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R. E. TRICK

3,402,736

FLUID CONTROL VALVES

Filed Aug. 18, 1966

2 Sheets-Sheet 1

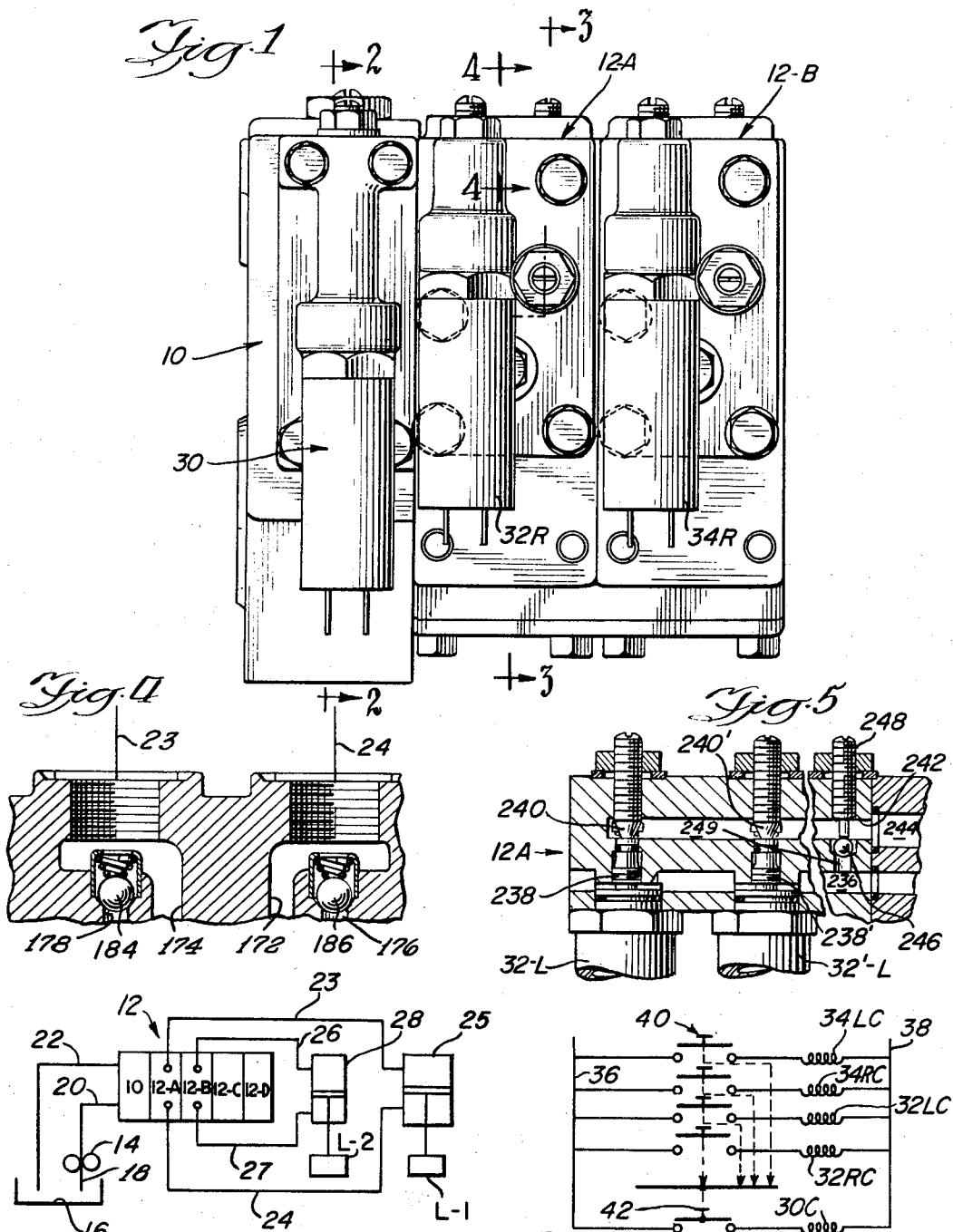


Fig. 6

Fig. 7
INVENTOR
Robert E. Trick

Mason, Koldmainen, Roldman & Hyos

BY

ATTORNEYS

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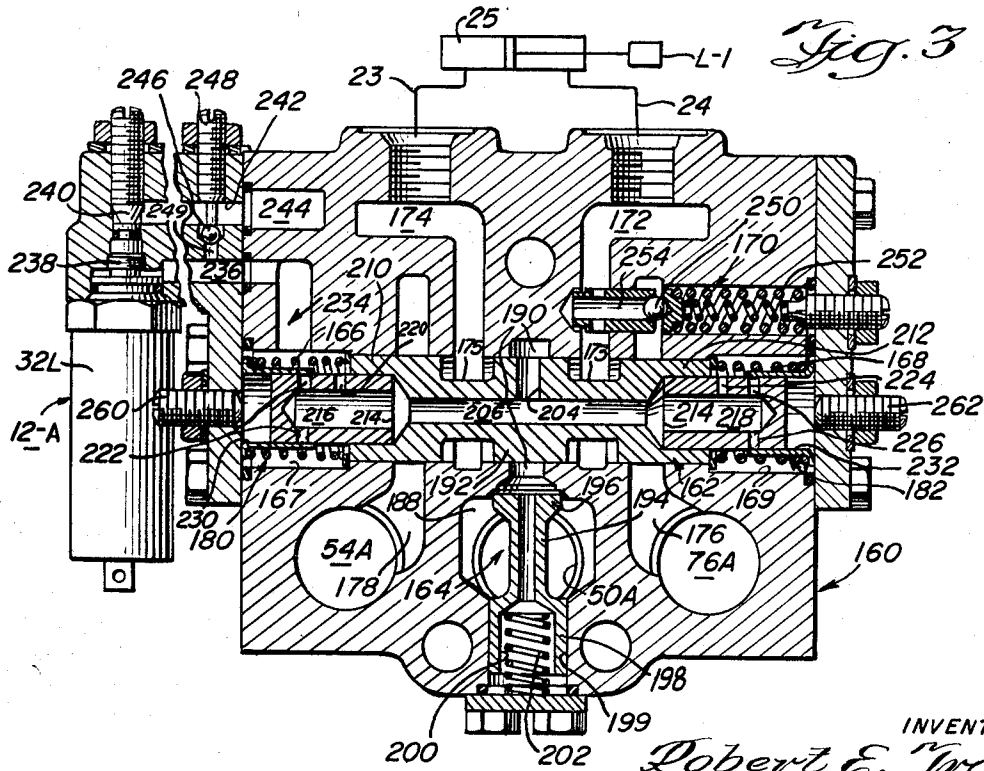
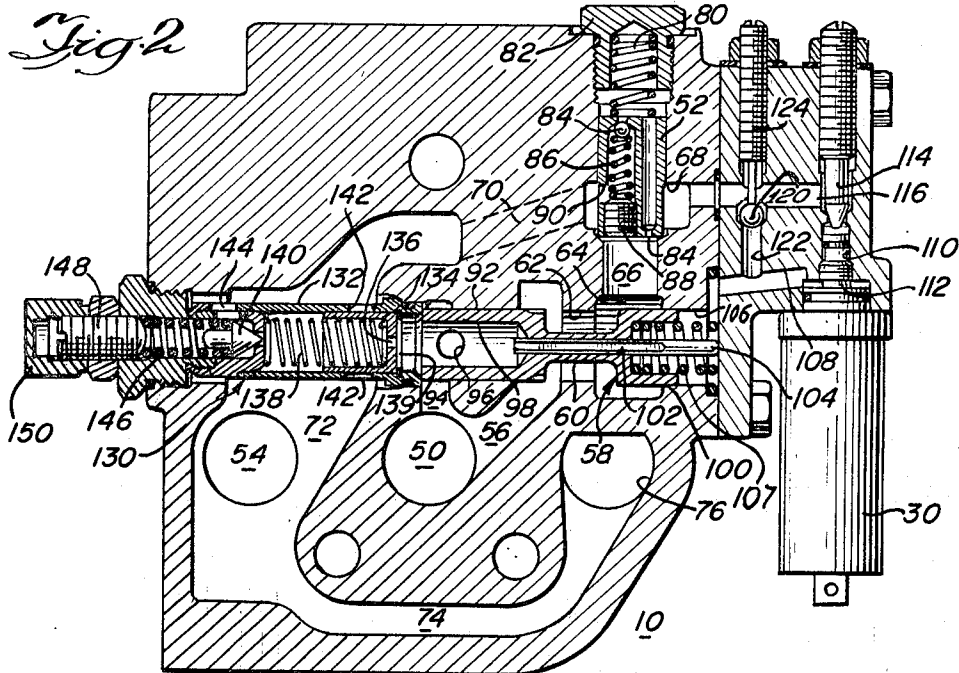
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2 Sheets-Sheet 2



INVENTOR

Robert E. Trick

By Mason, Lehmann, Rottburn & Wyse

ATTORNEYS

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FLUID CONTROL VALVES

Robert E. Trick, Racine, Wis., assignor to Webster Electric Company, Inc., Racine, Wis., a corporation of Delaware

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ABSTRACT OF THE DISCLOSURE

A hydraulic control valve including a spool valve movable in opposite directions from a central load circuit closing position, spring means acting on opposite ends of the spool for holding it in its central position, pressure chambers at opposite ends of the spool, passageways in the spool providing communication from a pressure port to the pressure chambers, orifices at the ends of the spool opening to an adjacent pressure chamber, exhaust passageways from each of the pressure chambers and means including selectively operable electromagnetic pilot valve means providing predetermined restricted flow through an exhaust passageway to effect movement of the spool to selected intermediate positions. The valve has also a pilot pressure providing bypass valve, an unloading valve upstream of the bypass valve and selectively operable electromagnetic pilot valve means for effecting movement of the unloading valve. A plurality of the control valves may be assembled in a system and control means provided for simultaneously effecting operation of the unloading valve and a selected one or ones of the plurality of control valves.

The present invention relates to fluid control valves and has for its primary object the provision of a remotely controllable, fast acting and adjustable pressure compensated full flow directional valve that can be installed in a multiple valve bank and supplied with liquid through a new and improved inlet valve unit or section.

In the growing mobile equipment field, there exists a real need for a truly mobile type valve with electrical actuation. Various factors have contributed to this need, a most significant one being that the operators are located farther away from the points of hydraulic power application and also from the main power plant. The main power plant area, of course, represents the location for the hydraulic power pump and in many cases the ideal location for the valve bank. If the valve bank is located near the operator, as is almost mandatory with manually actuated valves, it can readily be realized that excessively long hydraulic lines will have to be employed. The only practical way to circumvent all of the disadvantages inherent to these conditions is to utilize electrical actuation of the valve bank. This allows the operator effectively to control the valve bank with only a small electrical control cable acting as the sole link between the control panel and valve bank. Considerable savings will be possible because of the reduction in footage of hydraulic lines required because the lines do not have to be routed to the operator. A further result of this approach is the savings in area and the enhancing of the aesthetic appearance of the vehicle.

