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(71) Applicant: **KAYABA INDUSTRY CO., LTD.**
Sekai Boeki Center Bldg.,
2-4-1, Hamamatsu-cho
Minato-Ku,
Tokyo 105 (JP)

(72) Inventor: **Koiwai, Hideshi**
2-27-15 Kami-Toda
Toda-shi,
Saitama-ken (JP)
Inventor: **Naito, Hisato**
C-202 Shimokuzawa
Sagamihara-shi,
Kanagawa-ken (JP)
Inventor: **Yonekubo, Yoshitake**
2-15-15 Tamagawa
Setagaya-ku,
Tokyo (JP)
Inventor: **Nishiumi, Kenichi**
1-19-7-304 Terao-Dai
Tama-ku,
Kawasaki-shi,
Kanagawa-ken (JP)
Inventor: **Hasegawa, Yoshimi**
8-22-7 Nishi
Yachiyo-Dai,
Yachiyo-shi,
Chiba-ken (JP)

(74) Representative: **Carpmael, John William Maurice**
CARPMAELS & RANSFORD
43 Bloomsbury Square
London, WC1A 2RA (GB)

(54) Load-sensing active hydraulic control device.

(57) A load-sensing active hydraulic control device which includes a service port connected to a breaker or scissors in a power shovel of the like and is capable of preventing an excessive increase in pressure on the side of the service port from affecting other actuators. An on-off valve (34) is connected to a load detecting passage (33) of a change-over valve (3) connected to the side of a service port and is closed to prevent a pressure on the side of the service port from being fed back to a regulator when a load pressure on the side of a main actuator (2) is equal to or exceeds a set pressure.

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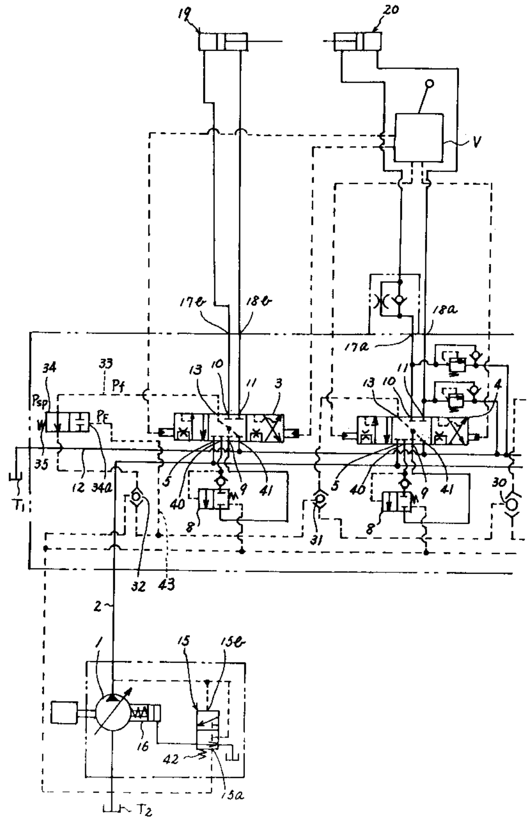
F I G. 1



This invention relates to a load-sensing active hydraulic control device which is suitable for use for a power shovel or the like and includes a service port which permits a specific actuator as well as a main actuator constantly used to be connected thereto.

It is an object of the present invention to provide a load-sensing active hydraulic control device which is capable of preventing excessive energy loss when an additional actuator is used under a pressure lower than a maximum circuit pressure of the system.

In accordance with the present invention, there is provided a load-sensing active hydraulic control device which comprises a variable discharge pump, a plurality of actuators provided with change-over valves, respectively, which are connected in parallel to each other with respect to the variable discharge pump, a variable orifice of which a degree of opening is controlled depending on the amount of changing-over of each of the change-over valves, pressure compensating valves each arranged on a downstream side of the variable orifice for keeping a pressure difference between a load pressure and a pressure on the downstream side of the variable orifice constant, a load detecting passage for detecting a load pressure of each of the actuators, and shuttle valves each connected to the load detecting passage, whereby a maximum pressure selected by the shuttle valves is introduced into a regulator of the variable discharge pump to keep an output of the pump constant.

