

[54] **LOAD RESPONSIVE CONTROL VALVE**[76] Inventor: **Tadeusz Budzich**, 80 Murwood Dr.,
Moreland Hills, Ohio 44022[22] Filed: **July 6, 1973**[21] Appl. No.: **377,044****Related U.S. Application Data**[63] Continuation-in-part of Ser. No. 185,146, Sept. -30,
1971, Pat. No. 3,744,517.[52] U.S. Cl. **137/596.1; 91/446; 91/451**[51] Int. Cl. **F16k 11/00**[58] Field of Search..... 137/596.2, 504, 596.1,
137/596, 625.3, 613, 501, 503, 115, 116,
117; 91/411 R, 411 A, 411 B, 445, 446, 420,
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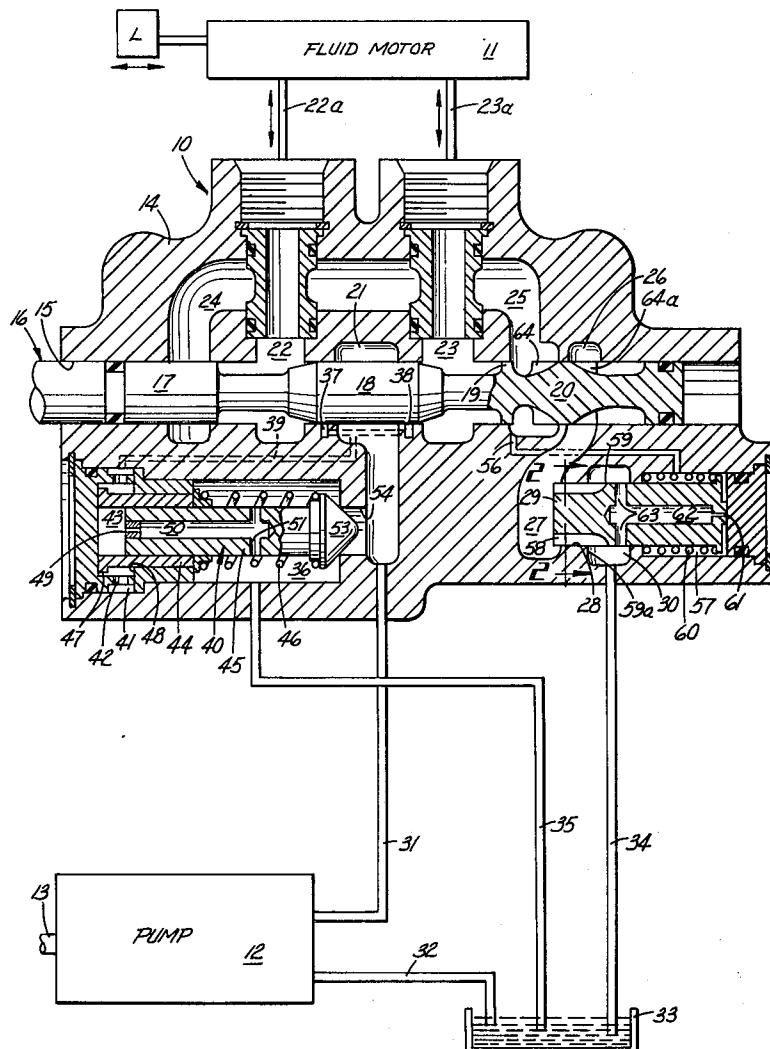
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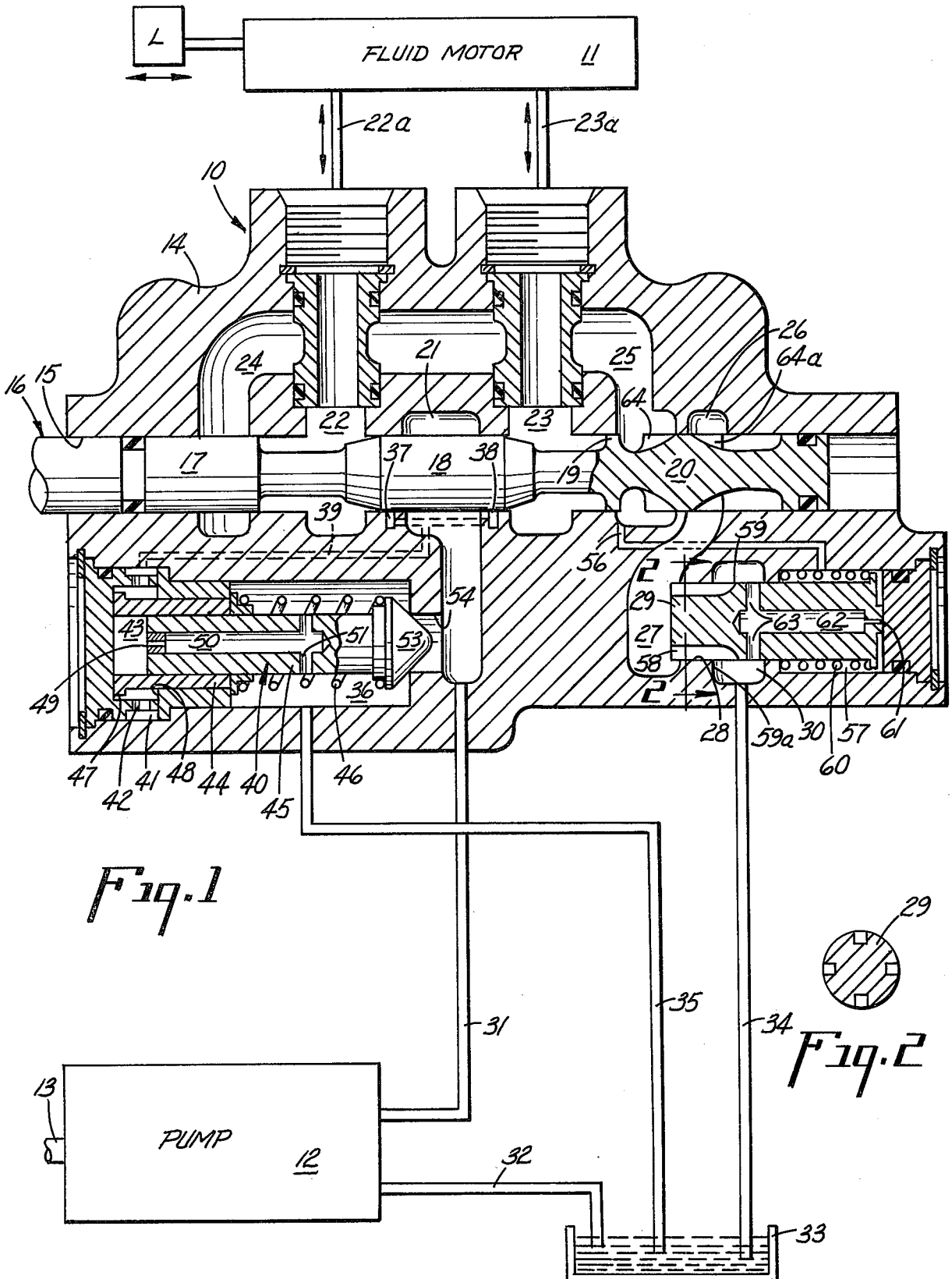
Primary Examiner—Martin P. Schwadron