Accordingly, an object of the present invention is the provision of new and improved flow control valves adapted to be installed in a multiple bank arrangement and which enables the main spool of any selected single or plurality of sections in a bank to be pressure compensated flow control valves, thereby to satisfy with one valve section the requirements of a low pressure and/or low flow demand while another section or sections may

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supply oil at maximum pressure to meet other demand requirements.

A further object of the present invention is the provision of a control set forth in the preceding paragraph providing step type throttling of the flow to a load device and making it possible to use any practical number of steps.

A further object is to provide fluid control valves of the character set forth above enabling high inertia loads readily to be accelerated to move from rest in a manner tailored to a particular installation and wherein the control valves can be made to repeat the desired performance indefinitely with only push button actuation.

Another object of the invention is to provide pilot controlled directional flow valves of the spool type in which full system pressure is utilized to effect desired and controlled movement of the spools.

A further object of the invention is to provide a new and improved hydraulic control including remotely controllable electromagnetically actuated pilot valve type inlet and directional flow control valves in which the electromagnetic valve means for both the inlet and a selected direction of operation of a directional flow control valve are simultaneously energized.

In brief, the present invention provides pilot controlled directional flow valves of the spool type in which full pressure is utilized to effect desired and controlled movement of the spool and which provide pressure compensated operation. Further, the control valves have associated with them a novel pilot controlled inlet section or valve. Both the inlet section and control valves are arranged for remote operation through electric circuits controlled by push button switches and so arranged that operation of a single push button energizes solenoid valves controlling operation of both the inlet section and a selected directional operation of a selected control valve. The control valves for the inlet section includes a pilot controlled pilot pressure providing poppet valve determining the pilot pressure and an unloading valve bypassing system fluid when none of the associated directional flow control valves is energized. The inlet section also includes a system pressure responsive pilot controlled relief valve. A solenoid operated pilot valve controls the unloading valve and when the solenoid is operated the pilot valve opens a passage through which a controlled and adjustable flow takes place to effect opening of the unloading valve at a desired rate in response to the pilot pressure. In the event operation of the unloading valve becomes necessary independently of the solenoid valve, this can be done by a manually operable valve in parallel with the solenoid valve. The unloading valve, when operated, closes the system bypass provided by the poppet valve and system fluid is transmitted by the pump to a load device through a directional control valve. The latter directs the fluid to the cylinder of a hydraulic motor to operate the latter in a desired direction and at a flow rate which is pressure compensated and maintained within close limits. Each control valve comprises a spool selectively movable from a central spring biased position in one direction or another to other positions to control the flow of system fluid to and from the hydraulic motor. A load check valve associated with a fluid inlet to the spool or a pressure port prevents reverse flow of liquid. The check valve is normally closed but opens at a relatively low pressure and closes in the event of a reverse flow. The system pressure is supplied to opposite ends of the spool valve through small orifices and exhausted from a selected end through an exhaust outlet passageway when a solenoid valve is operated. The exhaust is at an adjustable and desired rate, thereby to effect shifting of the spool by pilot pressure in a se-

lected direction and to a selected position to provide a given flow rate. The pilot controlled exhaust takes place through a pre-adjusted reduced flow orifice in or associated with the body which, acting in combination with system pressure on the opposite end of the spool, provides pressure compensation maintaining the given flow rate despite fluctuations or variations in load pressure. Each end of the valve spool has second orifice means associated therewith, which orifice means is normally closed when the spool is centered. When the spool is moved in one direction, say to the left, the second orifice at the right is uncovered, thereby to supply high volume pilot oil to that end and provide rapid shift of the spool.

Anticavitation means are provided and, if desired, step type operation of the spool may be obtained by utilizing a plurality of solenoid valves in each exhaust line to withdraw oil at different rates from an end of the spool, thereby to move it into different throttling positions, in all of which the above described self-compensation takes place.

Other objects and advantages of the present invention will become apparent from the ensuing description of illustrative embodiment thereof, in the course of which reference is had to the accompanying drawings, in which:

FIG. 1 is a side view of an embodiment of the present invention illustrating a multiple valve bank including an inlet section and a plurality, two being shown, of directional control valves; although any desired number of valves can be used with the single inlet section;

FIG. 2 is a cross sectional view taken along the line 2—2 of FIG. 1 and illustrating details of the inlet section;

FIG. 3 is a view taken along the broken line 3—3 of FIG. 1 and illustrating details of one of the directional control valves;

FIG. 4 is a fragmentary view taken along the line 4—4 of FIG. 1 illustrating anticavitation means embodied in a directional control valve.

FIG. 5 is a fragmentary partly sectional view of an alternative construction of a directional control valve, such as illustrated in FIG. 2, to provide step type throttling of flow to a load device;

FIG. 6 is a schematic and diagrammatic representation of a control system embodying the present invention; and

FIG. 7 is a circuit diagram illustrating a control circuit utilized with the system of FIG. 6.

Referring now to the drawings and first to FIG. 1, the embodiment of the invention therein illustrated comprises a bank of valves including an inlet section 10 and a plurality of directional flow control valves 12 including individual valves 12A and 12B arranged in side by side relation with the inlet section. While but two control valves have been illustrated, a larger number may be utilized, as illustrated for example in FIG. 6 where four of the valves are shown in conjunction with a single inlet section 10.

