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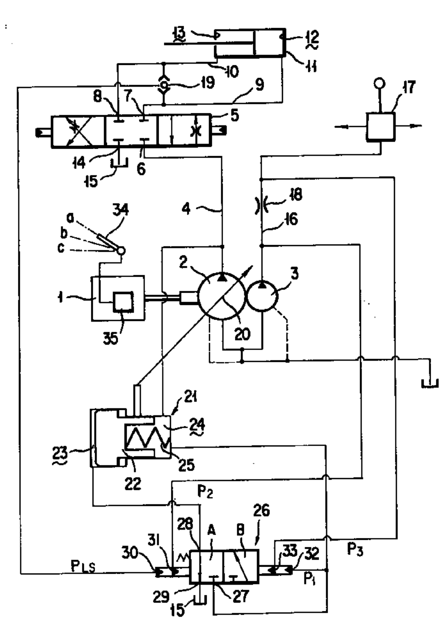
(54) **DISPLACEMENT CONTROLLING DEVICE FOR A VARIABLE DISPLACEMENT TYPE HYDRAULIC PUMP**

(57) A displacement controlling device for a variable displacement type hydraulic pump comprising a cylinder (21) for varying the displacement of a variable displacement type hydraulic pump (2), a control valve (26) for controlling the supply and discharge of discharging pressure of the variable displacement type hydraulic pump to and from the cylinder, a fixed displacement type hydraulic pump (3) driven together with the variable displacement type hydraulic pump by the same engine (1), and a throttle valve (18) provided along a discharge line of the fixed displacement type hydraulic pump, wherein the control valve is changed over by virtue of comparison between a differential pressure between the discharge pressure and a loading pressure and a set differential pressure, which is a differential pressure across the throttle valve so as to control the displacement of the variable displacement type hydraulic pump via the cylinder, whereby the differential pressure is made to be a certain value that matches the set differential pressure.

FIG. 1



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Description

TECHNICAL FIELD

The present invention relates to a displacement control system for controlling a displacement of a variable displacement type hydraulic pump to be employed in a hydraulic circuit supplying a pressurized fluid to an actuator of a construction machine or so forth.

BACKGROUND ART

As a hydraulic circuit supplying a pressurized fluid to an actuator of a construction machine and so forth, one supplying a discharged pressurized fluid of the hydraulic pump to the actuator through the operation valve, has been known. When a closed center type operation valve which shuts off a pump port at a neutral position, is employed as the operation valve in such hydraulic circuit, a discharge passage of the hydraulic pump becomes dead ended while the operation valve is in the neutral position, to make the discharged pressurized fluid at high pressure. Thus, a driving horse power consumption of an engine driving the hydraulic pump becomes large.

As a hydraulic circuit resolving this problem, there has been known a circuit, in which a variable displacement type hydraulic pump (hereinafter referred to as a variable hydraulic pump) is employed as the hydraulic pump, a displacement (a discharge amount per one revolution cycle) of the variable hydraulic pump is controlled to make the displacement smaller when a differential pressure between an inlet side pressure (pump discharge pressure) and an outlet side pressure (load pressure) is large and to make the displacement larger when the differential pressure is small and whereby to make the differential pressure constant, and a discharge flow rate (displacement x number of revolution per unit period) of the variable hydraulic pump can be a value corresponding to an opening degree (a communication area between a pump port and an actuator port) of the operation valve, as disclosed in Japanese Unexamined Utility Model Publication (Kokai) No. Heisei 5-86003.

However, in such a hydraulic circuit, the discharge flow rate of the variable hydraulic pump is displacement x number of revolution per unit period, and thus becomes small when an engine speed is low and large when the engine speed is high even when the displacement is constant, to differentiate the differential pressure of the inlet side pressure and the outlet side pressure even when the opening degree of the operation valve is the same. Therefore, even when the opening degree of the operation valve is the same, the displacement of the variable hydraulic pump is controlled to be larger when the engine speed is low and to be smaller when the engine speed is high so that the discharge flow rate becomes a value corresponding to the opening degree.

Therefore, even if the engine speed is lowered when fine operation for fine actuation of the actuator by reducing a supply flow rate to the actuator is desired, the displacement of the variable hydraulic pump becomes larger to increase discharge flow rate of the variable hydraulic pump to make it impossible to perform fine operation.

Therefore, as a solution for such drawback, in the foregoing Japanese Unexamined Utility Model Publication No. Heisei 5-86003, a fixed displacement type hydraulic pump (hereinafter referred to a fixed displacement hydraulic pump) is driven by the engine which drives the variable hydraulic pump, and a drain circuit including a restriction and a relief valve is connected to a discharge passage of the fixed displacement hydraulic pump, a pressure on the side of the fixed displacement hydraulic pump in relation to a junction in the discharge passage is detected to control the displacement of the variable pump depending upon the detected pressure.

Thus, since the detected pressure becomes a value corresponding to the engine speed, the displacement of the variable hydraulic pump can be controlled with taking the engine speed into account. As a result, the discharge flow rate of the variable hydraulic pump can be controlled with taking the engine speed into account, and whereby the differential pressure of the pump discharge pressure and the load pressure becomes a value corresponding to the engine speed.

