

[54] **CASE PRESSURIZATION CONTROL
FOR A POSITIVE DISPLACEMENT
DEVICE DRIVEN HYDRAULICALLY BY
A FOUR-WAY CONTROL VALVE**

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[51] Int. Cl.F01c 1/18, F01c 21/16

[58] Field of Search.....91/420, 442, 451, 452, 186;
418/196, 189, 71, 74; 92/82, 86; 137/117; 123/186

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[57] **ABSTRACT**

The case pressure in a chamber of a positive displacement hydraulically driven device having a pair of control ports operatively associated with a four-way control valve, is regulated always to be just below the lower pressure in the control ports by a constant amount so that backlash between the mechanically coupled movable members of the device is unidirectionally preloaded out at all times and breakout friction and internal leakage are reduced.

9 Claims, 13 Drawing Figures

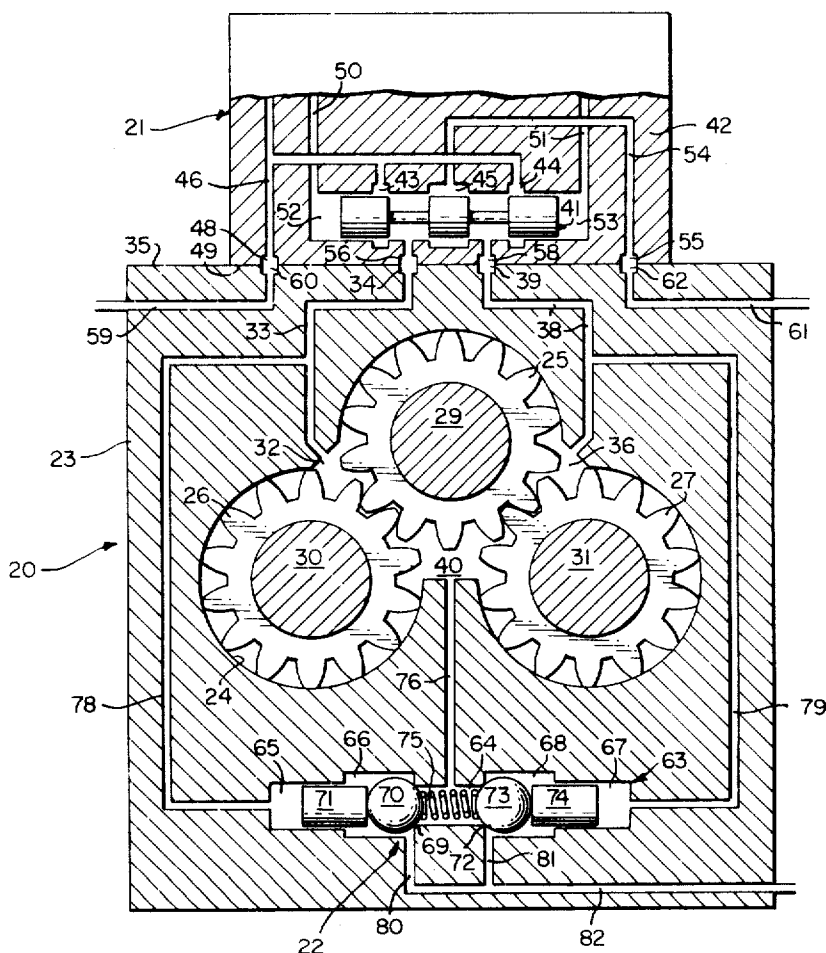


Fig. 1.

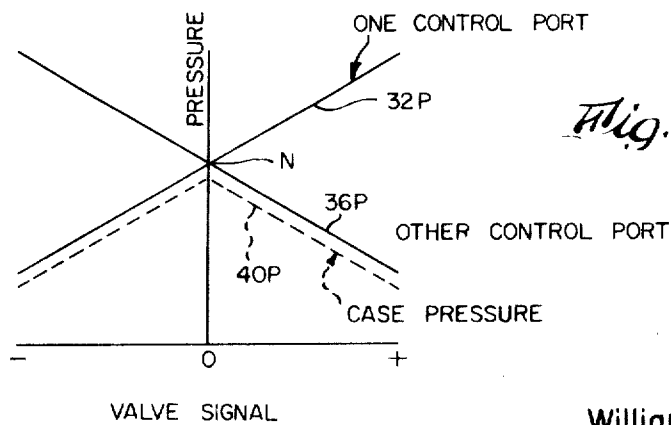
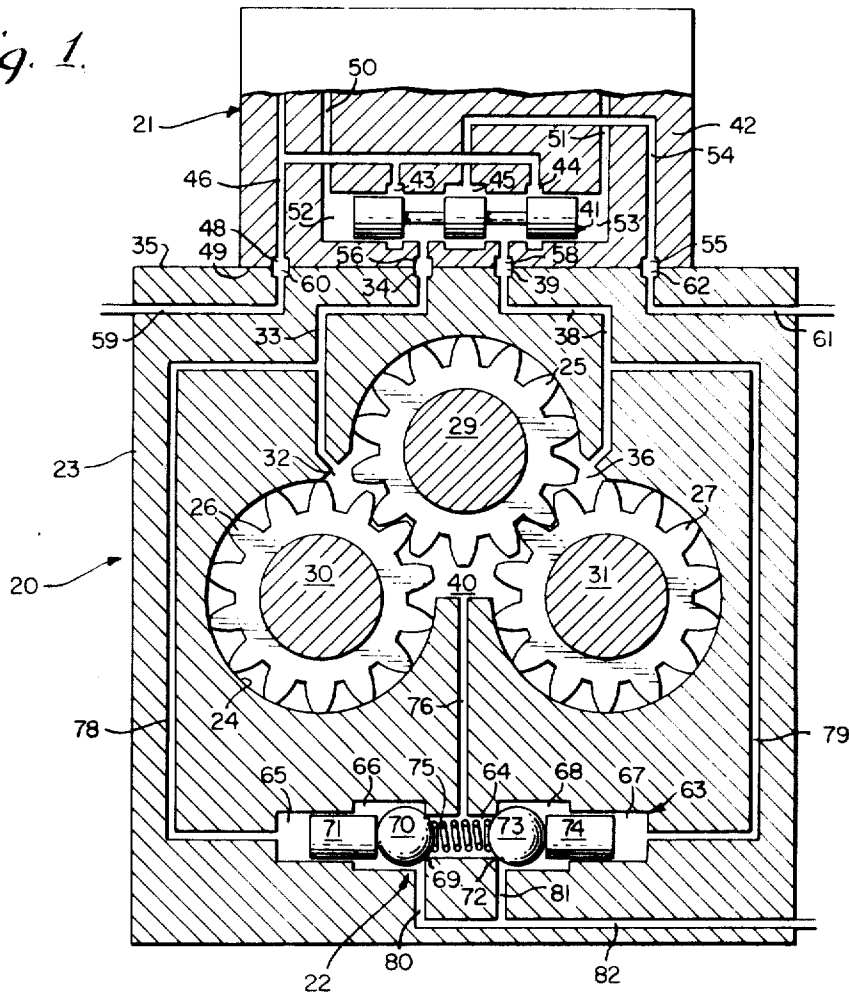


Fig. 2.