The load-sensing active hydraulic control device of the present invention generally constructed as described above is characterized in that the load detecting passage communicating directly with the shuttle valve positioned on the most downstream side is provided with an on-off valve which is closed when a pressure on an upstream side of the shuttle on the most downstream side is equal to or exceeds a set pressure.

In the load-sensing active hydraulic control device of the present invention constructed as described above, when a pressure on the upstream side of the shuttle valve positioned on the most downstream side is equal to or exceeds a set pressure, the on-off valve is closed. Closing of the on-off valve prevents, even when a load pressure of the actuator connected to the shuttle valve on the most downstream side is increased, the load pressure from being introduced into the regulator of the variable discharge pump. Therefore, after the on-off valve is closed, the pressure compensating valve is caused to operate to exhibit a load-sensing function.

These and other objects and many of the attendant advantages of the present invention will be readily appreciated as the same becomes better understood by reference to the following detailed description when considered in connection with the accompanying drawings in which like reference numerals designate like or corresponding parts throughout, wherein:-

FIGURE 1 is a circuit diagram showing an embodiment of a load-sensing active hydraulic control device according to the present invention;

FIGURE 2 is a graphical representation showing control characteristics of keeping an output of a variable discharge pump constant; and

FIGURE 3 is a circuit diagram showing a conventional load-sensing active hydraulic control device.

Referring first to Figure 3, a conventional load-sensing active hydraulic control device includes a main actuator 20 connected to normal actuator ports 17a and 18a and a specific sub-actuator 19 connected to service ports 17b and 18b. Also, an actual power shovel includes in addition to the main actuator 20, a plurality of additional main actuators connected thereto, which are not shown in Figure 3 for the sake of brevity.

The conventional control device further includes a variable discharge pump 1 provided with a main passage 2 to which a change-over valve 4 of the main actuator 20 and a change-over valve 3 of the sub-actuator 19 are connected in parallel. The change-over valves 3 and 4 are constructed in substantially the same manner, therefore, corresponding parts of the valves 4 and 3 are designated by like reference numerals. Likewise, pressure compensating valves 8 respectively connected to the valves 3 and 4 are constructed in substantially the same manner, therefore, corresponding parts of the pressure compensating valves 8 are designated by the same reference numerals.

The above-described change-over valves 3 and 4 are constructed into a seven-port and three-position structure. More particularly, the change-over valves 3 and 4 each include a first port 5 connected to the main passage 2, a second port 40, a third port 9 connected through the pressure compensating valve 8 to the second port 40, and a fourth port 41 connected to a return passage 12 connected to a tank T1.

The change-over valves 3 and 4 each include fifth and sixth ports 10 and 11 connected to the main actuator 20 or sub-actuator 19 through the actuator ports 17a and 18a or service ports 17b and 18b. Further, the change-over valves 3 and 4 each include a seventh port 13 connected to shuttle valves 30, 31 and 32. The shuttle valves 30-32 are connected to all the change-over valves 3 and 4 one by one and adapted to select a high pressure from the change-over valves. The shuttle valve 30 positioned on the right side in Figure 3 is defined as being an upstream side of the system.

In Figure 3, reference character V designates a pilot valve, which functions to control a pilot pressure for changing over the change-over valve 4.

A control cylinder 16 and a valve 15 co-operate with each other to constitute a regulator. Also, changing-over of each of the change-over valves 3 and 4 causes communication of the first port 5 and a second port 40 and a degree of opening of each of the ports 5 and 40 is determined depending on the amount of changing-over of the change-over valves 4 and 3. The communication of the first and second ports 5 and 40 forms a variable orifice.

The above-described variable discharge pump 1 is adapted to control its tilting angle which is proportional to its discharge quantity depending on operation of the control cylinder 16. The control cylinder 16 is controlled depending on a change-over position of the valve 15. The valve 15 includes a first pilot chamber 15a connected to the shuttle valve 32 on the most downstream side and a second pilot chamber 15b connected to a discharge side of the variable discharge pump 1.

The valve 15 thus constructed is kept at a normal position shown in Figure 3 by the action of a spring 42. The valve causes the control cylinder 16 to communicate with the tank T1 to keep the discharge quantity minimum when it is at the normal position.