However, in the construction set forth above, since a part of the discharged pressurized fluid of the fixed displacement hydraulic pump is flowed into a tank through the restriction and the relieve valve, the discharged pressurized fluid cannot be used effectively. Also, when the discharged pressurized fluid is supplied to other hydraulic device, a flow rate to be flowed into the tank is reduced to cause a variation of the differential pressure between the upstream side and the downstream side of the restriction to cause a variation of the detected pressure to vary displacement control characteristics of the variable hydraulic pump.

On the other hand, when the engine speed is extremely low, the discharge flow rate of the fixed displacement hydraulic pump can be too small to elevate the hydraulic pressure to a set pressure of the relief valve. In such case, the displacement of the variable hydraulic pump cannot be controlled with taking the engine speed into account.

Therefore, in view of the problems set out above, it is an object of the present invention to provide a displacement control system for a variable displacement type hydraulic pump, in which a discharge flow rate of the variable displacement type hydraulic pump becomes extremely small at low engine speed to improve operability in fine operation, a discharged pressurized fluid of a fixed displacement type hydraulic pump can be effectively used without flowing out to a tank, a displacement control characteristics of the variable displacement type hydraulic pump can be maintained constant even when the discharged pressure of the fixed displacement type hydraulic pump is supplied to other hydraulic device, and the displacement of the variable displacement type hydraulic pump can be controlled with taking an engine speed into account even when the engine speed is extremely low.

DISCLOSURE OF THE INVENTION

In order to accomplish the above-mentioned object, according to one aspect of the present invention,

a displacement control system for a variable displacement type hydraulic pump comprising a cylinder for varying a displacement of the variable displacement type hydraulic pump, a control valve for controlling supply and drain of a discharge pressure of the variable displacement type hydraulic pump to and from the cylinder, a fixed displacement type hydraulic pump driven simultaneously with the variable displacement type hydraulic pump by a common engine and a restriction provided in a discharge passage of the fixed displacement type hydraulic pump,
the control valve being operated for switching by comparison of a differential pressure between the discharge pressure and a load pressure and a differential pressure between upstream and downstream of the restriction as a set differential pressure, for controlling a displacement of the variable displacement type hydraulic pump via the cylinder so that the differential pressure becomes a value corresponding to the set differential pressure.

With the construction set forth above, when the engine speed is low, the discharge flow rate of the fixed displacement type hydraulic pump becomes smaller to make the differential pressure between upstream and downstream of the restriction smaller to make the set differential pressure smaller. On the other hand, when the engine speed is high, the discharge flow rate of the fixed displacement type hydraulic pump becomes larger to make the differential pressure between upstream and downstream of the restriction larger to make the set differential pressure larger.

Accordingly, when the engine is in low speed revolution, the discharge flow rate of the variable displacement type hydraulic pump becomes extremely small to improve operability in fine operation.

On the other hand, since the set differential pressure of the control valve is varied by the differential pressure between upstream and downstream of the restriction provided in the discharge passage of the fixed displacement type hydraulic pump, the discharged pressurized fluid of the fixed displacement type hydraulic pump can be used effectively without flowing out to the tank. Also, when the engine speed is constant, the differential pressure between upstream and downstream of the restriction is not varied even when the discharged pressurized fluid of the fixed displacement type hydraulic pump is supplied to other hydraulic device, the displacement control characteristics of the variable displacement type hydraulic pump can be constant. Furthermore, the differential pressure between upstream and downstream of the restriction is generated even at extremely low engine speed to enable control with taking the engine speed into account.

According to the second aspect of the present invention, a displacement control system comprises a cylinder for varying a displacement of the variable displacement type hydraulic pump, a control valve for controlling supply and drain of a discharge pressure of the variable displacement type hydraulic pump to and from the cylinder, a fixed displacement type hydraulic pump driven simultaneously with the variable displacement type hydraulic pump by a common engine and a restriction provided in a discharge passage of the fixed displacement type hydraulic pump, a first additional restriction provided in a pilot circuit between the downstream side of the restriction and the control valve and a second additional restriction provided in a drain passage connected to downstream side of the first additional restriction,

the control valve being operated for switching by comparison of a differential pressure of the discharge pressure and a load pressure and a differential pressure between the upstream side of the restriction and the downstream side of the first additional restriction as a set differential pressure, for controlling a displacement of the variable displacement type hydraulic pump via the cylinder so that the differential pressure becomes a value corresponding to the set differential pressure.