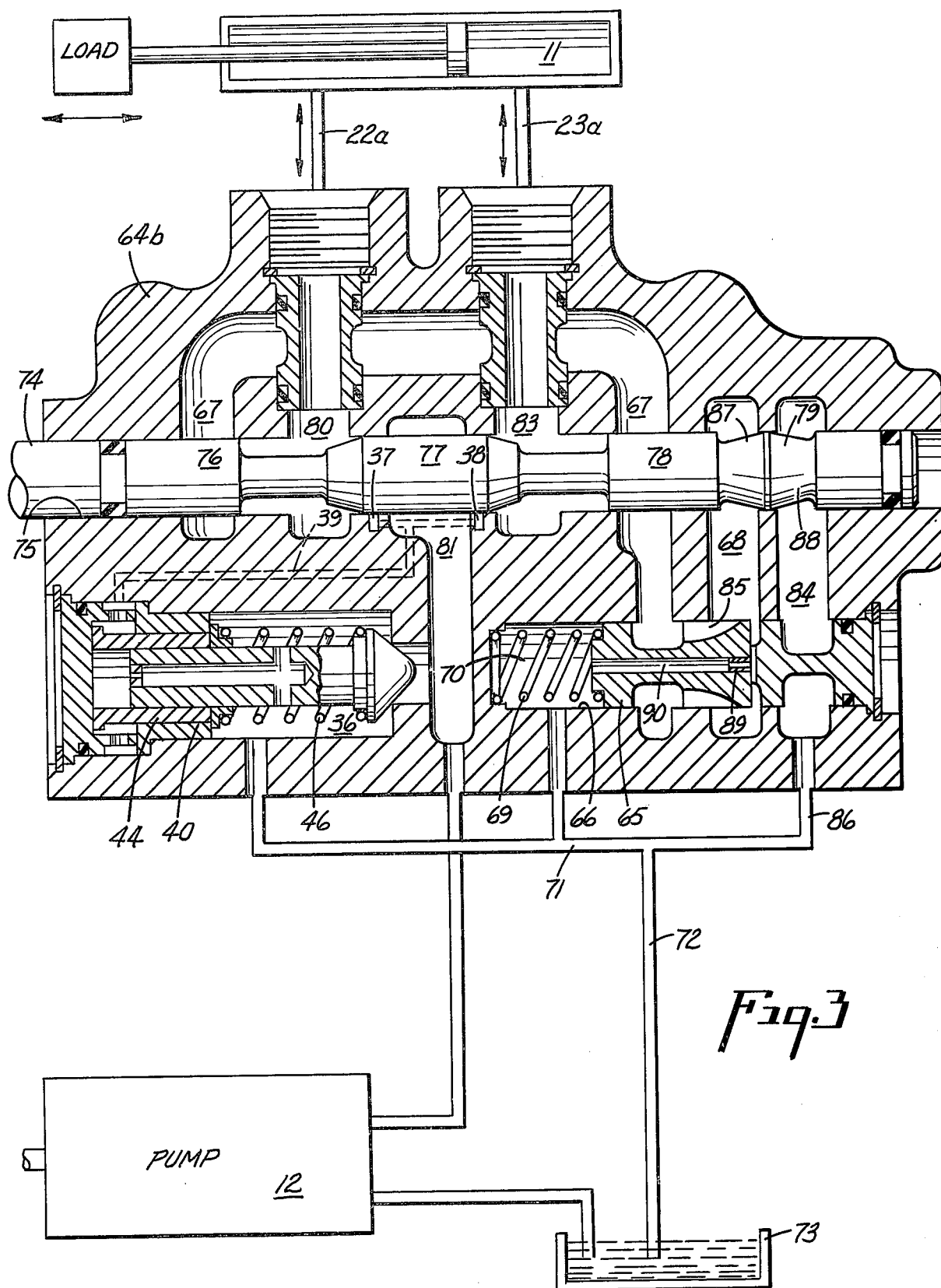
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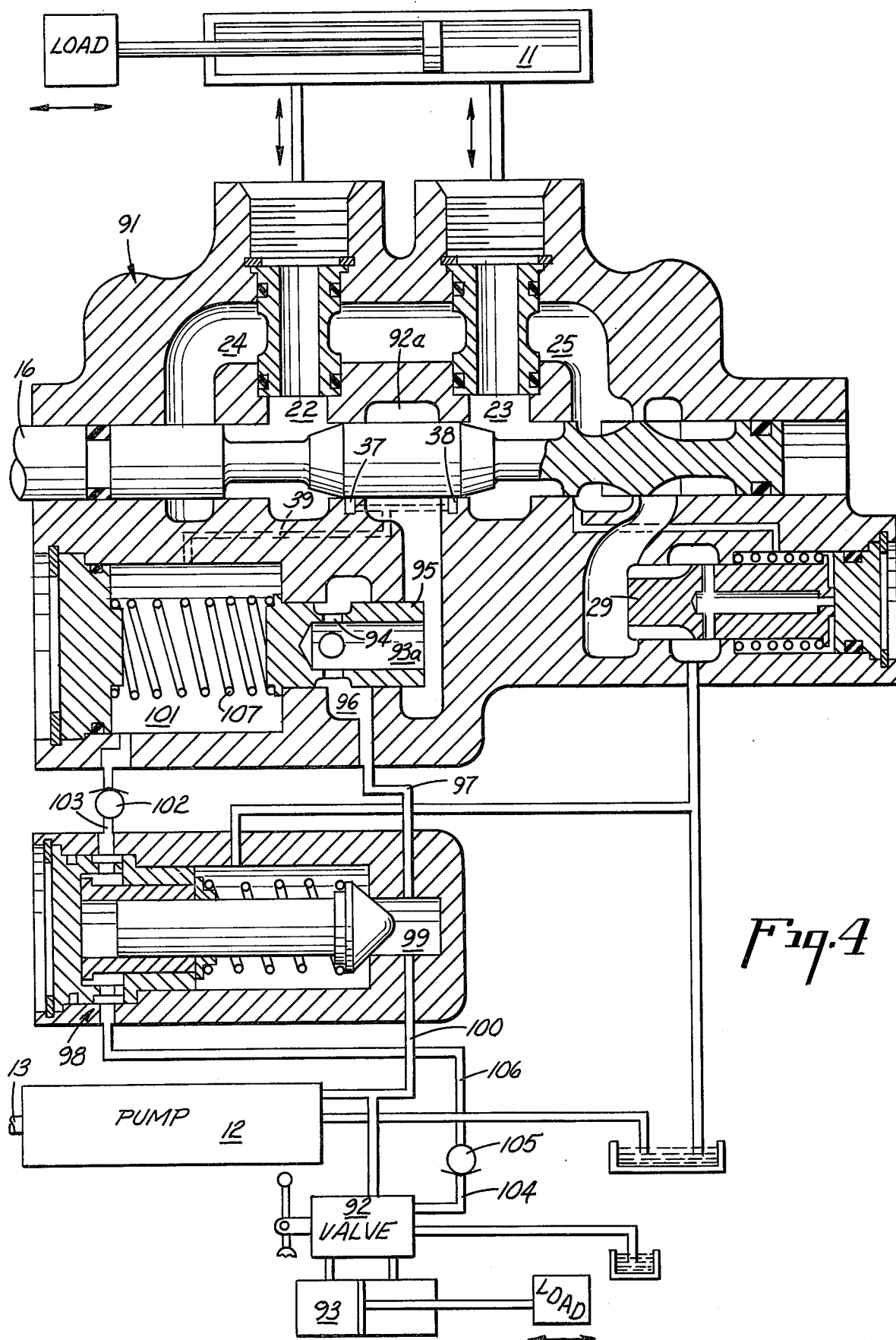
[57] **ABSTRACT**

