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Description

The present invention relates to an automatic supply and exhaust valve assembly comprising at least one control valve comprising a tubular element having a source port, a load port and at least one drain port and further comprising a reciprocative valve member, with said tubular element and said valve member having axially spaced first and second mating pairs of valving surfaces which are mechanically coupled and arranged for connecting said source port with said load port as fluid pressure supplied to the source port exceeds pressure at the load port and for connecting said load port with said drain port so as to relieve pressure at said load port.

A valve assembly and, more specifically, an automatic supply and exhaust valve of this general type is known from US—A—2 702 044. The known valve comprises oppositely directed pairs of valving surfaces, one of which is closed while the other is open and vice versa, so that the load port being arranged intermediate said source and drain ports is either in fluid connection with the source port or with the drain port. thus, the designation "automatic valve" refers to the fact that any device being connected to the load port, especially the air brakes of a truck, are automatically vented via the drain port as soon as the pressure at the source port decreases below a predetermined value at which the source port is closed.

Other known valve assemblies, such as described in GB—A—597 103 and FR—A—2 298 043 comprise reciprocative valve members for opening and closing in synchronism fluid connections between one single source port and two separate load ports. These valves are serving the purpose of subdividing an inlet fluid stream at the source port into two equal outlet fluid streams at the load ports.

As is understood by those skilled in the art, large process control valves, e.g. such as those employed in petroleum refineries and chemical and power plants are often driven by electrically controllable, hydraulic positioning systems. Such hydraulic positioning systems commonly includes a powerful, single-acting, spring-return hydraulic piston and a constant speed positive displacement pump which provides a source of hydraulic power, both for stroking the piston and for holding same at any selected position within its stroke. Typically the pump is run continuously and the pressure to the actuator is modulated by a conventional three way servo valve or equivalent systems means such as a flapper nozzle or jet pipe system which relieves excess pressure to the sump. The servo valve, in turn, is responsive to an electrical command signal employed in conjunction with a position feedback loop.

While the actuator is immobile, the servo valve throttles the pump output in order to create the proper back pressure as required to hold the piston in position and the totality of the flow is returned to the pump sump when the piston is immobile. As a result of the continuous pump operation, the efficiency of present state-of-the-art hydraulic actuators system is, in the large majority of applications, in the order of five percent or less. Inherently a majority of the hydraulic energy generated by the pump is wasted as heat while the actuator is immobile at any intermediate position. As is understood by those skilled in the art, the actuator is, in fact, immobile much of the time in most valve applications, particularly in large and rather stable processes. Not only is the loss of energy wasteful, the heat created is itself troublesome.

Proceeding on the basis of the prior art and considering the problems discussed above, it an object of the present invention to provide an improved valve assembly, especially a reciprocative check valve which selectively allows free flow through a single line port in either direction in order to selectively fill or unfill a variable volume load, e.g. to extend or retract the piston of a single-acting, spring-return actuator.

This object according to the invention is accomplished by means of an automatic supply and exhaust valve assembly of the type indicated at the outlet and being characterized in that said pairs of valving surfaces are arranged to open and close in synchronism, and that said drain port is provided in said tubular element at a position intermediate said source and load ports so as to allow for simultaneous matching flows from both said load port and said source port to said at least one drain port when said valving surfaces are open and so as to be disconnected from said source and load ports when said valving surfaces are closed.

It is an advantage of such a reciprocative check and control valve that it maintains the piston position's volume with a single positive acting check valve when the piston is immobile. Further, the valve adjusts the flow returning from the actuator to the flow from the hydraulic power source. Moreover, pump means need be operated only when the piston is moving. Additionally, a system comprising an inventive valve assembly may be precisely controlled and especially has a symmetrical response while being reliable and of relatively simple and inexpensive construction. Other objects and features will be in part apparent and, in part, pointed out hereinafter.

Preferred embodiments of the invention are the subject matter of dependent claims.

A hydraulic system in accordance with one aspect of the present invention employs a pump which draws fluid from a pump to provide fluid under pressure. Located between the pump and the load is a reciprocative check valve having first and second mating pairs of valving surfaces which are mechanically coupled and arranged to open in synchronism when the pump pressure exceeds the load pressure. These valving surfaces are connected in series between the pump and the load with the connection bezween the two pairs of valving surfaces being connected also, through a release valve, to the sump. When the release valve is closed, operation of the pump will expand the load volume and, when the release valve is open, operation of the pump will contract the load volume.

In accordance with another aspect of the invention, the reciprocative check valve or flow matching valve is a device employing a tubular body structure having a source port at a first axial position along the body and a load port at a second position along the body which is axially displaced from the first position. A drain port is located between the source and load ports. In the body, there is a plug member which is movable axially in response to any difference in the pressures at the source and load ports. The plug member includes rigidly connected mating surfaces which progressively open the source and load ports in synchronism. When opened, the source and load ports each connect with a drain port. A common or separate drain port may be provided depending on the end use application. When a common drain port is used and is closed, a hydraulic flow into the source port will exit through the load port and, when the drain port is open, a hydraulic flow into the source port will produce a controlled flow through the load port, both flows exiting the valve through the drain port.

The invention will now be further explained in connection with the accompanying drawings.

Figure 1 is a somewhat diagramatic illustration of a conventional circuit hydraulic actuator, i.e. in accordance with the prior art;

Figure 2 is a cross-sectional view of a reciprocative check valve or flow matching valve constructed in accordance with the present invention;

Figure 3 is a diagrammatic illustration of a hydraulic actuator constructed in accordance with the present invention and employing a control valve in accordance with the present invention, that is, of the type illustrated in Figure 2 or 4;

Figure 4 is a cross sectional view of a preferred mechanical construction of a control valve in accordance with the present invention;

Figure 5 is a diagrammatic illustration of a hydraulic actuator system constructed in accordance with the present invention and employing a bi-directional variable speed pump in cooperation with a flow matching valve;

Figure 6 is a block diagram of control electronics suitable for use in the actuator system of Figure 5; and

Figure 7 is a diagrammatic illustration of a double acting hydraulic actuator system in accordance with the present invention.

Corresponding reference characters indicate corresponding parts throughout the several views of the drawings.

As indicated above, hydraulic actuators operating process control valves typically employ a relatively massive hydraulic prime mover. Referring now to Figure 1, such a mover is indicated generally by reference character 11 and comprises a piston 13 and a cylinder 15. To provide a return force and a modicum of fail safe operation, the piston is normally biased by a heavy spring, as indicated at 17, toward a return position. Hydraulic fluid from a sump 19 is provided under pressure suitable for operating prime mover 11 by a unidirectional pump 21 which runs constantly. In the supply line 24 to the cylinder 15, a control valve, e.g. an electrically operated, spool type servo valve 25, is provided for modulating the pressure of the fluid as suitable for moving the prime mover piston or holding it at any position within the stroke range. The excess flow is returned to the pump sump 19 through line 28.

In such prior art systems, the pump cannot be stopped since it is normally impossible to put a positive acting check valve between the servo valve and the actuator to prevent drifting of the piston.

This problem is alleviated by employing in such an hydraulic actuator circuit a flow-matching or reciprocative check valve in accordance with the prsent invention. A relatively simple version of such a valve is illustrated in Figure 3 and serves well for the purpose describing the basic valve function and overall system operation. Referring now to Figure 2, the valve illustrated there comprises a generally cylindrical or tubular body portion 33 within which operates a plug member 35. The overall control valve structure is designated by reference character 31. The valve body 33 provides a first valve seat 37 and a second valve seat 39 which is axially displaced along the body from the first valve seat. Both valve seats face in the same direction and are of the same diameter.

The plug member 35 includes a first valving surface 41 and a second valving surface 43 which mate with the seats 37 and 39 respectfully. The axial displacement between the valving surfaces 41 and 43 matches the axial spacing of the seats 37 and 39 so that the two ports open synchroniously. The port controlled by the valving surface 41 in cooperation with the seat 37 may be considered the source or supply port while the port controlled by the second valving surface 43 in conjunction with the second valve seat 39 may be considered the load port. While the plug member 35 is preferably lightly biased in the direction tending to close the ports, e.g. by a spring 45, the plug member is essentially floating in the body so as to be responsive to any difference in pressure between the supply side and the load side.