The system illustrated in FIG. 6 includes a pump 14 which may be of the positive displacement or pressure compensated type and supplied with fluid from a suitable reservoir or tank 16 through an inlet conduit 18. The discharge from the pump is supplied through a discharge or outlet line 20 to the inlet section 10. When none of the control valves is in use, the inlet section, as will be described in greater detail hereinafter, bypasses the pump output and returns it to tank 16 through return line 22. Each of the control valves is connected to a suitable load device. For example, first valve 12A is connected by lines 23 and 24 to a cylinder 25 operating a load device L-1 and a second valve 12-B is connected by lines 26 and 27 to cylinder operating load L-2, the cylinders shown to be of different size to indicate different load requirements. Inasmuch as the load devices and connections thereto are known in the art, they will not be further described in detail.

In order that the inlet and control valves constituting a bank of valves may be located at a point near the power

plant and controlled from a distance by an operator, the inlet section 10 is constructed and arranged with a single solenoid valve 30 arranged to control the supply of fluid to the control valves whereas the control valves 12 are controlled by one or more solenoid valves for selected directional movement of a control valve to the right or left, such as the solenoid valves 32-R and 34-R for the right, as illustrated in FIG. 1, and valves 32-L and 34-L, of which only the former is illustrated in FIG. 3 for the left. For step type throttling by the valves 12, additional solenoid valves may be provided as will be described later in conjunction with FIG. 5.

The solenoid valves may be selectively operated through suitable circuits and controls, such, for examples, as those illustrated in FIG. 7. The various solenoid valves are supplied with associated electromagnetic operating coils indicated by the reference characters 32-RC, 32-LC, 34-RC and 34-LC. The coils are energizable from lower lines 36 and 38 upon closure of a selected push button switch 40 for the particular coil solenoid valve for the selected valve. The arrangement is such that when any push button switch is closed, a switch 42 controlling the energization of the coil 30C for solenoid 30 of the inlet section is also energized. As a result, the inlet section is activated to supply fluid to the selected control valve 12 a valve spool associated with the selected valve is moved to the right or left depending upon the solenoid valve that is operated. The arrangement, as will appear shortly, is such that a self compensated flow of fluid under pressure is supplied to the load connected to a particular control valve. Also, the flow to one control valve 12 may be at low pressure and one flow rate and that to another control valve may be at a higher pressure and different flow rate.

Referring now particularly to FIG. 2, which illustrates the inlet section 10, the oil from the pump 14 enters the section through inlet port 50 which, as noted hereinafter communicates with aligned ports 50A, 50B, in the control valves 12A, 12B, etc. The fluid supplied to the port 50 is maintained at a predetermined pressure, which may be called pilot pressure and which is generally a pressure of about seventy p.s.i. and thus considerably less than system pressures encountered when a valve 12 is actuated which may be as high or in excess of twenty-five hundred p.s.i. This pilot pressure is maintained by bypass valve of the poppet type indicated as a whole by reference character 52. Valve 52 maintains the pressure available for pilot control of the inlet section within a limited and accurate range despite considerable variation in the rate of flow of the fluid from the pump that may result when one or more of valves 12 are activated. When none of the control valves is activated, valve 52 bypasses excess oil to the return line 22 through an outlet or tank port 54, which is aligned with similar ports 54A, 54B (latter shown) of the control valves 12A, etc. The bypass extends from port 50 to valve 52 through a body passage 56 leading to a pilot operated unloading valve 58 of the spool type having a small diameter intermediate portion 60 surrounded by valve chamber portion 62 in the body adjacent an annular recess 64 around the spool and connected by passageway 66 to poppet valve 52, the latter being shown seated in its closed position in FIG. 2. The bypass extends from valve 52 to tank port 54 through an annular chamber 68, near the lower end of the poppet valve 52, and connected by passageway 70 to a return or outlet chamber 72 communicating with tank port 54 which as heretofore indicated, is connected to the pump tank 16 through return line 22. At this point it may be noted further that the chamber 72 and tank port 54 are connected by body passageway 74 to a return port 76 aligned with return ports 76A, 76B, etc. in the valves 12A, etc.

Poppet valve 52 is biased toward its closed position lightly by a spring 80 between the upper end of the valve and a spring seating closure 82 and is arranged for quick opening at pilot pressure as of about seventy p.s.i. The

opposite ends of valve 52 communicate through an orifice 24 to equalize the pressure on both sides of the valve, when the pressure reaches approximately seventy p.s.i., an unloading relief valve comprising a ball 84, a ball biasing spring 86, a spring adjusting screw 88 is opened quickly to relieve the pressure from the top side of the poppet to tank through a side hole 90 leading to the chamber 68 and passage 70. The arrangement is such that the pilot pressure is maintained accurately despite considerable variations in flow of the main supply. The valve 52 thus maintains pilot pressure and bypasses the pump discharge except when a control valve 12 is activated by energization of an associated solenoid valve, whereupon the unloading spool 58 is moved to terminate the bypass flow and to direct flow to the valve bank 12.

Prior to energization of a solenoid valve associated with a control valve 12, the pilot pressure is communicated to the left and enlarged tubular end 92 of the unloading spool 58 through a branch body passageway 94 leading from inlet port 50 to a radial opening 96 in the end 92 leading to a recess 98 in the end 92. Pilot pressure is further communicated to the right of the opposite enlarged tubular end 100 of spool 58, the end 100 having a somewhat greater diameter than end 92. The pressure is communicated along the opposed flats 102 on a stationary cylindrical axial rod 104 extending through the valve spool 58 and along which valve 58 is relatively movable. The flats communicate with a chamber 106 in the indicated position of the unloading valve, but they end short of the right end of the rod so that fluid flow to the chamber 106 is cut off by the cylindrical end of the rod when the spool has moved its full stroke to the right against a valve biasing spring 107 in the chamber 106 and acting against the left end of the valve 58. Actually, spring 107 may be eliminated when the diameter of spool end 100 is greater than that of end 92. The pressure in chamber 106 is communicated through an exhaust passageway in the valve body including a portion 110 normally closed by a movable valve element 112 forming part of and adapted to be opened upon energization of solenoid valve 30. When passage 110 is opened and the unloading valve 58 moves to the right at a rate determined by the adjustment of a needle valve 114 controlling flow through the passageway to the tank port 54 through exhaust passageway portion 116, the previously described passageway 70 and chamber 72.