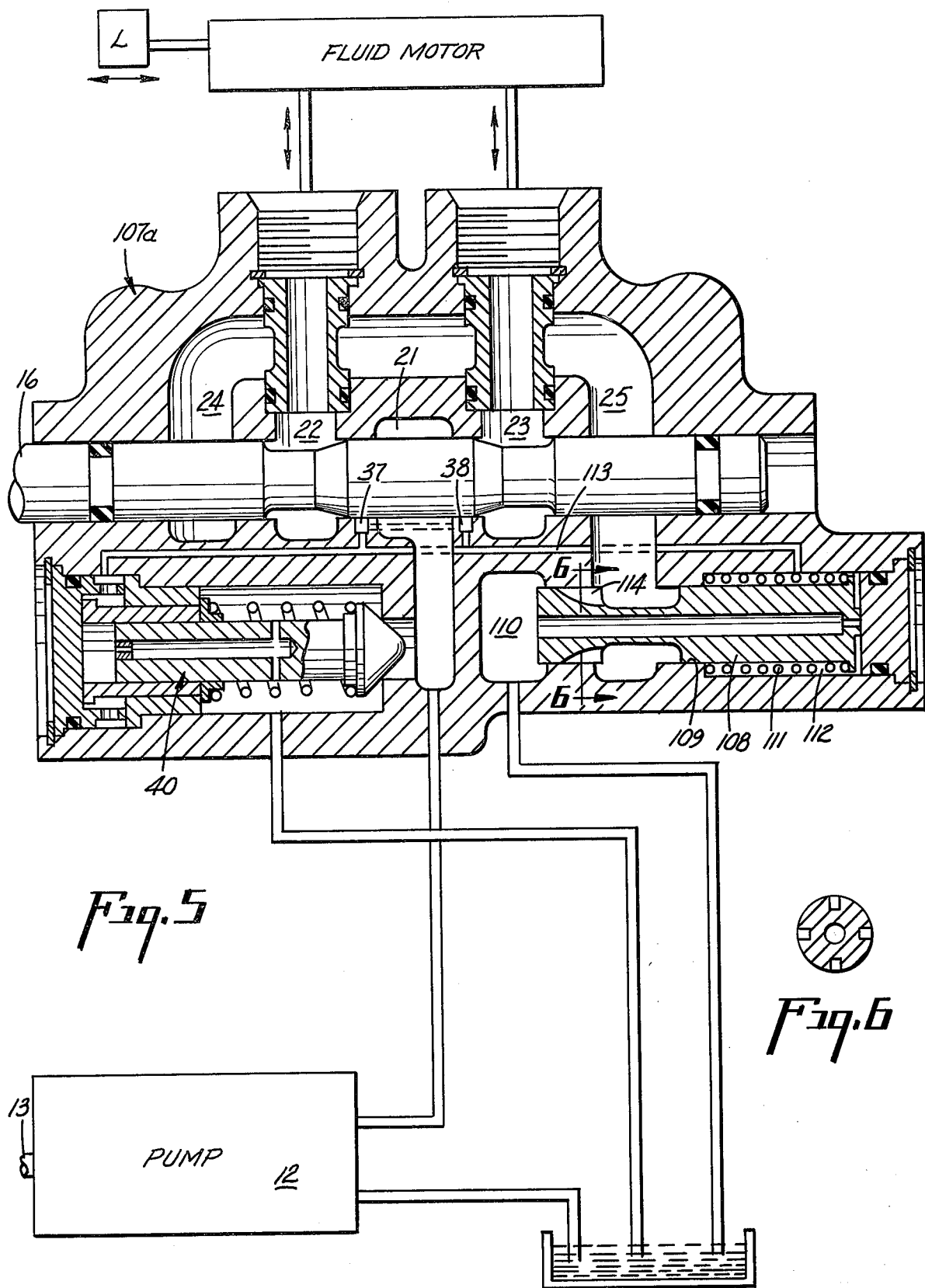
A direction and flow control valve for use in a central fluid power load responsive system having plurality of loads. The valve maintains a selected constant flow level for control of both positive and negative loads, irrespective of the change in the load magnitude or change in the fluid pressure, supplied to the valve. The system is powered by a single, fixed volume pump. The direction flow control valve is equipped with a load responsive control, which automatically maintains the pump discharge pressure at a level higher than the pressure required by the system's largest load. Each direction and flow control valve is equipped with a differential pressure regulating control, which is responsive to the load pressure for controlling the negative loads and preferably a second differential pressure regulating control for controlling the speed of positive loads which are smaller than the system's largest load.

27 Claims, 6 Drawing Figures









LOAD RESPONSIVE CONTROL VALVE

This is a continuation in part of application Ser. No. 185,146, filed Sept. 30, 1971 now U.S. Pat. No. 3,744,517, for "Load Responsive Fluid Control Valves".

BACKGROUND OF THE INVENTION

This invention relates generally to load responsive fluid control valves and to fluid power systems incorporating such valves which systems are supplied by a single fixed displacement pump. Such control valves are equipped with an automatic load responsive control, and can be used in a multiple load system, in which a plurality of loads is individually controlled under positive and negative load conditions by separate control valves.

In more particular aspects this invention relates to direction and flow control valves capable of controlling simultaneously a number of loads under both positive and negative conditions.

Closed center load responsive fluid control valves are very desirable for a number of reasons. They permit load control with reduced power losses and therefore, increased system efficiency and when controlling one load at a time provide a feature of flow control, irrespective of the variation in the magnitude of the load. Such valves include a load responsive control which automatically maintains pump discharge pressure at a level higher, by a constant pressure differential, than the pressure required to sustain the load. A variable orifice, introduced between the pump and the load, varies the flow supplied to the load, each orifice area corresponding to a different flow level, which is maintained constant irrespective of any variation in the magnitude of the load. The application of such a system is, however, limited by two basic system disadvantages. The valve control can maintain a constant pressure differential and therefore constant flow characteristics when operating only one load at a time. With two or more loads, simultaneously controlled, only the highest of the loads will retain the flow control characteristics, the speed of actuation of the lower loads varying with the change in magnitude of the highest load. This drawback can be overcome in part by the provision of a proportional valve as disclosed in my U.S. Pat. No. 3,470,694, dated Oct. 7, 1969. However, while this valve is effective in controlling positive loads it does not retain flow control characteristics when controlling negative loads, which instead of taking, supply the energy to the fluid system, and hence the speed of actuation of such a load in a negative load system will vary with the magnitude of the negative load. Especially with so-called overcenter loads where a positive load may become a negative load, such a valve will lose its speed control characteristics in the negative mode.

Load responsive flow control valves are disclosed in my pending patent application Ser. No. 185,146 now U.S. Pat. No. 3,744,517. While those valves are effective in controlling negative loads they require close synchronization of valve spool timing when controlling both negative and positive loads.

SUMMARY OF THE INVENTION

It is therefore a principal object of this invention to provide an improved load responsive fluid valve which will retain flow control characteristics when controlling both positive and negative loads.

It is another object of this invention to provide improved load responsive fluid valve which can control a multiplicity of positive and negative loads.

It is a further object of this invention to provide an improved load responsive fluid valve which converts the energy of negative load by throttling, automatically maintaining a constant pressure difference between the load pressure and the valve outlet pressure.

It is a further object of this invention to provide an improved, load responsive fluid valve which converts the energy of negative load by throttling while maintaining a constant pressure differential across a metering land irrespective of the direction of the operation of negative load.

Briefly the foregoing and other additional objects and advantages of this invention are accomplished by providing a novel load responsive flow control valve, constructed according to the present invention, for use in load responsive hydraulic systems. A load responsive flow control valve is positioned between a fixed displacement pump and each motor. Each valve has an automatic valve outlet throttling section and inlet bypass control section; and, when a plurality of valves are used, each also has an automatic inlet throttling section. Since the inlet bypass control section can only maintain a flow proportionality when one load is controlled at a time and since it cannot control flow proportionality in control of negative load, these valve inlet and outlet automatic throttling controls provide a constant pressure differential across the valve spool, permitting retention of flow control characteristics, with simultaneous control of all loads both positive and negative.