According to the third aspect of the present invention, a displacement control system for a variable displacement type hydraulic pump comprising a cylinder for varying a displacement of the variable displacement type hydraulic pump, a control valve for controlling supply and drain of a discharge pressure of the variable displacement type hydraulic pump to and from the cylinder, a fixed displacement type hydraulic pump driven simultaneously with the variable displacement type hydraulic pump by a common engine, a restriction provided in a discharge passage of the fixed displacement type hydraulic pump and a switching valve for switching supply and drain of the discharge pressure to and from the control valve and being associated with the cylinder via the spring,

the control valve being operated for switching by comparison of a differential pressure between the discharge pressure and a load pressure and a differential pressure between upstream and downstream of the restriction as a set differential pressure, for controlling a displacement of the variable displacement type hydraulic pump via the cylinder so that the differential pressure becomes a value corresponding to the set differential pressure, and

the switching valve being operated for switching by comparison of the discharge pressure and a mounting load of the spring, for controlling a displacement of the variable displacement type hydraulic pump via the control valve and the cylinder to maintain an input torque of the variable displacement type hydraulic pump constant.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will be understood more fully from the detailed description given herebelow and from the accompanying drawings of the preferred embodiment of the invention, which, however, should not be taken to be limitative to the present invention, but are for explanation and understanding only.

In the drawings:

Fig. 1 is a diagrammatic explanatory illustration of a construction of the first embodiment of a displacement control system for a variable displacement type hydraulic pump according to the present invention;
Fig. 2 is a chart showing a relationship between an engine speed and a discharge flow rate of the variable hydraulic pump in the first embodiment;
Fig. 3 is a section showing a particular construction of a control valve of the first embodiment;
Fig. 4 is a diagrammatic explanatory illustration of a construction of the second embodiment of a displacement control system for a variable displacement type hydraulic pump according to the present invention; and
Fig. 5 is a diagrammatic explanatory illustration of a construction of the third embodiment of a displacement control system for a variable displacement type hydraulic pump according to the present invention.

BEST MODE FOR IMPLEMENTING THE INVENTION

The preferred embodiment of a displacement control system for a variable displacement type hydraulic pump will be discussed hereinafter with reference to the accompanying drawings.

Fig. 1 shows the first embodiment of a displacement control system for a variable displacement type hydraulic pump according to the present invention. As shown in Fig. 1, by an engine 1, the variable displacement type hydraulic pump 2 (hereinafter referred to as a variable hydraulic pump 2) and a fixed displacement type hydraulic pump 3 (hereinafter referred to as a fixed displacement hydraulic pump 3) are driven. Then, a discharge passage 4 of the variable hydraulic pump 2 is connected to a pump port 6 of an operation valve 5. First and second actuator ports 7 and 8 of the operation valve 5 are respectively connected to a first chamber 12 and a second chamber 13 of an actuator 11 via respective of first and second circuits 9 and 10. A tank port 14 is connected to a tank 15.

A discharge passage 16 of the fixed displacement hydraulic pump 3 is connected to an inlet side of a hydraulic pilot valve 17. By operating the hydraulic pilot valve 17, a discharged pressurized fluid of the fixed displacement hydraulic pump 3 is supplied to the other hydraulic device. The discharge passage 16 is provided with a restriction 18.

The reference numeral 21 denotes a cylinder which has a piston 22, a large diameter pressure receiving chamber 23 and a small diameter pressure receiving chamber 24 defined at both sides of the piston 22, and the piston 22 is connected to a swash plate 20 of the variable hydraulic pump 2. Then, the swash plate 20 for varying a displacement of the variable hydraulic pump 2 is designed to vary an angle thereof by being pivotally tilted by a piston 22 of a swash plate control cylinder 21. The piston 22 of the cylinder 21 is moved in a displacement reducing direction by a pressurized fluid of the large diameter pressure receiving chamber 23 and moved in a displacement increasing direction by a pressurized fluid of the small diameter pressure receiving chamber 24 and a spring 25.

The large diameter pressure receiving chamber 23 is selectively connected to the tank 15 or the discharge passage 4, by a control valve 26 and the small diameter pressure receiving chamber 24 is connected to the discharge passage 4.

The control valve 26 has a first port 27 connected to the discharge passage 4 via the small diameter pressure receiving chamber 24 and a second port 28 connected to the large diameter pressure receiving chamber 23 and a tank port 29. The control valve 26 is changed over to a drain position A to shut off the first port 27 and to communicate the second port 28 with the tank port 29 by a pressurized fluid of a first pressure receiving portion 30 and a pressurized fluid of a first auxiliary pressure receiving portion 31, and is changed over to a supply position B to communicate the first portion 27 with the second port 28 and to shut off the tank port 29 by the pressurized fluid of a second auxiliary pressure receiving portions 32 a pressurized fluid of a second auxiliary pressure receiving portion 33.

The first pressure receiving portion 30 is connected to an outlet side of a shuttle valve 19 detecting a higher pressure of the first circuit 9 and the second circuit 10 to be supplied with the outlet side pressure (load pressure) of the operation valve 5. On the other hand, the second pressure receiving portion 32 is communicated with the discharge passage 4 via the small diameter pressure receiving chamber 24 to be supplied with the inlet side pressure (pump discharge pressure) of the operation valve 5. Furthermore, the first auxiliary pressure receiving portion 31 is connected to a upstream side of the restriction 18, and the second auxiliary pressure receiving portion 33 is connected to a downstream side of the restriction 18.

A revolution speed of the engine 1 is controlled by feeding an engine speed command signal generated from an operation member 34, such as an accelerator pedal or the like, to a control governor 35 of the engine 1. For example, when the operation member 34 is placed at a low speed position a, the engine speed becomes low speed, and at a medium speed position b, the engine speed becomes medium speed, and at a high speed position c, the engine speed becomes high speed.

