(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number: **0 081 941**
A2

(12) **EUROPEAN PATENT APPLICATION**

(21) Application number: **82306373.0**

(22) Date of filing: **01.12.82**

(51) Int. Cl.³: **F 15 B 13/04**

(30) Priority: **11.12.81 US 329960**

(43) Date of publication of application: **22.06.83**
Bulletin 83/25

(84) Designated Contracting States: **BE DE FR GB NL SE**

(71) Applicant: **FORD MOTOR COMPANY LIMITED, Eagle Way, Brentwood Essex CM13 3BW (GB)**

(84) Designated Contracting States: **GB**

(71) Applicant: **FORD-WERKE AKTIENGESELLSCHAFT, Ottoplatz 2 Postfach 21 03 69, D-5000 Köln 21 (DE)**

(84) Designated Contracting States: **DE**

(71) Applicant: **FORD FRANCE SOCIETE ANONYME, 344 Avenue Napoléon Bonaparte B.P. 307, F-92506 Rueil Malmaison Cedex (FR)**

(84) Designated Contracting States: **FR**

(71) Applicant: **Ford Motor Company, The American Road, Dearborn MI 48121 (US)**

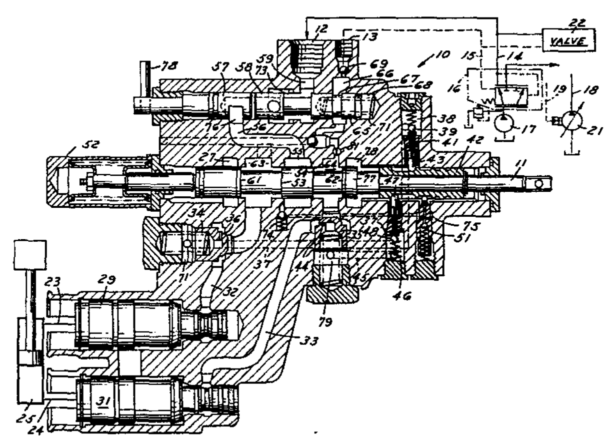
(84) Designated Contracting States: **BE NL SE**

(72) Inventor: **Rajagopal, Ganesh, 1608 Mayflower, Richardson Texas 75081 (US)**
Inventor: **Tallian, Richard, 8780 Mohican, Clarkston Michigan 48016 (US)**

(74) Representative: **Messulam, Alec Moses et al, A. Messulam & Co. 24 Broadway, Leigh on Sea Essex SS9 1BN (GB)**

(54) **Hydraulic valve.**

(57) A hydraulic valve for controlling the flow of hydraulic fluid from a source of fluid under pressure (17, 21) to and from at least one load shifting hydraulic motor (25). The valve comprises a valve body (10) housing an axially movable spool (11) and a spool manipulator means actuable to shift the spool (11) in opposite directions from a neutral position to cause fluid flow to a hydraulic motor to move a load coupled thereto in a desired direction. Incorporated in the valve are pilot operated poppet valves (34, 35) operative to substantially prevent fluid leakage past the main spool (11) when the latter is in its neutral position.



EP 0 081 941 A2

0081941

HYDRAULIC VALVE

This invention relates to a hydraulic valve.

Known hydraulically actuated load check systems are able to function only when hydraulic system pressure is available. This means that loads can be lowered only when the hydraulic pump is supplying fluid under pressure to the system. For example, the hydraulic pump for operating accessories coupled to an agricultural tractor is driven from an engine power take-off and, therefore, is operative only when the engine is running. If an operator forgets to lower a load completely before shutting off the engine, and if the control handle of the remote valve controlling the load raising and lowering mechanism is then put into detent at the lowering position, a dangerous situation could arise upon the operator starting up the machine. The load would suddenly fall because the control handle had been placed in detented position while the engine was off.

According to the present invention, there is provided a hydraulic valve for controlling the flow of hydraulic fluid from a source of fluid under pressure to and from at least one load shifting hydraulic motor, the valve comprising a valve body housing an axially movable main spool for directing fluid through selective ones of a plurality of passages extending through the valve body from a fluid input connection to the fluid source to one or more fluid output connections, characterised by spool manipulator means actuable to shift said valve main spool in either axial direction from a neutral position to cause actuation of the hydraulic motor to move a load coupled thereto in a desired direction, and a load position holding system having pilot operated poppet valves operative to minimise fluid leakage past the main spool when the latter is in a neutral position.