Additional objects of the invention will become apparent when referring to the preferred embodiments of the invention as shown in the accompanying drawings and described in the following detailed description.

DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal sectional view of one embodiment of a flow control valve including the control mechanism used in control of negative loads and bypass control used in control of positive loads with system lines, pump and reservoir shown diagrammatically.

FIG. 2 is a sectional view taken substantially along the plane designated by the line 2—2 of FIG. 1;

FIG. 3 is a longitudinal sectional view of flow control valve of FIG. 1 including another embodiment of a control mechanism used in control of a negative load with pump, power lines and reservoir shown diagrammatically;

FIG. 4 is a longitudinal sectional view of another embodiment of a flow control valve including two control mechanisms used in control of multiple positive and negative loads with additional valve, pump, and power lines shown diagrammatically;

FIG. 5 is a longitudinal sectional view of still another embodiment of flow control valve; and

FIG. 6 is a sectional view taken substantially along the plane designated by the line 6—6 of FIG. 5.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to the drawings, and for the present to FIG. 1, one embodiment of a flow control valve, generally designated as 10, is shown interposed between diagrammatically shown fluid motor 11 driving a load L and

a fixed flow pump 12. The fixed flow pump 12 is driven through shaft 13 by a suitable prime mover, not shown.

The flow control valve 10 is a four-way type and has a housing 14 provided with a bore 15, axially guiding a valve spool designated generally as 16. The valve spool 16 is equipped with lands 17, 18, 19 and 20, which in the position shown will isolate from each other fluid inlet chamber 21, load chambers 22 and 23 and outlet chambers 24 and 25 and first exhaust chamber 26 formed in the housing 14. The first exhaust chamber 26 is cross-connected through passage 27 and bore 28 and guiding control spool 29, to a second exhaust chamber 30.

The outlet of the pump 12 is connected through discharge line 31 to inlet chamber 21. The inlet of pump 12 is connected through line 32 to diagrammatically shown reservoir 33. Reservoir 33 is also connected by line 34 to the second exhaust chamber 30 and by line 35 to exhaust space 36 formed in the housing 14. Pressure sensing passages 37 and 38 communicate with the bore 15 between inlet chamber 21 and load chambers 22 and 23 respectively and are blocked by land 18 of the valve spool 16 in its neutral position, as shown in FIG. 1. The pressure sensing passages 37 and 38 are connected through passages 39 to differential pressure relief valve generally designated as 40. Movement of the valve spool 16 to the right, from the position as shown, will connect first the pressure sensing passage 37 to the load chamber 22 and then connect the load chamber 22 with the inlet chamber 21. Movement of the control spool 16 to the left will first connect the pressure sensing passage 38 to the load chamber 23 and then connect the load chamber 23 with the inlet chamber 21.

The differential pressure relief valve 40 bypasses pump flow from inlet chamber 21 to the exhaust space 36 to regulate the pump discharge pressure in response to the load pressure signal transmitted through passage 39. In absence of any signal, corresponding to the blocked position of sensing passages 37 and 38 as shown in FIG. 1, the differential pressure relief valve 40 automatically diverts all of the flow of pump 12 to exhaust space 36 by virtue of the pressure opening the valve 40 against the bias of spring 46, and therefore maintaining inlet chamber 21 at a minimum preselected pressure level corresponding to the preload in spring 46, thus operating at minimum standby power loss. Movement of the land 18 of valve spool 16 to right will connect pressure sensing passage 37 to load chamber 22 and transmit the load pressure signal through passage 39 to an annular chamber 41 of differential pressure relief valve 40.

From the annular chamber 41 the pressure signal is transmitted through passage 42 to control chamber 43 where it acts on the cross-sectional area of force sleeve 44 and control plunger 45. Force generated by pressure signal on force sleeve 44 will move it from left to right compressing spring 46 until stop 47 of force sleeve 44 engages surface 48. Fluid in control chamber 43 is connected through leakage orifice 49 and passages 50 and 51 with exhaust space 36. Control plunger 45 is equipped with conical head 53 which in the modulating position creates a bypass orifice cross-connecting opening 54 and exhaust space 36. The cross-sectional area of opening 54 is made the same as cross-sectional area of control plunger 45. Control plunger 45 is subjected to a control signal pressure in control chamber 43 and

force of spring 46 in direction to maintain conical surface 53 in contact with opening 54 and is subjected to pressure in inlet chamber 21 which creates a force in a direction to move conical surface 53 away from opening 54 and therefore to create a flow passage between inlet chamber 21 and exhaust space 36. Subjected to these forces the control plunger 45 will control the bypass flow of fluid from pump 12 to exhaust space 36, to maintain the inlet chamber 21 at a pressure higher than pressure in load chamber 22, the difference between these pressures being always constant and proportional to preload in spring 46. Therefore with the force sleeve 44 in position as shown in FIG. 1, equivalent to minimum preload in spring 46, the constant pressure difference between inlet chamber 21 and load chamber 22 will be maintained constant by the differential pressure relief valve 40 at minimum level. With the force sleeve 44 fully out, engaging with stop 47 surface 48 and therefore with preload in the spring 46 at maximum level, the constant pressure differential between inlet chamber 21 and load chamber 22 will be maintained constant at maximum level. The differential pressure relief valve 40 will then modulate to automatically adjust the bypass flow from pump 12 to exhaust space 36 to maintain the pump discharge pressure and therefore pressure in the inlet chamber 21 at a level higher, by a fixed pressure differential, than the load signal pressure in the load chamber 22.

Further movement of land 18 to the right will connect inlet chamber 21 to load chamber 22, resulting a fluid flow from chamber 21 through line 22a to the fluid motor. Since the pressure in chamber 21 is maintained at a fixed pressure differential with respect to pressure in load chamber 22 by the differential pressure relief valve the volume of flow per unit time will be proportional to the area of the orifice opened between chambers 21 and 22 and constant for each specific area, irrespective of the actual pressure level in the load chamber 22. Since the area of the orifice between chambers 21 and 22 is proportional to the travel of the valve spool 16, the fluid flow from inlet chamber 21 to load chamber 22 will also be proportional to the displacement of the valve spool 16, each specific position of the valve spool 16 corresponding to a specific constant fluid flow level, irrespective of the magnitude of the pressure level required to operate the load.

In a similar fashion, with movement of the valve spool 16 in the opposite direction, the fluid flow from inlet chamber 21 to load chamber 23 is controlled and supplied to the opposite side of the fluid motor through line 23a.

The foregoing regulation is in the positive load mode and in its general aspects does not per se constitute the present invention. The regulation in the negative load mode will now be described.

Pressure sensing passage 56 connects outlet chamber 25 with a fluid receiving space 57. Control spool 29, guided in bore 28, is equipped with longitudinally extending grooves 58 terminating in metering edge 59 providing communication between the first exhaust chamber 26, the passage 27 and the second exhaust chamber 30. (See also FIG. 2) Movement of control spool 29 from right to left will gradually reduce the effective area of the grooves 58, eventually metering edge 59 cutting off communication between passage 27 and second exhaust chamber 30.

Movement of the control spool 29 from right to left is opposed by differential spring 60, normally biasing control spool 29 into the fully open position shown in FIG. 1. Space 57 is connected through resistance orifice 61, drillings 62 and 63 to second exhaust chamber 30. Movement of the valve spool 16 from left to right will open at first pressure sensing passage 37 to load chamber 22 and through land 19 load chamber 23 to outlet chamber 25. Pressure in load chamber 22, transmitted through pressure sensing passage 37 will activate, through passage 39, the differential pressure relief valve 40, in the manner as previously described. Simultaneously, pressure signal from load chamber 23 will be supplied through sensing passage 56 and will be transmitted to space 57.