Next, operation of the first embodiment set forth above will be discussed.

The control valve 26 is switched into the supply position B to pivoted the swash plate 4 in a direction to reduce the tilt angle thereof when the differential pressure between the inlet side pressure (pump discharge pressure) and the outlet side pressure (load pressure) of the operation valve 5 is greater than the set pressure, and is switched into the drain position A to tilt the swash plate 4 in a direction to increase the tilt angle thereof when the differential pressure is smaller than the set pressure to make the differential pressure between the inlet side pressure and the outlet side pressure of the operation valve 5 constant. By this, the discharge flow rate of the variable hydraulic pump 2 becomes a value corresponding to an opening degree (a communication area of the pump port 6 and the first or second actuator ports 7 or 8) of the operation valve 5.

On the other hand, a set differential pressure of the control valve 26 is varied depending upon a differential pressure between upstream and downstream of the restriction 18. The differential pressure between upstream and downstream of the restriction 18 is proportional to a square of a discharge flow rate (engine speed) of the fixed displacement hydraulic pump 3.

Thus, the set differential pressure of the control valve 26 becomes small when the engine speed is low, and becomes large when the engine speed is high. On fine operation at low engine speed, the discharge flow rate of the variable hydraulic pump 2 is reduced than that at the high speed to improve a fine operation ability.

Namely, the differential pressure between upstream and downstream of the restriction 18 is proportional to a square of the engine speed (discharge flow rate of the fixed displacement hydraulic pump 3) as shown in Fig. 2(a). The set differential pressure between control valve 26 is linearly proportional to the differential pressure of the upstream and downstream of the restriction 18 as shown in Fig. 2(b). The displacement of the variable hydraulic pump 2 is proportional to a square of the set differential pressure as shown in Fig. 2(c). Therefore, the discharge flow rate of the variable hydraulic pump 2 is linearly proportional to the engine speed as shown in Fig. 2(d).

Next, a particular construction of the control valve 26 will be explained.

As shown in Fig. 3, a sleeve 42 is threadingly inserted into a sleeve bore 41 formed in a valve body 40, such as a housing or the like of the variable displacement pump 2. The sleeve bore 41 is formed with a first inflow port 65, a flow out port 66, a control port 67, a pump pressure supply port 68 and a second inflow port 69. The sleeve 42 is formed with a first port 43, a second port 44, a third port 45, a fourth port 46 and a fifth port 47. Within a spool insertion bore 42a located at an axial center portion of the sleeve 42, a spool 48 is slidably disposed.

The spool 48 has a first small diameter portion 50, a second small diameter portion 51 and a third small diameter portion 52. In an axial bore 48a formed at one end portion of the spool 48, a small diameter portion 54 of a stationary piston 53 inserted in the spool insertion bore 42a is inserted to define a first pressure receiving chamber 55 (the second pressure receiving portion 32 of Fig. 1) and a second pressure receiving chamber 56 (the second auxiliary pressure receiving chamber 33 of Fig. 1). Then, the first pressure receiving chamber 55 is communicated with the second small diameter portion 51 through a fluid conduit 57 formed in the axial center portion of the spool 48. Also, by projecting the third small diameter portion 52 into the spring insertion hole 42b through an axial center bore 42c of the sleeve 42, a third pressure receiving chamber 58 is defined.

To one end of the spring insertion hole 42b of the sleeve 42, a cylindrical tip end portion 60 of the threaded rod 59 is engaged to define a fourth pressure receiving chamber 61. Furthermore, the threaded rod 59 is threadingly engaged with the sleeve 42 and fixed by tightening a lock nut 62. Then, in the axial center portion of the threaded rod 59, a bore 64 communicated with the fourth pressure receiving chamber 61 via the cylindrical tip end portion 60, is formed and extended from a piping joint portion 63 at the other end. Also, in the spring insertion hole 42b, the spring 49 is disposed between the other end of the spool 48 and the cylindrical tip end 60 of the threaded rod 59.

It should be noted that a pressure receiving diameter d_1 of the first pressure receiving chamber 55 and a pressure receiving diameter d_2 of the fourth pressure receiving chamber 61 are equal to each other.

The first port 43 is communicated with the second pressure receiving chamber 56 and the first inflow port 65. The first inflow port 65 is connected to the downstream side of the restriction 18 shown in Fig. 1. The second port 44 is connected to the flow out port 66 (the tank port 29 in Fig. 1). The flow out port 66 is communicated with the tank 15 shown in Fig. 1. The third port 45 is connected to a control port 67 (the second port 28 in Fig. 1). The control port 67 is connected to the large diameter pressure receiving chamber 23. The fourth port 46 is connected to a pump pressure supply port 68 (the first port 27 in Fig. 1). The pump pressure supply port 68 is connected to the discharge passage 4 via the small diameter chamber 24. The fifth port 47 is connected to the third pressure receiving chamber 58 (the first auxiliary pressure receiving portion 31 in Fig. 1) and a second inflow port 69. The second inflow port 69 is connected to the upstream side of the restriction 18 shown in Fig. 1. To the piping joint portion 63, a not shown hose is connected. Through these, the outlet side pressure (load pressure) of the operation valve 5 is supplied to the fourth pressure receiving portion 61 (first pressure receiving portion 30 in Fig. 1).