Prior to energization of the solenoid 30, the unloading spool 58 is held in its indicated open position primarily due to the different effective areas acting upon the left and right ends of the end 100 of the spool subject to equal pressures, plus slight biasing exerted upon the spool by the previously described spring 107.

When the solenoid 30 is energized, oil flows at a rate determined by adjustment of the needle valve 114 from the chamber 106 at the right of the spool to the tank. As a result, the unloading spool 58 is moved to the right at a controlled rate to close the bypass flow past the poppet valve 52. The flow past poppet valve is terminated by the entry of the spool portion 92 into valve chamber or recess portion 62. As earlier indicated, when the unloading valve is fully stroked, the circular end of rod 104 cuts off flow of oil to the solenoid valve controlled passageway, thereby to reduce power losses as well as the amount of oil that has to flow through critical flow areas.

In the event it is desired manually to effect movement of the unloading valve 58, this can be accomplished by a manual actuator comprising a ball valve 120 normally closing a passage 122 shunting the solenoid control bypass and a screw 124 holding the ball seated. The screw is loosened approximately one turn, whereby the unloading valve 58 is shifted in the same manner as under solenoid valve control.

The inlet section is also provided with system oil pressure relief through a relief valve indicated as a whole by reference character 130 of the pilot actuated type and

adapted to relieve excess system pressure by bypassing system fluid to the tank port 54. The relief valve comprises a tubular carrier 132 fixedly secured in the body so as to extend across the outlet chamber 72 into abutting relation with unloading spool 58 in its unloading or left position. Adjacent the abutting end is a movable bypass valve element 134 having a central orifice 136 and biased by a spring 138 to the right to a limit position determined by a ring 139 in the carrier 132. System pressure is communicated to valve element 134 through the branch passageway 94 and the port 96 when the unloading valve is in its indicated position or directly from 94 when the valve is moved to the right. When the pressure exceeds a predetermined safe value such as 2500 or more p.s.i. the system pressure acts on poppet valve 140 through orifice 136 and when valve 140 opens the pressure back of valve element 134 is quickly reduced so that the latter rapidly opens a bypass to tank port 54 through a plurality of openings 142 in tube 132. Poppet valve 140 relieves system pressure to tank through port 144, is biased toward closed position by a spring 146, the compression of which is adjustable by a screw 148 covered by a cap 150.

As indicated earlier, the inlet section solenoid valve 30 is energized simultaneously with any one of the control valve solenoids 32 and 34. As described, actuation of the solenoid valve 30 results in closing of the unloading valve and the establishment of system pressure in the inlet port 50 and the ports 50A, 50B, etc. aligned therewith and extending to the valves 12 in the bank of which only 50A is shown. It will also be recalled that tank ports 54 and return ports 76 in the inlet valve section 10 are in communication with ports 54A and 76A of the control valve, as indicated in FIG. 3 with respect to valve 12-A. Each of the control valves 12 of the invention includes a body 160, illustrated in connection with valve 12A, having a valve chamber in which is mounted spool type valve 162 selectively movable in opposite directions from a normal control position upon operation of solenoid pilot valves 32-L or 32-R, of which the former only is shown in FIG. 3 to which reference will now be had. In the main, each of the control valves 12 also includes a tubular load check valve, indicated as a whole by reference character 164, in the inlet port 50-A, centering springs 166 (in pressure chamber 167) and 168 (in pressure chamber 169) in the opposite ends of spool 162, and load or cylinder pressure relief valve 170 associated with a so-called motor passageway-port 172 connected by line 24 to cylinder 25 operating load L-1. A similar pressure relief valve, not shown, is provided for the motor passageway-port 174 connected by line 23 to the load device. Passageways 172 and 174 lead to annular recesses 173 and 175 on the spool at opposite sides of the control land 192 which are adapted when the spool 162 is shifted in one direction or the other to direct fluid from pressure port 190 to a selected end of a load cylinder and to return it from the other end to tank port 54 through one of passageways 176 or 178 leading to tank ports 54 and 76A, respectively.

Additionally, the spool valve 162 has associated with it structures indicated by the reference characters 180 and 182 at its ends constructed and arranged to provide pressure compensation and to prevent excessive power loss due to wasting high pressure pilot oil when shifting the spool 162. Means comprising ball valves 184 and 186, spring biased to closed position shown in FIG. 4, are provided for anticavitation purposes. The ball valves normally close passageways leading from the cylinder passageways 172 and 174 to the passageways 176 and 178 connected to the tank ports 76A and 54. Accordingly, in the event of a system condition requiring additional oil to prevent cavitation, the ball valves are free to be opened and supply tank oil to alleviate the condition. The valves are lightly biased to closed position.

