United States Patent

[11] 3,628,567

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[21] Appl. No. **46,109**
[22] Filed **June 15, 1970**
[45] Patented **Dec. 21, 1971**
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[32] Priority **June 14, 1969**
[33] **Germany**
[31] **P 19 30 387.8**

[54] **POWER CONTROL VALVE**
9 Claims, 4 Drawing Figs.

[52] U.S. Cl. **137/596.13,** 137/596.2, 137/625
[51] **Int. Cl.** **F16k 11/07**
[50] **Field of Search** 137/596.13, 596.2, 625.6, 625.63, 625.64, 625.65, 625.68, 596.12, 596.14, 625.69

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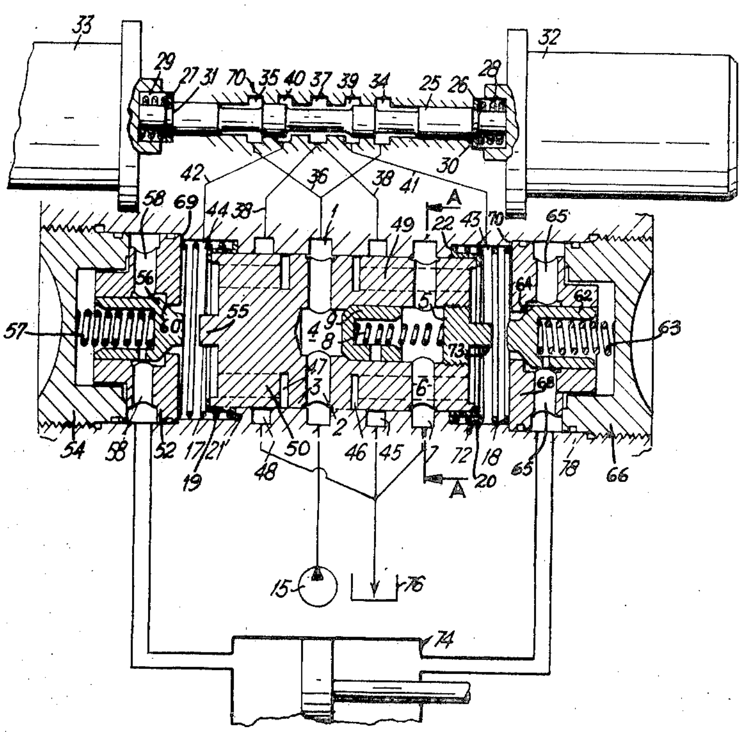
ABSTRACT: A two-stage hydraulic valve system for controlling a double-acting load the system having a three-position first stage valve for supplying pressure medium to pressure chambers on either end of a piston valve slide in a power control valve having twin lift valves for connecting the pressure source to the double-acting load. The lift valves, which are in the ends of the piston valve cylinder connect the load to the pressure chambers when the valves are opened, either by the movement of the piston valve slide against the lift valve, or by the increased pressure in the pressure chamber when the valve slide is at one end of its stroke and the pump pressure is applied to the pressure chamber at the other end of the valve slide. In the neutral position of the power valve there is free circulation of pressure medium from the source to the reservoir through a pressurizing valve. This circulation is cut off when the valve is actuated and the full pump pressure applied to one of the pressure chambers by means of valving passages and ports in the cylinder and slide.