The valve body 33 and plug member 35 provide, between them, an intermediate chamber 47. A drain port opens into chamber 47, as indicated at reference character 49. While the valve body 33 and plug member 35 are illustrated as integral structures for the purpose of explanation, it will be understood by those skilled in the mechanical arts that these parts are necessarily assembled of multiple components so as to permit the construction of interlocking assembly shown in the drawings.

The valve 31 of Figure 2 is functional to provide flow matching characteristics and, in effect, reciprocative check valving. This operation may best be understood in conjunction with the description of an overall system, such as that illustrated in Figure 3. As may be seen, the control valve of Figure 2 is installed in the supply line 24. However, the return line 28 is also connected to the drain port of the valve 31. In series with the drain port is a simple two way directional valve, e.g. a solenoid operated on-off valve, designated by reference character 51. As is explained in greater detail hereinafter, the utilization of a simple on/off valve for unfilling the cylinder is possible since the single control valve 25 is functional during both filling and emptying of the pcston 13.

At this point, it is useful to consider Figures 2 and 3 together. Assuming that the drain port 49 is closed off, i.e. the solenoid valve 51 is closed, it can readily be seen in that a hydraulic flow into the source port of the valve 31 will proceed through the intermediate chamber 47 and on out through the load port. As will be understood, the filling of the cylinder 15 in this situation can be controlled by the operation of the throttling valve 25. Further, when the supply pressure drops below the pressure in the cylinder 15, the control valve 31 acts as a positive operating check valve to prevent any backflow from the piston.

To reduce the volume of hydraulic fluid in the cylinder 15, the drain port 49 must be opened i.e., by opening the on/off valve 51. However, the mere opening of this value will not permit the piston to retract if the servo valve is not delivering pressure to the source port. If, though, the servo valve is then operated so that a hydraulic flow is introduced into the source port of the control valve 31 while its drain port is open, it can be seen that this flow will return back to the sump as soon as the source pressure equals the load pressure and moves the plug member 35 sufficiently to open the source port.

However, once the plug member 35 moves, the load port is also opened by the same amount as the source port. As noted previously, the valving surface in the two ports are of equal diameter and, therefore, of equal area. Further, since the pressure on the load side is necessarily about equal to that on the source side and since the drain pressure is the same for the two flows, it can be seen that the pressure drop across the two valving surfaces will be equal. Accordingly, it will be understood that substantially equal flows will occur from the source and load sides. In this way the control valve 31 operates as a flow matching device, that is, the flow out of the piston will be equal to the flow into the source port of the control value 31. As in the filling mode, this flow is controllable by means of the throttling valve 25. In addition, the control valve 31 also operates as a check valve with respect to the return line 30 since, even if the solenoid valve is open and the pump pressure drops below the load pressure, no back flow will take place.

Since the flow out of the piston 31 is controlled by the same servo or throttling valve 25 which controls filling flow and, since the servo valve is operating under essentially the same pressure differential conditions in both situations, it can be seen that essentially symmetrical operation is attained for both filling and emptying of the cylinder and that this will facilitate implementation of an overall servo control system within which such hydraulic actuators are typically employed. Further, if desired, valves 51 and 25 can be combined in a single spool valve type structure.

While the operation of the relatively simple valve shown in Figure 2 is readily understood so that it serves well for the purpose of illustration, it will be understood by those skilled in the art that the balanced operation of such a construction at small flows becomes highly dependent on the accuracy with which the critical dimensions may be matched, i.e. the length of the plug member 35 between the surfaces 41 and 43 as compared with the actual separation between the valve seats 37 and 39. Maintenance of critical dimensions is facilitated with the arrangement shown in Figure 4 and this construction is presently preferred.

With reference to the device shown in Figure 4, it will be apparent to those skilled in the art that the constructional techniques are quite similar to those employed in the making of spool valves where close tolerances are regularly achieved. Fitting within an overall body assembly 61 is a sleeve 63 and a piston 65. Sleeve 65 is stationary within the body member 61 while the piston 65 is slidable axially within the sleeve 63, i.e. similar to the manner in which the spool element in a spool valve is slidable. Preferably, the piston is lapped to the sleeve to provide a close, low leakage fit. The sleeve 63 is provided with a pair of internal annular grooves 67 and 69 with a precise axial separation between them. The piston 65 is provided with a matching pair of external annular grooves 71 and 73 wth an axial separation between these grooves which matches the axial separation between the grooves 67 and 69 on the sleeve.

Within the piston 65 a first passageway system 70 connects the groove 71 with the source port while a second passageway system 77 provides communication from the groove 73 to the load port end of the sleeve 63. Cross ports 78 and 79 in the sleeve connect the grooves 67 and 69, to a respective pair of drain ports 82 and 84 in the valve body 61. When employed in the system shown in Figure 3, the drain ports 82 and 84 are connected together externally of the body 61 to form a common drain functioning in the same manner as the single drain of the sample valve of Figure 2. However, in other systems, such as the double acting cylinder system described hereinafter with respect to Figure 7, it is advantageous to utilize separate drain paths from the source and load with a pressure barrier, i.e. a sealed land, therebetween. These are referred to hereinafter as the source drain and the load drain respectively.

The upper end of the sleeve 63 provides a valve seat, as indicated by reference character 85 and a spherical valving element 87 is lightly biased into contact with this seat by a spring 89. A projecting portion 91 of the piston 65 is formed to lift the valving element 87 from the seat 85 just as the annular grooves on the piston come adjacent the respective annular grooves on the sleeve 63.

Ignoring for a moment the action of the spherical valving element 87, it can be understood that the cooperative action of the piston and sleeve portion of the valve is essentially similar to the action provided by the valve of Figure 2. As the pressure at the source port comes equal to that at the load port, the piston moves upwardly, opening the two valving sections in synchronism. If the drain ports are connected but closed, fluid flow introduced into the source port will proceed, through the drain ports, on to the load port. If the drain ports are open, however, matching flows from the source port and the load port will exit through the drain ports. These flows will be well matched in volume since the valve openings are closely matched and since the pressure drop in each channel will be equal.

This basic operation is not changed by the presence of the spherical valving element at the top of the sleeve since the spherical valving element 87 is lifted from the valve seat at the same time or slightly before the annular grooves open to each other. However, any time the source pressure drops significantly below the load pressure the spherical valving element acts as a simple but highly effective check valve eliminating backflow from the load. Since the desired sealing requirement is met by this element, there is no requirement for an absolute seal between the piston and the sleeve. Since the overall operation of the valve device of Figure 4 in the system of Figure 3 is basically the same as that of the valve device of Figure 2, it will also be seen that the valve device of Figure 4 may be directly substituted in the novel hydraulic system of Figure 3 which will continue to provide the desired function and advantages.

The flow matching characterisics of values constructed in accordance with the present invention may also be advantageously utilized in systems where a variable speed bidirectional pump is available. Referring now to Figure 5, a bidirectional positive displacement pump 100 is utilized for providing hydraulic fluid under a pressure suitable for operating a cylinder, again indicated by reference character 15. A pressurized accumulator 101 provides a reservoir of hydraulic fluid. This reservoir is connected, through respective check valves 102 and 103, to both sides of pump 100. Pump 100 is preferably of the positive displacement, meshing gear type and is driven in either direction by a stepping motor 105. Movement of the piston 13 is tracked by a suitable transducer e.g. a side wire potentiometer as indicated at 106, so as to provide a suitable feedback signal or voltage. A pressure release valve is provided, as indicated by reference character 108, for limiting the maximum pressure which can be applied to the cylinder 15. The cylinder 15 is connected, through a check valve 109, to one side of the pump and, through a flow matching control valve 31, to the other side of the pump.

The relatively simple version of the flow-matching control valve is illustrated in Figure 5 and serves well for the purpose describing the overall system operation but it should be understood that the construction shown in Figure 4 is presently preferred. When the pump 100 is driven in a direction producing flow from right to left as seen in Figure 5, flow from the high pressure side of the pump is blocked by the check valve 103 but is passed by the check valve 109 so that flow increases the operating volume of the cylinder 15. In this situation, the control valve 31 acts simply as a positively operating check valve.