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Fig. 4.

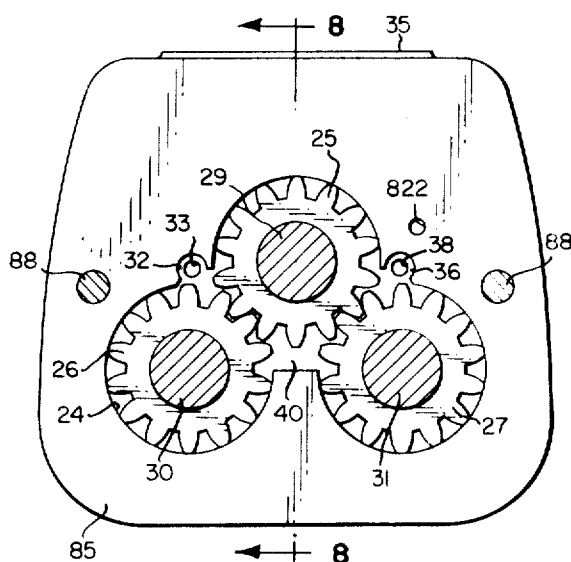
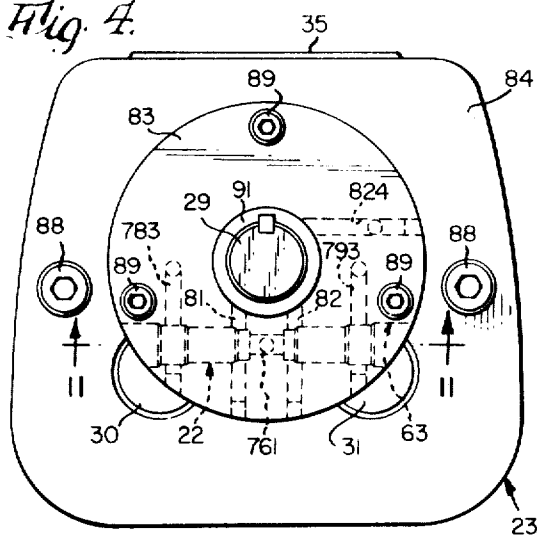


Fig. 5.

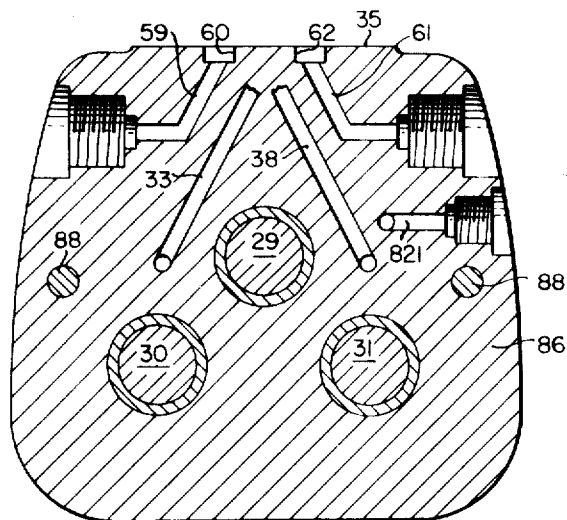


Fig. 6.

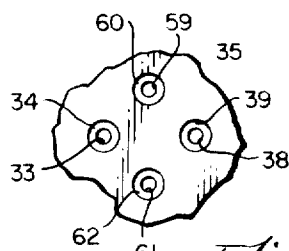


Fig. 7.

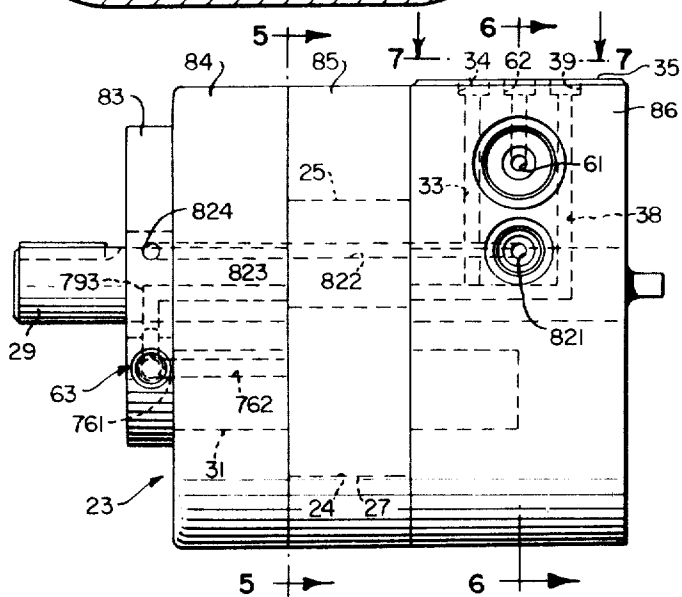


Fig. 3.

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Fig. 8.

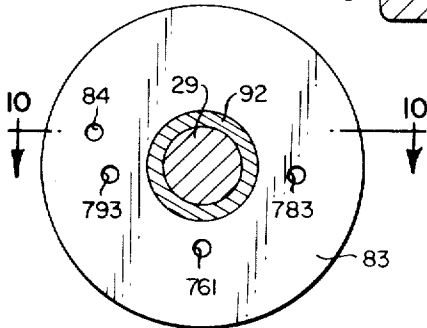
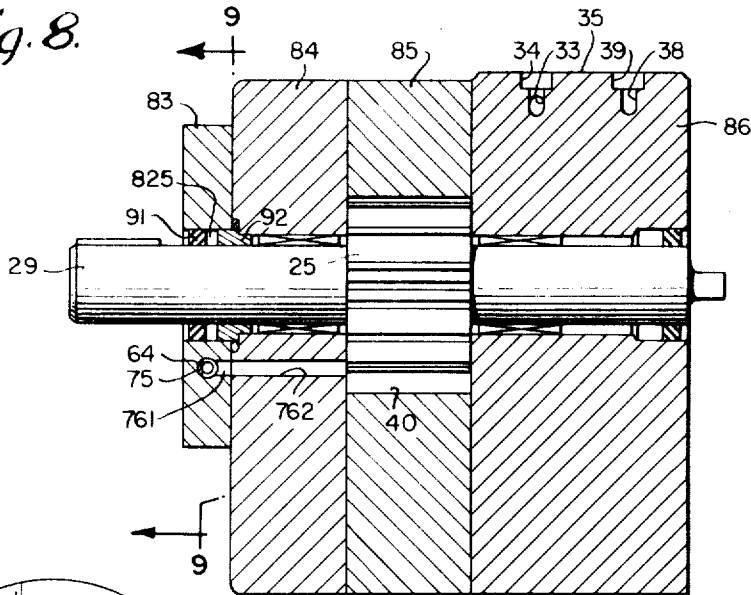


Fig. 9.

