 31. A by-pass 32 is provided with a pressure relief valve 33 arranged to hold discharge pressure constant at a preselected value, which may be of the order of 250 lb./in.² Control oil at this pressure is supplied from the conduit 31 both to a regulator represented by the box 34, and to the pump control device 25 through branch conduit 31a. Box 34 represents any suitable means for establishing a fluid pressure signal which is proportional to the discharge pressure desired from the pump 1. This regulator may be in accordance with the above-mentioned Edwards-Garr-Ogle application Serial No. 605,960, now Patent No. 2,622,393, or the Ogle-Garr-Edwards application Serial No. 697,058, filed September 14, 1946, now Patent No. 2,645,293.

The device 25 for controlling the stroke of the pistons 4 comprises a casing 36 having an inlet port 37 for receiving high pressure oil from the branch conduit 31a and another port associated with conduit 35 for receiving

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oil from regulator 34 at the variable control pressure. The pressure of the oil in supply conduit 31 is relatively constant, except for slight variations due to the inherent characteristics of pump 26 and valve 33, whereas the control pressure signal in conduit 35 varies in a manner determined by variable throttling means in the regulator 34.

A bushing 38 is slidably disposed in the casing 36 and has axially spaced ports 39 and 40 controlled by valve discs 41 and 42 secured to a rod 43. Port 40 is always in communication with inlet port 37. In the steady state position shown in Fig. 1, disc 41 covers the port 39, thereby interrupting communication between the high pressure port 37 and the conduit 13 leading to the chamber 10 of the servo-motor. The disc or piston 42 and the casing 36 define a control pressure chamber 44 communicating with the conduit 35. The force produced by the fluid pressure in the chamber 44 against the piston 42, is opposed by a pressure-sensing piston 45 slidably disposed in a cylinder 46 and connected to an extension of the valve rod 43. The cylinder 46 is connected to the main discharge conduit 21 of the pump 1 so that piston 45 is responsive to the gage pressure of the liquid in conduit 21. Piston 45 and the valve plunger 41, 42, 43 are biased to a neutral position by two opposed centering springs 47, 48 engaging end faces of the pistons 42, 45.

Bushing 38 is engaged by a follow-up lever 49 which has an intermediate portion held on a fixed fulcrum 50 and an end portion engaging the axially slidable ring 16 of the pump. The pressure in chamber 44 acts on the end surface of bushing 38 so as to bias it to the left and thereby hold lever 49 in engagement with member 16.

The housing 36 has a conduit 51a through which oil drains back to the control oil reservoir 29.

During steady state conditions, the pump discharge pressure sensed by the piston 45 in the cylinder 46 balances the pressure signal in the control chamber 44, and the valve disc 41 covers the port 39 so that no liquid is admitted to conduit 13.

In connection with the following description of the operation, it will be noted that the dotted arrows in Fig. 1 represent the liquid being pumped, while solid arrows represent control liquid. Single-headed arrows represent flow; double-headed arrows represent merely changes in pressure without any material flow.

It will be observed that piston 42 is appreciably larger in diameter than piston 45; therefore a relatively higher pump discharge pressure is required in cylinder 46 to balance the lower control signal pressure in chamber 44. With this arrangement a preselected ratio is maintained between the pressure signal supplied to the device 25 and the discharge pressure produced by the pump 1, the ratio depending on the relation between the areas of the pistons 45 and 42.

The operation of the arrangement is as follows. For any given pump displacement, the pressure in the servo-motor cylinder 10 of the pump 1 remains constant. Upon increase of the control pressure signal in chamber 44, the rod 43 is forced to the left against the pressure in the cylinder 46 and the bias of spring 47, effecting the flow of control oil from the conduit 31a through the ports 37, 40, 39, conduit 13, and the bore 12 of the pump shaft and into the chamber 10, causing movement of the wobble member 5 to increase the piston stroke and accordingly the discharge pressure of the pump. The increased discharge pressure in conduit 21 is communicated to the cylinder 46 where the piston 45 is caused to move to the right, whereupon the valve disc 41 assumes its original aligned position with the port 39, and the supply of operating liquid to the servo-motor cylinder 10 is discontinued.