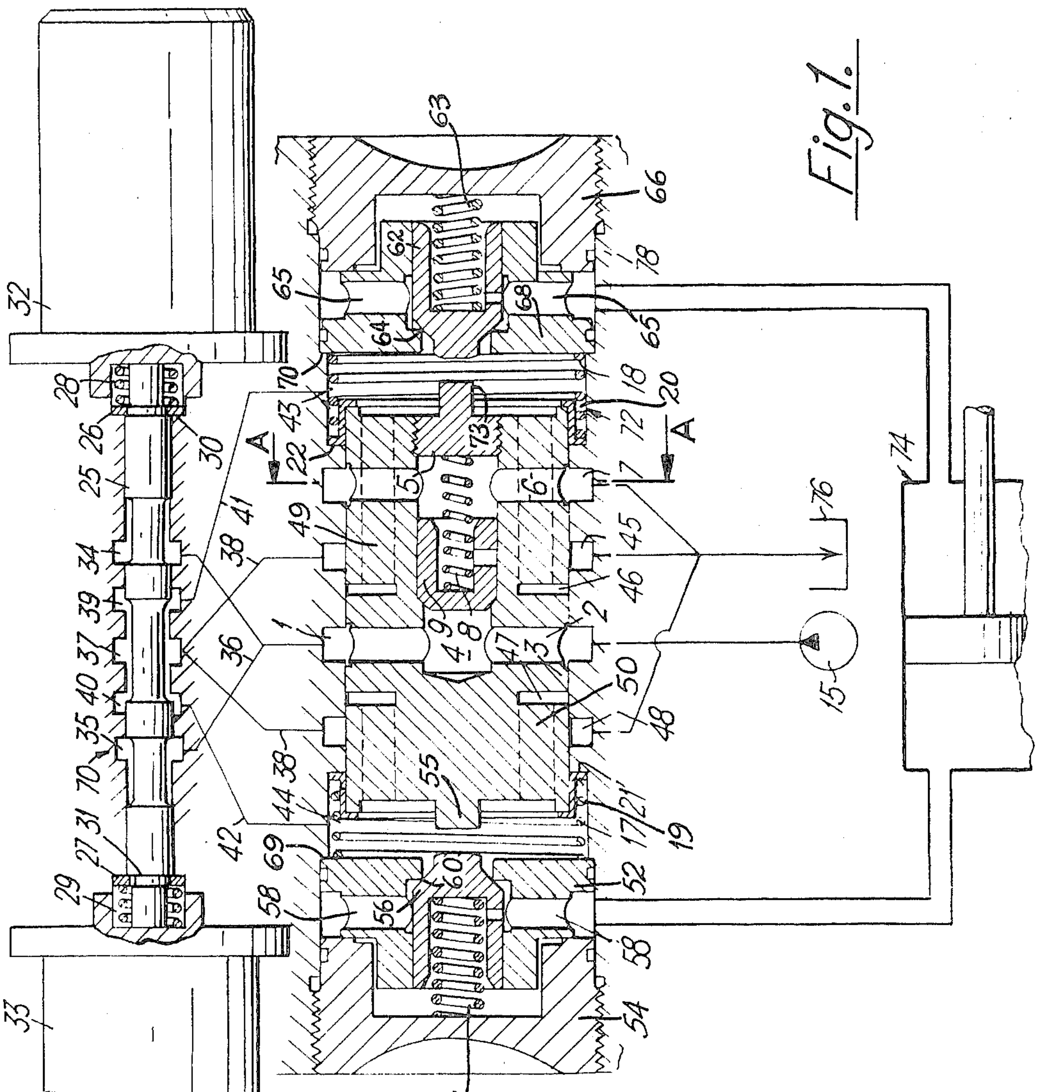


Fig. 1.

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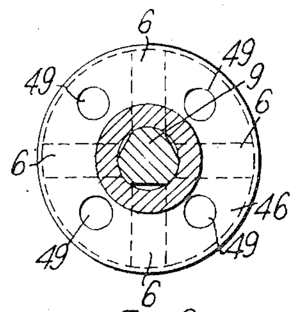

Fig.2.