Fig. 11.

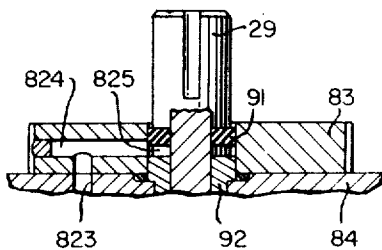
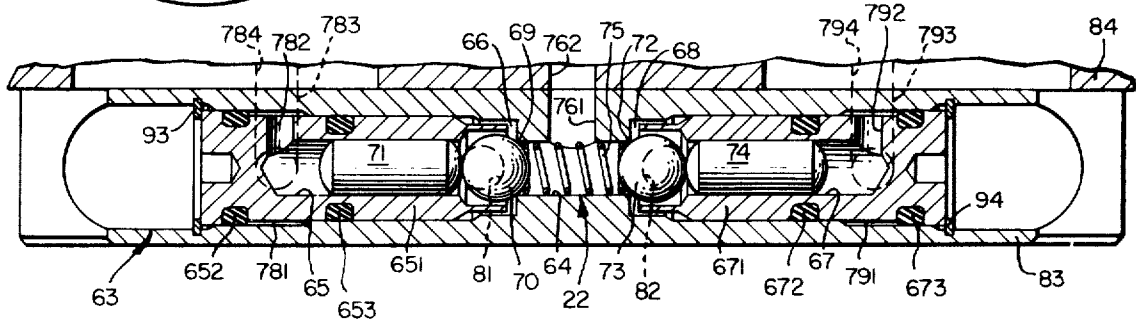


Fig. 10.

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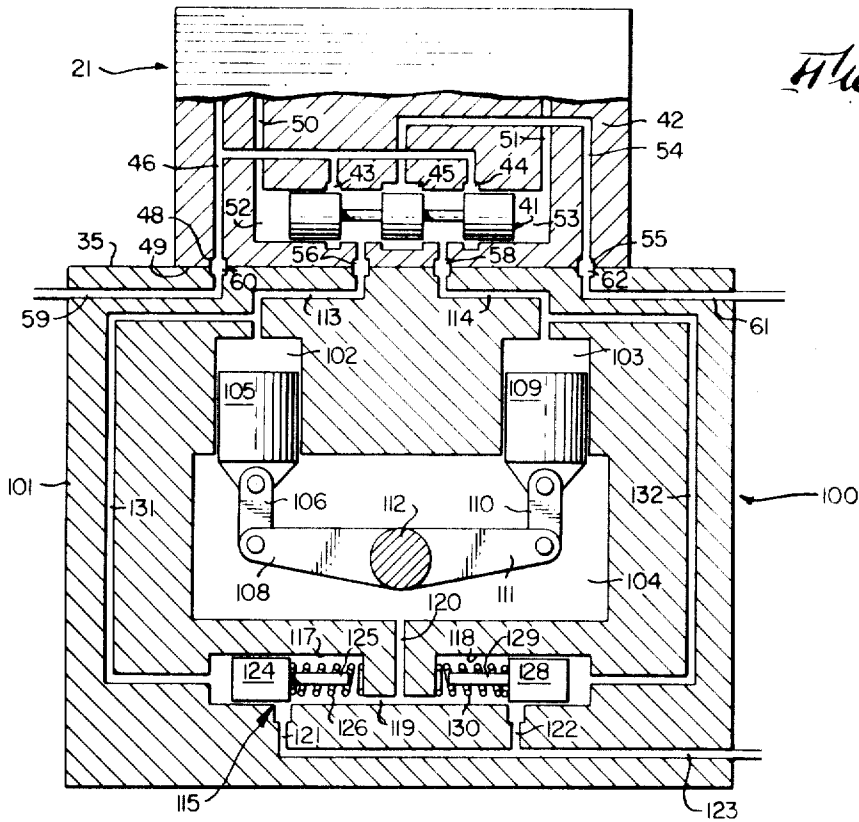


Fig. 12.

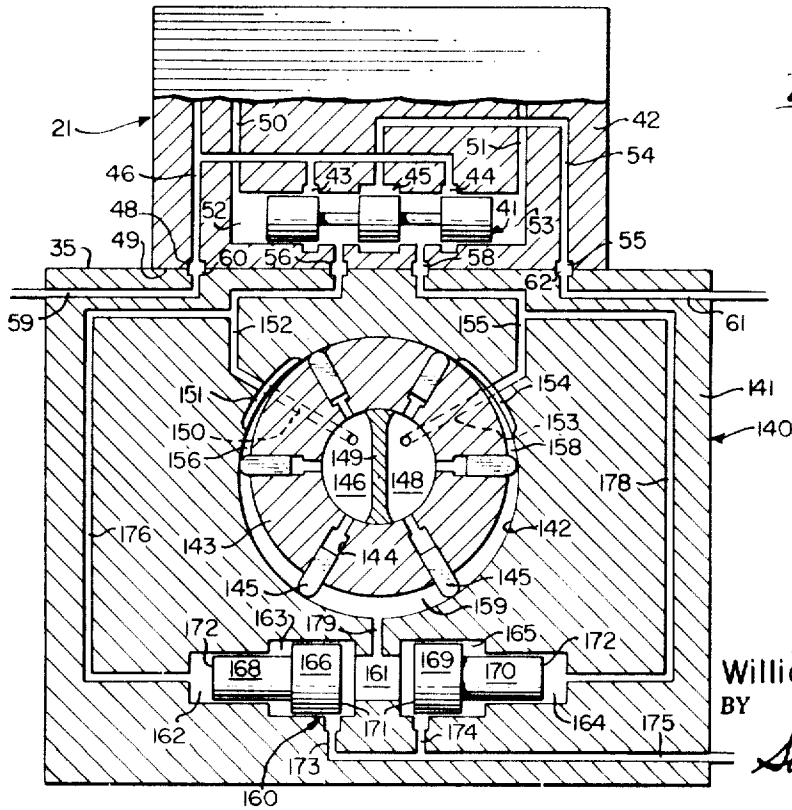


Fig. 13.

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CASE PRESSURIZATION CONTROL FOR A POSITIVE DISPLACEMENT DEVICE DRIVEN HYDRAULICALLY BY A FOUR-WAY CONTROL VALVE

BACKGROUND OF THE INVENTION

The advantages of maintaining high-case pressurization in a positive displacement hydraulically driven device such as a servomotor controlled by a four-way control valve such as an electrohydraulic servovalve so as to reduce bearing loads and leakage are known.

