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(54) HYDRAULIC APPARATUS FOR THE CONTROL OF AT LEAST TWO LOADS

(71) We, ROBERT BOSCH GmbH, a German company of Postfach 50, 7000 Stuttgart 1, Germany., do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:

This invention relates to hydraulic apparatus for the control of at least two loads.

Apparatus for that purpose is described in United States patent specification No. 3 722 543 in which a first control member is provided for controlling direction and hydraulic throttling of each load and a second control member is provided for hydraulic pressure compensation. The second control members are all connected in series in a supply duct and each control an additional connection to the following second control member. This contruction, in which the second control members control three paths, is relatively expensive. Moreover this arrangement increases the danger of mutual influence between the loads in parallel operation thereof and leads to energy losses. Furthermore, with this control apparatus, a maximum working pressure must be transmitted to a pressure limiting valve for the system, through throttles and non-return valves arranged in a branch control line, the respective working pressure being derived from a duct between two of the first control members. Due to this, the construction of this control apparatus is relatively expensive.

In accordance with the invention, hydraulic control apparatus for the control of at least two hydraulic loads is provided, comprising at least two valve units each associated with one of the loads, an inlet for the supply of hydraulic fluid to the apparatus, an outlet for the discharge of hydraulic fluid to drain and a pressure limiting valve for controlling the pressure of the hydraulic fluid supplied to the apparatus through the inlet, each valve unit comprising a main valve for controlling the direction of operation of a load and for controlling the throttling of hydraulic fluid delivered to or discharged from that load and a control valve for influencing the operation of the pressure limiting valve and for controlling the pressure of the hydraulic fluid delivered to or discharged from the main valve, the control valves being permanently connected in parallel with the inlet or with the outlet and the inlet being connected to a chamber at one end of the pressure limiting valve, a first control line in communication with the inlet through a throttle and leading from a chamber at the other end of the pressure limiting valve, through each of the control valves in series, to the outlet to drain and each valve unit being provided with a second and a third control line through which the pressure difference across its main valve due to the throttling of the hydraulic fluid thereby can be applied to pressure chambers at opposite ends of its control valve.

Even when the loads are operated in parallel, such a control apparatus is so matched to the power demands in accordance with the delivery output from the pump, that energy losses are completely avoided and control of the loads independently of each other is possible. Furthermore, there is no danger of mutual influence between the loads.

Moreover, since the control valves are permanently connected in parallel, the load pressure can be selected and measured indirectly at the control valves and can be used for controlling the pressure limiting valve to raise or lower the supply pressure.

Preferably, each control valve comprises a slide valve member provided with a control edge for influencing the passage of hydraulic fluid through the first control line.

Furthermore, the valve bore for the slide valve member may be provided with two circulation chambers arranged one on each side of the control edge, the circulation chambers being connected to opposite sections of the first control line.

In a preferred form of control apparatus, the valve bore for the slide valve member is also provided with an inlet chamber and an outlet chamber, the slide valve member is provided with an annular groove in the vicinity of the inlet and outlet chambers and a spring is provided for retaining the slide valve member in an initial position in which it provides communication between the inlet and outlet chambers whilst communication between the circulation chambers is closed by the control edge, the slide valve member being movable against the force of the spring into a position in which communication between the circulation chambers is opened and communication between the inlet and outlet chambers is closed. The annular groove may be provided with at least one control edge.

In one form of control apparatus in accordance with the invention, each control valve is arranged between the inlet for the supply of hydraulic fluid and a main valve.

In another form of control apparatus in accordance with the invention, each control valve is connected between a main valve and the outlet for the discharge of hydraulic fluid to drain. This arrangement provides for good fluid flow regulation even when operating tensile loads.

A fourth control line leading to drain may be provided so that, in the neutral position of the main valve, the third control line is in communication with the fourth control line and is connected to the load on movement of the main valve out of the neutral position.

The apparatus in accordance with the invention may be used to control single or double acting loads. When controlling double acting loads, the third control line is connected to one side or the other of the load depending on the direction of movement of the main valve out of the neutral position.

In order that the invention will be well understood and readily carried into effect, two embodiments thereof will now be described by way of example, with reference to the accompanying drawings in which:

Figure 1 is a longitudinal section through one form of control apparatus in accordance with the invention;

Figure 2 is the hydraulic circuit diagram of the control apparatus of Figure 1; and

Figure 3 is the hydraulic circuit diagram of another form of control apparatus in accordance with the invention.