When the pump is driven in the opposite direction, producing flow from left to right as seen in Figure 1, the flow from the high pressure side of the pump is blocked by the check valve 102 but is admitted into the source port of the central valve 31. As soon as the pressure at the high pressure side of the pump equals the load pressure, i.e. the pressure in the cylinder 15, the plug member is lifted from the valve seats opening the source port. However, at the same time that the source port of the control valve is open, the load port will also be opened by a like amount. Since the intermediate chamber is vented back to the accumulator 101, it can be seen that the flow generated by the pump will cause an equal flow to pass through the load port of the control valve, with both flows exiting the control valve through the drain port so as to return to the accumulator or sump. Because the emptying flow from the cylinder 15 is maintained substantially equal to the flow out of the pump, it can be seen that the overall sensitivity or "gain" of the hydraulic system is the same for both filling or emptying, a highly desirable attribute as will be understood by those skilled in the servo-control art.

While various control schemes for controlling the operation of the pump 100 may be implemented, one particular such scheme is shown by way of example in Figure 6. The feedback signal obtained from the potentiometer 106 is compared, in a differential amplifier 111, with a reference voltage representing the desired position of the piston thereby to generate an error signal representing the difference between the desired and actual positions for the piston. A zero crossing detector circuit 113 provides a signal indicating the sense or polarity of the error and this signal is provided to the direction control input of a conventional stepper motor driving circuit, indicated by reference character 115. A signal proportional to the amplitude of the error, independent of polarity, is provided by an absolute value detector circuit, indicated by reference character 117. As is understood, this circuitry may be constituted by a simple array of diodes. The signal proportional to the absolute value of the error is provided to enable a voltage-to-frequency converter circuit 119 whose output is, in turn, applied to the step signal input of the stepper motor driving circuit 115.

From the foregoing, it will be understood by those skilled in the art that the stepper motor 105 will be energized in a direction which reverses in accordance with the sensed direction of the error and at a speed which is proportional to the magnitude of the error. This operation thus closes the servo-loop so that the position of the piston will follow variations in the set point reference signal, as desired. However, as compared with conventional electro-hydraulic actuators in which the pump runs continuously, the stepper motor 105 is energized only when an error exists and the level of energization is proportional to the error. In relatively stable overall systems therefore, the motor is energized only intermittently.

As will be understood, this intermittent energization only when needed both reduces the average power requirement and the amount of heat dissipated in the system. Further, when the motor is not energized, the position of the piston 13 is maintained by positive acting check valve structures and is not a function of the leakage or backflow which would occur through the pump 19 if the load pressure were maintained across the pump itself. It may also be pointed out that, other than the the control valve 31, all of the valves in the system are essentially simple check valve constructions and no elaborate reversing or four-way valves are required, as would typically be the case in conventional hydraulic servo control systems.

The flow directing concepts of the present invention may also be advantageously applied to the control valving for a double acting or bidirectional hydraulic cylinder system if separate drain ports are utilized for the source and load ports. Referring now to Figure 7, a prime mover is indicated generally by reference character 121 and comprises piston 123 and cylinder 125. The double rod ended piston provides equal annular area on both faces of the piston.

A bi-directional, positive displacement pump 127 is utilized for providing hydraulic fluid under a pressure suitable for operating the cylinder. A pressurized accumulator 131 provides a reservoir for the hydraulic fluid. This reservoir is connected through respective check valves 132 and 133 to both sides of the pump 127. Pump 127 is preferably of the positive displacement meshing gear type and is driven in either direction by a stepper motor 135 whose speed can be varied from zero to a maximum by means of suitable control electronics 137. Movement of the piston is tracked by a suitable transducer; e.g., a slide wire potentiometer indicated at 138 so as to provide a suitable feedback voltage or signal.

One side of the pump is connected to one side of the cylinder 121 through a hydraulic circuit which includes the source/source-drain path of a flow matching valve 139 and a check valve 147. The other side of the pump 127 is symmetrically connected through a hydraulic circuit which includes the source/source-drain path of a flow matching valve 141 and a check valve 145. Both flow matching valves 139 and 141 are identical in construction and size. The construction is preferably that illustrated in Figure 4 with separate drain ports being maintained for the source and load ports.

Each side of the cylinder 121 is also cross connected to the load port of the opposite flow matching valve 139 or 141. Similarly, the load-drain port of each of the flow matching valves is cross connected to the source port of the other flow matching valve. In the following description of operation, it is assumed that a load is being applied to the piston 123 from the left side so that the right side of the cylinder is under greater pressure than the left side.

In order to drive the piston against the load, the pump 127 is driven so as to produce a flow from left to right as seen in the drawings. When the pressure at the outlet of the pump exceeds that of the low pressure side of the cylinder, initial opening of the paths through the flow matching valve 139 will occur and the pressure at the outlet side of the pump will then equalize with the high pressure side of the cylinder. Under continued pumping, flow will then occur through the source/source-drain path of the valve 139 and the check valve 147, driving the piston to the left. At the same time, an equal flow will return from the left side of the piston through the load/load-drain path of the control valve 139 to the low pressure side of the pump in a relatively straightforward manner.

When the pump 127 is operated in the opposite direction, i.e. producing flow from right to left as seen in Figure 7, an essentially similar operation takes place but additional flow matching effects may come into play. Initially, when the pump output pressure reaches that on the high pressure side of the cylinder, the paths through the flow matching control valve 141 will start to open. Flow from the high pressure side of the cylinder will pass through the load/load-drain path of the control valve 141 back to the intake side of the pump allowing the piston to move to the right. At the same time, a matching flow will tend to pass through the source/source-drain path of the valve 141 and the check valve 145 to fill the expanding low pressure side of the piston. However, as may be seen, the load is aiding the motion and the pump 127 may be considered to be in an overrunning condition where the intake pressure may tend to exceed that on the low pressure side of the cylinder. Under this situation, the other, normally passive, flow matching valve 139 may open slightly. The opening of the load drain/load path will bypass some of the pump's output allowing a partial reclosing of the flow matching valve 141 which therefore increases the restriction and reduces the flow from the high pressure side of the cylinder, thereby restoring balance.

Since the hydraulic circuit is entirely symmetrical, it can be seen that complementary actions are obtained if the load is applied to the piston in the opposite direction. An additional advantage of the symmetrical design of Figure 7 is that, when the pump is stopped, the high pressure trapped under the plug of the active valve 131 will cause the passive valve plug to lift and relieve the said trapped pressure which would otherwise delay the closing of the high pressure load port causing the piston to creep. In addition, the pump is unloaded and ready for the next start thereby substantially eliminating any risk of the motor stalling.

Summarizing, it can be seen that the flow pumped into one side of the cylinder is always matched by the flow returned to the pump intake regardless of the direction of rotation of the pump and regardless of the direction of the load. Further, since this hydraulic circuit is entirely symmetrical, it follows that the high pressure and low pressure sides of the cylinder are only dictated by the direction of the load vector. Conversely, the response or sensitivity of the actuator is identical in both directions regardless of the direction of the load, a highly desirable attribute as will be understood by those skilled in the servo control art.

It may also be pointed out that, in relatively stable overall systems, the motor is energized only intermittently. As is understood, this both reduces the average power requirement and the amount of heat dissipated in the system. Furthermore, when the motor is not energized, the piston position is maintained by positive acting check valve structures and is not a function of the leakage or back flow which would occur through the pump if a load pressure were maintained across the pump itself. It may also be pointed out that all of the valves in this system, other than flow matching valves 139 and 141, are essentially simple check valve construction and no elaborate reversing four way valves, or counter balancing valves are required as would be the case in conventional hydraulic actuators.