Returning now to detailed consideration of the various portions of a valve 12, when the inlet section 10 un-

loading valve 58 is actuated and the pressure reaches a predetermined valve adequate to open check valve 164, oil flows from port 50-A and the valve inlet chamber 188 to an annular pressure port or passageway 190 above the check valve and surrounding a central land portion 192 of the spool valve. The check valve 164 has a reduced diameter elongated portion 194 extending substantially across the chamber 188. At the upper end, as viewed in FIG. 3, of the reduced portion is an enlarged head 196 normally closing passageway 190. At the lower end of the reduced diameter portion is an enlarged head portion 198 somewhat greater in diameter than 196 extending into a closed chamber 199 and biased upwardly by a spring 200 located in a cavity 202. The load check valve opens when the force due to pressure in chamber 188 acting on the area of 198 subject to the chamber pressure minus the area of 196 subject to the chamber pressure is greater than the force in the pressure chamber 190 acting on top of head 198 minus the area of head 196 subject to the chamber pressure plus the force of spring 200. The tubular construction of the load check valve ensures that the pressure at the underside of the load check valve, once it is opened, is equal to the pressure above the load check valve. When the pressure relationship set forth above is reversed, the load check valve closes and will effectively block any reverse flow from the load cylinder. The check valve opens but a slight amount and never bottoms in chamber 199. Pilot fluid to opposite ends of the spool valve 162 is supplied from annular pressure port 190 through radial and axial passageways 204 and 206, which lead to the pilot flow controlling means 180 and 182, respectively. The latter includes closed end port defining inserts 210 and 212 at opposite ends, the open ends of which communicate with opposite ends of passageway 206 through filter screens at locations indicated by reference characters 214. The inserts have axial passageways 216 and 218, respectively, which project somewhat beyond the ends of the spool whereat they have first and second radial orifices 220 and 222 in insert 210 and 224 and 226 in insert 212 and of which the outermost orifices 222 and 226 are larger than the inner ones. Actually there are two of each of the latter orifices, for balancing pressures acting on the inside of the spring guides 230 and 232 which are encircled by the spool centering springs 166 and 168. Each of the inserts is encircled for part of its length by the associated spring guide 230 or 232. The spring guides cover the larger outer orifices 222 and 226 when the spool is centered. When a spool is shifted in one direction or another, the smaller orifice 220 (upon a shift to the left) will also be covered to terminate flow of pilot oil to be exhausted and thus save power and reduce flow while the larger port 226 at the opposite end will be uncovered to deliver a high volume of pilot oil to chamber 167 to actuate the spool to the left more rapidly under system pressure. Pilot flow supplied through the orifices in the inserts to the spring cavities enter, at the left, an exhaust passageway 234 in the valve body and a similar passageway (not shown) at the right. From passageway 234 a passageway 236 leads to the solenoid valve 32-L. When the solenoid valve which comprises a normally closed movable valve element 238 is energized the movable valve element 238 is retracted and oil exhausts at a rate determined by the by the adjustable needle valve 240 to a passageway 242 leading to a body passageway 244 communicating in a manner not shown with the tank port 54. Similar passageways controlled by the solenoid valve 32R communicate with the tank port 76A.

In the event manual shifting of the spool is desired, a ball valve 246 is adapted to be opened upon rotation of a ball holding screw 248 thereby to bypass the solenoid valve through passageway 249.

The cylinder pressure relief valve 170 comprises a ball 250 biased by a spring 252 against a tubular insert 254 the inner end of which communicates with the cylinder passageway 172 and the outer end with passage 176.

The cylinder passageways 172 and 174 are selectively connected to the pressure port 190 beyond the load check valve 164 upon shifting of the spool in one direction or another. The communication is effected through the previously mentioned annular passageways 173 and 175, the inner edges of which provide pressure compensation with the pressure port 190 as described later herein.

In the event it becomes necessary for any reason to effect manual shifting of a control spool, the spool can be so shifted in one direction or another by screwing in a set screw 260 to engage the left end of the spool and shift it to the right. Similarly, the spool can be shifted to the left by screwing in the set screw 262.

In the event it is desired to have a single directional flow control valve operate to provide different flow rates, the valve may be provided, as indicated in FIG. 5, with a second solenoid valve 32'-L with its movable valve element 238' and adjustable valve seat 240' adapted to be energized by an associated switch in the same manner as solenoid valve 32-L.

In the description of the operation it should be noted that the valve sections 10 and 12A are both illustrated in positions occupied by them when there is no flow of fluid. When installed and ready for operation the pump 14 is energized to supply fluid through its discharge line 22 to the inlet port 50 and unloading valve 58 of inlet section 10. When the pump pressure reaches some relatively low value such as 70 p.s.i. the pilot pressure providing bypass valve 52 is opened following the opening of valve 84 associated therewith. The entire pump output is bypassed through passageways 70 and 72 to the tank port 54 from whence it is returned to tank through line 22. Pilot pressure exists in recess 98 of the unloading valve and in the chamber 106 at the end of the valve spool, the pressure being communicated thereto by means of the flats 102 on the rod 104. Pressure is maintained in chamber 106 by reason of the closure of the exhaust passageways 108, 110 by the movable valve element 112 of the solenoid valve 30 and by the manually operable ball valve 120. Pressure is supplied to the system relief valve 130 but the latter will not open because it is intended to open at maximum pressure for which the system is intended.

When it is desired to actuate a load, such as load L-1, in one direction valve 12-A is actuated by operation of solenoid valve 32-L by closure of switch 40 to energize the coil 32-LC. The coil for solenoid valve 30 of the inlet valve section is simultaneously energized by the switch 42.

When solenoid valve 30 of the inlet section is operated its valve element 112 is moved to exhaust fluid from chamber 106 at a predetermined rate determined by the setting of the needle valve 114, the exhaust being to the tank port 54 through the passageway 116, recess 68 and passageways 70 and 72. As a consequence of the reduction in pressure in chamber 106, the unloading spool moves to the right. Flow to the poppet valve 52 is terminated by the entry of spool portion 92 into passageway 62 and builds up the system pressure which is transmitted to the inlet port 50A of the control valve 12A, as well as the inlet port of the other valve sections. The wastage of pilot fluid past the solenoid valve 30 is prevented when the unloading spool moves to the right so that the circular end of rod 104 extends into the axial opening in the spool. Also, the system relief valve 130 remains subject to system pressure.