The invention will now be described further, by way of example, with reference to the accompanying drawings, in which :

Figure 1 is a sectional view of a hydraulic valve according to a mechanically operated embodiment of the present invention,

Figure 2 is a view in part similar to Figure 1 of a hydraulic valve adapted to be electronically controlled, and

Figure 3 is a schematic view of a prior art load check system.

The hydraulic valve described herein is used to control the flow of hydraulic fluid (which could be at high pressure) to and from hydraulic motors connected to the valve. Two embodiments of the valve are shown. The first embodiment is mechanically actuated and the second embodiment is electrically controlled. A majority of the parts are common to both valves. The primary application of these valves is in agricultural tractors.

Figure 1 shows the mechanical version of the valve 10 with the main spool 11 in the neutral position. The inlet port 12 and the pilot signal port 13 of the valve 10 can be connected either to the output port 14 and signal port 15 of an unloading valve 16 when the hydraulic supply is from a fixed displacement pump 17, or to the output port 18 and signal port 19 of a pressure flow compensated, variable displacement pump 21. It is possible to connect several valves similar to valve 10 simultaneously to the outlet ports 14 or 18 and the signal ports 15 or 19 of the two hydraulic sources. As an example, valve 22 is shown connected to ports 14 and 15. The output ports 23 and 24 of the valve are shown connected to any other hydraulic motor or actuator with two ports.

It is also possible to connect actuators with only one port to the valve as is shown in Figure 1a where a single acting cylinder 26 is connected to port 23. Single acting cylinders can extend and retract when connected to either outlet port 23 or 24. However, in the present design, there is an advantage when the single acting cylinder is connected to port 23. This will be explained later in this description.

The chambers 27 and 28 of the valve 10 are internally connected together, and by means of a return port, to the sump of the hydraulic system, (not shown). In Figure 1, the outlet ports 23 and 24 are formed by the quick-disconnect couplers 29 and 31 which are housed in the same body as valve 10. The couplers 29 and 31 act as through passageways between port 23 and passage 32 and between port 24 and passage 33, respectively, once they are connected to the coupling hoses from the hydraulic motor.

When the valve spool 11 is in the neutral position, the load check valves 34 and 35 are seated and prevent the load (in this case a weight on cylinder 25) from moving. The oil in passage 32 is also connected through orifice 36 and passage 37 to chamber 38. The oil in chamber 38 is checked by ball 39 which seals orifice 41. Sleeve 42, in the neutral position, allows pilot actuating pin 43 to exert no force on ball 39 so that the oil in chamber 38 is prevented from entering orifice 41 and from there draining to the chamber 28. Similarly, the oil in passage 33 is connected through orifice 44 and passage 45 to chamber 46. With the sleeve 42 in the neutral position, pin 47 exerts no force on ball 48, and the oil in chamber 46 is prevented from entering orifice 49 and thence draining into chamber 28.

Sleeve 42 is a separate part which is attached to the spool 11 by means of shims and a snap ring so that the relative motion of the pins 43 and 47 with respect to the opening and closing of the pressure and tank ports is in the right sequence. It is possible to make the sleeve a part of the spool itself in designs where a larger dead-band at neutral is permissible, or where other dimensions related to the opening and closing of ports are held to suitable tolerances. An advantage of the present design, where the sleeve 42 is a separate part from the spool, is that the side loads on the sleeve, exerted by the pins 43 and 47 and the detent plunger 51, are not transmitted to the spool but are directly transmitted to the body of the valve 10. This lessens the chances of binding for the main spool 11 since it is subjected only to axial loads.

In the absence of external forces, spool 11 is held at the neutral position by the spring arrangement in the housing 52. The edges 53 and 54 on the spool 11 are positioned so as to close off the chamber 55 which is connected through passage 56, the slot in the flow control valve 57 and holes in pressure compensating spool 58 to passage 59 and the inlet port 12 to which the pressure supply is connected. Edges 61 and 62 on the spool 11 have a small taper on their diameter so that at the neutral position chamber 63 drains into chamber 27 and chamber 64 drains into chamber 28. Chambers 63 and 64 are connected to the shuttle valve 65 which allows only the larger pressure of the two to be sensed at chamber 66 and through orifice 67 at chamber 68. At neutral, the pressure in all these chambers is at tank pressure, chamber 66 is connected to the load sensing port 13 through the check valve 69. If valve 22 is also at neutral, the pressure in the load sensing port is reduced to the tank pressure and the pressure supply in the inlet port will be at standby pressure. At this condition, in the case of the fixed displacement pump 17, all the pump flow will be bypassed by the unloading valve 16 to either the tank or to some other circuit. In the case of the variable displacement pump 21, the output flow would be just sufficient to maintain the leakage at standby pressure.