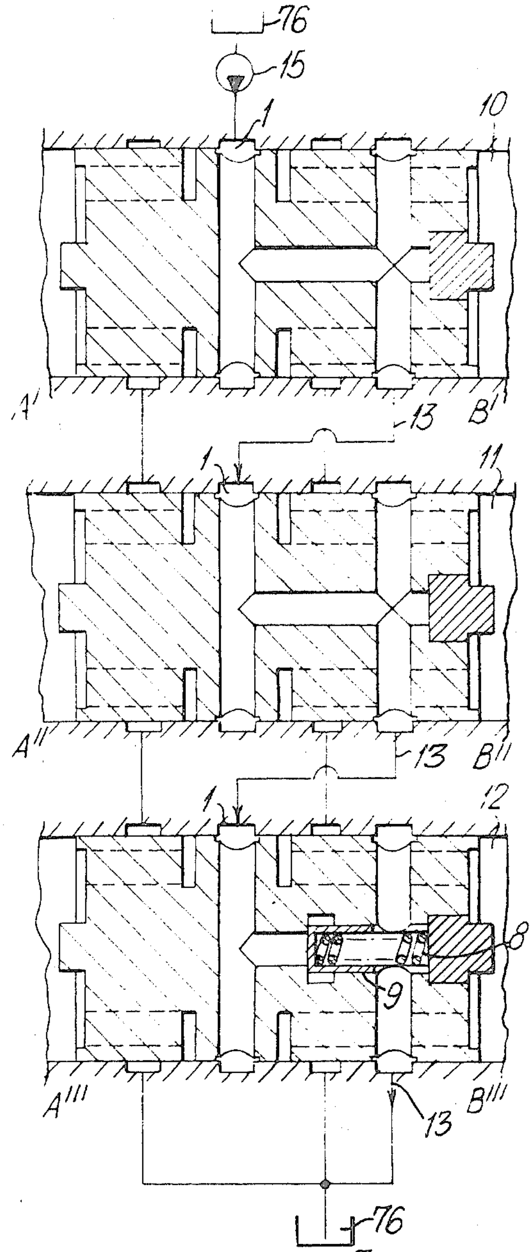

Fig.3.

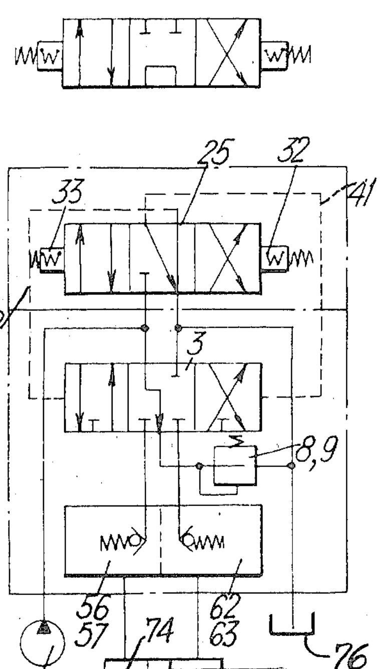

Fig.4.

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POWER CONTROL VALVE

BACKGROUND OF THE INVENTION

This invention relates to hydraulic control valves and particularly to the second stage power valve of a two-stage system for controlling a double-acting load or consumer.

There are many two-stage valve systems for controlling a double-acting load. One such system comprises a servocontrolled circulation valve which controls the circulation of pressure medium from a pump to a reservoir in the rest or neutral position of the system, two consumer valves which respond alternately according to a mutual servocontrol and a two-way twin check valve, the actuating member of which is applied by the pressure controlled by the consumer valves. The double acting consumer is applied by the pressure medium flow, the latter being released in one direction or the other by the twin check valve. The servocontrol of both the circulation valve and the consumer valves can be electric, mechanic, or hydraulic.

The separate construction of the various valves in this system increases the response time of the system as well as the space required to house the components.

According to another suggestion, the pressure resulting from a pressure medium circulation which is interrupted by a directly controlled slide valve, acts upon the actuating member of the twin check valve. The slide itself is kept in a floating position inside the valve chamber during rest position. Since the pressure medium flows through the chamber of the slide valve, the actuation of said valve will require a correspondingly great external force and consequently a larger electromagnet if the actuation is made electrically, the electromagnet requiring not only much more space but also a relatively long time to be energized, the result being a greater response time of the total system.

For the installation of a smaller magnet the slide valve could be servocontrolled and the pressure medium circulating through the slide valve prestressed by a pressurizing valve. The slide of the main valve is then controlled by actuation of the servocontrol valve, said slide interrupting the pressure medium circulation and deviating the pressure medium for the actuation of the twin check valve. This, however, would not sufficiently improve the total response time.

SUMMARY OF THE INVENTION

It is an object of the present invention to provide a compact and less costly power control valve assembly with twin lift valves for connecting the pressure source to a load.

It is a further object to provide a valve assembly of this type which has less response time than the prior known systems.