When a discharge pressure or the pump 1 is increased to cause a build up of a pressure at the pilot chamber 15b to overcome the pressure at the pilot chamber 15a and the spring 42, the valve 15 is changed over (i.e. moved down in Figure 3) to a position so that its upper side in Figure 3 is operative, thereby to introduce the discharge pressure of the pump into the control cylinder 16. Such an action of the pressure on the control cylinder 16 causes the tilting angle of the variable discharge pump 1 to be reduced, resulting in the discharge quantity of the pump 1 being decreased.

Then, changing-over of each of the change-over valves 3 and 4 to a left side position causes communication of the first and second ports 5 and 40 through the variable discharge orifice. The degree of opening of the orifices, and hence of the communication depends on the amount of changing-over of each of the change-over valves 3 and 4.

Such communication between ports 5 and 40 causes hydraulic fluid to flow from the main passage 2 through the pressure compensating valves 8 (which are moved to the right) and then through the third port 9 to the fifth port 10. Also, on the side of the main actuator 20, the hydraulic fluid thus flowing to the fifth port 10 is then fed through the actuator port 17a to the main actuator 20. At this time, the fifth port 10 is caused to communicate with the seventh port 13 as well, so that a load pressure of the main actuator 20 is introduced through the seventh port 13 into the shuttle valve 31.

The shuttle valve 31 compares the load pressure of the main actuator 20 with a load pressure selected by the shuttle valve 30 positioned on an upstream side of the shuttle valve 31, selecting higher one of both load pressures, which is then introduced into the shuttle valve 32 positioned on a downstream side of the shuttle valve 31.

Likewise, on the side of the sub-actuator 19, hydraulic fluid flowing through the fifth port 10 is fed through the service port 17b to the sub-actuator 19. At this time, the fifth port 10 is caused to communicate with the seventh port 13 as well, therefore a load pressure of the sub-actuator 19 is introduced from the seventh port 13 into the shuttle valve 32.

The shuttle valve 32 connected to the seventh port 13 of the change-over valve 3 on the side of the sub-actuator 19 is positioned on the most downstream side of the system, so that a maximum pressure of the system selected by the shuttle valve 32 is introduced into the pilot chamber 15a of the valve 15 functioning to control the tilting angle and hence the output of the variable discharge pump 1.

Therefore, in the conventional device, the maximum pressure of the system and the discharge pressure of the variable discharge pump 1 are compared with each other to cause the discharge pressure of the variable discharge pump 1 to be kept increased in an amount of α corresponding to elastic force of the spring 42 as compared with the maximum pressure of the system. Also, the variable discharge pump 1 is controlled in such a manner that the power thereof is kept constant.

The hydraulic fluid fed to the actuator 20 and sub-actuator 19 as described above is then returned through the sixth port 11 and fourth port 41 to the tank T1.

Also, the sub-actuator 19 is connected to the service ports 17b and 18b may typically comprise a breaker or scissors in the case of a shovel.

The conventional control device constructed as described above exhibits the following disadvantages, for example, when a required maximum pressure or the sub-actuator 19 connected to the service ports 17b and 18b is lower than a required maximum pressure of the main actuator 20.

For example, when the sub-actuator 19 reaches a stroke end during use of the main actuator 20 within a range of the required maximum pressure of the sub-actuator 19, the load pressure on the side of the sub-actuator 19 is increased to a level exceeding the required maximum pressure. Concurrently, the load pressure on the side of the sub-actuator 19 is caused to increase to a set maximum pressure of the system or the required maximum pressure of the main actuator 20.

Thus, in the conventional control device, when the load pressure of the sub-actuator is increased due to accidental arriving of the sub-actuator at the stroke end or the like, a circuit pressure of the system is excessively increased, resulting in energy loss being increased correspondingly.

The present invention has been made in view of the foregoing disadvantage of the prior art, and is illustrated in Figure 1.

Referring now to Figure 1, a load detecting passage 33 through which the seventh port 13 of the change-over valve 3 is connected to the shuttle valve 32 is provided with an on-off valve 34. The on-off valve 34 is kept an open position shown in Figure 1 by an action of a spring 35 and changed over to a closed position when a pilot pressure acting on a pilot chamber 34a overcomes elastic force of the spring 35. The pilot chamber 34a is connected through a passage 43 to an upstream side of the shuttle valve 32.