When spool 11 moves to the left, the sleeve 42 also moves with it. After a certain small initial displacement, sleeve 42 contacts pin 43 and starts raising it. When the pin 43 rises, it contacts ball 39 and raises it off its seat on orifice 41. Once the ball 39 is off its seat, oil in the chamber 38, passageway 37 and chamber 71 can drain to the chamber 28 through orifice 36. This causes a pressure difference on either side of the poppet check valve 34 so that it is unseated and opens communication between channel 32 and chamber 63.

As the spool 11 moves further to the left, edge 54 opens a flow path between the chamber 55 and chamber 64. Also, edge 62 of the spool 11 closes off the flow path between chamber 64 and chamber 28. Edge 61 on the spool 11 opens up the flow path between chamber 63 and chamber 27. Oil flowing into chamber 64 will not be able to flow into passage 33 until the pressure in chamber 64 is greater than the pressure in passage 33. The pressure in chamber 64 builds up to the required level as follows.

The pressure in chamber 64 is transmitted through the shuttle valve 65, chamber 66, check valve 69, and load sensing port 13 to the port 15 on the unloading valve 16. This causes the pressure at port 14 to rise to the level of the pressure in port 15 plus the standby pressure level of the unloading valve. The pressure in port 15 is fed back into the valve 10 through port 12, passages 59, 56, 55 and into 64. Thus, the pressure in chamber 64 continues to rise until the check valve 35 is opened and flow can proceed to the cylinder 24. The return flow from the cylinder enters the valve 10 at port 23 and is drained in chamber 16 through passage 32 and chamber 63.

The pressure compensating spool 58 provides a means by which the flow of oil from chamber 55 into chamber 64 is made proportional to the area opened up by edge 54 of the spool. In the following discussion, the flow control valve 57 is assumed to be wide open so that the area of flow at the flow control valve is much larger than the flow area exposed by edge 54 of the spool. The flow area exposed by edge 54 may be made to follow any suitable function of the axial displacement of the spool 11. For example, the area may be directly proportional to the displacement. The flow across a sharp edge orifice generally follows the rule $Q = KA\sqrt{P}$, where Q = flow, K = constant, A = flow area, P = pressure drop across flow area. This formula indicates that the flow Q may be made proportional to the flow area A by holding the pressure drop P across the area constant. The pressure compensating spool 58 seeks to do this. The pressure drop P is the difference in pressure between the pressures in chambers 55 and 64. The pressure in chamber 55 is sensed at the left end of the spool 58 and the pressure in chamber 64 is sensed at the right end of the spool in the cavity 68. The spool is also subjected to a force by spring 71 which opposes and balances the force on the spool due to the pressure difference, P, across it. If the force due to the pressure difference exceeds the spring force, spool 58 chokes the incoming flow at the metering edge 73 so that the force balance is restored. Similarly, if the pressure drop falls below that needed to maintain balance against the spring force, the spool 58 attempts to restore the balance by reducing the restriction at edge 73. In this manner, the flow across the edge 54 of the main spool 11 is held almost constant for a given position of the spool, irrespective of fluctuations in supply or load pressures. One advantage of holding the pressure drop constant across the metering area of the main spool 11 is that the flow forces on the spool are also constant and usually lower than in designs where the maximum possible pressure drop is allowed to occur across the spool. Reduced flow forces mean that the effort needed to move the spool is reduced, which is an advantage during manual operation of the valve.

As the spool 11 continues to move further to the left, the output flow increases proportionally to the area opened up by edge 54. When the maximum flow position has almost been reached, sleeve 42 causes detent piston 51 to raise. This is sensed by a human operator as an increase in resistance to the further movement of the spool. This feature is provided so that the operator has a means of

0081941

sensing when the spool will be going into detent. If the operator wishes to put the spool into detent, he can force the spool a little further and the detent piston 51 will continue to hold the sleeve 42 and the spool 11 in that position. If the operator does not wish to detent the spool, he has an indication that he has reached the maximum flow position of the spool. This feature is desired when the valve is stroked continually as in loader operation.