Claims

1. An automatic supply and exhaust valve assembly comprising at least one control valve (31; 139, 141) comprising a tubular element (33; 63) having a source port, a load port and at least one drain port (49; 82, 84) and further comprising a reciprocative valve member (35; 65), with said tubular element (33; 63) and said valve member (35; 65) having axially spaced first and second mating pairs of valving surfaces (37, 41, 39, 43; 69, 71, 67, 73) which are mechanically coupled and arranged for connecting said source port with said load port as fluid pressure supplied to the source port exceeds pressure at the load port and for connecting said load port with said drain port (49; 82, 84) so as to relieve pressure at said load port, characterized in that said pairs of valving surfaces (37, 41, 39, 43; 69, 71, 67, 73) are arranged to open and close in synchronism, and that said drain port (49; 82, 84) is provided in said tubular element (33; 63) at a position intermediate said source and load ports so as to allow for simultaneous matching flows from both said load port and said source port to said at least one drain port (49; 82, 84) when said valving surfaces are open and so as to be disconnected from said source and load ports when said valving surfaces (37, 41, 39, 43; 69, 71, 67, 73) are closed.

2. The valve assembly according to Claim 1 wherein said valve member is a plug member (35) including conical valving surfaces (41, 43) which are actually spaced so as to mate simultaneously with said seats (37, 39) and to open synchronously.

3. The control assembly according to Claim 1 wherein said tubular element comprises a tubular sleeve (63) and wherein said valve member comprises a piston (65) actually slidable within said sleeve (63), said sleeve (63) and said piston (65) having valving surfaces comprising a first pair of mating valving grooves (67, 73) and, actually displaced from said first pair, a second pair of mating valving grooves (69, 71), said valving grooves being matched to open to each other in synchronism; and wherein in said piston (65) there is a first passageway (77) from said first valving groove (73) to one end of said piston (65) and a second passageway (70) opening from said second valving groove (71) to the other end of said piston (65) and wherein in said sleeve (63) there are passageways (78, 79) connecting each pair of said valving grooves to a drain port (82, 84).

4. The valve assembly according to Claim 2 wherein said tubular element (33) and said plug member (35) provide between them a chamber (47) which is located actually between said valve seats (37, 39) and wherein said drain port (49) is opening into said chamber (47) whereby, when said drain port (49) is closed, hydraulic flow in a direction tending to open said valving surfaces (37, 41, 39, 43) will proceed through said tubular element (33) and when said drain port (49) is open a hydraulic flow into said tubular elements (33) in said one direction will produce a controlled flow in the opposite direction with both flows exiting through said open drain port (49).

5. The valve assembly according to Claim 3 wherein at one end of said sleeve (63) there is a valve seat (85) and wherein a valving member (87) is provided, which is adapted to mate and close said valve seat (85), said piston (65) including a portion (91) which, during movement of the piston (65) engages said valving member (87) to lift it off said seat (85) substantially at the same time that said mating valving grooves (67, 73, 79, 81) open.

6. The valve assembly according to one of Claims 1 to 5 wherein said control valve is part of a hydraulic system comprising:

a variable volume load (11);

a sump (19);

a unidirectional pump for drawing fluid from said sump (19) and providing fluid under said fluid pressure; a release valve (51);

with said control valve (31, 139, 141) being arranged between said sump and said load so that its first and second mating pairs of valving surfaces (37, 41, 39, 43; 69, 71, 67, 73) which are mechanically coupled, which are arranged to open in synchronism when said fluid pressure provided by said pump (21) exceeds the pressure at said load port and which are connected in series between said pump and said load (11), and with said drain port between the two pairs of valving surfaces being connected also through said release valve (51) to said sump (19), whereby, when said release valve (51) is closed, operation of said pump (21) will expand the load volume and, when said release valve (51) is open, operation of said pump (21) will contract said load volume.

7. The valve assembly according to one of Claims 1 to 5 wherein said control valve is part of a hydraulic system comprising:

a fluid reservoir (101);

a bidirectional pump (100);

a variable volume load (11);

with said control valve (31; 139, 141) serving as a flow matching control valve wherein a hydraulic flow into the source port produces a controlled flow into the load port with both flows exiting through the drain port; and further comprising:

means connecting said reservoir (101) to both sides of said pump (100) through respective check valves (102, 103) permitting flow from the reservoir (101) to said pump (100);

means connecting one side of said pump (100) to the source port of said control valve (31; 139, 141);

means connecting the load port of the said control valve (31; 139, 141) to said load (11);

means connecting the drain port of said control valve (31; 139, 141) to said reservoir (101); and

means connecting the other side of said pump (100) to said load (11) through a check valve (109) permitting flow from said pump (100) to said load (11).

8. The valve assembly according to one of Claims 1 to 5 wherein a pair of said control valves is part of a hydraylic system comprising:

a bidirectional pump (127);

with said control valves (139, 141) serving as flow matching control valves each having a source port, a load port and a pair of drain ports wherein a hydraulic flow into the source port produces a controlled flow into the load port and with the flows exiting through respective drain ports; and further comprising

means connecting each side of said pump (127) to the source port of a respective control valve (139, 141);

means connecting the load port of each control valve (139, 141) to a respective one of said cylinder ports;

means connecting the load drain port of each control valve (139, 141) to the source port of the other control valve (139, 141); and

check valve means (145, 147) connecting the source drain port of each control valve (139, 141) to the load port of the other control valve.

Patentansprüche

1. Automatische Zu- und Ablaufventilanordnung mit mindestens einem Steuerventil (31; 139, 141), welches ein rohrförmiges Element (33; 63) umfaßt, welches eine Zulauföffnung, eine Arbeitsdrucköffnung und mindestens eine Ablauföffnung (49; 82, 84) umfaßt und welches ferner ein hin- und herbewegliches Ventilelement (35; 65) umfaßt, wobei das rohrförmige Element (33; 63) und das Ventilelement (35; 65) ein erstes und ein zweites Paar von axial im Abstand voneinander angeordneten, zusammenpassenden Ventilflächen (37, 41, 39, 43; 69, 71, 67, 73) aufweisen, welche mechanisch gekoppelt und zum Verbinden der Zulaufffnung mit der Arbeitsdrucköffnung überschreitenden Fluiddruck an der Zulauföffnung und zum Verbinden der Arbeitsdrucköffnung mit der Ablauföffnung (49; 82, 84) zum Abbauen des Druckes an der Arbeitsdrucköffnung angeordnet sind, dadurch gekennzeichnet, daß die Paare von Ventilflächen (37, 41, 39, 43; 69, 71, 67, 73) zum sychronen Öffnen und Schließen angeordnet sind, und daß die Ablauföffnung (49; 82, 84) in dem rohrförmigen Element (33; 63) in einer Position zwischen der Zulauföffnung und der Arbeitsdrucköffunung angeordnet ist, um gleichzeitig eine gegenseitige Anpassung der Ströme sowohl von der Arbeitsdrucköffnung als auch von der Zulauföffnung zu der mindestens einen Ablauföffnung (49; 82, 84) zu ermöglichen, wenn die Ventilflächen sich in der Offenstellung befinden, und um die Ablauföffnung von der Zulauföffnung und der Arbeitsdrucköffnung zu trennen, wenn sich die Ventilflächen (37, 41, 39, 43; 69, 71, 67, 73) in ihrer Schließstellung befinden.

2. Ventilanordnung nach Anspruch 1, bei der das Ventilelement ein stopfenförmiges Element (35) ist, welches konische Ventilflächen (41, 43) aufweist, die tatsächlich einen solchen Abstand voneinander haben, daß sie sich gleichzeitig passend an die Sitze (37, 39) anlegen und synchron öffnen.

3. Ventilanordnung nach Anspruch 1, bei der das rohrförmige Element eine rohrförmige Buchse (63) umfaßt und bei der das Ventilelement einen Kolben (65) umfaßt, der innerhalb der Buchse (63) tatsächlich gleitverschieblich ist, wobei die Buchse (63) und der Kolben (65) Ventilflächen haben, die ein erstes Paare von fluchtend ausrichtbaren Ventilnuten (67, 73) und tatsächlich im Abstand von diesem ersten Paar eine zweites Paar von fluchtend ausrichtbaren Ventilnuten (69, 71) aufweisen, wobei diese Ventilnuten fluchtend ausgerichtet werden, um sich synchron zueinander zu öffnen;

und bei dem in dem Kolben (65) ein erster Kanal (77) von der ersten Ventilnut (73) zu einem Ende des Kolbens (65) und ein zweiter Kanal (70) von der zweiten Ventilnut (71) vorgesehen ist, der sich zum anderen Ende des Kolbens (65) öffnet, und wobei in der Buchse (63) Kanäle (78, 79) vorhanden sind, die jedes Paar von Ventilnuten mit einer Ablauföffnung (82, 84) verbinden.