Thus, when a load pressure of a main actuator 20 other than a sub-actuator 19 is equal to or exceeds a set pressure determined by the spring 35 of the on-off valve 34, the on-off valve 34 is closed. Such closing of the on-off valve 34 prevents a load pressure of the sub-actuator 19 from affecting a discharge quantity of a variable discharge pump 1.

When the set pressure of the on-off valve 34, as shown in Figure 2, is set at a position x which is defined immediately before a pump output control area in which an output of the variable discharge pump 1 is kept constant, it is effectively prevented that a load pressure of the sub-actuator 19 is increased to an excessively high level to decrease a discharge quantity of the variable discharge pump 1 when the main actuator 20 and sub-actuator 19 are concurrently operated.

Now, the manner of operation of the load-sensing active hydraulic control device of the illustrated embodiment thus constructed will be described hereinafter.

The description will be made on the assumption that a pressure in the load detecting passage 33, a pressure on an upstream side of the shuttle valve 32, a set pressure of the on-off valve 34 and a final feedback pressure are indicated at P_f. P_F, $_P$sp and P_{lmax}, respectively.

First, P_F is equal to or below P_{sp} ($P_F \leqq P_{sp}$), the on-off valve 34 is kept open, so that the pressures P_f and P_F are compared with each other in the shuttle valve 32, resulting in higher one of the pressures P_f and P_F being selected, which is then fed back to the regulator. Thus, under such circumstances, a maximum value of a load pressure of each of all the actuators is selected.

Then, P_F is above P_{sp} ($P_F > P_{sp}$), the on-off valve 34 is kept closed, so that the pressure P_F is selected in the shuttle valve 32 and then fed back to the regulator.

Under the circumstances, a pressure fed back to the regulator depends on the pressure P_F on the upstream side of the shuttle valve 32 irrespective of a value of the load pressure P_f of the sub-actuator 19. Thus, in a situation that, for example, even when the sub-actuator 19 reaches a stroke end, resulting in P_f being larger than P_F, the load pressure P_f of the sub-actuator 19 is $P_F + \alpha$ which is a discharge pressure of the pump or is fixed to a load pressure of the sub-actuator 19 which is a pressure within the discharge pressure of the pump 1.

When the sub-actuator 19 is solely used, the on-off valve 34 is not changed over. Therefore, in this instance, a pressure compensating valve 8 functions to carry out load-sensing control. Thus, when it is required to use the sub-actuator 19 at a high pressure, sole operation of the sub-actuator 19 permits it to be used at a maximum pressure of the system.