Thus it is seen that the piston 45 provides "follow-up action" for the control valve member 41, 42, 43, 45. However, the time lag between a change in the pressure

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signal in the chamber 44 and the restoring movement effected by piston 45 may cause hunting and instability of operation. The lever 49 connecting the bushing 38 of the control device 25 with the ring 16 of the pump constitutes direct mechanical follow-up means for reducing the effects of this time lag, thereby preventing hunting.

As pointed out above, the movement of the valve disc 41, to the left causes increased stroke of the pump pistons 4, effected by increasing the angularity of the wobble member 5 whereby the latter forces the ring 16 toward the right against the biasing force of the spring 15. This in turn effects counterclockwise movement of the lever 49 about its fulcrum under the biasing influence of the pressure in chamber 44 acting on the right-hand end of bushing 38, whereby the latter is moved toward the left to effect at least partial restoring of the original relative position between the valve disc 41 and the port 39 in the bushing. In steady state operation bushing 38 is held stationary by the lever 49 at a position determined by member 16.

Conversely, when the pressure signal in chamber 44 decreases, the pump discharge pressure in cylinder 46 forces piston 45 to the right whereupon valve disk 41 uncovers the port 39 so as to effect communication with the drain port 51 and its associated drain conduit 51a. This permits operating liquid to drain from the servo-motor 10 through the bore 12 in shaft 6 and the conduit 13, whereupon the spring 15 causes the ring member 16 to move to the left so as to decrease the angularity of wobble member 5 and the stroke of pistons 4. This reduces the pump discharge pressure in conduit 20, causing piston 45 to move to the left so that valve disk 41 again covers the port 39.

When the pump is inoperative, spring 15 biases ring member 16 to the left so that wobble member 5 assumes the "neutral" position with the thrust surface 5a normal to the axis of shaft 6. The springs 47, 48 are so designed that with no pressure in either chamber 44 or cylinder 46 the control member 41, 42, 43, 45 is positioned with the disk 41 to the left of the port 39. Then when operation begins, there is free communication from conduit 31a through ports 37, 40, 39 and conduits 12, 13 to the servo-motor cylinder 10. It should also be noted that the characteristics of springs 47, 48 may be so selected that the pump characteristics curve can be made to either rise or fall, as desired by the designer, as a function of displacement, depending upon the spring gradients of the springs 47, 48. More specifically, springs 47, 48 introduce biasing forces which are a function of the displacement of ring member 16, i. e. a function of pump displacement. By properly selecting the spring gradients, the ratio of pump discharge pressure to control signal pressure can be made to progressively decrease or increase as pump displacement increases. By making spring 47 less stiff relative to spring 48, the ratio of discharge pressure to control pressure will increase, as displacement increases. Likewise, by making spring 47 stiffer, the pressure ratio will decrease as displacement increases.

Fig. 2 represents diagrammatically a modification of the pump control device 25 of Fig. 1. In this figure, parts performing the same function are given the same reference numbers as the corresponding parts in Fig. 1. A number of differences in the structure will be apparent. For instance, the piston 42 does not slide in the bushing 38, as in Fig. 1, but is arranged in a separate cylinder defining the chamber 44 and having a drain pipe 52 to prevent liquid trapped at the left-hand side of piston 42 from interfering with movement thereof. Also, the discharge pressure sensing piston 45 is formed separately from rod 43 and has a spherical socket receiving the rounded end of spindle 43. Operating liquid under pressure is supplied through conduit 31a to the chamber 53, where it exerts a biasing force to the left on the

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exposed end surface of bushing 38. This biasing force serves to maintain bushing 38 in contact with the upper end of the follow-up lever 49, as described below.