The problem is unique to control by a four-way control valve such as a servo valve which operates typically on a 3,000 p.s.i. fluid supply system. At null the servo valve's two control ports severally are subjected to the still high pressure of, say, 1,500 p.s.i. When the device is to be driven, the servo valve raises the pressure at one control port and lowers it an equal amount at the other. Static friction loads must be overcome before the pressure reaction members of the driven device can move. Also such high pressures increase fluid leakage through clearances inevitably present between relatively movable parts. These undesirable conditions of high-breakout friction and high leakage are caused not by the differential pressure between the control ports but by the differential between the pressure in a control port on one side of the pressure reaction members and the case pressure in a chamber on the other side of such members.

One known system of reducing bearing loads and leakage in such a control mechanism is to provide means for maintaining the case pressure at one-half the sum of the two control pressures. This arrangement is effective to unidirectionally preload out backlash between mechanically coupled movable members in the driven device but only so long as the direction of drive of such device does not reverse or the pressure in one control port does not drop below case pressure. If either of these conditions occurs, i.e., reversal of drive direction or polarity change in the control port to case chamber differential pressure, the mechanical backlash evidences itself. Thus, with a driven device that is bidirectional in operation, i.e., one that starts, stops, backs up, etc., the problem of backlash is pronounced. For example, a hydraulic motor operates in this bidirectional fashion when used in a servo positioning system, such as to position the work table of a milling machine through rotation of a lead screw.

SUMMARY OF THE INVENTION

The present invention provides a control mechanism in which case pressurization is controlled not only to have the advantages of reducing friction and leakage but also overcomes the disadvantages mentioned above for the prior art system which operates on the principle of maintaining case pressure at one-half the sum of the two control pressures at the control ports of the control valve.

In accordance with the present inventive concept the mechanical backlash is unidirectionally preloaded out even with bidirectional drive of the driven device. This is an outstanding advantage, achieved by always maintaining the case pressure at a level below the lower of the control pressures, and renders the control suitable for many practical applications for which the aforementioned prior art system is unsatisfactory.

Another salient advantage of the present invention is that the case pressure can be maintained at a level below the lower of the control pressures by a predetermined, i.e., a constant, amount at all times, and preferably at an optimum lower level thereby to reduce friction and leakage to the minimums consistent with the differential control pressures required to drive a load on the output member of the driven device while still ensuring finite unidirectional preloading of the mechanical backlash.

A further advantage of the present invention is that the inherent leakage causing fluid flow toward the case chamber is utilized as the source of pressurized fluid for such chamber in which the pressure is controlled by means which valves the

flow toward a drain. In this manner separate connections to supply pressure are avoided, these being required by the aforementioned prior art system, and therefore the present invention results in a simpler construction.

Other advantages of the present invention will be apparent from the following detailed description of preferred embodiments illustrated in the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a partially sectional and schematic view vertically of a control mechanism embodying a preferred form of the present invention and showing a four-way control valve operatively associated with a positive displacement hydraulically driven device of the three-gear servomotor type having a case chamber in which the pressure is controlled by one form of means which valves leakage flow collected in such case chamber toward a drain.

FIG. 2 is a diagram plotting pressure against valve signal and graphically showing the relation between pressure in the case chamber and the pressures in the control ports, for the control mechanism depicted in FIG. 1.

FIG. 3 is a side elevational view of commercial apparatus embodying the three-gear servomotor and case pressurization control means shown in FIG. 1, the four-way control valve being omitted.

FIG. 4 is a front elevational view of the apparatus shown in FIG. 3, viewed from the left thereof.

FIG. 5 is a vertical transverse sectional view thereof taken on line 5—5 of FIG. 3.

FIG. 6 is another vertical transverse sectional view thereof taken on line 6—6 of FIG. 3.

FIG. 7 is a fragmentary top plan view thereof taken on line 7—7 of FIG. 3.

FIG. 8 is a vertical longitudinal sectional view thereof, on a slightly enlarged scale, taken on line 8—8 of FIG. 5.

FIG. 9 is a vertical transverse sectional view thereof taken on line 9—9 of FIG. 8.

FIG. 10 is a fragmentary horizontal sectional view thereof taken on line 10—10 of FIG. 9.

FIG. 11 is a fragmentary horizontal sectional view thereof, on a greatly enlarged scale, taken on line 11—11 of FIG. 4.

FIG. 12 is a schematic view similar to FIG. 1 but showing a control mechanism embodying another form of the present invention, specifically illustrating a push-pull actuator as a different type of positive displacement hydraulically driven device and also specifically illustrating a different form of case pressurization control means.

FIG. 13 is another schematic view similar to FIG. 1 but showing a control mechanism embodying still another form of the present invention, specifically illustrating a vane motor as a different type of positive displacement hydraulically driven device and also specifically illustrating yet another form of case pressurization control means.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIGURES 1-11

The control mechanism schematically shown in FIG. 1 comprises three subassemblies including a positive displacement hydraulically driven device 20, specifically illustrated as being of the three-gear servomotor type; a four-way control valve 21; and case pressurization control means indicated generally at 22.

The three-gear servomotor 20 is of the general type illustrated and described in greater detail in U.S. Pat. No. 3,447,422, there described as a zero backlash fluid motor, to which patent reference is made for a complete explanation of its construction and operation. Suffice it to say here, the servomotor 20 comprises a body or case 23 formed internally with a compartment 24 in which are arranged a driven gear 25 and two drive gears 26 and 27 meshing therewith. Driven gear 25 is suitably fast to a shaft 29 which extends outwardly from

the case and constitutes a movable output member. The drive gears 26 and 27 are suitably fast to shafts 30 and 31, respectively. All shafts 29-31 are suitably journaled in antifriction bearings supported by case 23.

A portion of compartment 24 providing a space 32 between left drive gear 26 and driven gear 25 constitutes a first control chamber serviced by a port passage 33. The upper end of this passage leads to a port 34 formed as a recess in an upper flat surface 35 on case 23. A similar space portion 36 of compartment 24 between right drive gear 27 and driven gear 25 constitutes a second control chamber serviced by a port passage 38 which leads to a port 39 provided as a recess in surface 35. Still another portion 40 of compartment 24 immediately below driven gear 25 and between drive gears 26 and 27 constitutes a sump or case chamber.

As fully explained in said U.S. Pat. No. 3,447,422, so long as the pressure in sump chamber 40 is below the pressure in either control chamber 32 or 36, the pressure differential between chambers 32 and 40 urges left drive gear 26 to rotate in a counterclockwise direction, and the pressure differential between chambers 36 and 40 urges right drive gear 27 to rotate in a clockwise direction, as viewed in FIG. 1. The result of these drive urgings is to load out unidirectionally backlash or dimensional clearances that are present between the interengaging teeth among the gears 25-27.