Next, operation of the control valve 26 is discussed.

A pump discharge pressure P_1 of the variable hydraulic pump 2 is supplied to the fourth port 46 through the pump pressure supply port 68, and supplied to the first pressure receiving chamber 55 via the second small diameter portion 51 and the fluid conduit 57 to thrust the spool 48 toward right. The outlet side pressure (load pressure) P_{LS} of the operation valve is supplied to the fourth pressure receiving chamber 61 to thrust the spool 48 toward left.

Here, when the differential pressure of the pump discharge pressure P_1 and the load pressure P_{LS} is zero ($P_1 = P_{LS}$), namely when the opening degree of the operation valve is maximum, since the pressure receiving diameter d_1 of the first pressure receiving chamber 55 and the pressure receiving diameter d_2 of the fourth pressure receiving chamber 61 are equal to each other, the spool 48 is thrust toward left by the spring 49, as shown in Fig. 3.

By this, through the first small diameter portion 50 of the spool 48, the second port 44 communicates with the third port 45 to establish communication of the control port 67 with the tank 15 via the flow out port 66 to place the control valve 26 at the drain position A in Fig. 1. Then, the pressurized fluid of the large diameter pressure receiving chamber 23 of the swash plate control valve 21 flows out, the displacement of the variable hydraulic pump 2 is increased, accordingly.

Once the displacement of the variable hydraulic pump 2 becomes large to increase the discharge flow rate, the flow rate of the pressurized fluid flowing from the pump port 6 of the operation valve 5 to the first or second actuator port 7 or 8 is increased. Thus, when the opening degree of the operation valve 5 is maintained as that in the condition set forth above, a pressure loss of the operation valve 5 is increased to make the differential pressure of the pump discharge pressure P_1 and the load pressure P_{LS} large.

By this, a force acting on the spool 48 becomes

$$\pi/4d_1^{\,2} \times P_1 - \pi/2d_2^{\,2} \times P_{LS} = \pi/4d_2^{\,2} \times \Delta P_{LS} > 0$$

When the force becomes greater than a mounting load of the spring 49, the spool 48 is moved toward right to be placed in a condition shown in Fig. 3 to block the communication between the second port 44 and the third port 45. Thus, the communication between the control port 67 and the flow out port 66 is blocked.

On the other hand, since $d_1 = d_2$ as set forth above, if $\pi/4d_2^{\,2} \times P_{LS}$ and an initial mounting load of the spring 49 are equal to each other, the spool 48 is constantly stopped at the position shown in Fig. 3. Thus, the differential pressure ΔP_{LS} between the discharge pressure P_1 of the pump and the load pressure P_{LS} is always maintained constant. The differential pressure (the set differential pressure) is determined by the initial mounting load of the spring 49.

On the other hand, the spool 48 is thrust toward right by the pressure of the downstream side of the restriction 18 acting in the second pressure receiving chamber 56 and thrust toward left by the pressure of the upstream side of the restriction 18 acting in the third pressure receiving chamber 58. Thus, when the spool 48 is thrust toward left by the differential pressure between upstream and downstream of the restriction 18, the set differential pressure is greater than that in the case set forth above.

Namely, a condition upon stopping of the spool 48 at the position shown in Fig. 3 is $\pi/4d_2^{\,2} \times \Delta P_{LS}$ = mounting load of the spring 49 + $\pi/4(D^2 - d_1^{\,2}) \times (P_2 - P_3)$. It should be noted that D is a diameter of the spool 48, P_2 is the pressure at upstream side of the restriction 18, and P_3 is the pressure at the downstream side of the restriction 18.

When the desired differential pressure ΔPLS is set at the engine speed set in the foregoing condition, when the engine speed is low, the discharge flow rate of the fixed displacement hydraulic pump 3 is decreased to reduce the differential pressure (P_2 - P_3) at upstream and downstream of the restriction 18 to lower the set differential pressure. Thus, the spool 48 is moved toward right to establish communication between the third port 45 and the fourth port 46. Thus, a condition where the control valve 26 of Fig. 1 is placed at the supply position B, in which the discharge pressure fluid of the variable hydraulic pump 2 flows to the control port 67 and then supplied to the large diameter pressure receiving chamber 23, is established. By this, the swash plate 20 is tilted in the displacement reducing direction to reduce the displacement of the variable hydraulic pump 2 to significantly smaller the discharge flow rate of the variable hydraulic pump 2 than that at high speed.

Fig. 4 shows the second embodiment of the displacement control system for the variable displacement type hydraulic pump according to the present invention. This is constructed by providing a first restriction 71 in a pilot circuit 70 connecting the downstream side of the restriction 18 in the discharge passage 16 of the fixed displacement hydraulic pump 3 and the second auxiliary pressure receiving portion 33 of the control valve 26, and connecting the downstream side of the first restriction 71 with the tank 15 through a drain circuit 73, and providing a second restriction 73 in the drain circuit 72.