Assume that the positive load generates a positive sustaining pressure in load chamber 22. Then the load chamber 23 will be at zero pressure and therefore zero pressure signal is transmitted through pressure sensing passage 56 to space 57. Further movement of valve spool 16 from left to right will connect load chamber 22 to inlet chamber 21 and load chamber 23 and outlet chamber 25 through metering groove 64 with first exhaust chamber 26. Since in the position as shown control spool 29 connects first exhaust chamber 26 with second exhaust chamber 30, the fluid flows from inlet chamber 21 to load chamber 22 and through line 22a to fluid motor 11 and out of fluid motor 11 through line 23a to load chamber 23 and outlet chamber 25, which through grooves 64, first exhaust chamber 26, passage 27, and control spool 29 are connected to second exhaust chamber 30, which in turn through line 34 is connected to reservoir 33. For the positive load condition as just described the speed of the fluid motor is fully controlled by the area of the orifice between the inlet chamber 21 and the load chamber 22 and the fixed pressure differential, maintained between these two chambers, by the differential pressure relief valve 40.

However, assume that during the above actuation of the fluid motor the load characteristics changed from positive to negative. With the valve spool 15 displaced to the right the pressure in the load chamber 22 would drop to zero. This pressure signal, transmitted through the pressure sensing passage 37 would permit retraction of force sleeve 44, with corresponding reduction in preload in spring 46 and would bring the pump discharge pressure to the minimum standby level through the differential pressure relief valve, the pump automatically supplying required flow at this minimum pressure level. Due to the action of the negative load the pressure would be generated in load chamber 23 and outlet chamber 25 and this pressure signal transmitted through pressure signal passage 56 to space 57. This pressure, reacting on the cross-section area of control spool 29, overcomes the preload force of differential spring 60, moving the control spool 29 from right to left. This movement will reduce the effective area of grooves 58 as the metering edge 59 approaches cut-off face 59a. Resistance to flow through grooves 58 will raise the pressure in the passage 27 and first exhaust chamber 26 until a condition of force equilibrium is achieved. Under this condition of equilibrium, force generated due to pressure in load chamber 23 transmitted to space 57 through passage 56, reacting on the cross-section area of valve spool 29, is balanced by the force generated due to the pressure in passage 27, acting on the cross-section area of control spool 29 plus

the biasing force of differential spring 60. Therefore under these conditions control spool 29 will automatically assume a throttling position, maintaining a constant pressure differential between load chamber 23 and first exhaust chamber 26. This constant pressure differential is equal to preload of differential spring 60 divided by cross-section area of the control spool 29.

Since modulating control spool 29 maintains by throttling action a constant pressure differential between outlet chamber 25 and first exhaust chamber 26, flow between these two chambers will be directly proportional to the area of orifice created by the grooves 64 and 64a in metering land 20 between these chambers and constant for each particular value of this area irrespective of the pressure level in load chamber 23 and outlet chamber 25 sustaining the negative load. Preload in the differential spring 60 can be so selected that the equivalent constant pressure differential is the same as the constant pressure differential, regulated by the differential pressure relief valve. In this way the flow control characteristics of the valve can be maintained in both directions of fluid motor operation irrespective of the magnitude of the positive or negative load.

When starting with a negative load, displacement of the valve spool 16 in the appropriate direction will transmit first a zero pressure signal to the differential pressure relief valve 40 and a load sustaining pressure signal to the control spool 29. Since at the time the load chamber and outlet chamber sustaining the negative load pressure are still isolated from the first exhaust chamber by the metering land 20, the control spool 29 will move all the way from right to left under the generated load pressure, isolating passage 27 from the second exhaust chamber 30. Further movement of valve spool 16 will connect the pressurized outlet chamber through grooves 64 or 64a with the first exhaust chamber, gradually increasing the pressure in passage 27, until control spool 29 moves to its modulating position maintaining a constant pressure differential in a manner as already described. In this way flow control feature will be retained during operation of negative load.

Movement of the valve spool 16 from right to left from the position shown will actuate the fluid motor 11 in the opposite direction, with the chamber 23 becoming the inlet chamber and chamber 24 becoming the outlet chamber. Thus the valve is double acting in that it controls negative loads in either direction of movement.

To allow for leakage and increase stability for the control the resistance orifice 61 is provided which connects space 57 through drillings 62 and 63 with the second exhaust chamber 30. The area of resistance orifice 61 is very much smaller than the area of pressure signal passage 56.

Referring now to FIG. 3 another load responsive valve generally designated as 64b is shown. This valve is similar to that of FIG. 1 with respect to control of positive loads but has a modified negative load control section. A negative load control spool 65 is located in bore 66 connecting outlet chamber 67 and first exhaust chamber 68. In its normal position control spool 65 under action of differential spring 69 maintains the passage between the outlet chamber 67 and first exhaust chamber 68 open. Space 70 in communication with control spool 65 is connected by passages 71 and 72 to reservoir 73. Valve spool generally designated as 74 is

guided in bore 75 and is equipped with isolating lands 76, 77 and 78 and metering land 79. With the valve spool 74 in neutral position as shown in FIG. 3, land 76 isolates load chamber 80 from outlet chamber 67, land 77 isolates inlet chamber 81 from load chambers 80 and 83, land 78 isolates outlet chamber 67 from load chamber 83, and first exhaust chamber 68 and metering land 79 isolates first exhaust chamber 68 from second exhaust chamber 84. Assume that the valve spool 74 is moved from left to right connecting first pressure sensing passage 37 to differential pressure relief valve 40 while simultaneously connecting load chamber 83 with outlet chamber 67. Assume also that load chamber 80 is subjected to a pressure of positive load. In a manner, as previously described, a pressure signal is transmitted through passage 39 to differential pressure relief valve 40 increasing through force sleeve 44 preload in spring 46. The differential pressure relief valve 40 will regulate the bypass flow from pump 12 to exhaust space 36 to maintain inlet chamber 81 at a pressure higher by a constant pressure differential than the pressure in load chamber 80. If the valve spool 74 is further moved from left to right, load chamber 80 will be connected to inlet chamber 81 and metering land 79 will connect the first exhaust chamber 68 to the second exhaust chamber 84. Fluid under a constant pressure differential will be supplied from inlet chamber 81 and will flow through line 22a to fluid motor 11 and out of fluid motor 11 through line 23a to load chamber 83 and outlet chamber 67 which through grooves 85, first exhaust chamber 68, valve spool bore 75, second exhaust chamber 84 and lines 86 and 72 is connected to reservoir 73. Since differential relief valve 40 will maintain a constant pressure differential between inlet chamber 81 and load chamber 80, as previously described, the flow into fluid motor 11 will be proportional to the axial displacement of valve spool 74 irrespective of the load pressure.

Assume that midway through the actuation the positive load becomes a negative load; the pressure in the load chamber 80 would then start dropping below a level equivalent to the setting of the spring 46. The force sleeve 44 would reduce the preload in spring 46 to minimum level and the differential pressure relief valve would regulate the bypass flow to maintain inlet chamber 81 at this minimum pressure level, equivalent to minimum preload of spring 46. Due to action of negative load the flow from first exhaust chamber 68 to second exhaust chamber 84 will tend to increase. Since the resistance to fluid flow through an orifice created by metering land 79 will increase with flow, the pressure in first exhaust chamber 68 will also increase until a pressure level is reached, when the force generated by this pressure level acting on the cross-section area of control spool 65 will overcome the preload in spring 69 moving the control spool 65 from right to left and reducing the effective area of flow through grooves 85 between the outlet chamber 67 and the first exhaust chamber 68. The control spool 65 will modulate throttling fluid flow and regulating the area of flow between outlet chamber 67 and the first exhaust chamber 68 to maintain in the first exhaust chamber 68 a constant pressure level, equivalent to the preload in the differential spring 69. Since the second exhaust chamber 84 is directly connected to reservoir 73 and therefore is maintained at atmospheric pressure and since the first exhaust chamber 68 is maintained through throttling

action of control spool 65 at constant pressure, the pressure differential existing between the first and second exhaust chambers will remain constant, when controlling a negative load. Therefore the flow through the orifice created by displacement of metering land 79 will be proportional to its area, which in turn is proportional to displacement of valve spool 74 from its neutral position. The metering surfaces 87 and 88 of metering land 79 permit regulation of the area of the orifice in respect to movement of metering land 79 and therefore valve spool 74, in turn permitting control of the speed of the load determined by the movement of valve spool 74.