4. Ventilanordnung nach Anspruch 2, bei der das rohrförmige Element (33) und das stopfenförmige Element (35) zwischen sich eine Kammer (47) bilden, welche tatsächlich zwischen den Ventilsitzen (37, 39) angeordnet ist und bei der sich die Ablauföffnung (49) in diese Kammer (47) öffnet, wodurch eine hydraulische Strömung in einer Richtung, die die Tendenz hat, die Ventilflächen (37, 41, 39, 43) in ihre Offenstellung zu bringen, wenn diese Ablauföffnung (49) geschlossen wird, durch das rohrförmige Element (33) weiterfließen wird, während eine hydraulische Strömung, welche in dieser einen Richtung in das rohrförmige Element (33) hineinfließt, dann wenn diese Ablauföffnung (49) offen ist, eine kontrollierte Strömung in der entgegengesetzten Richtung erzeugt, wobei beide Strömungen durch die offene Ablauföffnung (49) austreten.

5. Ventilanordnung nach Anspruch 3, bei der an einem Ende der Buchse (63) ein Ventilsitz (85) vorgesehen ist, und bei der ein Ventilelement (87) vorgesehen ist, welches passend an den Ventilsitz (85) anlegbar ist, um diesen zu schließen, wobei der Kolben (65) ein Teilstück (91) umfaßt, welches während er Bewegung des Kolbens (65) das Ventilelement (87) erfaßt, um es im wesentlichen zu dem gleichen Zeitpunkt, zu dem die fluchtend ausrichtbaren Ventulnuten (67, 73, 79, 81) öffnen, von seinem Ventilsitz (85) abzuheben.

6. Ventilanordnung nach einem der Ansprüche 1 bis 5, dadurch gekennzeichnet, daß das Steuerventil Teil eines hydraulischen System ist, welches umfaßt:

eine mit dem Arbeitsdruck beaufschlagbare Arbeitsvorrichtung (11) mit variablem Volumen;

einen Sumpf (19);

eine in einer Richtung arbeitende Pumpe zum Ansaugen eines Fluids aus dem Sumpf (19) und zum Liefern von unter dem Fluiddruck stehendem Fluid;

ein Druckabbauventil (51);

wobei das Steuerventil (31; 139, 141) zwischen dem Sumpf und der Arbeitsvorrichtung angeordnet ist, derart, daß sein erstes und sein zweites Paar von fluchtend ausrichtbaren Ventilflächen (37, 41, 39, 43; 69, 71, 67, 73), die mechanisch gekoppelt sind, derart angeordnet sind, daß sie synchron öffnen, wenn der von der Pumpe (21) gelieferte Fluiddruck den Druck an der Arbeitsdrucköffnung übersteigt, wobei die beiden Paare von Ventilflächen in Serie zwischen der Pumpe und der Arbeitsvorrichtung (11) liegen und wobei die Ablauföffnung zwischen den beiden Paaren von Ventilflächen außerdem über das Druckabbauventil (51) mit dem Sumpf (19) verbunden sind, wodurch beim Schließen des Druckabbauventils (51) ein Arbeiten der Pumpe (51) zu einer Ausdehnung des Volumens der Arbeitsvorrichtung führt, während ein Arbeiten der Pumpe (21) bei geöffnetem Druckabbauventil (51) zu einer Verringerung des Volumens der Arbeitsvorrichtung führt.

7. Ventilanordnung nach einem der Ansprüche 1 bis 5, bei der das Steurventil Teil eines hydraulischen Systems ist, welches umfaßt:

einen Fluidspeicher (101);

eine in zwei Richtungen arbeitende Pumpe (100);

eine Arbeitsvorrichtung (11) mit variablem Fluidvolumen;

wobei das Steuerventil (31; 139, 141) als Steuerventil zur Strömungsanpassung dient, bei dem eine hydraulische Strömung in die Zulauföffnung eine kontrollierte Strömung in die Arbeitsdrucköffnung bewirkt, wobei beide Strömungen durch die Ablauföffnung austreten; und ferner umfassend:

Einrichtungn, die den Flüssigkeitsvorrat (101) jeweils über Rückschlagventile (102, 103) mit beiden Seiten der Pumpe (100) verbinden und eine Strömung von dem Vorrat (101) zu der Pumpe (100) ermöglichen;

Einrichtung, die die eine Seite der Pumpe (100) mit der Zulauföffnung des Steuerventils (31; 139, 141) verbinden;

Einrichtungen, die die Arbeitsdrucköffnung des Steuerventils (31; 139, 141) mit der Arbeitsvorrichtung (11) verbinden;

Einrichtungen, die die Ablauföffnung des Steuerventils (31; 139, 141) mit dem Vorrat (101) verbinden; und

Einrichtung, die die andere Seite der Pumpe (100) mit der Arbeitsvorrichtung (11) über ein Rückschlagventil (109) verbinden, welches eine Strömung von der Pumpe (100) zu der Arbeitsvorrichtung (11) gestattet.

8. Ventilanordnung nach einem der Ansprüche 1 bis 5, bei der ein Paar der Steuerventile Teil eines hydraulischen Systems ist, welches umfaßt:

eine in zwei Richtungen arbeitende Pumpe (127);

wobei die Steuerventile (139, 141) als Steuerventile zur Strömungsanpassung dienen, von denen jedes eine Zulauföffnung, eine Arbeitsdrucköffnung und zwei Ablauföffnungen hat, wodurch eine hydraulische Strömung in die Zulauföffnung zu einer kontrollierten Strömung in die Arbeitsdrucköffnung führt und wobei die Strömungen durch betreffende Ablauföffnungen abfließen; und ferner umfassend:

Einrichtungen, welche jede Seite der Pumpe (127) mit der Zulauföffnung jeweils eines der Steuerventile (139, 141) verbinden;

Einrichtungen, die die Arbeitsdrucköffnung jedes Steuerventils (139, 141) mit jeweils einer der Zylinderöffnungen verbinden;

Einrichtungen, die die Arbeitsdruck/Ablauf-Öffnung jedes Steuerventils (139, 141) mit der Zulauföffnung des jeweils anderen Steuerventils (139, 141) verbinden; und

Rückschlagventileinrichtungen (145, 147), welche die Zulauf/Ablauf-Öffnung jedes Steuerventils (139, 141) mit der Arbeitsdrucköffnung des jeweils anderen Steuerventils verbinden.

Revendications

1. Ensemble valve d'alimentation et d'évacuation automatique comprenant au moins une valve

de commande (31; 139, 141) comprenant un élément tubulaire (33; 63) avant un orifice de source un orifice de charge et au moins un orifice de drainage (49; 82, 84) et comprenant on outre un élément obturateur (35; 65) manoeuvrable dans deux sens opposés, ledit élément tubulaire (33; 63) et ledit élément obturateur (35; 65) ayant une première et une deuxième paire de surfaces obturatrices conjugées (37, 41, 39, 43; 69, 71, 67, 73), espacées axialement, qui sont mécaniquement couplées et agencées pour relier ledit orifice de source audit orifice de charge lorsque la pression de fluide fournie à l'orifice de source excède la pression à l'orifice de charge, et pour relier ledit orifice de charge audit orifice de drainage (49; 82, 84) de manière à relâcher la pression audit orifice de charge, caractérisé en ce que lesdites paires de surfaces obturatrices (37, 41, 39, 43; 69, 71, 67, 73) sont agencées de manière à ouvrir et fermer en synchronisme, et en ce que ledit orifice de drainage (49; 82, 84) est prévu dans ledit élément tubulaire (33; 63) à une position intermédiaire entre lesdits orifices de source et de charge, de manière à autoriser des écoulements appariés simultanés, provenant dudit orifice de charge et dudit orifice de source, allant audit orifice de drainage au nombre d'au moins un (49; 82, 84) lorsque lesdites surfaces obturatrices sont ouvertes, et de manière à être déconnecté desdits orifices de source et de charge lorsque lesdites surfaces obturatrices (37, 41, 39, 43; 69, 71, 67, 73) sont fermées.