The resulting increased pressure in the inlet port 50A and chamber 188 effects downward or opening movement of the load check valve 164 with the result that pressure is supplied to the pressure port 190 of valve 12-A. This pressure is applied to opposite ends of the spool 162 through the passageways 204 and 206 and the smaller orifices 220 and 224 at opposite ends of the spool to the pressure chambers 167 and 169, the larger orifices 222 and 226 being closed by the spring guides and to the exhaust passageway, such as the passageways 234, 236, 244 associated with solenoid valve 32-L.

When the solenoid valve 32-L is operated there is a slow and adjusted flow of fluid from the pressure port 190 to tank at a rate determined by the setting of the needle valve 240 whereby the control spool 162 shifts to the left to a predetermined position. The pressure port 190 is then connected through spool recess 173 to the motor port 172 and thence through line 24 to one end of cylinder 25 to move the cylinder piston to the load L-1 to the left as viewed in FIG. 3 and the fluid at the other end of the cylinder is exhausted through line 23 motor port 174 recess 175 and passageway 178 to the tank port 54A. Excess motor pressure line is relieved through relief valve 250 which is adapted to open at some predetermined pressure above the normal system pressure.

The arrangement is such that a predetermined pressure compensated constant rate of flow is supplied to the load through the spool recess 173 regardless of the load demand pressure. This operation is accomplished by the cooperative action of full system pressure at the right end of spool 162 in pressure chamber 169 applied through radial port 204 and axial passageway 206 and the exhaust of fluid past orifice 220 and the restricted flow past the needle valve 240. The orifice 220 which is smaller than 222 normally remains open and is covered partly by the spring guide only when the valve 240 is adjusted to be substantially wide open and the control spool moves to its extreme left position. It should also be mentioned that the clearance between the spring guides and the spool inserts is so close as to maintain the slip losses to a bare minimum.

In practice it has been found that the control valve of the present invention substantially maintains a given flow rate determined by the adjustment of orifice 240 regardless of the load pressure. The flow rate is determined by the position to which the spool 162 is moved and this is determined by adjustment of orifice 240. This adjustment establishes a percentage of the system pressure acting on the left end, as illustrated, of the spool in conjunction with the associated spring 180. The opposite end of the spool is acted upon by the system pressure or more accurately, the system pressure less the pressure drop from pressure port 190 to motor port 172. Both ends of the spool have the same area so that a balance occurs at the predetermined flow which takes place from the pressure port 190 to the recess 173 and motor port 172. Assuming the load pressure to increase with its accompanying decrease in rate of flow, the spool is shifted to the left to increase the flow rate and, thus, compensates for restricting valve 240. Accordingly, the spool shifts to the left to increase the flow rate and, thus, compensates for the reduction in flow rate resulting from an increase in load pressure in the motor port. A reverse action takes place with an increase in flow rate, i.e., there is a resulting shift of the spool to the right to decrease the flow rate.

The larger orifices 222 and 226 are provided in order to make the spool fast acting. While normally closed by the spring guides one or the other of the larger orifices become uncovered as the spool moves in one direction or the other. A greater fluid flow is thus supplied to the pressure chamber, such as to chamber 169 upon movement of the spool to the right, so that the spool is moved rapidly by system pressure.

In the event the spool is approximately at its limit of movement the smaller orifice at the end is partly covered by the spring guide. As a result the flow of pilot oil through the valve is terminated to reduce the power losses and wastage of pilot oil.

If step type operation is required or desired, then two or more of the solenoid valves may be provided in place of the single solenoid valve 32-L.

In the event conditions in the lines to the cylinder require additional oil to prevent cavitation, the lightly biased anticavitation valve 184 or 186 will open at a low pressure to supply the requirements. An advantage of the present system is that each of the valves 12 is operable independently of the other to provide the predetermined self

compensating flow to its associated load. In other words, the requirements of load L-1 and the operation of valve 12-A are independent of the requirements of load L-2 and the operation of valve 12-B—or other loads and valves.

In the event the solenoid valves are not operable to effect shifting of the spool in the inlet and control valve sections, the solenoid valves may be bypassed. For the inlet section 10 the valve 120 may be opened by loosening screw 124 and for the control valves the ball valves corresponding to ball valve 246 may be opened by unloosening screw 248. As earlier explained, the opening of these valves exhausts the pressure chambers associated with the spools, thereby to effect shifting of spools.

In the event a positive manual shift of the control valve spool 162 is necessary, this can be accomplished by movement of the screw 260 or 262 to engage and positively move the spool in the desired direction.

Additionally, it may be mentioned that a single inlet section 10 may be utilized in an arrangement such as illustrated in FIG. 6. Additional banks of control valves 12 may be mounted in the same manner as the illustrated bank 12 but without the use of an additional inlet section, simply by connecting a supply line 20 to the inlet port 50 and return line 22 to the tank ports 54 and 76A.

While the present invention has been described in connection with the illustrative embodiment, these details are not intended to be limitative of the invention except insofar as set forth in the accompanying claims.