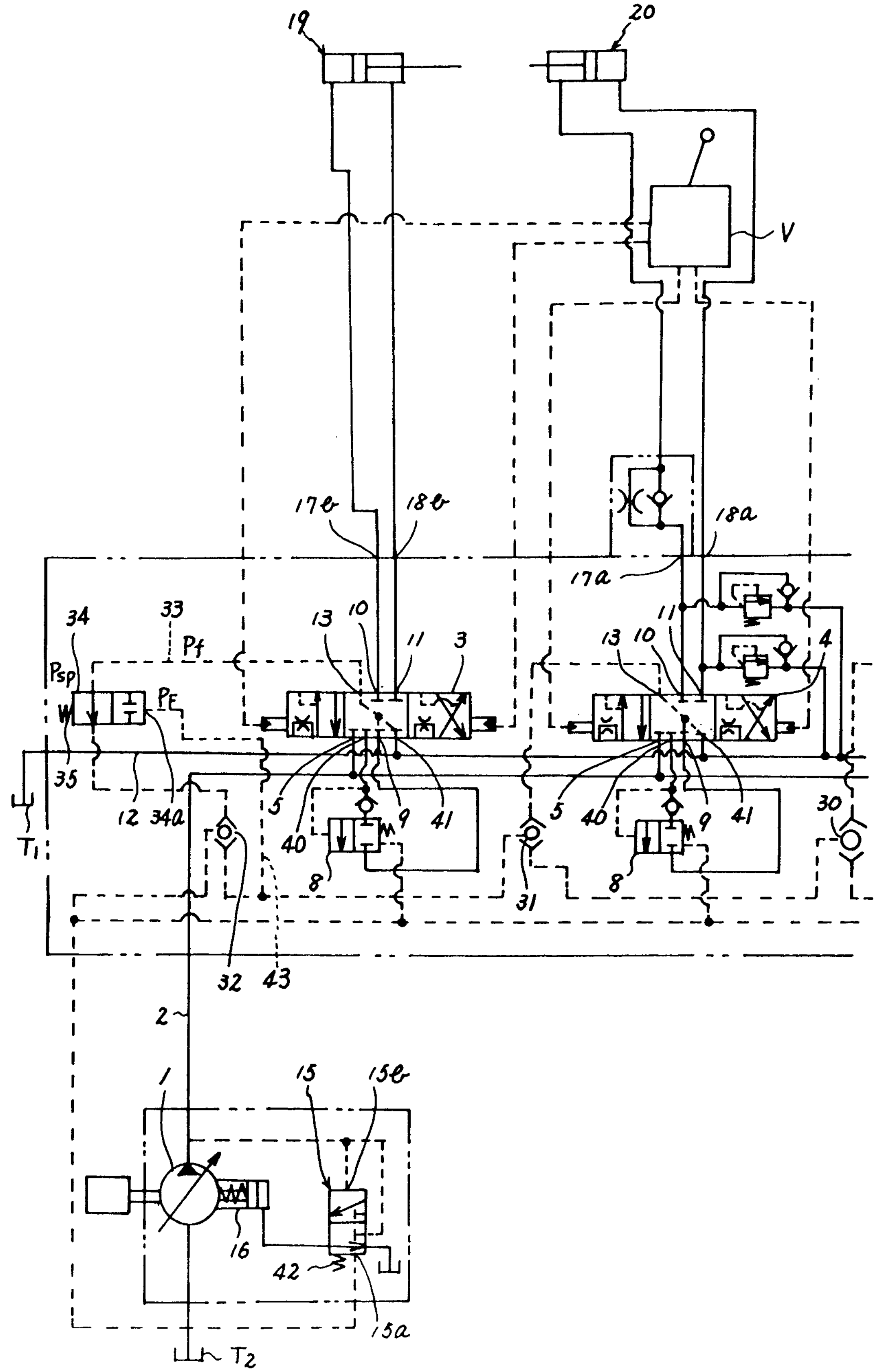
As can be seen from the foregoing, the load-sensing active hydraulic control device of the present invention effectively prevents excessive consumption of energy when an additional actuator is used at a pressure lower than a maximum circuit pressure of the system, resulting in improving economical efficiency.

While a preferred embodiment of the invention has been described with a certain degree of particularity with reference to the drawings, obvious modifications and variations are possible in light of the above teachings. It is therefore to be understood that within the scope of the appended claims, the invention may be practised otherwise than as specifically described.

Claims

1. A load-sensing active hydraulic control device comprising a plurality of actuators provided with change-over valves (3,4) respectively, a variable discharge pump (1), said change-over valves being connected in parallel to each other with respect to the variable discharge pump, and having a variable orifice of which a degree of opening is controlled depending on the amount of changing-over of each of the change-over valves, pressure compensating valves (8) each arranged on a downstream side of the variable orifice for keeping a pressure difference between a load pressure and a pressure on the downstream side of the variable orifice constant, a load detecting passage (33) for detecting a load pressure of each of the actuators, and shuttle valves (30,31, 32) each connected to the load detecting passage, whereby a maximum pressure selected by the shuttle valves is introduced into a regulator of the variable discharge pump to keep an output of the pump constant, characterised in that the load detecting passage communicating directly with the shuttle valve positioned on the most downstream side is provided with an on-off valve (34) which is closed when a pressure on an upstream side of the shuttle valve on the most downstream side is equal to or exceeds a set pressure.