A two-stage hydraulic valve system for controlling a double-acting load, the system having a three-position first stage valve for supplying pressure medium to pressure chambers on either end of a piston valve slide in a power control valve having twin lift valves for connecting the pressure source to the double-acting load. The lift valves, which are in the end of the piston valve cylinder, connect the load to the pressure chambers when the valves are opened, either by the movement of the piston valve slide against the lift valve, or by the increased pressure in the pressure chamber when the valve slide is at one end of its stroke and the pump pressure is applied to the pressure chamber at the other end of the valve slide. In the neutral position of the power valve there is free circulation of pressure medium from the source to the reservoir through a pressurizing valve. This circulation is cut off when the valve is actuated and the full pump pressure applied to one of the pressure chambers by means of valving passages and ports in the cylinder and slide.

Specifically, this invention provides a piston-type valve slidably arranged in the valve housing and controlling the connections between valve pressure chambers on either side of the piston and the pressure medium source, the high-pressure outlet and the return line; said valve piston or slide opening in its two end positions, the valve bodies of two oppositely

disposed lift valves in the housing, said lift valves being arranged in the connection lines of the valve pressure chambers to the consumer connections.

Preferably parallel radial bores which align with circular grooves of ports in the housing as well as a longitudinal bore connecting the radial bores, are arranged in the valve piston, the valving piston acting in both directions and providing actuating members for the seat valve bodies of the twin lift valve. In the neutral position of the valve slide there is a free pressure medium circulation from the pressure medium source, via a circular groove in the housing, the radial bores and the longitudinal bore of the piston and a second circular groove in the housing to the reservoir. A pressurizing valve is arranged in one of the passageways. Axial bores in the valve piston are provided for the flow to and from the consumer or load, which bores are connected to the reservoir via circular grooves in the piston and in the housing upon actuation of the valve slide.

The pressurizing valve is preferably arranged in the longitudinal bore connecting the radial bores of the valve slide.

The effective surfaces of the valve slide are provided with pins for the actuation of the seat valves of the twin lift valves, the seat valves closing the consumer or load connections. The first stage pilot valve and the second stage power valve have three positions. Preferably both first stage valve and the valve slide piston of the twin lift valve are centered by helical springs and pressure-balanced in the neutral or zero position. The first stage pilot valve provides five circular grooves of which the three grooves in the center are interconnected in the neutral position of the valve, the center groove being connected to the circular grooves of the second stage which are connected to the reservoir, whereas the other two circular grooves of the first stage servo-pilot valve are connected to the actuating pressure chambers located between the valve piston and the seat valves of the second stage valve.

In the neutral position the two outer circular grooves of the first stage servo-pilot valve are separated from the three center grooves and connected to the center circular groove of the second stage power valve which is connected to the pressure medium source.

In the two actuated positions of the first stage pilot valve, one of the two outer circular grooves is connected to the adjacent circular groove which is connected to one of the pressure chambers of the second stage power and separated from the other grooves whereas the second outer circular groove is separated from the adjacent groove, which again is connected to the center groove.

First stage pilot valve is actuated either manually, hydropneumatically, mechanically, or electrically.

Advantageously several valves can be connected in series in such a way that the circulation lines of the various valves are hydraulically connected in series with only the last valve which is directly connected with the reservoir, having a pressurizing valve.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 shows a valve system embodying the present invention.

FIG. 2 shows a cross section through Section A—A of the power valve shown in FIG. 1.

FIG. 3 shows an arrangement of three power valves of type shown in FIG. 1 connected in series.

FIG. 4 is a schematic representation of the valve system shown at FIG. 1.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to FIGS. 1, 2 and 4 there is shown a two-stage pilot valve system comprising a three-position first stage pilot valve 70, a second stage power valve 72, and a load or consumer in the form of a double-acting power piston 74. A pump 15 supplies hydraulic liquid to power the system while the outlets from the valve system are connected to a reservoir 76.