The hydraulic control apparatus 10 of Figure 1 comprises a pressure limiting valve 11 arranged in a connector plate 12, a valve unit 13, a valve unit 14 and an end plate 15. The valve unit 13 is associated with a load 68 and the valve unit 14 is associated with a load 69.

The valve unit 13 has a housing 16 accommodating a main valve and a control valve. The main valve member 18 is arranged to slide in a valve bore 17 which is provided with a supply chamber 19, two load chambers 21 and 22, two discharge chambers 23 and 24, a control groove 25 and a control groove 26. Two piston sections 27 and 28 are provided on the main valve member 18 for controlling the connections to and from the chambers 19 to 24. Two further piston sections 31 and 32 are provided on the valve member 18 with an annular groove 29 between them in the vicinity of the control groove 25.

Bores 33, 34 and 35 are provided in the main valve member 18 and control ducts 38 and 39 leading to load ducts 36 and 37 are provided in the housing 16. In the illustrated central position of the valve member 18, the annular groove 29 is connected to the control groove 26 through the bores 33, 34 and 35, and the piston sections 31 and 32 block the control ducts 38 and 39.

The control valve member 43 is provided with a first control edge 53 and a second control edge 54. The control edge 53 is arranged at one end of an annular groove 40 formed in the valve member 43. The valve member 43 is arranged to slide in a valve bore 44 provided with an inlet chamber 45, an outlet chamber 46, a first circulation chamber 47 and a second circulation chamber 48. The annular groove 40 is arranged in the vicinity of the inlet and outlet chambers 45 and 46 and the circulation chambers 47 and 48 are arranged one on each side of the control edge 54. Thus, the control edge 53 controls communication between the inlet and outlet chambers and the control edge 54 controls communication between the circulation chambers. Pressure chambers 42 and 61 are provided by the bore 44 at opposite ends of the valve member 43 and a spring 63 arranged in the chamber 42 urges the valve member 43 into an initial position towards the left in Figure 1.

The inlet chamber 45 is in communication with a pump 52 through an inlet 30, a supply duct 51 and a non-return valve 49 safeguarding the chamber 45 and the duct 51. The outlet chamber 46 is connected to the supply chamber 19 of the main valve.

The pressure limiting valve 11 comprises a valve member 64 arranged to slide in a valve bore provided in the plate 12. A chamber 60 is provided at one end of the valve 11 and a chamber 66 is provided at the other end of the valve 11. A spring 65, arranged in the chamber 66, urges the valve member 64

towards the left in Figure 1 and the left hand end of the valve member 64 is provided with a control edge for controlling the amount of hydraulic fluid which is allowed to pass to an outlet 50 to drain through a discharge chamber 70 and a discharge line 71, under the action of the fluid pressure at the inlet 30 and against the action of the spring 65.

The inlet 30 is connected to a chamber 60 at one end of the valve member 64 of the pressure limiting valve 11. A first control line 56 is in communication with the inlet 30 through a throttle 55 and leads from the chamber 66 at the other end of the valve member 64, through each of the control valves associated with the valve units 13 and 14, to the outlet 50 to drain through discharge ducts 57 and 58 discharge chamber 70 and line 71. The control line 56 passes through the control valves by way of respective circulation chambers 47 and 48 and is under the influence of the control edges 54.

A second control line 59 connects the outlet chamber 45 of the main valve to the pressure chamber 61 of the control valve. A third control line 41 leads from the pressure chamber 42 to the control groove 25 and can either be subjected to load pressure through the control ducts 38 and 39 or relieved of pressure to the outlet 50 through the bores 33, 34 and 35, grooves 25 and 26 and discharge line 62, depending on the axial position of the main valve member 18. More specifically, the chamber 42 is influenced by load pressure in the working positions of the main valve member and relieved of pressure in the illustrated neutral position of the main valve member. Moreover, the pressure difference across the main valve due to throttling of the hydraulic fluid thereby, can be applied to the pressure chambers 61 and 42 through the control lines 59 and 41.