The spool 11 may be taken out of detent and returned to neutral either manually by overcoming the resistance of the detent piston 51 or automatically when the system pressure reaches a preset value. When the pressure in chamber 55 exceeds the pressure set at relief valve 74, oil flows into chamber 75 and pushes the detent piston back so that the sleeve 42 (and spool 11) may return to the neutral position. This feature is required when the spool is placed in detent and the cylinder 25 strokes until it reaches the end of its travel. The pressure then builds up to the pump relief pressure, unless it is relieved by returning the spool 11 to neutral. Once the spool 11 returns to neutral, the pressure in chamber 55 drops to the standby level, the relief valve 74 reseats, and the detent piston 51 is returned by its spring.

During the entire leftward movement of the spool 11 and the sleeve 42, pin 43 continues to keep ball 39 raised off its seat so that poppet check valve 34 also continues to be off its seat to allow return oil to freely flow from passage 32 into chamber 63 and chamber 27.

The function of the flow control valve 57 is to restrict the maximum flow from the valve to a prese t value when the spool 11 is put into its first detent position when moving to the left. This is done by varying flow area 76 by rotating the valve 57 through control lever 78. Since the flow through the valve is proportional to the flow area across which the pressure drop is maintained by the pressure compensating spool 58, varying area 76 effectively controls the flow through the valve when the spool 11 is set at its detent position.

When the spool 11 is moved further to the left after it gets to the first detent position, it reaches a second detent position that corresponds to the "float" mode of operation of the valve 10. Cam sleeve 42 has a groove corresponding to this position in which detent piston 51 will hold the sleeve and the spool in detent. In the "float" position of the spool 11, edge 77 of the spool 11 moves sufficiently to the left to close the passage between chambers 55 and 64. Edge 77 of the spool 11 then allows free flow between chamber 64 and the chamber 28. Free flow is also possible between chamber 63 and chamber 27. Edge 54 of the spool prevents any flow from chamber 55 into chamber 63. When the sleeve 42 is moved to the "float" position, pin 47 is forced to move so as to raise ball 48 off its seat so that flow is possible from chambers 79 through passage 45 and chamber 46, past the orifice 49 into the chamber 28. Pressure in channel 33 will then force the poppet 35 off its seat and allow free flow into chamber 64 and from there into chamber 28. In the "float" position, both the output ports 23 and 24 are connected to the chambers 27 and 28, respectively. This allows the piston in cylinder 25 to "float" to any position it is forced to occupy as a result of the external forces acting upon it.

The sequence of events that occur when the spool 11 moves to the right are similar to those that occur when the spool 11 moves to the left. There is only one detent position when the spool moves right. There is no float position in that direction.

Edge 53 of the spool moves to the right, allowing flow from the inlet chamber 55 to enter chamber 63. The pressure in chamber 63 is fed back to the pump through shuttle valve 65 and sensing port 13 so that the pressure starts building up until it is able to overcome pressure in passage 32 and open the poppet check valve 34. The flow can then proceed to the outlet port 23 and the rod end of the cylinder 25. When sleeve 42 moves to the right with spool 11, it causes pin 47 to move so as to raise ball 48 off its seat. This allows oil in chamber 79, passage 45 and chamber 46 to drain past orifice 49 into the chamber 28. If pressure is present in passage 33, flow occurs across orifice 44 and the resulting pressure difference on either side of poppet 35 moves the poppet off its seat and allows flow from passage 33 to enter chamber 64 and then be drained to the chamber 28, past edge 62 of the spool.

Flow control is obtained by varying the flow area between chambers 55 and 63 as the edge 53 moves to the right. Once the spool is in the detent position on the right, the maximum flow can be set by rotating flow control valve 57. When the pressure in chamber 55 exceeds the value set at relief valve 74, the detent piston 51 is kicked back and the spool 11 is released from the detented position.