Assume that the valve spool 74 is moved from left to right connecting pressure sensing passage 37 leading to the pressure differential relief valve 40 with load chamber 80, while simultaneously connecting load chamber 83 with outlet chamber 67. Assume also that load chamber 83 is subjected to pressure caused by a negative load. As previously described the differential relief valve 40 will maintain inlet chamber 81 at minimum pressure level. The increase in pressure in outlet chamber 67 will be transmitted through grooves 85 to the first exhaust chamber 68. Since the metering land 79 still isolates the exhaust chambers 68 and 84, an increase in pressure in the first exhaust chamber 68 will move the control spool 65 from right to left isolating the outlet chamber 67 from the first exhaust chamber 68. The first exhaust chamber 68 is connected through resistance orifice 89 and drilled passage 90 to space 70, which is at atmospheric pressure, a small flow through resistance orifice 89 will reduce pressure in the first exhaust chamber 68 which will be corrected by readjustment in position of control spool 65, at which point the control spool will modulate to maintain the first exhaust chamber 68 at a constant pressure level. Further movement of valve spool 74 from left to right will connect inlet chamber 81 to load chamber 80 and connect first exhaust chamber 68 to second exhaust chamber 84. Since the control spool 65 will maintain the first exhaust chamber 68 at a constant pressure level, the resulting flow through metering orifice at metering land 79 will be proportional to the orifice area irrespective of the pressure level generated by the negative load in the load chamber 83 and outlet 67. The metering orifice area varies with the movement of metering land 79 and valve spool 74, the velocity of the load also varying with the movement of valve spool 74, each position of valve spool 74 corresponding to specific speed of the load, irrespective of load magnitude. Displacement of the spool in opposite direction will control both positive and negative loads in an identical manner since the valve is double acting in that it can control the movement of the motor in either direction.

Referring now to FIG. 4, another embodiment of a flow control valve, generally designated as 91, is shown interposed between a fluid motor 11 and a fixed displacement pump 12. A second similar flow control valve represented by the box 92 is interposed between a second fluid motor 93 and the fixed displacement pump 12. The general configuration of the flow control valve 91, including the valve spool 16, control spool 29, position of the outlet and load chambers and location of the pressure sensing passages are identical to FIG. 1 and the valve operates in an identical manner to control negative loads. However, this valve will also con-

trol multiple positive loads. The mechanism for this function is as follows.

Supply chamber 92a is connected through passages 93a and 94, of inlet control valve 95, to inlet chamber 96, which through line 97 is connected to outlet port of differential pressure relief valve, generally designated as 98 and through space 99 of the valve 98 and line 100 to fixed displacement pump 12. Load sensing line 39, in communication with load sensing passages 37 and 38, is connected through space 101, check valve 102 and line 103 to the differential pressure relief valve 98. Similarly, control valve 92 is connected through load sensing line 104 and check valve 105 and line 106 to the differential pressure relief valve 98.

The control valve 95 which extends into space 101 is biased towards position as shown by differential spring 107. The control valve 95 is subjected in one direction to force developed on its cross sectional area by pressure existing in supply chamber 92a and in the opposite direction by force developed on its cross section area by pressure existing in space 101 and by force exerted by differential spring 107. Rising pressure in supply chamber 92a, above the level of the pressure in space 101 will overcome the load of differential spring 107 and move the control valve 95 from right to left, increasing the resistance to flow from pump 12 to supply chamber 92a. In its modulating position the control valve 95, by throttling fluid flow between chamber 92a and space 101 will maintain a constant pressure differential between supply chamber 92a and space 101, the pressure differential being proportional to preload in differential spring 107. Since, when the valve spool 16 is displaced from its neutral position either of the load chambers is connected by load sensing passages 37 and 38 and load sensing line 39 to space 101, the control valve 95 will maintain a constant pressure differential between appropriate load chambers and supply chamber 92a.

Assume the flow control valve 91 is actuated, the flow control valve 92 remaining in its neutral position. Assume that in actuating the flow control valve 91 the spool valve 16 is moved from left to right, connecting load sensing passage 37 to load chamber 22. Also, assume that load chamber 22 is subjected to a pressure sustaining positive load. A pressure signal from load sensing passage 37 will be transmitted through line 39, space 101 and through check valve 102 and line 103 to differential pressure relief valve 98, which in a manner as previously described with respect to differential pressure relief valve 40 will adjust the bypass flow of fixed volume pump 12 to maintain inlet chamber 96 at a pressure higher by a fixed pressure differential than the pressure in load chamber 22.

Control valve 95 is then subjected to pump discharge pressure, transmitted through passages 94 and 93a to supply chamber 92a, acting on the cross-sectional area of the control valve 95 in one direction and the pressure existing in load chamber 22, connected to space 101, acting on the cross-section of control valve 95, plus the preload of the differential spring 107 in the opposite direction. As already mentioned the pressure in space 101 is always smaller by a constant pressure differential than the discharge pressure of the pump and therefore pressure in inlet chamber 96 and supply chamber 92a. The preload in the differential spring 107 is so selected that it equals this constant pressure differential, multiplied by the cross-section area of control

valve 95 so the control valve 95, subjected to these forces, will remain in position as shown in FIG. 4. Therefore as long as differential pressure relief valve 98 controls the bypass flow of pump 12 to maintain a constant pressure differential between pump discharge pressure and the appropriate load chamber pressure, the control valve 95 will remain inactive. Under these conditions flow control valve 92 will perform in an identical way as control valve 91, when controlling both positive and negative loads.

However, assume that the flow control valves 91 and 92 are actuated simultaneously. In this case the flow control valve, having the highest load controls the output pressure from the pump due to well known action of check valves 102 and 105 whereby the higher load pressure signal will be transmitted from control valve 92 to differential pressure relief valve 98.

Assuming that the load from motor 93 is higher, the check valve 105 passes the signal; but check valve 102 isolates the higher pressure signal from space 101. In a manner as previously described, the differential pressure relief valve 98 will respond to the load control signal from flow control valve 92, the flow control valve 92 fully retaining flow control features in control of the higher load. However in the case of valve 91 the pump discharge pressure in line 97 will exceed the control fixed pressure differential, disturbing the equilibrium of the control valve 95. Under the action of unbalanced forces, control valve 95 will move from right to left, gradually reducing the effective area of flow between inlet chamber 96 and supply chamber 92a. The resulting increase in resistance to flow will reduce the pressure in supply chamber 92a until equilibrium is restored. Since, as previously described, the preload in the differential spring 107 is equal to the product of the constant differential pressure of the pump control and cross-section area of the control valve 95, the control valve 95 will modulate, throttling the excess pressure in the inlet chamber 96 and maintain the supply chamber 92a at a fixed pressure differential above the load signal from the appropriate load chamber. If the load should become negative, the pressure in the load chamber 22 would drop to or near zero. This would cause the valve 95 to move further from right to left, throttling most of the pressure of inlet chamber 96 to maintain constant pressure differential between supply chamber 92 and load chamber 22. Simultaneously the pressure in the load chamber 23 would increase actuating the valve 29 as previously described to control the negative load.