I claim:

1. A hydraulic control valve for a load device movable in opposite directions, including in combination, a valve body including a valve chamber and pressure, motor and tank ports communicating with the chamber, a spool valve slidably mounted in said chamber for movement from a central position in which the pressure and motor ports are closed in opposite directions to other positions in which the pressure port and tank ports are selectively connected to the motor ports to operate the load device in opposite directions, spring means acting on opposite ends of the spool valve for holding it in its central position, pressure chambers at the opposite ends of the spool, passageways in the spool providing communication from the pressure port to the pressure chambers, orifices at the ends of the spool opening to the pressure chamber at the adjacent end of the spool, an exhaust passageway from each of the pressure chambers, and means including selectively operable electromagnetic pilot valve means in the exhaust passageways providing predetermined restricted fluid flow through said exhaust passageways to effect movement of the spool to selected intermediate positions.

2. A hydraulic control valve as claimed in claim 1 wherein the pilot valve means is adjustable to provide a desired restricted flow.

3. A hydraulic control valve as claimed in claim 1, including a plurality of said electromagnetic pilot valve means in each of said exhaust passageways that are selectively operable to effect movement of the spool to different selected intermediate positions in opposite directions from its central position.

4. A hydraulic control valve as claimed in claim 1, wherein the passageways in said spool includes an axial passageway communicating with the orifices at the ends of the spool and a central radial port leading to said passageway from the pressure port.

5. A hydraulic control valve as claimed in claim 4 wherein the ends of the spools are provided with closed end inserts having open ends communicating with the spool passageways and the orifices are in said inserts.

6. A hydraulic control valve as claimed in claim 4 wherein spring guides closely encircle the opposite ends of the spools and are located in the pressure chambers and the spring means encircle the spring guides.

7. A hydraulic control valve as claimed in claim 4 wherein there are two axially spaced orifices at each end

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of the spool, said orifices being of different sizes and the outermost orifice being the larger.

8. A hydraulic control valve as claimed in claim 7 including spring guides in the pressure chambers overlying and closing the outer larger orifices when the spool is in its central position.

9. A hydraulic control valve as claimed in claim 8 wherein the spring guides almost entirely close also the smaller orifices at the limit of movement of the spool.

10. A hydraulic control valve as claimed in claim 4, including a load check valve upstream of the pressure port.

11. A hydraulic control valve as claimed in claim 10 wherein said load check valve operates on a differential area principle.

12. A hydraulic control valve, including in combination, a valve body having an inlet port and a tank port, a pilot pressure providing bypass valve between the inlet and tank ports, and means including an unloading valve upstream of the bypass valve movable from a first to a second position for terminating the bypass flow and having a pressure chamber at one end communicating with the inlet port, and selectively operable electromagnetic pilot valve means for exhausting the pressure chamber to effect movement of said unloading valve from its first to its second position.

13. A hydraulic control valve as claimed in claim 12 wherein said bypass valve includes a movable valve element having a closed end subject to pump pressure and having an orifice supplying the pressure to a chamber back of the element, spring means biasing said element to a position to close the bypass flow, and a spring pressed normally closed valve between the said chamber and the tank port.

14. A hydraulic control valve as claimed in claim 12 wherein said unloading valve is of the spool type, has first and second enlarged end portions communicating respectively with the pressure chamber and inlet port and an intermediate section of reduced diameter interconnecting the inlet port and bypass valve, and communication to the pressure chamber through the spool is cut off when the spool is moved to terminate bypass flow.

15. A hydraulic control valve as claimed in claim 14 wherein the communication through the spool to the pressure chamber is effected by flats on an axial rod in the spool.

16. A hydraulic valve as claimed in claim 14 wherein said second end portion of the unloading valve is tubular, the valve body has a branch inlet port leading to near the end of said second end portion and the latter has a radial port communicating with the branch inlet passage-way when the unloading valve is in its first position.

17. A hydraulic valve as claimed in claim 16 wherein the second end of said unloading valve communicates with a pilot operated relief valve relieving excess pressure to the tank port.

18. A hydraulic control system for interconnecting a pump having an outlet and return and a plurality of load devices, including in combination, a plurality of directional flow control valves of the spool valve type all having

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inlet ports connected to the pump outlet and return ports connected to the return, means for individually operating the spool valves selectively to connect an associated load device to the pump outlet and return for operation of the device, and said spool valves being of the pilot controlled type and each including pressure chambers at opposite ends of the spool subject to system pressure, and solenoid type pilot valve means selectively operable to exhaust the pressure in a chamber at a predetermined reduced rate to effect operation of the spool to selected intermediate positions.

19. A hydraulic control system as claimed in claim 18, including a pilot pressure providing bypass valve having inlet and return ports connected to the others and bypassing the pump outlet to return and controlled by solenoid type pilot valve means.

20. A hydraulic control system as claimed in claim 19, including switch means and circuits controlled thereby for simultaneously operating the solenoid type pilot valve means for said bypass valve and a selected one of the control valves.

21. A fluid control system for operating a load device, including in combination, a fluid pump having an outlet and return, and control means selectively supplying fluid from said pump outlet to the load device, said control means including first and second valve means between the pump and device, said first valve means normally bypassing the fluid flow from the pump outlet to return and maintaining the fluid at pilot pressure and including a pilot operated unloading valve and first controllable electromagnetic pilot valve means controlling said unloading valve for terminating the bypass flow, and said second valve means including pilot operated directional flow control valve means operable from one position in opposite directions to second and third positions for selectively supplying fluid from the pump outlet to said load device to operate the latter and a plurality of other selectively operable electromagnetic pilot valve means for effecting operation of said directional valve means in each of the opposite directions.

22. A fluid control system as claimed in claim 21 including switch means and circuits controlled thereby for simultaneously closing circuits for said first and selected ones of said plurality of electromagnetic pilot valve means.

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HENRY T. KLINGSIEK, *Primary Examiner.*