It is possible to operate the valve with a single-acting cylinder such as the cylinder 26 shown in Figure 1a. In this valve, it is preferable to attach such single-acting cylinders to port 23 of the valve. For the single-acting mode, valve 81 is turned through 90° so that communication is cut between the shuttle valve 65 and load sensing port 13 on the one side and chamber 64 on the other side. The advantage of this is that when the load is being lowered by moving spool 11 to the left, the oil from the inlet chamber 55 flows into chamber 64 from where it is prevented from being fed back to the load sensing port 13 by valve 81. Thus, the pump output pressure is not raised to the relief pressure of the system, as it would have been if valve 81 had been left open. This is because chamber 64 is deadheaded due to output port 24 being closed off. Thus, the pressure in chamber 64 cannot reach an equilibrium pressure corresponding to a load pressure. When the spool 11 is moved to the right, oil flows into the single-acting cylinder 26, causing the piston to be extended. The sequence of events is as described previously for the case when the spool moves to the right.

The hydraulic valve in Figure 1 that has been described so far is designed so that a majority of the parts and machining operations can be used in an electrohydraulic version shown in Figure 2. The sleeve 42 used in Figure 1 is replaced by a simpler sleeve 82 in Figure 2. The detent grooves are absent in this sleeve and O-ring grooves 83 have been added. The detent piston 51 is not used since the detent feature is provided at the control handle. The detent kickout relief valve 74 is also deleted. A pressure switch 84 is added in the electrohydraulic valve to provide a signal to kick out the detent at the control handle at a preset pressure. The manual flow control valve 57 is eliminated since it is possible to get the same function by electronic means. The ends 85 and 86 of the spool 11 are used as pistons when the spool is to be moved by flow controlled by electrohydraulic means.

One method of electrohydraulic actuation is shown in Figure 2. This has been described in detail in previous U.S. applications, namely, Serial No. 149,065, filed May 5, 1980, "Electro-Hydraulic Proportional Actuator", and Serial No. 301,789, filed September 14, 1981, "Electro-Hydraulic Remote Valve", both by G. Rajagopal and H.S. Basrai, and assigned to the assignee of this application. Solenoid type three-way valves 87 and 88 are used to direct oil into and out of the ends 85 and 86 so as to either move the spool 11 to the left, right, or to just hold it in position. Position transducer 89 is used to feed back the position of the spool 11 to the electronic control system. Pressure compensated flow control valve 91 is used to limit the speed of movement of spool 11 to a maximum value.

A feature of the valve embodying the present invention is the design of the load checks used to lock the hydraulic motors or cylinders attached to the valve at ports 23 and 24. The poppet checks 34 and 35 and pilot poppet balls 39 and 48, with their corresponding seats, have very low leakage so that loads attached to the valve are essentially held still and not allowed to leak down for long time periods when the valve spool 11 is at the neutral position. In valves without load checks, leak-down is minimized by holding the clearance between the spool 11 and its bore to the smallest feasible values. The use of load checks allows a larger clearance between spool and bore and, correspondingly, larger tolerances during manufacture without concern about leak-down. On the present valve, the main load checks 34 and 35 are controlled by a pilot system that is mechanically actuated by the sleeve 42 or 82 that moves with spool 11.

When raising or lowering a load, only the load check valve in the return oil path is mechanically actuated. The other load check valve is actuated by the buildup of inlet pressure in the hydraulic system sufficient to move the load. Thus, no sudden drop of the load due to insufficient inlet pressure is possible. Since the return load check valves are mechanically actuated, it is possible to lower a load even with the hydraulic pump shut off. This feature is helpful when an operator turns the engine off on his machine and finds that he still has a load or implement in the raised position. He can lower it to its safe position without having to turn the engine on again. In the case of the electrohydraulic version of the valve, the load cannot be lowered by remote electrical control since hydraulic pressure is required to move the spool 11. However, the load may be lowered by mechanically moving spool 11 if it is accessible.

Another feature of the present design can best be explained by comparison with the design approach shown schematically in Figure 3. This illustration shows a double-acting check valve 93 used to prevent a double-acting cylinder from slowly lowering a load due to leakage past the valve spool of a control valve (not shown). The diagram indicates the fluid flow when the control valve is moved to extend the piston 96 of the double-acting cylinder 94. Fluid pressure entering through conduit 95 is exerted against the floating piston 92 of the check valve 93 which, because of its greater area, overcomes the spring 97 and relieves pressure behind the ball 98 in the return passage 99. This permits return flow to the control valve through the return conduit 101. The fluid flowing into conduit 95 forces check ball 102 off its seat to extend the cylinder, that is, move the piston 96 upwardly as viewed in Figure 4. The direction of movement of the piston 96 is reversed by reversing the direction of the fluid into and out of conduits 95, 101.

followed by page 13a.........