The control valve member 43 is so formed that, with a lack of hydraulic fluid, it is urged by the spring 63 in the pressure chamber 42, into an initial position in which its first control edge 53 completely opens the communication between the inlet chamber 45 and the outlet chamber 46 whilst its second control edge 54 blocks the first control line 56. With an increasing displacement by the pressure in the chamber 61 against the force of the spring 63, the valve member 43 can take up intermediate positions in which it throttles the communication between the chambers 45 and 46 to a greater and greater extent or even completely blocks it, whilst it opens communication through the control line 56 more and more.

The valve unit 14 is of similar construction to the valve unit 13 described above so that further description in respect of the unit 14 is unnecessary.

As Figure 2 shows, the two valve units 12 and 13 are connected to the supply duct 51 in parallel with each other and are also connected in parallel with the pressure limiting valve 11. The two control valves 43 are connected in series in the first control line 56 and each in themselves forms a 2-way valve. They are also connected in parallel with the inlet 30. The pressure limiting valve 11 is protected in respect of pressure-limiting by a pilot valve 67. The valve unit 13 controls the load 68 and the valve unit 14 controls the load 69, both loads being shown as double-acting.

The method of operation of the control apparatus 10 in Figure 1 is as follows: It will be assumed that the main valve members 18 of the two valve units 12 and 13 have taken up their illustrated neutral positions. If the pump 52 delivers hydraulic fluid into the supply duct 51, then the greater part of this fluid will flow completely un-throttled through the pressure limiting valve 11, by displacing the valve member 64 against the force of the spring 65, and directly back into the discharge line 71. Simultaneously, only a small partial flow flows from the supply duct 51 through the throttle 55 and the first control line 56 to the discharge ducts 57 and 58. This partial flow occurs because the low pressure throttled by the pressure limiting valve 11 acts in the pressure chambers 61 through the supply duct 51, the non-return valves 49, the first control edges 53 of the control valve members 43 and the second control lines 59 and slightly displaces the valve members 43 against the force of the springs 63, so that the second control edges 54 each open a throttled communication through the first control line 56. Moreover, the spring chambers 42 are each relieved to the discharge line 71 through the discharge line 62. Furthermore, the valve member 43 does not close the communication between the inlet chamber 45 and the outlet chamber 46 but rather throttles it to some extent so that the member 43 takes up an intermediate position in which the first control line 56 is just throttled and is not completely blocked. Thus, the pump 52 operates against a pressure which is influenced on the one hand by the relatively weak spring 65 and is increased by a pressure component which depends on the stronger spring 63 biasing the valve member 43.

With this movement to the right of the valve member 43, the throttling in the first control line 56 by the second control edge 54 is simultaneously reduced. This leads to a lower closing pressure in the chamber 66 for the valve member 64 in the pressure limiting valve 11 so that the pressure in the supply duct 51 drops. Then, the valve member 43 can once again move nearer towards its initial position whereby the flow losses at the valve member 43 are maintained as low

as possible. On the contrary, a lowering of the pressure difference across the main valve member 18 leads to an opening of the communication through the first control edge 53 and a simultaneous greater throttling of the first control line 56 which leads to a pressure increase through the pressure limiting valve 11. This means that each pressure difference variation is then balanced by a reaction of the control valve, but that simultaneously pressure generated by the pump 52 is matched to the varied requirements by the pressure limiting valve 11 so that the valve member 43 can move back once again into its initial position. In this manner, the flow losses are maintained low. Moreover, when the second valve unit 13 is not actuated, the pressure chamber 42 is relieved to the return 71 through the third control line 41 and the discharge line 62, whilst the effective pressure in the supply duct 51 displaces the valve member 43 to such an extent that the first control line 56 is not blocked and the regulating operations in the valve unit 13 cannot be upset.

If both valve units 13 and 14 are actuated simultaneously, then both loads 68 and 69 can be controlled in a load pressure-compensated manner completely independently of each other. In so doing, the respective maximum load pressure is measured and selected indirectly by the position of the valve members 43 and is utilized for controlling the pressure of the pump 52. Thus, the pressure generated by the pump 52 is controlled at a value which is only slightly higher than the particular maximum load pressure plus the predetermined pressure difference across the main valve. If the amount of oil flowing to one load at low load pressure is greater than the quantity delivered by the pump 52, then the load at the higher load pressure is protected by the non-return valve 49.