Thus, left drive gear 26 constitutes a first pressure reaction means operatively arranged in case 23 between left control chamber 32 and sump chamber 40 and is coupled through mechanical backlash to driven gear 25 for driving the output shaft 29 in one direction. Similarly, the right drive gear 27 constitutes a second pressure reaction means operatively arranged in case 23 between right control chamber 36 and sump chamber 40 and is coupled through mechanical backlash to driven gear 25 for driving the output shaft 29 in the opposite direction. The effective direction of rotation of driven gear 25 and hence output shaft 29 fast thereto is determined by the drive dominance thereon of one of the drive gears 26 and 27 over the other, and this output shaft is capable of changing its speed and even its direction of rotation without changing the respective directions of bias of these drive gears.

Since a pressure differential exists between control chambers 32 and 36 and sump chamber 40, there is a frictional load on the bearings and peripheries for the various gears 25-27. Collectively all of these frictional loads on the gears constitute preload forces producing a breakout friction for movement of output shaft 29.

The necessary clearances between the various gears 25-27 and the opposing flat side and curved peripheral walls of the case compartment 24 provide paths for leakage flow of fluid from the higher pressured control chambers 32 and 36 toward the lower pressured sump chamber 40.

It is evident that both the breakout friction and the leakage flow are proportional to the pressure differentials between the control chambers 32 and 36 severally and the sump chamber 40.

The four-way control valve 21 is operatively associated with the three-gear servomotor 20 to control pressures differentially at ports 34 and 39. This control valve may be of any suitable construction, such as disclosed in any one of U.S. Pat. Nos. 2,767,689, 3,023,782 and 3,228,423. As shown in FIG. 1, control valve 21 is representative of an electrohydraulic servo valve and is illustrated as having an output stage valve spool 41 slidably arranged in a body 42 and operatively associated with a pair of metering pressure ports 43 and 44 and a metering return port 45. Ports 43 and 44 are shown manifolded together and connected to a passage 46 which leads to a supply pressure port 48 in the lower flat surface 49 of valve body 42. Surfaces 35 and 49 engage. This passage 46 is also shown as leading upwardly to another part of the control valve, suggestive of the supply for a first stage fluid amplifier (not shown) which in response to electrical command input signals produces a pressure differential applied via passages 50 and 51 to the opposite end chambers 52 and 53

for valve spool 41. Metering return port 45 is shown as serviced by a passage 54 leading to a return port 55 provided as a recess in lower surface 49 of valve body 42.

Intermediate metering ports 43 and 45 is a left control port passage leading to a left valve control port 56 provided as a recess in surface 49 of the valve body. Between metering ports 44 and 45 a right control port passage leads to a right control port 58 provided as another recess in lower surface 49 of the valve body. Left valve control port 56 sealingly communicates with left servomotor control port 34, while right valve control port 58 sealingly communicates with right servomotor control port 39.

A passage 59 is shown as arranged in and leading from the left side of the case 23 of the servomotor for supplying pressurized fluid from a suitable source (not shown) to a port 60 provided as a recess in upper surface 35 of this case and sealingly communicating with port 48 in valve body 42. Another passage 61 is shown provided in the servomotor case 23 leading from the right side thereof and terminating at its opposite end in a return port 62 in upper case surface 35 and sealingly communicates with return port 55 in valve body 42.

From the foregoing, it will be seen that in response to suitable command input signals to four-way control valve 21 valve spool 41 can be caused to move so as to control the flow of fluid through port passages 33 and 38 leading to control chambers 32 and 36 in servomotor 20.

A feature of the present invention is to provide a mechanism for controlling case pressurization in sump chamber 40 in such a way that the aforementioned breakout friction and leakage flow are reduced even though the driven gear 25 may be driven bidirectionally, i.e., it may start, stop, reverse, accelerate or decelerate, and while maintaining the unidirectional preloading of the backlash associated with each of the drive gears 26 and 27. The case pressurization control mechanism 22 performs this function and keeps the case pressure in sump chamber 40 at a level below the lower of the control pressures in control chambers 32 and 36 by a predetermined, i.e., a constant, amount at all times.

Case pressurization control mechanism 22 will now be described. Case 23 is shown as provided with a horizontal cylindrical chamber indicated generally at 63 including a central portion 64, a left end portion 65, a left intermediate portion 66 of enlarged diameter between portions 64 and 65, a right end portion 67, and a right intermediate portion 68 of enlarged diameter between portions 64 and 67. The step between the bores of portions 64 and 66 provides an annular left seat 69 on which a left ball valve 70 can be seated. A left plunger 71 slidable in the bore of left end portion 65 is arranged on the side of such ball opposite from left seat 69. A right annular seat 72 is provided by the shoulder between the different diameters of the central portion 64 and right intermediate portion 68. A right ball valve 73 is arranged to engage this seat and is backed up by a right plunger 74 slidably arranged in right end portion 67.

A helical compression spring 75 is shown as arranged in central chamber portion 64 and at opposite ends bears against the balls 70 and 73. A vent passage 76 is shown as provided in case 23 to establish communication between sump chamber 40 and central chamber portion 64. The left end of left end chamber portion 65 communicates with port passage 33 via a conduit 78 provided in case 23. A similar conduit 79 in the case establishes communication between the right end of right end chamber portion 67 and right port passage 38.

Left intermediate chamber portion 66 has an outlet passage 80 arranged on the downstream or left side of left seat 69. A similar outlet passage 81 is arranged on the downstream or right side of right seat 72 and communicates with right intermediate portion 68. These outlet passages 80 and 81 are manifolded to a return conduit 82 shown provided in case 23 and leading to the right side thereof.

The cross-sectional area of left plunger 71 corresponds to the cross-sectional area of central chamber portion 64 at left seat 69. Likewise, the cross-sectional area of right plunger 74

corresponds to that of the central chamber portion 64 at right seat 72.

The left assembly comprising ball valve 70 and plunger 71 moves horizontally in response to a rightwardly directed force applied by the pressure in left control port chamber 32 acting against the outer or left end of plunger 71 and to a leftwardly directed force representing the sum of the force exerted by spring 75 and the force applied by the case pressure in sump chamber 40 acting against that portion of ball valve 70 which is exposed in central chamber portion 64 when this ball valve engages left seat 69. The right assembly comprising ball valve 73 and plunger 74 moves horizontally in response to a leftwardly directed force applied by the pressure in right control port chamber 36 acting against the outer or right end of plunger 74, and to a rightwardly directed force representing the sum of the force exerted by spring 75 and the force applied by the case pressure in sump chamber 40 acting against that portion of ball valve 73 which is exposed in central chamber portion 64 when this ball valve engages right seat 72. If one control port pressure dominates over the sum of the spring force and the force due to case pressure, the corresponding ball valve remains seated. On the other hand, if the sum of the spring force and the force due to case pressure dominates over the control port pressure, the corresponding ball valve will lift off its seat to vent some of the fluid from the sump chamber and thereby lower the case pressure until the ball valve can reseat under the influence of new force balance acting thereon. The vented fluid flows from sump chamber 40 through vent passage 76, through one end of central chamber portion 64, passed the now uncovered valve seat 69 or 72 into the corresponding chamber portion 66 or 68, thence out the corresponding outlet passage 80 or 81 into return conduit 82. This return conduit 82 is preferably separate from return conduit 61 for control valve 21 to insure that the servomotor bearings and seals will never be exposed to pressure surges that may occur in servovalve return conduit 61 due to transient flow. Inasmuch as the effective areas of seats 69 and 72 correspond to the effective outer end areas of plungers 71 and 74, spring 75 acts as a bias force means which assures that the lower control pressure acting on the outer end of one plunger will always remain higher than the case pressure acting against the corresponding ball valve on the upstream side of the seat for such ball valve.