This hydraulically actuated load check system will function only when hydraulic system pressure is available, which means that loads can be lowered to a safe position only when the engine is on. If an operator forgets to lower the load before shutting off the engine, and if the control handle of the valve is then put into detent at the lowering position, a dangerous situation arises when the operator starts up the machine again. The load would suddenly fall because the control handle had been placed in............................

followed by page 14........

a detented position while the engine was off. When lowering a heavy load using the load check system shown in Figure 3, it is possible to encounter problems such as chattering and even no response at all if the ratio of the areas of the cylinder end to rod end are greater than or equal to the ratio of the areas of the actuating piston 92 to the ball poppet seat. This limitation is not present in the valve shown in Figures 1 and 2.

Another feature of the valve described here is the use of sleeve 42 for many functions. This saves on the overall length of the valve and reduces the number of parts. Sleeve 42 provides the following functions.

1. It actuates the pilot check pins 43 and 47, which actuate the main poppets 34 and 35.

2. It provides grooves for detents at three positions: "raise", "lower" and "float". Dual groove angles are used on some faces so as to give a low resistance to initial movement yet provide a detent against the return spring force.

3. The sleeve provides a feel of resistance before dropping into detent so that an operator who does not want to put the spool into detent has a means of sensing when to stop.

4. The same sleeve also provides for actuation of both pilot check pins 43 and 47 when the float mode of operation is required.

5. The sleeve isolates the main spool 11 from side loads caused by actuation of the pilot check pins and the detent piston.

Sleeve 82, used in the electrohydraulic valve, provides an additional functin.

6. It acts as a piston.

Yet another feature of the present valve is that most of the mechanical components can be used in both the mechanical and the electrohydraulic version of the valve.

A further feature is the provision of a single-acting mode in which the system pressure is not raised up to relief pressure when lowering a single-acting cylinder.

CLAIMS

1. A hydraulic valve for controlling the flow of hydraulic fluid from a source of fluid under pressure to and from at least one load shifting hydraulic motor, the valve comprising a valve body (10) housing an axially movable main spool (11) for directing fluid through selective ones of a plurality of passages (56,32,33) extending through the valve body from a fluid input connection (12) to the fluid source to one or more fluid output connections (23,24), characterised by spool manipulator means actuable to shift said valve main spool (11) in either axial direction from a neutral position to cause actuation of the hydraulic motor to move a load coupled thereto in a desired direction, and a load position holding system having pilot operated poppet valves (34,35) operative to minimise fluid leakage past the main spool (11) when the latter is in a neutral position.

2. A hydraulic valve according to Claim 1, in which the valve spool (11) is in communication with at least two chambers (27,28) in the valve body, said two chambers (27,28) being internally connected to each other and to a sump, each of said chambers (27,28) being in communication with separate passages (63,64) in the valve body (10) which passages lead to the one or more fluid connections (23,24) to the hydraulic motor, and one of the pilot operated poppet valves (34,35) being located in each passage between a chamber (63,64) and a related fluid output connection (32,33).

3. A hydraulic valve according to Claim 2, in which the main valve spool (11) has a plurality of axially spaced circumferential grooves, selected ones of said grooves being alignable with selected fluid passageways (27,63,55,64,28) in the valve body (10) to control fluid flow therethrough, and another of said grooves being a

cam groove that coacts with spring-biased actuating pins (43,47) in said pilot valves (39,48), said actuating pins (43,47) being operative to open selectively the pilot valves (39,48) to permit fluid flow to the related poppet valves (34,35) to actuate the latter in turn to provide fluid under pressure to the fluid output connections (23,24).

4. A hydraulic valve according to Claim 3, in which yet others of the grooves of the main valve vpool (11) are detent grooves, said detent grooves coacting with a biased detent (51) for holding the valve spool (11) at raise, lower and float modes.

5. A hydraulic valve according to Claim 3 or Claim 4, in which the main valve spool (11) carries at one end a cylindrical sleeve (42), said cylindrical sleeve having the axially circumferential cam and detent grooves in its surface and isolating the main spool from side loads caused by actuation of the pilot check pins (43,47) and the detent (51).