The discharge passage 16 of the fixed displacement hydraulic pump 3 is connected to a swiveling hydraulic motor 76 via an auxiliary operation valve 75. By switching the auxiliary operation valve 75 to the supply position, a swiveling hydraulic motor 76 is driven.

Next, operation of the second embodiment will be discussed.

By providing the first restriction 71 in the pilot circuit 70 and, in conjunction therewith, connecting the downstream side of the restriction 71 to the tank 15 via the second restriction 73, a pressure P_4 acting in the second auxiliary pressure receiving portion 33 of the control valve 26 is lowered upon passing through the first restriction 71 to be lower than the downstream side pressure P3 of the restriction 18 to establish $P_3 > P_4$.

A ratio of lowering of pressure becomes a given ratio determined by the flow area of the first restriction 71 and the flow area of the second restriction 73.

Upon driving the swiveling hydraulic motor 76 by supplying the discharged pressurized fluid of the fixed displacement hydraulic pump 3 by placing the auxiliary operation valve 75 at the supply position, since a start up torque of the swiveling hydraulic motor 76 is large, the discharge pressure of the fixed displacement hydraulic pump 3 upon starting up becomes significantly high, and during steady swiveling action, the discharge pressure of the fixed hydraulic pump 3 becomes low.

On the other hand, the fixed displacement hydraulic pump 3 is constructed with a gear pump, for example. When the discharge pressure becomes high, internal leakage amount is increased to lower efficiency. Therefore, even at the same revolution speed, the discharge flow rate of the fixed displacement hydraulic pump 3 in high pressure is reduced than that in the low pressure.

Once the discharge flow rate of the fixed displacement hydraulic pump 3 is reduced, the differential pressure (P_2 - P_3) between upstream and downstream of the restriction 18 becomes smaller. Therefore, upon starting up of the swiveling hydraulic motor 76, the differential pressure (P_2 - P) between upstream and downstream of the restriction 18 becomes smaller. Also, during steady revolution, the differential pressure (P_2 - P_3) between upstream and downstream of the restriction 18 becomes large.

Therefore, if the first and second restrictions 71 and 73 are not provided in the pilot circuit 70 and when the pressure P_3 at the downstream side of the restriction 18 directly acts on the second auxiliary pressure receiving portion 33 of the control valve 26, the differential pressure between the pressure acting in the first auxiliary pressure receiving portion 31 of the control valve 26 and the pressure acting in the second auxiliary pressure receiving portion 33 is varied between that upon starting up of the swiveling hydraulic motor 76 and that during steady revolution to differentiate the discharge flow rate of the variable hydraulic motor 2.

However, as shown in Fig. 4, by providing the first and second restrictions 71 and 73 in the pilot circuit 70, the pressure P_4 acting in the second auxiliary pressure receiving portion 33 of the control valve 26 becomes lower pressure at a given ratio than the pressure P_3 of the downstream side of the restriction 18. Therefore, the discharge flow rate of the fixed displacement hydraulic pump 3 as set forth above is reduced by lowering of efficiency due to the discharge pressure, the differential pressure between the pressures in the first auxiliary pressure receiving portion 31 and the second auxiliary pressure receiving portion 32 of the control valve 26 becomes substantially constant so that the discharge flow rate of the variable hydraulic pump 2 will not be varied between the start-up and by steady revolution of the swiveling hydraulic motor 76.

For example, it is assumed that the discharge pressure of the fixed displacement hydraulic pump 3 is 50 kg/cm^2 and the discharge flow rate is 20 l/min, the pressure P_3 at the downstream side of the restriction 18 is 40 kg/cm^2, and the pressure P_4 in the second auxiliary pressure receiving portion 33 is 39.5 kg/cm^2 during steady revolution of the swiveling hydraulic motor 76, the differential pressure (P_2 - P_4) between the pressure P_2 acting in the first auxiliary pressure receiving portion 31 and the pressure P_4 acting in the second auxiliary pressure receiving portion 33 of the control valve 26 becomes 10 kg/cm^2.

In the condition set forth above, upon starting up of the swiveling hydraulic motor, when the discharge pressure of the fixed displacement hydraulic pump 3 is 200 kg/cm^2 and the discharge flow rate is 18 l/min, the downstream side pressure P_3 of the restriction 18 becomes 192 kg/cm^2 and the pressure P_4 acting in the second auxiliary pressure receiving portion 33 becomes substantially 189.5 kg/cm^2. Thus, the foregoing differential pressure (P_2 - P_4) becomes substantially 10 kg/cm^2.

Fig. 5 shows the third embodiment of the displacement control system of the variable displacement type hydraulic pump according to the present invention. The drain port 29 of the control valve 26 is selectively connected to one of the tank 15 and the discharge passage 4 by the switching valve 80 for an input torque control.

The switching valve 80 is changed over to a drain position C by the spring 81 and is changed over to a supply position D by the discharge pressure of the variable displacement pump 2 acting in the first pressure receiving portion 82, and an external pressure acting in the second pressure receiving portion 83. Furthermore, the spring 81 is associated with the piston 22 by a link 84.