2. Ensemble valve selon la revendication 1, dans lequel ledit élément obturateur est un organe obturateur (35) incluant des surfaces obturatrices coniques (41, 43) qui sont effectivement espacées de manière à se conjuguer simultanément auxdits sièges (37, 39) et à ouvrir en synchronisme.

3. Ensemble de commande selon revendication 1, dans lequel ledit élément tubulaire comprend un manchon tubulaire (63) et dans lequel ledit élément obturateur comprend un piston (65) pouvant coulisser dans ledit manchon (63), ledit manchon (63) et ledit piston (65) ayant des surfaces obturatrices qui comprennent une première paire de gorges à effet de valve conjugées (67, 73) et, décalée par rapport à ladite première paire, une deuxième paire de gorges à effet de valve conjuguées (69, 71), lesdites gorges à effet de valve étant conjugées pour ouvrir l'une vers l'autre en synchronisme;

et dans lequel il y a, dans ledit piston (65), un premier passage (77) allant de ladite première gorge à effet de valve (73) à une extrémité dudit piston (65) et un deuxième passage (70) ouvrant de ladite deuxième gorge à effet de valve (71) jusqu'à l'autre extrémité dudit piston (65), et dans lequel il y a, dans ledit manchon (63), des passages (78, 79) reliant chaque paire desdites gorges à effet de valve à un orifice de drainage (82, 84).

4. Ensemble valve selon revendication 2, dans lequel ledit élément tubulaire (33) et ledit élément obturateur (35) établissent entre eux une chambre (47) qui est située entre lesdits sièges de valve (37, 39), et dans lequel ledit orifice de drainage (49) s'ouvre dans ladite chambre (47), ce qui a pour effet que, lorsquefledit orifice de drainage (49) est fermée, le courant hydraulique dans un sens tendant à ouvrir lesdites surfaces obturatrices (37, 41, 39, 43) passera par ledit élément tubulaire (33) et que, lorsque ledit orifice de drainage (49) sera ouvert, un écoulement hydraulique dans lesdits éléments tubulaires (33) dans ledit sens produira un écoulement commandé dans le sens opposé, less deux écoulements sortant par ledit orifice de drainage ouvert (49).

5. Ensemble valve selon revendication 3, dans lequel il y a, à une extrémité dudit manchon (63), un siège de valve (85), et dans lequel un élément obturateur (87) est prévu qui est adapté pour se conjuguer audit siège de valve (85) et le fermer, ledit piston (65) incluant une partie (91) qui, au cours du mouvement du piston (65), recontre ledit élément obturateur (87) pour le décoller dudit siège (85), sensiblement un même temps que lesdites gorges à effet de valve conjugées (67, 73, 79, 81) ouvrent.

6. Ensemble valve selon l'une des revendications 1 à 5, dans lequel ladite valve de commande fait partie d'un système hydraulique comprenant:

une charge à volume variable (11);

un bac de dégorgement (19);

une pompe unidirectionnelle pour tirer du fluide dudit bac de dégorgement (19) et fournir du fluide sous ladite pression de fluide; une valve de libération (51);

ladite valve de commande (31; 139, 141) étant agencée entre ledit bac de dégorgement et ladite charge de manière que sa première et sa deuxième paire conjugées de surfaces à effet de valve (37, 41, 39, 43; 69, 71, 67, 73), qui sont couplées mécaniquement et sont agencées pour ouvrir en synchronisme lorsque ladite pression de fluide fournie par ladite pompe (21) excède la pression audit orifice de charge, et qui sont reliées en série entre ladite pompe et ladite charge (11), et ledit orifice de drainage entre les deux pairs de surfaces à effet de valve étant relié aussi via ladite valve de libération (51) audit bac (19), ce qui a pour effet que, lorsque ladite valve de libération (51) est fermée, le fonctionnement de ladite pompe (21) va augmenter le volume de la charge et, lorsque ladite valve de libération (51) sera ouverte, la fonctionnement de ladite pompe (21) contractera ledit volume de la charge.

7. Ensemble valve selon l'une des revendications 1 à 5 dans lequel ladite valve de commande fait partie d'un système hydraulique comprenant:

un réservoir de fluide (101);

une pompe bidirectionnelle (100);

une charge à volume variable (11);

ladite valve de commande (31; 139, 141) servant de valve de commande d'appariement d'écoulement, dans laquelle un écoulement hydraulique dans l'orifice de source produit un écoulement contrôlé dans l'orifice de charge, les deux écoulements sortant par l'orifice de drainage; et comprenant en outre:

des moyens reliant ledit réservoir (101) aux deux côtés de ladite pompe (100), via des valves de retenue respectives (102, 103) permettant l'écoulement du réservoir (101) à ladite pompe (100); des moyens reliant un côté de ladite pompe (100) à l'orifice de source de ladite valve de commande (31; 139; 141);

des moyens connectant l'orifice de charge de ladite valve de commande (31; 139, 141) à ladite charge (11);

des moyens reliant l'orifice de drainage de ladite valve de commande (31; 139, 141) audit réservoir (101); et

des moyens reliant l'autre côté de ladite pompe (100) à ladite charge (11) via une valve de retenue (109) autorisant l'écoulement de ladite pompe (100) à ladite charge (11).

8. Ensemble valve selon l'une des revendications 1 à 5, dans lequel une paire de dites valves de commande fait partie d'un système hydraulique comprenant:

une pompe bidirectionnelle (127);

lesdites valves de commande (139, 141) servant de valves de commande

et appariement d'écoulement ayant chacune un orifice de source, un orifice de charge et une paire d'orifices de drainage, dans lequel un écoulement hydraulique dans l'orifice de source produit un écoulement contrôlé dans l'orifice de charge, et les écoulements sortent par des orifices de drainage respectifs; et comprenant en outre

des moyens reliant chaque côté de ladite pompe (127) à l'orifice de source d'une valve de commande respective (139, 141);

des moyens reliant l'orifice de charge de chaque valve de commande (139, 141) à l'un desdits orifices de cylindre respectifs;

des moyens reliant l'orifice de drainage de charge de chaque valve de commande (139, 141) à l'orifice de source de l'autre valve de commande (139, 141); et

des moyens de valve de retenue (145, 147) reliant l'orifice de drainage de source de chaque valve de commande (139, 141) à l'orifice de charge de l'autre valve de commande.