The second stage power valve **72** has a valve slide or piston **3** arranged in the housing **78** and provides a diametrical bore **2** which is aligned with a circular groove or port **1** in the housing wall in the neutral or relaxed position shown. Groove **1** provides a direct connection to the pump **15** of the system. A diametrical bore **6** is preferably arranged in parallel to the bore **2**, the bore **6** being aligned with the circular groove **7** in the housing, the groove **7** being connected to the reservoir **76**. There are two additional circular grooves **45, 48** in the housing which are connected to the reservoir, these grooves being closed by the valve piston or slide **3** in the neutral position. If the valve slide **3** moves to the left, the circular groove **48** will cover a circular groove **47** of the valve slide **3** whereby the circular groove **48** is connected to the pressure chamber **44** via the longitudinal bores **50** opening into the groove **47**, the chamber **44** being located on the left surface of the valve slide as seen in FIG. **1**. Similarly, if the valve slide **3** moves to the right, the circular groove **45** of the housing will cover a circular groove **46** of the valve slide and is connected to the pressure chamber **43** located on the right surface of the slide **3** via the bores **49**.

The diametrical bores **2** and **6** are interconnected via a blind bore **4**, which is formed on the right face of the valve slide **3** and closed with a blind plug **5**. A seat valve body **9** is arranged in the connecting bore **4** between the bores **2** and **6**, said seat valve body being held in the closed position shown by a spring **8** which rests against the surface of the blind plug **5**. In the neutral position of the system the pressure medium flows to the reservoir from the pump via the circular groove **1**, the radial bores **2**, the longitudinal bore **4**, a passage in the valve body **9**, the radial bores **6** and the circular groove **7**. The seat valve body **9** prestressed by the spring **8**, establishes the operating pressure for the system which can be used for control purposes. In order to ensure that the valve slide **3** assumes the neutral position shown in which diametrical bores **2** and **6** are aligned with grooves **1** and **7** respectively, the valve slide **3** is positioned with respect to the housing abutments **21** and **22** on both sides by means of the springs **17** and **18** and by the sleeve-type catches or spring retainers **19** and **20**. The centering springs **17** and **18** bear upon the respective catch **19, 20** on the one hand and on a guide bushing **52, 68** on the other hand, said bushing limiting the chamber **43** or **44** respectively in the housing and resting by its step against threaded end plugs **54, 66**, the latter holding the guide bushing against the abutment **69, 70** of the housing. The end plugs **54, 66** provide a cylindrical recess into which projects the stepped end of the respective guide bushing **52, 68**. A lift or seat valve body **56** or **62** respectively, is disposed in a longitudinal bore in the guide bushings **52, 68**, each of the mentioned valve bodies being held in the initial closed position against the valve seat **60** or **64** by a spring **57** or **63** respectively, which rests against the bottom of the cylindrical recess of the respective end plugs **54, 66**, thereby keeping closed the passages **58** and **65** leading to the power piston **74**. In line with the seat valve bodies, axially extending pins **55, 73** are attached to both surfaces of the valve slide **3**, said pins opening the respective seat valve when the valve slide is displaced. The stroke of the seat valve body **56, 62** is limited by the end plugs **54, 66**.

The first stage pilot valve **70** is a three-position slide valve to be actuated in either direction by two electromagnets **32, 33**, or any other external force. The valve is held in the center or neutral position shown by means of the springs **28, 29** which press the abutment discs **30** and **31** against the abutments **26** and **27**. The housing of the first stage valve is provided with five circular grooves, the end two of which **34** and **35** are connected via the lines **36** to the circular groove **1** of the second stage power valve, thereby providing a direct connection to the pressure medium source. The groove **39** adjacent the circular groove **34** is connected via the line **41** to the chamber **43** at the right surface of the valve slide **3** whereas the groove **40** adjacent the circular groove **35** communicates with the chamber **44** via line **42**. The center circular groove **37** is connected with the reservoir via lines **38** and circular grooves **45** and **48** in the second stage valve. Upon actuation of the valve slide **25**, the hydraulic communication of the circular grooves **34, 35, 37, 39, 40** is controlled in such a way that power is transmitted to the power valve. In the neutral position the circular grooves **37, 39, 40** are interconnected and communicate with the pressure medium return whereas the circular grooves **34, 35** are connected to the pressure medium source.

OPERATION

Both the first stage pilot valve and the second stage pilot valve are pressure balanced in the neutral position. Movement of the power valve slide **3** is effected hydraulically via the first stage valve control slide **25** which is brought into position I in which the valve slide **25** is moved to the right by energizing of the electromagnet **32** and into position II in which the valve slide is moved to the left by energizing of the electromagnet **33**.