Since the shown embodiment is constructed as set forth above, when the pump discharge pressure of the variable displacement pump 2 becomes higher than a pressure corresponding to the mounting load of the spring 81, the switching valve 80 is changed over to the supply position D. Thus, the discharge pressure flows into the large diameter pressure receiving chamber 23 via the control valve 26. Therefore, the piston 22 is moved toward right to pivot the swash plate 20 to tilt in the direction of smaller displacement. By movement of the piston 22, the mounting load of the spring 81 is increased via the link 84 to change over the switching valve 80 back to the drain position C.

Thus, by repeating such operation, the displacement of the variable hydraulic pump 2 is controlled so that an input torque (pump discharge pressure x displacement) becomes constant.

As set forth above, according to the present invention, when the revolution speed of the engine 1 is low, the discharge flow rate of the fixed displacement type hydraulic pump 3 is reduced. By this, the differential pressure between upstream and downstream of the restriction 18 becomes smaller to make the set differential pressure of the control valve 26 smaller. On the other hand, when the revolution speed of the engine 1 is high, the discharge flow rate of the fixed displacement type hydraulic pump 3 is increased. Associating therewith, the differential pressure between upstream and downstream of the restriction 18 becomes larger to make the set differential pressure of the control valve 26 larger.

Accordingly, when the engine 1 is in low revolution speed, the discharge flow rate of the variable displacement type hydraulic pump 2 is significantly smaller to improve the operability in fine motion.

On the other hand, since the set differential pressure of the control valve 26 is varied by the differential pressure between upstream and downstream of the restriction 18 provided in the discharge passage 16 of the fixed displacement type hydraulic pump 3, the discharged pressurized fluid of the fixed displacement type hydraulic pump 3 does not flow out to the tank 15 and thus can be used effectively. Also, as long as the engine speed is constant, the differential pressure between upstream and downstream of the restriction 18 will not be varied even when the discharged pressurized fluid of the fixed displacement type hydraulic pump 3 is supplied to other hydraulic device. Thus, the displacement control characteristics of the variable displacement type hydraulic pump 2 can be made constant. Furthermore, even if the engine 1 is in extremely low speed, the differential pressure between upstream and downstream of the restriction 18 is caused to make it possible to control the displacement of the variable displacement type hydraulic pump 2 with taking the engine speed into account.

On the other hand, according to the present invention, the pressure P_4 acting in the second auxiliary pressure receiving portion 33 of the control valve 26 becomes a value corresponding to the pressure P_3 at downstream of the restriction 18 provided in the discharge passage 16 of the fixed displacement type hydraulic pump 3 lowered by a given ratio. When the revolution speed of the engine 1 is constant and the discharge flow rate is varied associating with the variation of efficiency due to the discharge pressure of the fixed displacement type hydraulic pump 3, the pressure difference between the first auxiliary pressure receiving portion 31 and the second auxiliary pressure receiving portion 33 of the control valve 26 becomes substantially constant.

Accordingly, even when the revolution speed of the fixed displacement type hydraulic pump 3 is the same and the discharge pressure thereof is varied between high pressure and low pressure, the set differential pressure of the control valve 26 can be constant to make the discharge flow rate of the variable displacement type hydraulic pump 2 can be substantially constant.

Although the invention has been illustrated and described with respect to exemplary embodiment thereof, it should be understood by those skilled in the art that the foregoing and various other changes, omissions and additions may be made therein and thereto, without departing from the spirit and scope of the present invention. Therefore, the present invention should not be understood as limited to the specific embodiment set out above but to include all possible embodiments which can be embodied within a scope encompassed and equivalents thereof with respect to the feature set out in the appended claims.

Claims

1. A displacement control system for a variable displacement type hydraulic pump comprising a cylinder for varying a displacement of said variable displacement type hydraulic pump, a control valve for controlling supply and drain of a discharge pressure of said variable displacement type hydraulic pump to and from said cylinder, a fixed displacement type hydraulic pump driven simultaneously with said variable displacement type hydraulic pump by a common engine and a restriction provided in a discharge passage of said fixed displacement type hydraulic pump,

   said control valve being operated for switching by comparison of a differential pressure between said discharge pressure and a load pressure and a differential pressure between upstream and downstream of said restriction as a set differential pressure, for controlling a displacement of said variable displacement type hydraulic pump via said cylinder so that said differential pressure becomes a value corresponding to said set differential pressure.

2. A displacement control system for a variable displacement type hydraulic pump as set forth in claim 1, wherein said cylinder includes a large diameter pressure receiving chamber supplied and drained with said discharge pressure by said control valve, and a small diameter pressure receiving chamber supplied said discharge pressure, said cylinder being moved in a smaller displacement direction, when a pressurized fluid is supplied to said large diameter pressure receiving chamber and is moved in a larger displacement direction when the pressurized fluid is supplied to said small diameter pressure receiving chamber, and

   said control valve comprises a first pressure receiving portion being supplied with said load pressure, a second pressure receiving portion being supplied with said discharge pressure, a first auxiliary pressure receiving portion being supplied with a pressure at the upstream side of said restriction, and a second auxiliary pressure receiving portion being supplied with a pressure at the downstream side of said restriction, said control valve being changed over to a drain position for draining said discharge pressure from said large diameter pressure receiving chamber by the pressure to said first pressure receiving portion and said first auxiliary pressure receiving portion, and to a supply position for supplying said discharge pressure to said large diameter pressure receiving chamber by the pressures to said second pressure receiving chamber and said second auxiliary pressure receiving chamber.