6. A hydraulic valve according to any one of Claims 3 to 5, in which said cam groove is axially elongated to provide for simultaneous actuation of said pilot valve actuating pins (43,47) when the float mode of operation is required.

7. A hydraulic valve according to any preceding claim, in which a biasing means (52) acts on one end of the axially movable main spool (11) to hold the latter at a neutral position.

8. A hydraulic valve according to Claim 7 connected between a pressure fluid source (17,21) and a load shifting hydraulic motor (25), in which upon the hydraulic motor being actuated by the hydraulic valve to raise or lower a load, only the poppet valve (34 or 35)

in the return fluid path is mechanically actuated by the cylindrical sleeve (42,82) engaging the pilot valve means (43,47), the poppet valve (35 or 34) in the fluid inlet of the hydraulic valve being actuated by the buildup of inlet pressure sufficient to move the load.

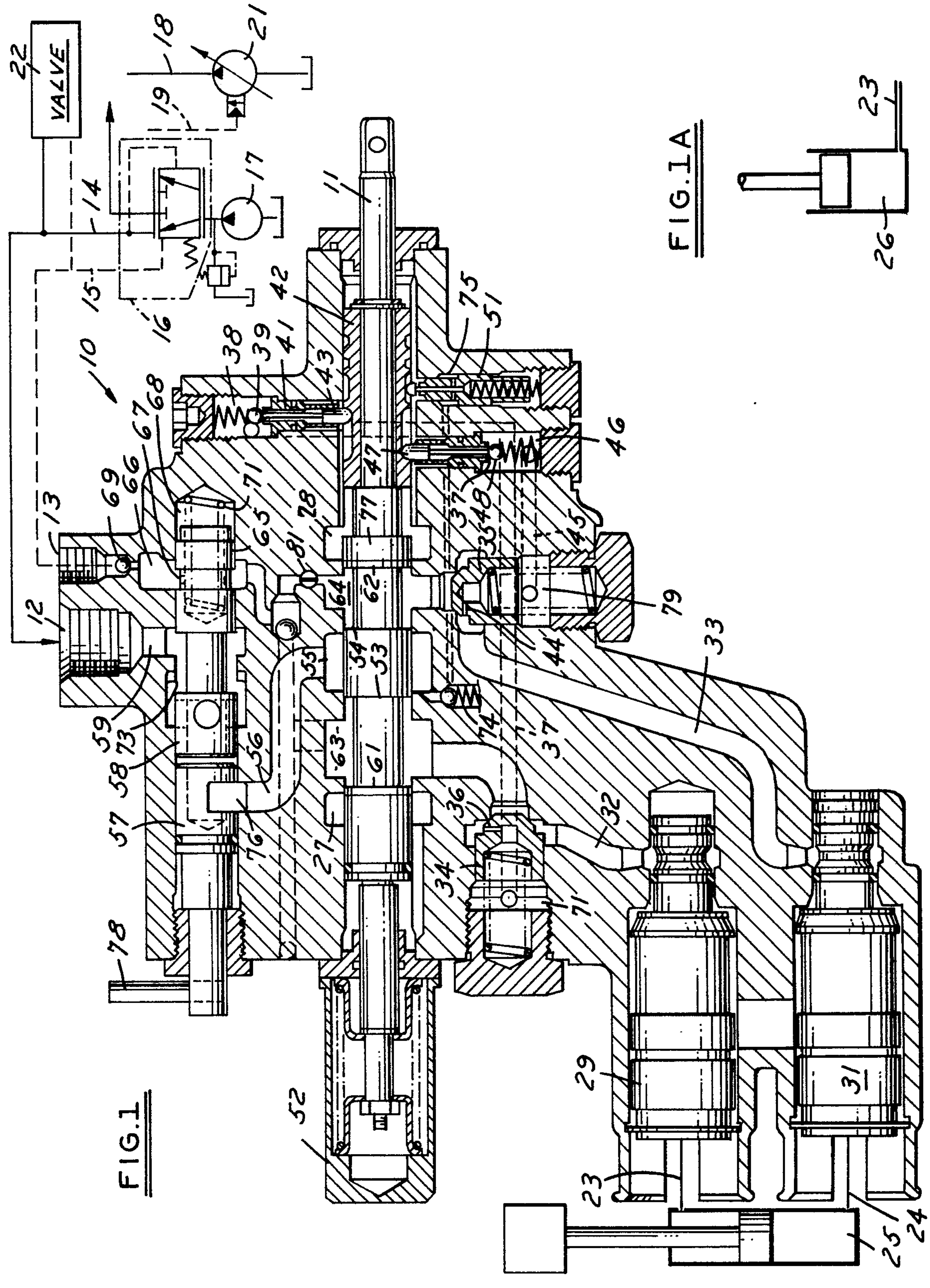


FIG.1
FIG.1A

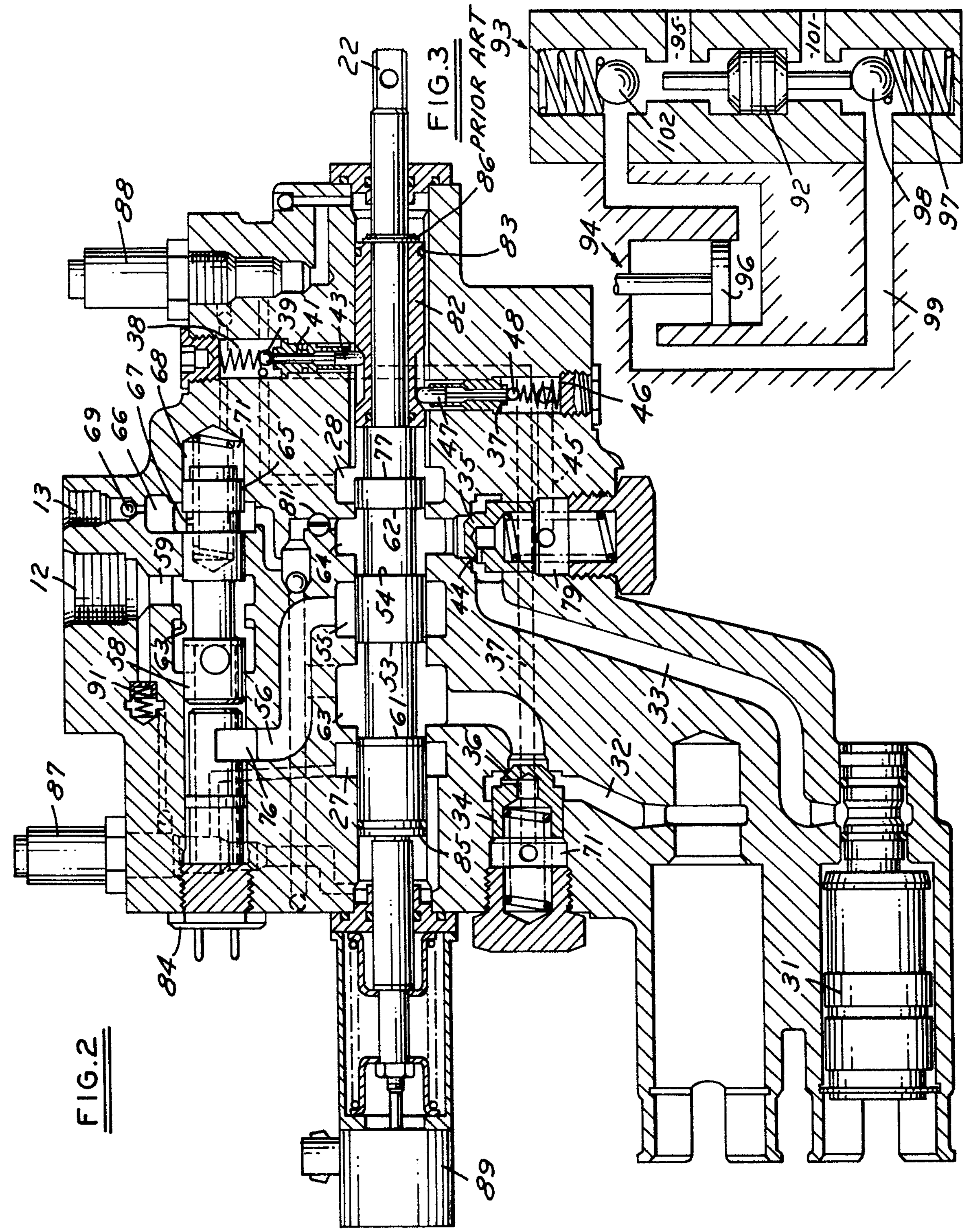


FIG.2

FIG.3 PRIOR ART