Preferably spring 75 is selected so that it corresponds to a pressure force against the inner exposed face of the ball valve upstream of its seat of from 50 to 100 p.s.i. Thus the pressure in sump chamber 40 will always be maintained at a level of from 50 to 100 p.s.i. below the pressure level in the lower of the two control chambers 32 and 36.

The effect of the case pressurization control mechanism illustrated schematically in FIG. 1 is graphically illustrated in the diagram of FIG. 2 in which pressure as an ordinate is plotted against valve signal as the abscissa. The pressure for left control port 32 is represented by the inclined line 32P and that for the other control port 36 is represented by the inclined line 36P. These two lines intersect on the zero signal line at some null pressure indicated at intersection point N. When the command signal has one polarity, represented as positive, the pressure in control port 32 rises with increased positive signal while that in the other control port 36 decreases. The reverse is true when the pressure relationship in ports 32 and 36 reverses due to reversal of the polarity of the command signal. The case pressure in sump chamber 40 is represented by the broken line 40P in FIG. 2. It will be seen that this case pressure 40P always remains below the lower of the control port pressures 32P and 36P.

While maintaining case pressure level in sump chamber 40 below the lower of the control pressures in control chambers 32 and 36 by a predetermined, i.e., a constant, amount at all times reduces friction and leakage while maintaining the unidirectional preloading of the backlash associated with each drive gear 26 or 27 and driven gear 25, it is preferred to reduce the breakout friction and leakage to the minimums

consistent with the differential control pressures required to drive a load on output shaft 29 while still ensuring finite unidirectional preloading of the backlash between the drive gears and the driven gear. This is achieved typically by selecting the bias force spring so as to produce the effect of 50 to 100 p.s.i. in the case pressure in sump chamber 40.

The operation of the case pressurization control mechanism 22 will now be illustrated by a consideration of specific pressure values. Accordingly, assume that with no command signal input into control valve 20 the null pressure in each control chamber 32 and 36 is 1,500 p.s.i. Assume further that the bias spring 75 is selected so as to exert the equivalent of 100 p.s.i. This will produce a case pressure in the sump chamber 40 of 1,400 p.s.i.

Assume now that a command signal is put into control valve 21 such that the pressure in left control chamber 32 rises to 1,800 p.s.i. and that in right control chamber 36 falls to 1,200 p.s.i., thus producing a 600 p.s.i. pressure differential. Considering the left portion of the case pressurization mechanism, it will be seen that 1,800 p.s.i. is applied to the left end of left plunger 71 whereas the sum of the force exerted by spring 75, equivalent to 100 p.s.i., and the initially assumed 1,400 p.s.i. pressure in sump chamber 40 applied via vent passage 76 and central chamber portion 64 to the right end portion of left ball valve 70 exposed upstream of left valve seat 69, will operate to urge this left ball valve firmly against its seat.

On the other hand, the pressure against the right end of right plunger 74 is now 1,200 p.s.i. which is dominated by the sum of the spring force assumed to correspond to 100 p.s.i. and the initially assumed 1,400 p.s.i. in sump chamber 40. This will unseat right ball valve 73 and allow fluid to flow from the sump chamber via vent passage 76, central chamber portion 64, passed valve seat 72 into outlet passage 81 and drain conduit 82. This venting of pressurized fluid in sump chamber 40 operates to lower the case pressure therein until a new force balance is struck on the right ball valve and right plunger which will allow this ball valve to reseat. This occurs when the pressure in sump chamber 40 drops to a level of 1,100 p.s.i. so that the sum of the force exerted by this pressure and that exerted by the bias spring 75, 100 p.s.i., equals the oppositely directed force applied to the right end of plunger 74 by the 1,200 p.s.i. pressure in the right end chamber portion 67.

If the command signal input to control valve 21 is now changed so as to reverse the polarity of the pressure differential between control ports 32 and 36, it will be seen that the right ball valve and plunger assembly will remain seated whereas the left ball valve and plunger assembly will unseat to vent the excess of pressure that may tend to build up in the sump chamber so as to drop the level of the case pressure in the sump chamber to that predetermined, i.e., a constant, amount, assumed to be 100 p.s.i., below the now lower control pressure in the left control chamber 32.

It will also be seen that inasmuch as the case pressure in sump chamber 40 constantly remains below the lower of the pressures in the control chambers 32 and 36, the unidirectional force acting in a counterclockwise direction preloading out the backlash between left drive gear 26 and driven gear 25 remains undisturbed, as also remains undisturbed the unidirectional force but acting in the opposite or clockwise direction preloading out the backlash between right drive gear 27 and driven gear 25. Thus case pressure in the sump chamber is kept at a level below the lower of the control pressures in the control chambers by a predetermined, i.e., a constant, amount at all times thereby to reduce breakout friction and leakage while maintaining the unidirectional preloading of the backlash associated with each drive gear.

A commercial form of the apparatus is illustrated in FIGS. 4-11 and whenever possible the same numerals are employed to identify the corresponding parts shown in schematic FIG. 1.

The case 23 is shown as comprising four sections, a front cover plate 83, a front end section 84, an intermediate section 85, and a rear end section 86. The intermediate section 85 is

provided with the three-lobed compartment 24 in which the drive gears 26 and 27 and the driven gear 25 are severally arranged, as shown in FIG. 5. These gears 25-27 are nonrotatively mounted on shafts 29-31, respectively, severally suitably journaled in case sections 84 and 86. So as to be accessible from the exterior of the case output shaft 29 is shown in FIGS. 3 and 8 as extending outwardly through front cover plate 83 and also rearwardly out rear section 86. The stack of sections 84-86 are secured together by a pair of machine screws 88. Cover plate 83 is shown as secured to front case section 84 by three machine screws 89.