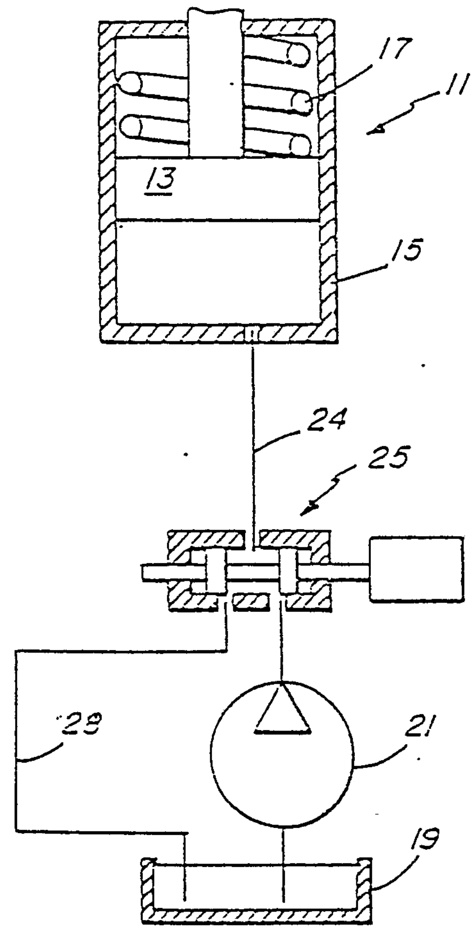

FIG. 1
PRIOR ART

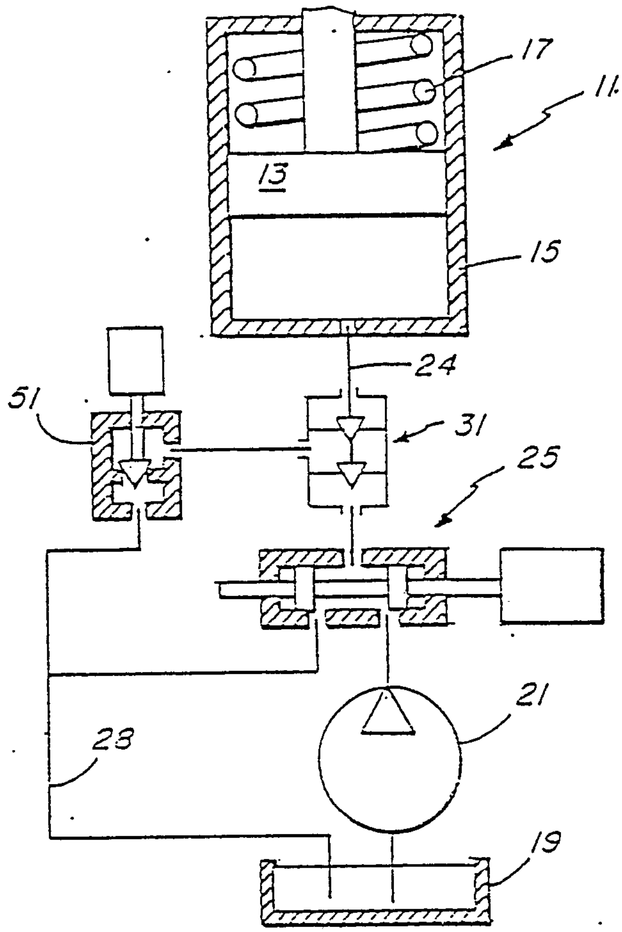

FIG. 3

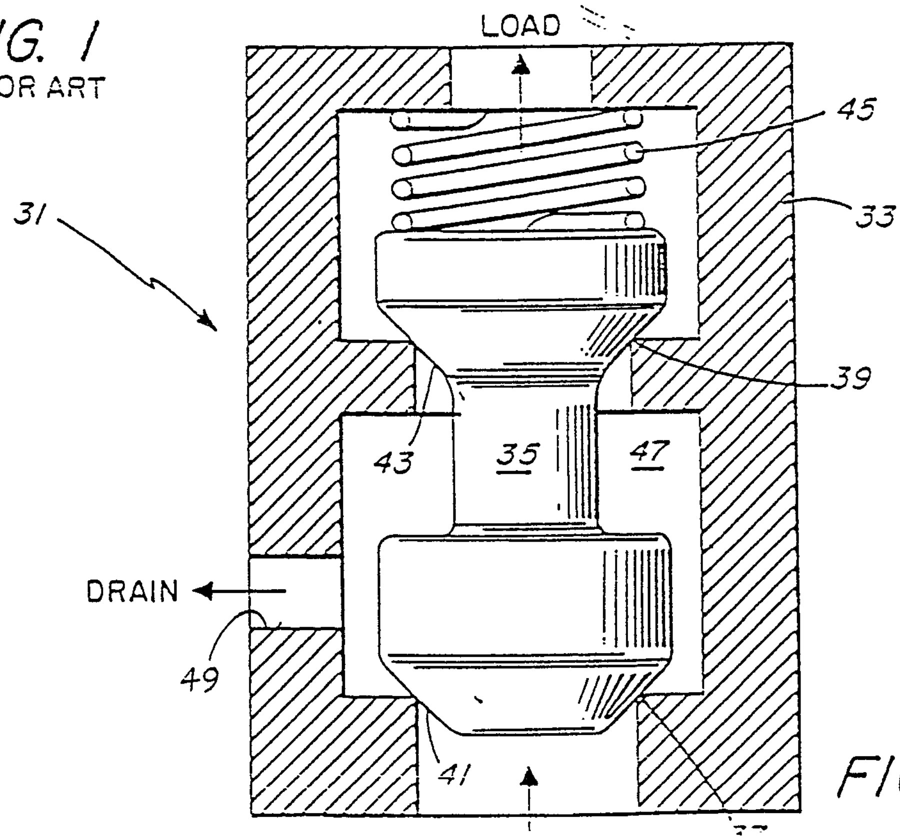

FIG. 2

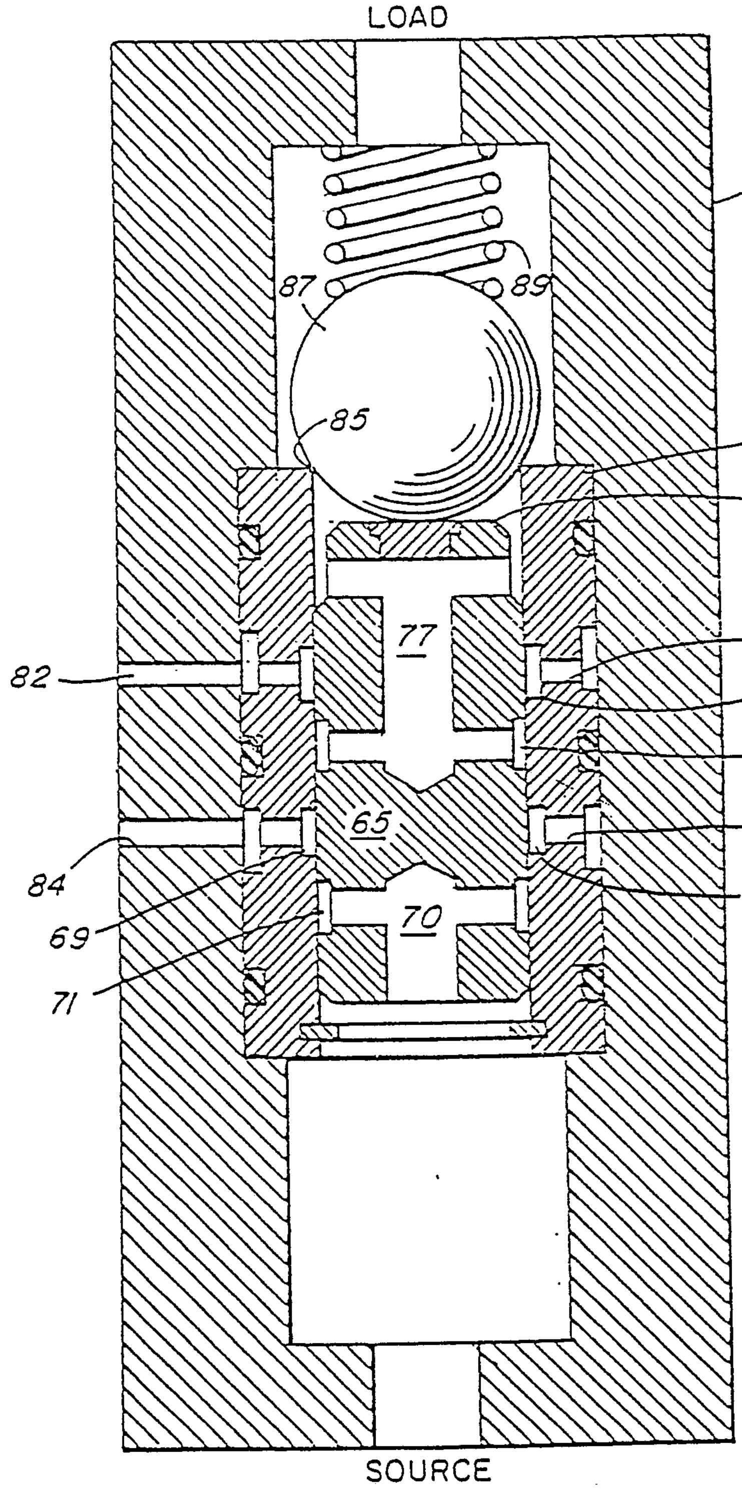

FIG. 4

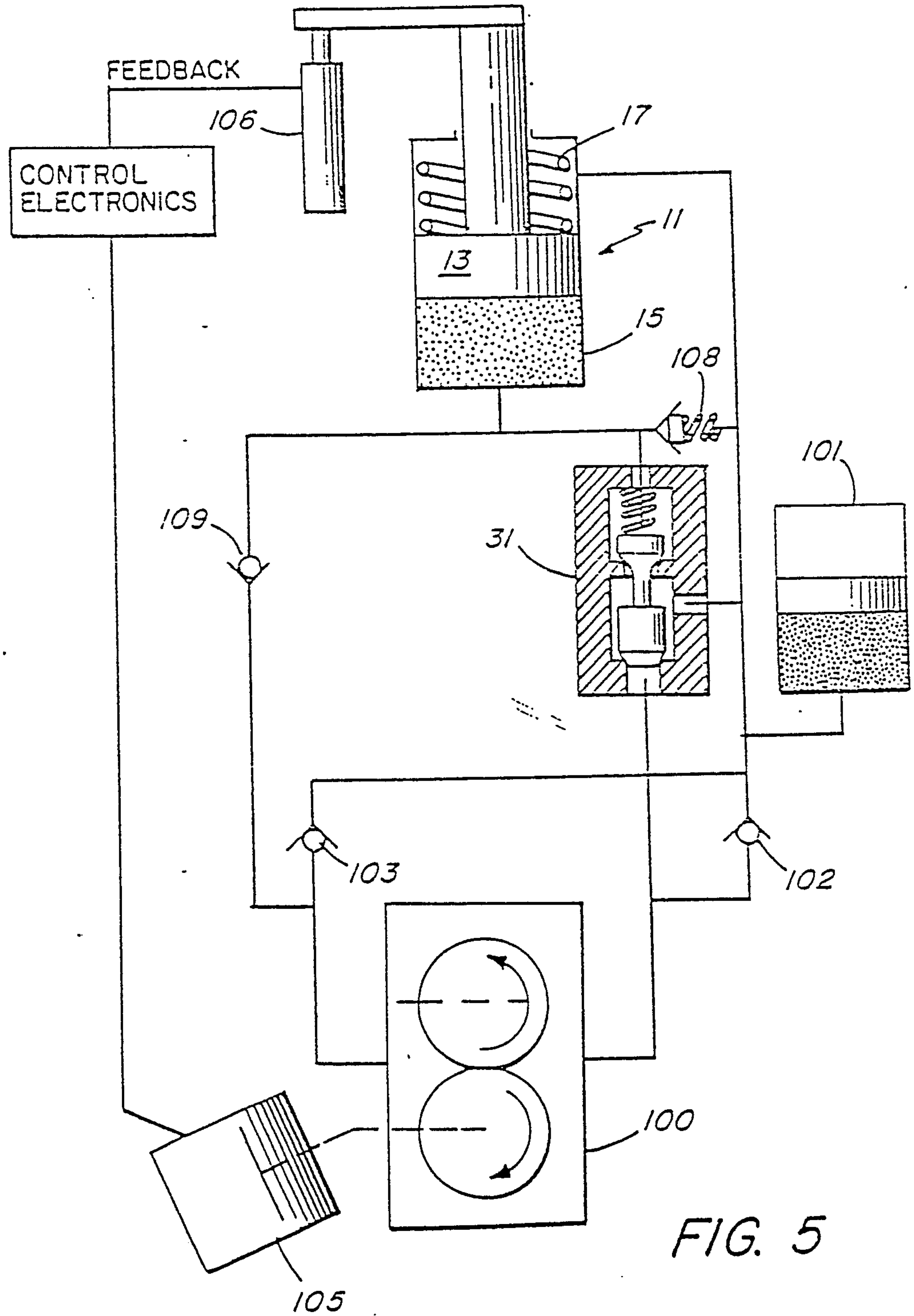
FEEDBACK
CONTROL ELECTRONICS
106
17
11
13
15
108
101
109
31
103
102
100
105

FIG. 5