Figure 3 is the hydraulic circuit diagram of a second embodiment of control apparatus in accordance with the invention. The control apparatus 80 in Figure 3 differs chiefly from that of Figure 1 in the arrangement of the valve units 81 and 82, in which each of the valve members 43 of the control valves is connected between a discharge port 83 of the main valve 18 and the discharge line 71. In this case, the control valves are permanently connected in parallel with the outlet 50.

In principle, this construction operates in the same manner as the control apparatus 10 of Figure 1. However, due to the throttling on the discharge side, the flow regulation in the valve unit 81 is effective even with a tensile force at the load 68 and is achieved in an advantageous manner.

Modifications to the illustrated control apparatus are, of course, possible without departing from the concept of the invention. Thus, for example, the flow forces occurring at the control valves can be used so as to support the force of the springs 63 so that the circulation pressure can be lowered still further without reducing the value of the predetermined pressure difference. Also, the second control edges 54 can be arranged on separate valve members which are coupled to the valve members proper in some way or other.

A further advantage of the control apparatus 10 consists in the fact that no control fluid flowing to or fro parallel to the load, towards it or away from it, flows back to the pressure controlled pressure limiting valve 11 in parallel with the main fluid flowing to the load. Furthermore, the control apparatus 10 has an advantageous amplifying effect. Since the quantity of control oil for controlling the valve member 64 is not removed from the load circuit proper and the latter is therefore not influenced either, that quantity can be made considerably greater than with the previously proposed apparatus. This favours a rapid response of the pressure regulating pressure limiting valve 11.

WHAT WE CLAIM IS:-

1. Hydraulic apparatus for the control of at least two hydraulic loads, comprising at least two valve units each associated with one of the loads, an inlet for the supply of hydraulic fluid to the apparatus, an outlet for the discharge of hydraulic fluid to drain and a pressure limiting valve for controlling the pressure of the hydraulic fluid supplied to the apparatus through the inlet, each valve unit comprising a main valve for controlling the direction of operation of a load and for controlling the throttling of hydraulic fluid delivered to or discharged from that load and a control valve for influencing the operation of the pressure limiting valve and for controlling the pressure of the hydraulic fluid delivered to or discharged from the main valve, the control valves being permanently connected in parallel with the inlet or with the outlet and the inlet being connected to a chamber at one end of the pressure limiting valve, a first control line in communication with the inlet through a throttle and leading from a chamber at the other end of the pressure limiting valve, through each of the control valves in series, to the outlet to drain and each valve unit being provided with a second and a third control line through which the pressure difference across its main valve due to the throttling of the hydraulic fluid thereby can be applied to pressure chambers at opposite ends of its control valve.

2. Hydraulic apparatus according to Claim 1, in which each control valve com-

prises a slide valve member provided with a control edge for influencing the passage of hydraulic fluid through the first control line.

3. Hydraulic apparatus according to Claim 2, in which a valve bore for the slide valve member is provided with two circulation chambers arranged one on each side of the control edge, the circulation chambers being connected to opposite sections of the first control line.

4. Hydraulic apparatus according to Claim 3, in which the valve bore for the slide valve member is also provided with an inlet chamber and an outlet chamber, the slide valve member is provided with an annular groove in the vicinity of the inlet and outlet chambers and a spring is provided for retaining the slide valve member in an initial position in which it provides communication between the inlet and outlet chambers whilst communication between the circulation chambers is closed by the control edge, the slide valve member being movable against the force of the spring into a position in which communication between the circulation chambers is opened and communication between the inlet and outlet chambers is closed.

5. Hydraulic apparatus according to Claim 4, in which the annular groove is provided with at least one control edge.

6. Hydraulic apparatus according to any preceding claim, in which each control valve is arranged between the inlet for the supply of hydraulic fluid and a main valve.

7. Hydraulic apparatus according to any one of Claims 1 to 5, in which each control valve is connected between a main valve and the outlet for the discharge of hydraulic fluid to drain.

8. Hydraulic apparatus according to any preceding claim, in which, in the neutral position of the main valve, the third control line is in communication with a fourth control line leading to drain and is connected to the load on movement of the main valve out of the neutral position.

9. Hydraulic apparatus according to Claim 8, in which the load is double acting and the third control line is connected to one side or the other of the load depending on the direction of movement of the main valve out of the neutral position.

10. Hydraulic apparatus for the control of at least two hydraulic loads, substantially as herein described with reference to Figure 1, Figure 2 or Figure 3 of the accompanying drawings.