Rear case section 86 has a flat upper surface 35 provided with four ports 34, 39, 60, and 62 corresponding to those described in connection with FIG. 1. This rear case section is also formed to provide control valve pressure supply conduit 59 and control valve return conduit 61. It is also formed to provide elbow-shaped port passages 33 and 38 leading from control ports 34 and 39, respectively, in surface 35 to control chambers 32 and 36, respectively. This rear case section 86 also has formed therein part 821 of the separate return conduit 82 which part communicates with another part 822 provided as a horizontal opening extending through intermediate section 85, in turn registering with another part 823 provided as a horizontal opening extending through front case section 84. The front end of return conduit part 823 communicates with one end of an elbow-shaped passage part 824 provided in front cover plate 83. The other end terminates in the cylindrical wall of a central horizontal hole 90 in plate 83 and through which output shaft 29 extends in spaced relation thereto. Axially spaced seal rings 91 and 92 arranged in hole 90 and surrounding shaft 29 provide an annular space 825. The various passages 821-825 correspond collectively to return conduit 82 shown in schematic FIG. 1.

Cover plate 83 serves as the housing for the case pressurization control mechanism 22 shown in detail in FIG. 11. This cover plate is shown as having a horizontal transverse through passage 63 of cylindrical configuration arranged below output shaft 29. Passage 63 includes a central chamber portion 64 that has a lateral branch passage 761 which communicates with a horizontal passage 762 provided in front case section 83. The rear end of passage 762 communicates with sump chamber 40. The two passages 761 and 762 collectively correspond to vent passage 76 shown in schematic FIG. 1.

At the opposite ends of central chamber portion 64 are the left and right seats 69 and 72, respectively, for the left and right balls 70 and 73, respectively, with which the left and right plungers 71 and 74, respectively, are associated. Left plungers 71 is slidably arranged in the bore 65 of a tubular guide 651 arranged in passage 63 and held there by a retaining ring 93. Axially spaced O-rings 652 and 653 near the outer end of such guide provide a sealed annular chamber 781 communicating via radial passage 782 in the guide with chamber 65. This annular chamber 781 also communicates with an elbow-shaped passage 783 provided in cover plate 83 above passage 63 which in turn communicates with a horizontal passage 784 extending through front case section 84 and leading to left control chamber 32. The passages 781-784 correspond to the conduit 78 in schematic FIG. 1.

Similarly there is a right guide 671 retained by ring 94 for accommodating the right plunger 74 and this guide is surrounded by O-rings 672 and 673 at axially spaced locations to provide an annular space 791. This space communicates with the right end chamber 67 for right plunger 74 by a radial passage 792 in guide 671. Also, this annular space communicates with one end of an elbow-shaped passage 793 in turn communicating with a horizontal through passage 794 provided in front case section 83 and leading to right control chamber 36. The passages 791-794 correspond to the conduit 79 in schematic FIG. 1.

Arranged between the ball valves 70 and 73 is spring 75. On the downstream side or to the left of left seat 69 the space 66 around left ball valve 70 communicates via a vertical outlet passage 80 with annular space 825. Similarly, the space 68

surrounding right ball valve 73 downstream of its seat 72 communicates via a vertical outlet passage 81 with space 825.

The mechanism illustrated in FIGS. 4-11 has the advantages of and operates in the same manner as that described for the schematic mechanism illustrated in FIG. 1.

FIGURE 12

While FIG. 12 is a schematic view similar to FIG. 1, a different form of control mechanism is shown in FIG. 12. In FIG. 12 the positive displacement hydraulically driven device is illustrated as a push-pull actuator 100. More specifically, it comprises a case 101 having a left cylinder 102 and a right cylinder 103 both of which communicate at their lower ends with a sump chamber 104. Arranged in left cylinder 102 is a piston 105 pivotally connected via a link 106 to one end of a left rocker arm 108. A piston 109 is arranged in the right cylinder 103 and is also pivotally connected via a link 110 to the end of a right rocker arm 111. The rocker arms 108 and 111 are fast to an output shaft member 112 which is accessible from the exterior of case 101. Arranged on top of case 101 is a control valve 21 which may be of the same type as the one described in connection with FIGS. 1-11. This valve controls the flow of fluid with respect to the upper ends of cylinders 102 and 103 through left and right port passages 113 and 114, respectively.

Thus a push-pull actuator is provided in which, for example, pressure of equal amount in the upper ends of the cylinders 102 and 103 will hold the actuator in a null position but when a pressure differential is developed in the upper ends of these cylinders one piston will move downwardly and the other will move upwardly causing pivotal movement of output shaft 112. In view of the pivotal connections between the arms, links and pistons, there is lost motion or backlash in the mechanical train. However, this backlash can be preloaded out by the unidirectional force due to the difference between the individual control pressure in the upper end of each cylinder and the case pressure in sump chamber 104. Such preload forces produce in this illustration also a breakout friction for movement of the output shaft and the difference in pressures between the upper ends of the cylinders and the sump chamber produce leakage flow from these cylinders to the sump chamber. Likewise such breakout friction and leakage flow is proportional to the pressure differentials between the cylinder upper end control chambers and the sump chamber.

The present invention contemplates providing a case pressurization control mechanism 115 operatively between the push-pull actuator and the sump chamber. For this purpose, case 101 is shown as provided with a pair of aligned horizontal left and right cylindrical chambers 117 and 118, respectively, connected by an eccentric intermediate passage 119, in turn communicating with sump chamber 104 via a vertical vent passage 120. Left chamber 117 intermediate its ends has an outlet passage 121 and right chamber has a similar outlet passage 122, both such passages being connected to a return conduit 123. Arranged in left chamber 117 is a piston valve 124 having a stop pin 125 on its right end face urged away from the right end wall of this chamber by a helical compression bias spring 126. In right chamber 118 there is a piston valve 128 from the left end face of which projects a stop pin 129 urged away from the left end of this chamber by a helical compression bias spring 130. The left end of left chamber 117 communicates with left port passage via conduit 131. The right end of right chamber 118 communicates with right port passage 114 via conduit 132.

The setting of springs 126 and 130 determines the bias force corresponding to a pressure at which the fluid in sump chamber 104 is maintained at a level a constant amount below the lower pressure in either control port passage 113 or 114.

With no command signal input to control valve 21 a pressure in sump chamber 104 will be maintained at a predetermined, i.e., a constant, level according to the setting of springs 126 and 130 below the null pressure in control port passages

113 and 114. Should the control valve produce a pressure differential in these control port passages, the higher pressure in one of these passages will maintain a closed condition of the corresponding piston valve 124 or 128 by moving its stop pin against the corresponding cylinder and wall, whereas a decrease in the pressure in the other control port passage will upset the force balance on the associated piston valve causing the same to move so as to open its outlet 121 and 122 and thereby allow fluid under excessive pressure to be vented from the sump chamber 104 through vent passage 120, passage 119, the adjacent portion of chamber 117 or 118 and the outlet 121 or 122 into return line 123. This venting will continue until a new force balance is achieved with the pressure level of the lower control pressure and the piston valve recloses its outlet so that the case pressure in the sump chamber will be below such lower control pressure by the predetermined, i.e., a constant, amount corresponding to the spring setting.