F I G. 1


19
20
V
17b
18b
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Psp
PE
13
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35
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34a
5
40
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32
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30
T1
2
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15
15b
16
42
15a
T2

FIG. 2

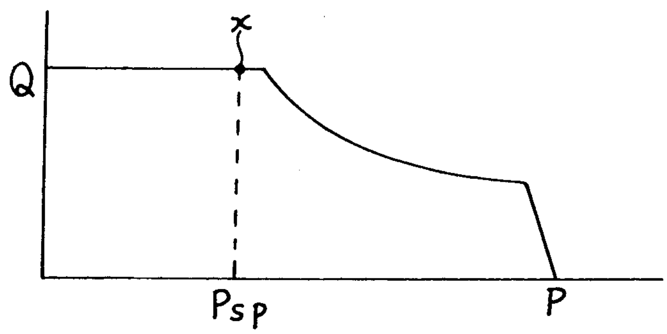
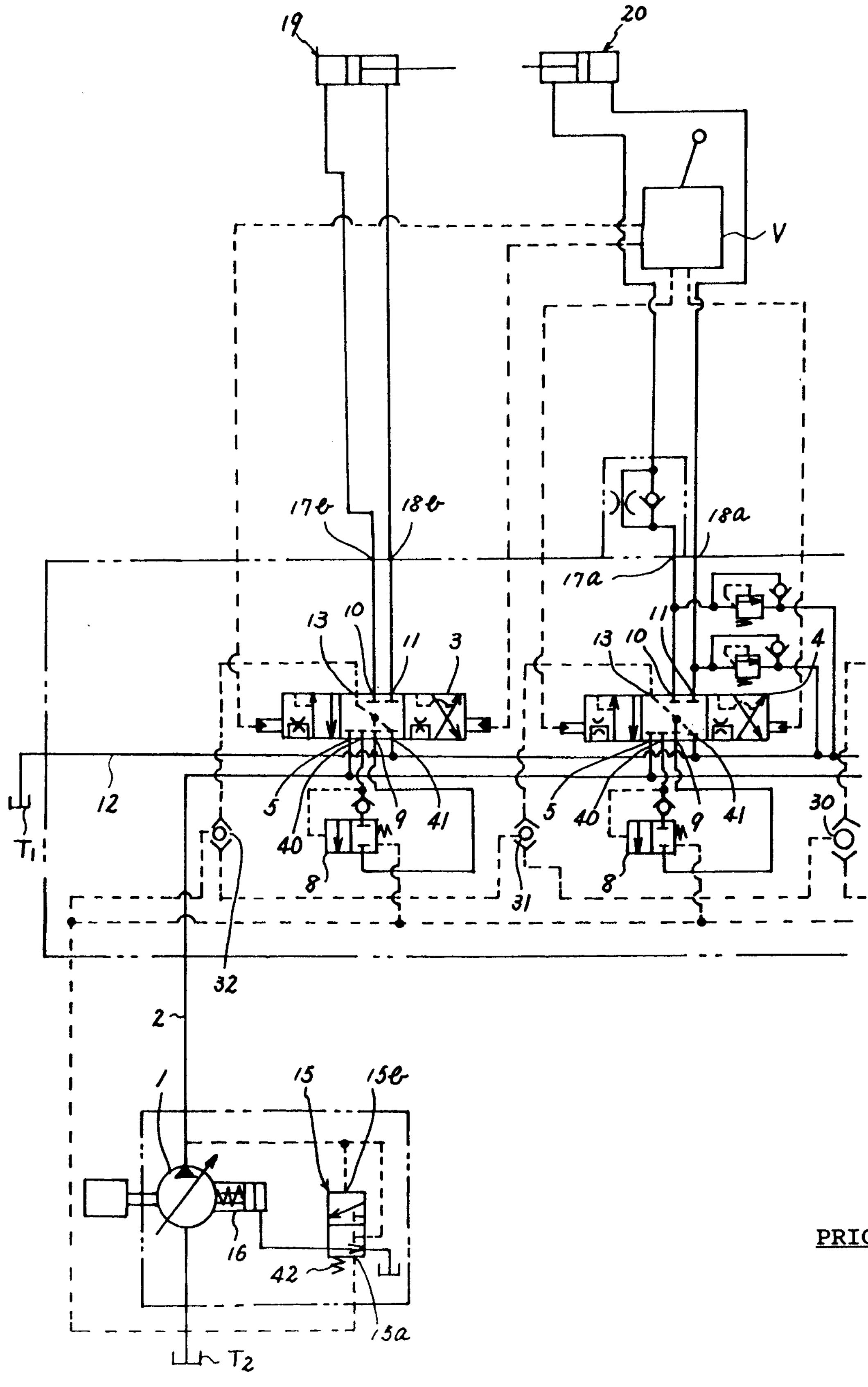


FIG. 3



PRIOR ART