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Sheet 1

Fig. 1

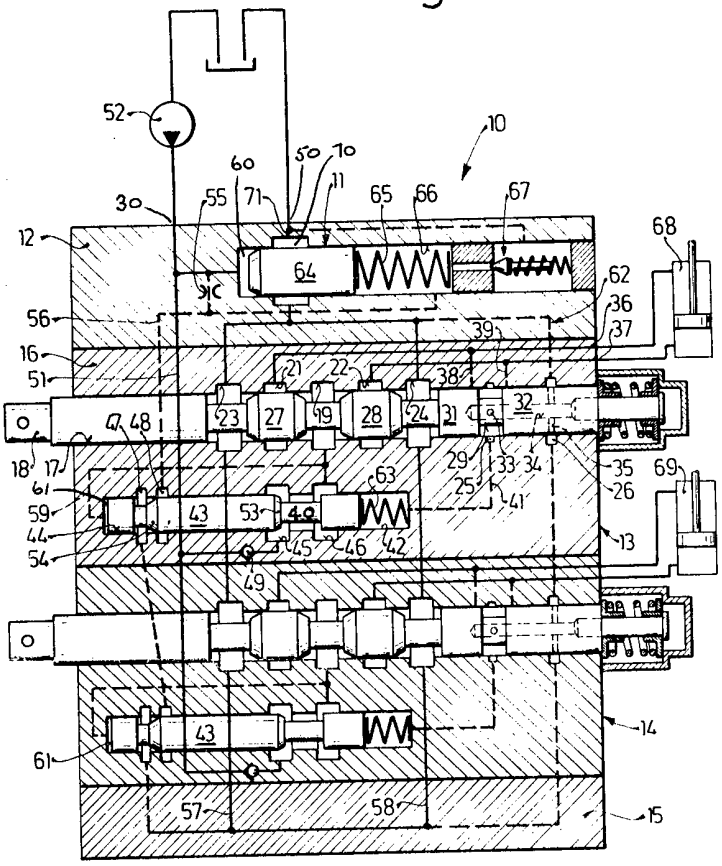
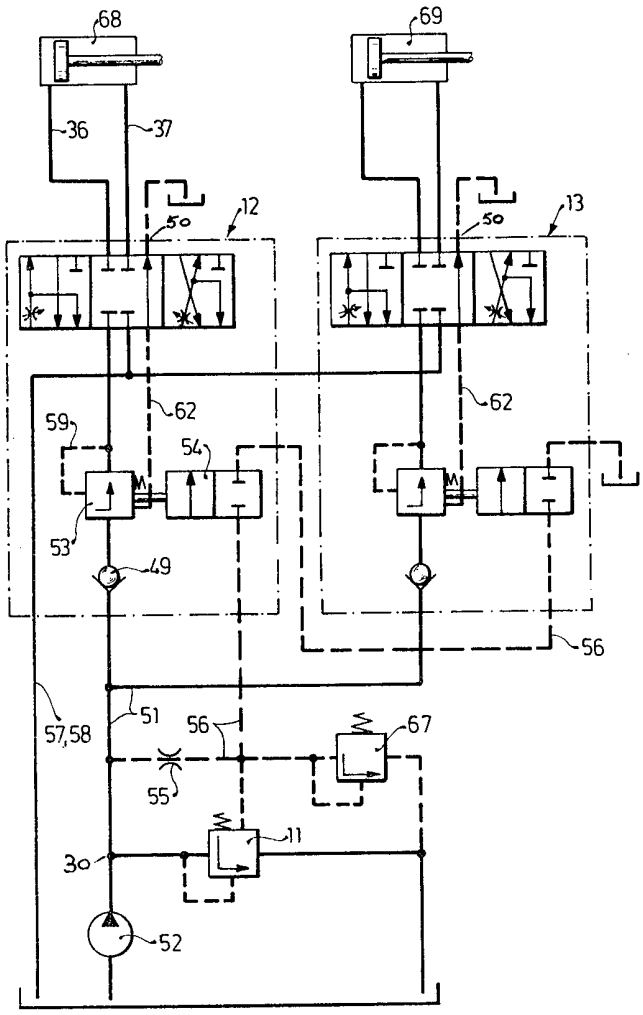


Fig. 2



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Sheet 3

Fig.3