If the first stage pilot valve slide **25** is brought from the neutral position to position I by the electromagnet **32**, the groove **34** and **39** as well as the grooves **37** and **40** are interconnected whereby the pump pressure or the prestressed pressure respectively, which is generated by the seat valve **9** during circulation, is conveyed through the connections **36** and **41**, respectively, to the power valve pressure chamber **43** whereas the main power valve pressure chamber **44** communicates with the pressure medium return via the connections **38** and **42**.

The valve piston or slide **3** of the second stage power valve **72** is moved against the centering spring **17** by the force pressure created by the pump and valve **9** multiplied by the cross-sectional area of the valve slide **3**. Due to the surface or ratio of the surfaces on the valve slide **3** and the check valve body **62**, the force acting upon the valve slide **3** will always prevail and move the valve slide **3**, whereas the seat valve is kept closed by spring pressure and the pressure from the load or consumer. As the valve moves to the left, the circulation bores **2** and **6** withdraw with their pressure relief grooves from the area of the housing grooves **1** and **7** causing the pressure medium circulation to be interrupted and the pressurizing valve **9** to close. Upon further movement of the control piston **3**, the control piston groove **47** registers with the housing groove **48** which is also connected to the pressure medium return and then the control piston groove **46** registers with the housing groove **1** at the pressure medium source. After the pressurizing valve **9** has closed, the full pump pressure will reach the chamber **43** via the actuated servocontrol valve. Since the housing groove **1** positively overlaps the control piston groove **46**, the same will happen via the eccentrically disposed axial bores **49**. The flow returns from the chamber **44** via the bores **50**, which are symmetrically arranged towards the bores **49**, the control piston groove **47** and the housing groove **48**. Before the valve slide **3** abuts against the surface of the guide bushing **52** which rests against the end plug **54**, the pin **55** on the left side of the valve slide **3** presses against the front surface of the seat piston **56**.

The seat piston **56** which is guided in the guide bushing **52** is kept in closed position by the compressing spring **57** which rests against the end plug **54**, and by the force resulting from the consumer or load pressure —the latter reaching the seat edge **60** or the reverse side of the seat piston **56** respectively via the bores **58** —multiplied by the cross sectional area as limited by the circumference of the seat edge **60**. The resulting seat force must be exceeded by the force moving the valve slide **3**. The valve slide **3** opens the seat valve piston **56**. The pressure medium which had so far collected in the consumer behind the seat piston **56**, flows via the bores **58**, the seat bore above the seat edge **60**, the chamber **44**, the bores **50** and the groove **47** in the valve slide **3** and the housing groove **48**, into the pressure medium return. If the consumer is a double-acting piston **74**, as shown, a pressure difference will align between the seat valves **56** and **62** since the other chamber of the consumer is connected to the hydraulic line behind the seat valve **62**. If this pressure difference has dropped below the pressure in the chamber **43** and if the pressure in the chamber 43 is in a position to overcome the additional counteracting force of the spring 63, the seat valve 62 will lift off the seat edge 64. The hydraulic fluid which had so far flown from the pump into the chamber 43 now flows through the bore which is limited by the circumference of the seat edge 64, the bores 65, and on the the load, the seat valve being further opened against the force of the spring 63 by the forces resulting from the flow of the pressure medium.

Due to the surface ratio of the surfaces on the valve slide 3 and on the seat valve 62, the force acting on the valve slide is always greater than that on the seat valve 62 so that the seat valve 62 cannot open unless the valve slide 3 first opens the seat valve 56.

If the electromagnet deenergizes, the centering spring 28 which has been compressed, returns the first stage valve slide 25 to the neutral or zero position. The chamber 43 is again connected to the return flow via the connection 41 which results in a pressure drop in the mentioned chamber. The spring 63 causes the seat valve 62 to close the consumer chamber on the seat sealing edge 64, said chamber being exposed to pressure. The pressure in the consumer chamber supports the force of the spring 63 in the sense of a check valve. Since both chambers 43 and 44 are now connected to the return flow, the valve slide 3 is pressure balanced. The centering spring which was compressed causes the valve slide 3 to return to the neutral or zero position as described before. The seat valves 56 and 62 return to their initial positions, too.