The major difference in the arrangement of Fig. 2 lies in the connection between follow-up lever 49, bushing 38, and valve rod 43. As will be seen from the drawing, the upper end of lever 49 is bifurcated, having portions 49b straddling the rod 43. Each of the portions 49b is of a transverse thickness sufficient to engage both the end of the bushing 38 and also a floating washer 54 disposed in an enlarged end portion 55 of the bore of bushing 38. Washer 54 has a central aperture through which the rod 43 freely passes. A similar washer 56 is disposed around rod 43 and in engagement with the annular seat forming the end of the bore portion 55. Fixed to the valve rod 43 is a flange 57 disposed intermediate the washers 54, 56. A comparatively stiff cup-shaped spring washer 58 is arranged between the washer 54 and the flange 57; and a similar spring washer 59 is interposed between flange 57 and washer 56. This spring assembly replaces, and serves somewhat the same function as the springs 47, 48 of Fig. 1. An important difference is that the pump discharge pressure ratio is not affected by the position of the wobble plate. With this arrangement the ratio of the pump discharge pressure to the control pressure is constant over the complete flow range of the pump. The gradients of the cup springs are so chosen as to give the pump transient pressure regulation under transient changes of displacement. Thus if the forces on pistons 42 and 45 are not equal, the difference between these forces will act on springs 58, 59. The spring gradients are selected so that the displacement of valve disk 41 with respect to port 39 is proportional to the difference in the net force on spindle 43.

The lower end of lever 49 is provided with a ball portion 49a contacting the flange of ring 16.

The operation of the arrangement of Fig. 2, which represents the steady state condition with maximum displacement, is as follows. In the inoperative condition, the ring member 16 will be in its extreme left-hand position, corresponding to zero angularity of the wobble member 5, with the follow-up lever 49 in its extreme clockwise position. The spindle 43, bushing 38, and associated parts will be in their extreme right-hand position. The spring washers 58, 59 are quite stiff, and are so designed that the "free length" of the spring assembly, is somewhat greater than in the condition shown in Fig. 2, which means that sleeve 38 is displaced to the right, relative to the spindle 43, so that port 39 is in free communication with port 37 and conduit 13. If now the pump shaft is caused to rotate, the normal pressure of the operating liquid supplied by the auxiliary pump 26 through conduit 31a to the chamber 53 biases the bushing 38 to the left to the position shown in the drawing, in which valve disk 41 is in aligned position with the port 39, the left-hand end of bushing 38 engages the fork 49b, and springs 58, 59 are both somewhat compressed, being equally stressed.

If now, a control signal pressure is supplied by the regulator 34 through conduit 35 to chamber 44, the resulting force on the pressure sensing piston 42 will cause spindle 43 to move to the left, during which movement any liquid trapped at the opposite side of piston 42 will be expelled through drain conduit 52. The displacement of valve disk 41 to the left causes the port 39 to be uncovered so that operating liquid is supplied from chamber 53 through port 39 to the conduit 13, and thence through the bore 12 in the pump shaft 6 to the servo-motor chamber 10. During this movement, sleeve 38 is held stationary by arm 49. The pressure now built up in the servo-motor causes the wobble member 5 to tilt on its trunnions 7 so as to produce effective displacement of the pump pistons 4.

It will be observed that this deflection to the left of

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the flange 57 causes the cup-shaped spring 58 to be further compressed while the other spring 59 is somewhat relieved. Thus, the combined effect of the springs 58, 59 is to introduce a net spring force tending to move spindle 43 back to the right relative to bushing 38. This new force is proportional to the relative displacement between spindle 43 and bushing 38 and tends to restore disk 41 to its aligned position with port 39 and thereby reduces the tendency to "over-shoot" and "hunt."