In this way both valves 92 and 91 when operated simultaneously will fully retain the flow control characteristics in control of both positive and negative loads. Valve 91 also is double acting and will control both positive and negative loads in either direction depending upon the direction of movement of valve spool 16.

Referring now to FIGS. 5 and 6, yet another flow control valve generally designated as 107a is shown. This valve is similar to that of FIG. 1 with respect to control of positive loads but has a modified negative load control section. A negative control spool 108 is located in a bore 109 connecting outlet chambers 25 and 24 and an exhaust chamber 110. In its normal position control valve 108 is located in a bore 109 connecting outlet chambers 25 and 24 and an exhaust chamber 110. In its normal position control valve 108 under the action of differential spring 111, maintains the passage between outlet chambers and exhaust chamber 110

closed. Space 112 is connected with pressure sensing passages 37 and 38 through line 113.

Assume that valve spool 16 is moved from left to right connecting first pressure sensing passage 37 to differential pressure relief valve generally designated as 40 and then connecting load chamber 22 with inlet chamber 21 and load chamber 23 with outlet chamber 25. Assume also that chamber 22 is subjected to pressure of a positive load. In a manner as previously described, the differential relief valve 40 will regulate the bypass flow from fixed displacement pump 12 to maintain a constant pressure differential between the inlet chamber 21 and the load chamber 22. Also, as long as there is pressure in pressure sensing port 37 and passage 113, the control valve 108 will be moved fully from right to left against bias of spring 111. This will cross connect the chamber 25 with exhaust chamber 110 through slots 114 in valve 108.

Assume that midway through the actuation the positive load would become a negative load and the pressure in the load chamber 22 would start dropping to zero level. The differential pressure relief valve will maintain, as previously described, a constant minimum pressure in inlet chamber 21 and due to pressure drop proportionally lower pressure in load chamber 22. However, since load chamber 23 is connected to outlet chamber 25 and since load chamber 23 is now pressurized, due to the action of negative load, the speed of actuation of the negative load will tend to increase, increasing pressure drop between inlet chamber 21 and load chamber 22, further reducing pressure in load chamber 22. Since load chamber 22 is connected with space 112 through pressure sensing passage 37 and line 113 and the pressure here drops, the differential spring 11 will move the control valve 108 from left to right. This will restrict the passage opening around the bore 109 in the region of slots 114, thus increasing the pressure in the outlet chamber 25. This increase in pressure in outlet chamber 25 will reduce the pressure differential between load chamber 23 and outlet chamber 25, proportionally reducing the flow between these two chambers. This in turn will tend to increase the pressure in the load chamber 22 and therefore pressure in space 112, modulating the throttling action of the control valve 108, to maintain flow control during operation of negative load. In this way control valve 107a of FIG. 5 is capable of controlling both positive and negative loads. This valve also is double acting in that it can control in either direction of movement of the motor.

Although preferred embodiments of this invention have been shown and described in detail it is recognized that the invention is not limited to the precise forms and structure shown and various modifications and rearrangements as will readily occur to comprehension skilled in the art upon full comprehension of this invention may be resorted to without departing from the scope of the invention as defined in the claims.

What is claimed is:

1. A valve assembly comprising a housing having a fluid inlet chamber, a fluid load chamber, a fluid outlet chamber and fluid exhaust means, first valve means for selectively interconnecting said load chamber with said inlet chamber and said outlet chamber, variable orifice means between said outlet chamber and said exhaust means and responsive to movement of said first valve means, and second valve means disposed to throttle fluid between said outlet chamber and said exhaust

means and means to operate said second valve means to maintain a constant pressure difference across said variable orifice means when said load chamber and said outlet chamber are interconnected and said outlet chamber is pressurized.

2. A valve assembly as set forth in claim 1 wherein said fluid throttling means are positioned between said variable orifice means and said fluid exhaust means.

3. A valve assembly as set forth in claim 1 wherein said fluid throttling means are positioned between said outlet chamber and said variable orifice means.

4. A valve assembly as set forth in claim 1 wherein said first valve means includes a valve spool axially guided in a spool bore and movable from a neutral position to at least one actuated position, said valve spool isolating said load chamber from said fluid inlet chamber and said fluid outlet chamber and said fluid outlet chamber from said fluid exhaust means when in neutral position.

5. A valve assembly as set forth in claim 4 wherein said variable orifice means are varied in relation to axial displacement of said spool with respect to said bore.

6. A valve assembly as set forth in claim 5 wherein said variable orifice means includes a metering land and metering surface means, said metering surface means determining the variation in said variable orifice means in respect to movement of said spool.

7. A valve assembly as set forth in claim 6 wherein said metering surface means have means operable to meter fluid flow between said fluid outlet chamber and said fluid exhaust means when said spool is displaced in both directions from neutral position.

8. A valve assembly comprising a housing having a fluid inlet chamber, a fluid load chamber, a fluid outlet chamber, a first fluid exhaust chamber and second fluid exhaust chamber, first valve means for selectively interconnecting said load chamber with said inlet chamber and said outlet chamber, said first valve means including a valve spool axially guided in a spool bore and movable from a neutral position to at least one actuated position, said valve spool isolating said load chamber from said fluid inlet chamber and said fluid outlet chamber and said fluid outlet chamber from said fluid exhaust chamber when in neutral position, variable orifice means interconnecting said outlet chamber and said first exhaust chamber and responsive to movement of said first valve means, and second valve means interconnecting said first exhaust chamber and said second exhaust chamber operable to maintain a constant pressure difference across said variable orifice means when said load chamber and said outlet chamber are interconnected and said outlet chamber is pressurized.

9. A valve assembly as set forth in claim 8 wherein said second valve means is responsive to pressure in said fluid outlet chamber.

10. A valve assembly as set forth in claim 8 wherein said second valve means has throttling means to throttle fluid from said first fluid exhaust chamber to said second fluid exhaust chamber.

11. A valve assembly as set forth in claim 8 wherein said valve spool has a fluid metering means between said fluid outlet chamber and said first fluid exhaust chamber, said fluid metering means movable from a neutral position to at least one control position and when in neutral position said metering means isolating

said fluid outlet chamber from said first fluid exhaust chamber.

12. A valve assembly as set forth in claim 8 wherein said second valve means includes a control spool guided in a bore interconnecting said first fluid exhaust chamber and said second fluid exhaust chamber said control spool being biased in one direction by pressure in said first exhaust chamber and by spring means and in the other direction by pressure in said outlet chamber.

13. A valve assembly comprising a housing having a fluid inlet chamber, a fluid load chamber, a fluid outlet chamber, a first fluid exhaust chamber, a second fluid exhaust chamber and fluid exhaust means said exhaust chambers being between said outlet chamber and said exhaust means, first valve means for selectively interconnecting said load chamber with said inlet chamber and said outlet chamber, said first valve means including a valve spool axially guided in a spool bore and movable from a neutral position to at least one actuated position, said valve spool isolating said load chamber from said fluid inlet chamber and said fluid outlet chamber and said first fluid exhaust chamber from said second fluid exhaust chamber when in neutral position, variable orifice means interconnecting said first and second exhaust chambers and responsive to movement of said first valve means, and second valve means interconnecting said outlet chamber and said first exhaust chamber having throttling means, and means to operate said second valve means to maintain a constant pressure difference across said variable orifice means when said load chamber and said outlet chamber are interconnected and said outlet chamber is pressurized.

14. A valve assembly as set forth in claim 13 wherein said second valve means is responsive to pressure in said first exhaust chamber.

15. A valve assembly as set forth in claim 13 wherein said second valve means has throttling means to throttle fluid from said outlet chamber to said first fluid exhaust chamber.