When the valve slide reaches the neutral position, the pump circulation will resume.

The operation of the system when the left-hand electromagnet 33 is activated is analogous to the operation described above with the movements reversed so that the second stage valve piston 3 will move to the right in the drawing and connect the left side of the load 74 to the supply pressure.

FIG. 3 shows an arrangement for connecting three power valves 10, 11 and 12 in series. The pressure inlet 1 of the valve 10 communicates with the pressure medium source 15. The high-pressure outlet 13 of each of the valves connected in series communicates with the high-pressure inlet of the subsequent valve. The high-pressure outlet of the last valve is connected to the return line and hence to the reservoir 16. The pressurizing valve 9 is installed in the last valve 12 only whereby a prestressed pressure is forced upon the pressure medium flow by all valves together. A separate control has to be provided for each valve. If the valve slide 3 of one of the valves 10, 11, 12 is moved axially as described above, the circulation of the medium conveyed by the pump 15 is interrupted and the consumers of the respective valves will be actuated.

A special advantage of the power valve is that there is a free pressure medium circulation in zero position which means that a valve, viz. the circulation valve and the servocontrol thereof, conforming to valve arrangements known in the art, can be omitted. By arrangement in a mutual housing of the valve slide and of the twin lift valve designed as a multiway seat valve, space is saved and the operating time is reduced since the seat valves are opened by the valve slide directly and not via a hydraulic pressure which must be built up. Movable parts are saved which also favorably influence the response time and reduces wear. The valve according to the invention requires only a small number of seals of equal size. The advantages referred to above also favorably influence the costs of manufacture.

While we have described above the principles of our invention in connection with specific apparatus it is to be clearly understood that this description is made only by way of example and not as a limitation to the scope of my invention as set forth in the objects thereof and in the accompanying claims.

We claim:

1. A servo valve comprising; a housing having a closed cylinder therein with axially spaced supply and discharge ports in the cylinder wall, a double-acting piston in the cylinder forming a pair of pressure chambers one between each end of the piston and an end of the cylinder, a pair of normally closed lift valves one in each end of the cylinder, each valve connecting one of the pressure chambers to a load passage, means on each end of the piston for opening one of the valves as the piston approaches the end of the cylinder, a pair of passages in the piston, each passage leading from one pressure chamber to an opening in the piston wall, the passage openings being axially spaced from each other, means for holding the piston in a neutral position intermediate the ends of the cylinder in which the passage openings in the piston are each positioned between a discharge port means which is nearer to the pressure chamber to which the passage is connected and a supply port means which is further from the pressure chamber, each passage opening being spaced the same distance from its supply port means and the same distance from its discharge port means.

2. The valve of claim 1 wherein the mean on each end of the piston for opening the lift valve is an axially extending stud adapted to lift the valve before the piston abuts against the cylinder end wall.

3. The valve of claim 2 including recirculating means comprising a passage in the piston connecting the supply port means to the discharge port means in the neutral position of the valve and a spring-loaded check valve in said passage.

4. The valve of claim 3 wherein the ports in the cylinder are annular grooves in the cylinder wall.

5. The valve of claim 3 wherein the means for holding the piston in the neutral position comprise compression springs in each pressure chamber acting between the end of the cylinder and the piston to bias the piston toward the neutral position and abutments in the cylinder for limiting the spring force on the piston in the neutral position.

6. A valve system comprising a plurality of valves as defined in claim 1 connected in series with the discharge port means of the preceding valve connected to the supply port means of the next valve.

7. The valve of claim 1 including recirculating means comprising a passage in the piston connecting the supply port means to the discharge port means in the neutral position of the valve and a spring-loaded check valve in said passage.

8. The valve of claim 1 wherein the ports in the cylinder are annular grooves in the cylinder wall.

9. The valve of claim 1 wherein the mean for holding the piston in the neutral position comprise compression springs in each pressure chamber acting between the end of the cylinder and the piston to bias the piston toward the neutral position and abutments in the cylinder for limiting the spring force on the piston in the neutral position.

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