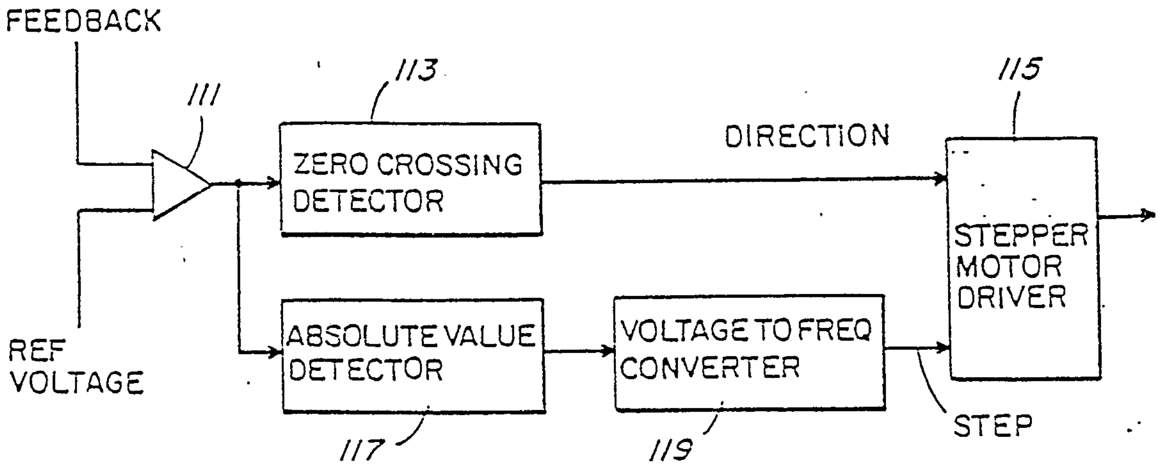
FEEDBACK
111
113
ZERO CROSSING DETECTOR
DIRECTION
115
STEPPER MOTOR DRIVER
REF VOLTAGE
ABSOLUTE VALUE DETECTOR
VOLTAGE TO FREQ CONVERTER
STEP
117
119

FIG. 6

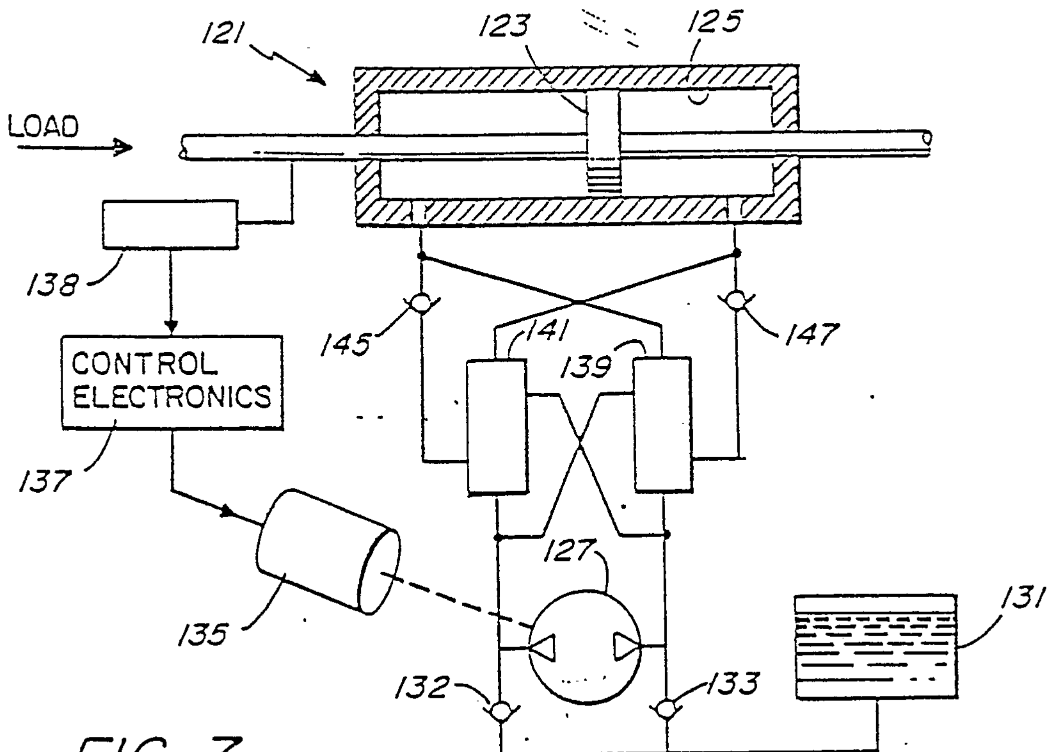
121
123
125
LOAD
138
CONTROL ELECTRONICS
137
145
141
139
147
127
135
131
132
133

FIG. 7