Another result is that the degree to which port 39 is opened by disk 41 is proportional to the net force acting on spindle 43. That is, with a large sudden increase in control pressure in chamber 44, spindle 43 will be moved a considerable distance to the left, against the bias of spring 58, causing port 39 to be opened wide. On the other hand, a smaller increase in control pressure causes port 39 to be only partly uncovered. Since the rate of response of servo-motor 8, 10 depends on the rate of supply of operating liquid thereto, it will be apparent that the pump discharge pressure will be increased more rapidly with large increases in control pressure, and more slowly with smaller increases in control pressure.

As described in connection with Fig. 1, the increased discharge pressure, acting on piston 45, causes the rod 43 and bushing 38 to resume their steady state relation illustrated. During this movement the bushing is held stationary, because biased solidly against fork 49b by the pressure in chamber 53.

When the control pressure in chamber 44 decreases, the pump discharge pressure acting on piston 45 causes the spindle 43 to move to the right with the result that valve disk 41 uncovers port 39 so that liquid is drained from the servo-motor cylinder 10 through conduits 12, 13, port 39, to the left of disk 41 and out through the generous clearances defined between the members 54, 56, 58, 59 and the spindle 43 and bore 55. This drainage liquid may be removed through the conduit 51a, in the event the pump is mounted with the control device 25 lowermost, or it may be permitted to drain into the pump housing 2 as indicated by the arrow 60, if the control device is mounted above the pump. It will be observed that movement of the valve spindle 43 to the right compresses the spring washer 59 and relieves the spring 58, so that a net spring force is introduced tending to move bushing 38 to the right and resist the movement of rod 43 to the right. Here again the degree of opening of port 39, and therefore the rate of response of the servo-motor 8, 10 is proportional to the magnitude of the change in control pressure.

Thus, it will be seen that in normal operation the springs 58, 59 always tend to resiliently resist and displacement of disk 41 in either direction relative to the bushing 38 and tend to keep disk 41 aligned with port 39. This spring restoring arrangement has been found to provide extremely stable operation of the pump, even with very sudden changes in pump displacement, over a wide range of delivery flow rates. As noted above, the arrangement of Fig. 2 provides a constant steady state ratio between pump discharge pressure and control pressure over the whole range of pump displacement.

Our invention provides a comparatively simple, yet extremely effective, arrangement for controlling the displacement of an infinitely variable pump of the type described. It has been found particularly effective for use with a fuel pump supplying oil to spray nozzles in the combustion system of a gas turbine powerplant.

What we claim as new and desire to secure by Letters Patent of the United States, is:

1. In a hydraulic relay valve, the combination of a housing defining a cylindrical chamber containing an axially slidable bushing and a spindle disposed in the bushing and axially slidable relative thereto, said spindle having portions cooperating with said bushing and housing to define axially spaced pressure liquid inlet and outlet chambers, the spindle and bushing having cooperating

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portions adapted to meter the flow of liquid between said inlet and outlet chambers, and means for positioning said bushing relative to the spindle including first abutment means adapted to engage the bushing directly, second abutment means on the spindle, and spring means interposed between said first abutment and second abutment and between the second abutment and the bushing, whereby displacement of the spindle relative to the bushing in either direction is resiliently resisted with a force proportional to said displacement.

2. In a hydraulic relay valve, the combination of a housing defining a cylindrical chamber containing an axially slidable bushing, a spindle disposed in the bushing and axially slidable relative thereto, the spindle having portions cooperating with the bushing and housing to define pressure liquid inlet and outlet chambers including a first chamber disposed adjacent a first end portion of the bushing whereby fluid pressure therein biases the bushing axially, the spindle and bushing having cooperating portions for metering the flow of liquid between said inlet and outlet chambers, and means for positioning the bushing relative to the spindle including first

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abutment means adapted to engage a second end portion of the bushing remote from said first end portion, second abutment means on the spindle, first spring means interposed between said first and second abutments, and second spring means interposed between said second abutment and a seat portion formed in the bushing, the free length of said spring members being such that they are compressed when the fluid pressure in said first chamber biases the bushing into contact with the first abutment member, whereby displacement of the spindle relative to the bushing in either direction is resiliently resisted.

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