FIG. 13

The positive displacement hydraulically driven device illustrated in FIG. 13 is a rotary vane motor 140 which includes a case 141 having a cylindrical compartment 142 in which a cylindrical rotor 143 is eccentrically arranged. This rotor is an output member suitably accessible from the exterior of case 141 and has a series of circumferentially spaced radial slots 144 in each of which a vane 145 is slidably arranged. The axial ends as well as the radial outer ends of vanes 145 are intended to engage sealingly the opposing wall surfaces of compartment 142. These vanes are urged radially outwardly by fluid derived from a pair of chambers 146 and 148 formed in a central case part 149 surrounded by rotor 143. Left chamber 146 communicates via passage 150 with a left control port 151 serviced by a left control port passage 152. Right chamber 148 communicates via passage 153 with a right control port 154 serviced by a right control port passage 155.

As in the other forms of the invention illustrated a control valve 21 is operatively associated with vane type servomotor 140 to control the flow of fluid with respect to control port passages 152 and 155.

The unoccupied portions of compartment 143 adjacent control ports 151 and 154 provide left and right control or work chambers 156 and 158, between which at the bottom of this compartment is a sump chamber 159. Pressure differentials between control chambers 156 and 158 severally and sump chamber 159 provide breakout friction, leakage flow and preloads out backlash in the vane servomotor.

Operatively interposed between control chambers 156, 158 and sump chamber 159 is a case pressurization control mechanism 160 which is illustrated in FIG. 13 of still a different form from the corresponding mechanisms illustrated in connection with FIGS. 1-11 and FIG. 12.

In FIG. 13, the case pressurization control mechanism 160 comprises a cylindrical chamber including a central portion 161, a left end portion 162, an enlarged left intermediate portion 163 therebetween, a right end portion 164, and a right enlarged intermediate portion 165. Slidably arranged in left intermediate portion 163 is a left piston 166 having a piston rod 168 slidably arranged in left end portion 162. Slidably arranged in right intermediate portion 165 is a right piston 169 having a piston rod 170 slidably arranged in right end portion 164. The outer or opposing end faces 171 of pistons 166 and 169 are larger in area than the outer or remote end faces 172 of rods 168 and 170. Left enlarged chamber portion 163 has a left outlet passage 173 and right enlarged chamber portion 164 has a right outlet passage 174, both manifolded to a return conduit 175. Left end chamber portion 162 communicates with left port passage 152 via left conduit 176. Right end chamber portion 164 communicates with right port passage 155 via right conduit 178. Central chamber portion 161 communicates with sump chamber 159 via vent passage 179.

In the form of case pressurization control mechanism 160 shown in FIG. 13 the bias force means are not provided by a

spring but rather by the difference in the end areas 171 and 172. The case pressure in sump chamber 159 is applied to the large piston end areas 171 so that this pressure need not be as high as that operating against the opposite small end areas 172 of the piston rods in order to achieve a force balance on the corresponding piston.

It is evident that if the case pressure in the sump chamber 159 is too high with respect to either control pressure in chamber 156 and 158 one of the pistons 166 and 169 will move to uncover its outlet 173 or 174 so as to vent or dump some of the pressurized fluid from the sump chamber thereby lowering its pressure until a few force balance is established on the piston which is so moved. While this occurs, the other piston is maintained in a closed condition by the higher pressure acting against its smaller end 172.

From the foregoing, it will be seen that the present invention can be embodied in specifically different types of control mechanisms while achieving all of the stated advantages.

What is claimed is:

1. In a control mechanism comprising a positive displacement hydraulically driven device including a case having a first control port, a second control port and a chamber, a movable output member accessible from the exterior of said case, first pressure reaction means operatively arranged in said case between said first port and chamber and coupled through mechanical backlash to said member for driving the same in one direction, and second pressure reaction means operatively arranged in said case between said second port and chamber and coupled through mechanical backlash to said member for driving the same in the opposite direction, and a four-way control valve operatively associated with said device to control pressures differentially at said ports, said backlash for each of said reaction means being preloaded out by the unidirectional force due to the difference between the individual control pressure in the corresponding one of said ports and the case pressure in said chamber, such preload forces producing a breakout friction for movement of said output member, and the difference in pressures between said ports and chamber producing leakage flow from said ports to said chamber, both such friction and leakage flow being proportional to the pressure differentials between said ports and chamber, the improvement which comprises means for controlling the case pressure in said chamber so as to keep it at a level below the lower of the control pressures in said ports the difference between such case pressure level and said lower of the control pressures being a constant amount at all times thereby to reduce friction and leakage while maintaining the unidirectional preloading of said backlash associated with each of said reaction means.

2. A control mechanism according to claim 1 wherein said constant amount falls in the range of from 50 to 100 p.s.i. whereby the friction and leakage are reduced to levels consistent with the differential control pressures required to drive a load on said output member while still ensuring finite unidirectional preloading of the backlash.

3. A control mechanism according to claim 1 wherein said means for controlling the case pressure in said chamber includes means providing variable orifices severally associated with said control ports and operatively arranged to restrict the leakage from said chamber to a drain.

4. A control mechanism according to claim 1 wherein said means for controlling the case pressure in said chamber includes first means providing a first variable orifice operatively interposed between said chamber and a drain and operatively responsive to the control pressure in one of said control ports and also including second means providing a second variable orifice operatively interposed between said chamber and a drain and operatively responsive to the control pressure in the other of said control ports.

5. A control mechanism according to claim 4 wherein each of said first and second means includes an outlet and valve closure means movable with respect thereto, said valve means being urged to move to close its said outlet in response to the

control pressure in the corresponding one of said control ports and being urged to move to open its said outlet in response to the case pressure in said chamber.

6. A control mechanism according to claim 5 wherein each such valve means moves in response to the balance of forces acting thereon and bias force means are provided whereby such control pressure always remains higher than said case pressure.

7. A control mechanism according to claim 6 wherein said bias force means comprises a spring.

8. A control mechanism according to claim 6 wherein each such valve means has opposite effective end areas and said bias force means comprises a difference in the size of said end

areas.

9. A control mechanism according to claim 6 wherein said case includes a pair of seats, a ball valve for each seat, a spring between such ball valves and arranged to urge them off their seats, a plunger for each ball valve engaging the same on the side opposite from its seat, said plunger having a cross-sectional area corresponding to that of such seat, an outlet to drain communicating with each of said seats on the downstream side thereof, means for applying the case pressure in said chamber to said balls on the upstream sides of said seats, and means for applying said control pressures severally to the ends of said plungers remote from said seats.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,639,088 Dated February 1, 1972

Inventor(s) William C. Moog, Jr.

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 8, line 11, "illustrates" should read --illustrated--.

Column 10, line 12, "few" should read --new--.

Column 10, line 35, "different" should read --difference--.

Signed and sealed this 11th day of July 1972.

(SEAL,
Attest:

EDWARD M. FLETCHER, JR.
Attesting Officer

ROBERT GOTTSCHALK
Commissioner of Patents