16. A valve assembly as set forth in claim 13 wherein said valve spool has a fluid metering means between said first fluid exhaust chamber and said second fluid exhaust chamber, said fluid metering means movable from a neutral position to at least one control position and when in neutral position said metering means isolating said first fluid exhaust chamber from said second fluid exhaust chamber.

17. A valve assembly as set forth in claim 13 wherein said second valve means includes a control spool guided in a bore interconnecting said fluid outlet chamber and said first fluid exhaust chamber said control spool being biased in one direction by pressure in said first exhaust chamber and in the other direction by spring means.

18. A valve assembly comprising a housing having a fluid inlet port, a fluid inlet chamber, a fluid load chamber, a fluid outlet chamber, a first fluid exhaust chamber and fluid exhaust means, first valve means for selectively interconnecting said load chamber with said inlet chamber and said outlet chamber, first variable orifice means interconnecting said outlet chamber and said exhaust chamber and responsive to movement of said first valve means, and a second valve means interconnecting said exhaust chamber and said fluid exhaust means having throttling means, and means to operate said second valve means to maintain a constant pressure difference

across said variable orifice means when said load chamber and said outlet chamber are interconnected and said outlet chamber is pressurized and second variable orifice means interconnecting said load chamber and said inlet chamber and responsive to movement of said first valve means, and third valve means interconnecting said inlet port and said exhaust means and operable to maintain a constant pressure difference across said second variable orifice means when said inlet chamber and said load chamber are interconnected and when said load chamber is pressurized.

19. A valve assembly as set forth in claim 18 wherein a fourth valve means interconnects said inlet port and said inlet chamber and operable to maintain a constant pressure difference across said second variable orifice means when said inlet chamber is open to said load chamber and the pressure difference between said load chamber and said inlet chamber is greater than said constant pressure difference.

20. A valve assembly as set forth in claim 18 wherein said third valve means includes a control member biased in one direction by pressure in said inlet and in the opposite direction by pressure in said load chamber and by spring means and operable to bypass fluid flow from said inlet port to said exhaust means to maintain a constant pressure difference across said second variable orifice means.

21. A valve assembly as set forth in claim 20 wherein said third valve means is connected by passage means with said load chamber, said passage means being closed by said first valve means when said first valve means is in neutral position, said first valve means when moved from its neutral position first interconnecting by said passage means said load chamber to said third valve means and then connecting said load chamber to said inlet chamber.

22. A valve assembly comprising a housing having a fluid inlet port, a fluid inlet chamber, a fluid load chamber, a fluid outlet chamber, a first fluid exhaust chamber, a second fluid exhaust chamber and fluid exhaust means said exhaust chambers being between said outlet chamber and said exhaust means, first valve means for selectively interconnecting said load chamber with said inlet chamber and said outlet chamber, first variable orifice means interconnecting said first and second exhaust chambers and responsive to movement of said first valve means, second valve means interconnecting said outlet chamber and said first exhaust chamber having throttling means, and means to operate said second valve means to maintain a constant pressure difference across said variable orifice means when said load chamber and said outlet chamber are interconnected and said outlet chamber is pressurized, second variable orifice means interconnecting said load chamber and said inlet chamber and responsive to movement of said first valve means, and third valve means interconnecting said inlet port and said exhaust means and operable to maintain a constant pressure difference across said second variable orifice means when said inlet chamber and said load chamber are interconnected and when said load chamber is pressurized.

23. A valve assembly as set forth in claim 22 wherein a fourth valve means interconnects said inlet port and said inlet chamber and operable to maintain a constant pressure difference across said second variable orifice means, when said inlet chamber is open to said load chamber and the pressure difference between said load

chamber and said inlet chamber is greater than said constant pressure difference.

24. The valve assembly as set forth in claim 22 wherein said third valve means includes a control member biased in one direction by pressure in said inlet port and in the opposite direction by pressure in said load chamber and by spring means and operable to bypass fluid flow from said inlet port to said exhaust means to maintain a constant pressure difference across said second variable orifice means.

25. A valve assembly as set forth in claim 24 wherein said third valve means is connected by a passage means with said load chamber, said passage means being closed by said first valve means when said first valve means is in neutral position, said first valve means when moved from its neutral position first interconnecting by said passage means said load chamber to said third valve means and then connecting said load chamber to said inlet chamber.

26. A fourway fluid control valve assembly comprising a housing having an inlet chamber, first and second load chambers, an outlet chamber and an exhaust chamber, a valve bore in direct communication with said aforementioned chambers, said valve bore axially guiding a valve spool having sequencing lands and a metering land, said sequencing lands isolating said inlet chamber said outlet chamber and said first and second load chambers, and said metering land isolating said outlet chamber from said exhaust chamber, exhaust means in said housing, valve means interconnecting said exhaust chamber and said exhaust means and operable to maintain a constant pressure differential across said metering land when one of said load chambers is connected to said outlet chamber and said outlet chamber is pressurized, said valve means including a control spool guided in a control bore, fluid throttling means on said control spool to control fluid flow between said exhaust chamber and said exhaust means, said control spool being biased in one direction to increase fluid flow by pressure in said exhaust chamber when said control bore is connected thereto and by spring means, and in the other direction by pressure in said outlet chamber, when the control bore is connected thereto and said outlet chamber is pressurized, a signal passage connecting said control bore and said outlet chamber, said valve spool when displaced from its neutral position in one direction first interconnecting said first load chamber to said inlet chamber and said second load chamber to said outlet chamber and then connecting said exhaust chamber to said outlet chamber by said metering land, said metering land having metering surface means to regulate the area of fluid flow between said outlet chamber and said exhaust chamber with respect to movement of said valve spool in said one direction, said valve spool when displaced from its neutral position in the opposite direction first interconnecting said first load chamber to said outlet

chamber and said second load chamber to said inlet chamber and then connecting said exhaust chamber to said outlet chamber by said said metering land, said metering land having metering surface means to regulate the area of flow between said outlet chamber and said exhaust chamber with respect to movement of the valve spool in said opposite direction, whereby valve will control a load under negative load conditions.

27. A fourway fluid control valve assembly comprising a housing having an inlet chamber, first and second load chambers, an outlet chamber and first and second exhaust chambers, a valve bore in direct communication with said aforementioned chambers, said valve bore axially guiding a valve spool having sequencing lands and a metering land, said sequencing lands isolating said inlet chamber said outlet chamber and said first and second load chambers, and said metering land isolating said first exhaust chamber from said second exhaust chamber, exhaust means in said housing, valve means interconnecting said outlet chamber and said first exhaust means and operable to maintain a constant pressure differential across said metering land when one of said load chambers is connected to said outlet chamber and said outlet chamber is pressurized, said valve means including control spool guided in a control bore, fluid throttling means on said control spool to control fluid flow between said outlet chamber and said first exhaust chamber, said control spool being biased in one direction to increase fluid flow by pressure in said second exhaust chamber and by spring means, and in the other direction by pressure in said first exhaust chamber when the control bore is connected thereto and said outlet chamber is pressurized, a signal passage connecting said control bore and second exhaust chamber, said valve spool when displaced from its neutral position in one direction first interconnecting said first load chamber to said inlet chamber and said second load chamber to said outlet chamber, and the connecting said first exhaust chamber to said second exhaust chamber by said metering land, said metering land having metering surface means to regulate the area of fluid flow between said first exhaust chamber and said second exhaust chamber in respect to movement of said valve spool in said one direction, said valve spool when displaced from neutral position in the opposite direction first interconnecting said first load chamber to said outlet chamber and said second load chamber to said inlet chamber, and the connecting said first exhaust chamber to said second exhaust chamber by said metering land, said metering land having metering surface means to regulate the area of flow between said first and second exhaust chambers in respect to movement of the valve spool in said opposite direction, whereby valve will control a load under negative load conditions.

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