3. A displacement control system for a variable displacement type hydraulic pump comprising a cylinder for varying a displacement of said variable displacement type hydraulic pump, a control valve for controlling supply and drain of a discharge pressure of said variable displacement type hydraulic pump to and from said cylinder, a fixed displacement type hydraulic pump driven simultaneously with said variable displacement type hydraulic pump by a common engine and a restriction provided in a discharge passage of said fixed displacement type hydraulic pump, a first additional restriction provided in a pilot circuit between the downstream side of said restriction and said control valve and a second additional restriction provided in a drain passage connected to the downstream side of said first additional restriction,

   said control valve being operated for switching by comparison of a differential pressure between said discharge pressure and a load pressure and a differential pressure between the upstream side of said restriction and the downstream side of said first additional restriction as a set differential pressure, for controlling a displacement of said variable displacement type hydraulic pump via said cylinder so that said differential pressure becomes a value corresponding to said set differential pressure.

4. A displacement control system for a variable displacement type hydraulic pump as set forth in claim 3, wherein said cylinder includes a large diameter pressure receiving chamber supplied and drained with said discharge pressure by said control valve, and a small diameter pressure receiving chamber supplied with said discharge pressure, said cylinder being moved in a smaller displacement direction when a pressurized fluid is supplied to said large diameter pressure receiving chamber and is moved in a larger displacement direction when the pressurized fluid is supplied to said small diameter pressure receiving chamber, and

   said control valve comprises a first pressure receiving portion being supplied with said load pressure, a second pressure receiving portion being supplied with said discharge pressure, a first auxiliary pressure receiving portion being supplied with a pressure at the upstream side of said restriction, and a second auxiliary pressure receiving portion being supplied with a pressure at the downstream side of said first additional restriction, said control valve being changed over to a drain position for draining said discharge pressure from said large diameter pressure receiving chamber by the pressure to said first pressure receiving portion and said first auxiliary pressure receiving portion, and to a supply position for supplying said discharge pressure to said large diameter pressure receiving chamber by the pressures to said second pressure receiving chamber and said second auxiliary pressure receiving chamber.

5. A displacement control system for a variable displacement type hydraulic pump comprising a cylinder for varying a displacement of said variable displacement type hydraulic pump, a control valve for controlling supply and drain of a discharge pressure of said variable displacement type hydraulic pump to and from said cylinder, a fixed displacement type hydraulic pump driven simultaneously with said variable displacement type hydraulic pump by a common engine, a restriction provided in a discharge passage of said fixed displacement type hydraulic pump and a switching valve for switching supply and drain of said discharge pressure to and from said control valve and being associated with said cylinder via said spring,

   said control valve being operated for switching by comparison of a differential pressure between said discharge pressure and a load pressure and a differential pressure between upstream and downstream of said restriction as a set differential pressure, for controlling a displacement of said variable displacement type hydraulic pump via said cylinder so that said differential pressure becomes a value corresponding to said set differential pressure, and

said switching valve being operated for switching by comparison of said discharge pressure and a mounting load of said spring, for controlling displacement of said variable displacement type hydraulic pump via said control valve and said cylinder to maintain an input torque of said variable displacement type hydraulic pump constant.

6. A displacement control system for a variable displacement type hydraulic pump as set forth in claim 5, wherein said cylinder includes a large diameter pressure receiving chamber supplied and drained with said discharge pressure by said control valve, and a small diameter pressure receiving chamber supplied with said discharge pressure, said cylinder being moved in a smaller displacement direction when a pressurized fluid is supplied to said large diameter pressure receiving chamber and is moved in a direction for larger displacement when the pressurized fluid is supplied to said small diameter pressure receiving chamber, and

said control valve comprises a first pressure receiving portion being supplied with said load pressure, a second pressure receiving portion being supplied with said discharge pressure, a first auxiliary pressure receiving portion being supplied with a pressure at the upstream side of said restriction, and a second auxiliary pressure receiving portion being supplied with a pressure at the downstream side of said restriction, said control valve being changed over to a drain position for draining said discharge pressure from said large diameter pressure receiving chamber by the pressures to said first pressure receiving portion and said first auxiliary pressure receiving portion, and to a supply position for supplying said discharge pressure to said large diameter pressure receiving chamber by the pressure to said second pressure receiving chamber and said second auxiliary pressure receiving chamber, and

said switching valve comprises a third pressure receiving portion being supplied with said discharge pressure, and a fourth pressure receiving portion being acted with an external force, said switching valve being changed over to a drain position for draining the discharge pressure from said control valve by a resilient force of said spring, and to a supply position for supplying the discharge pressure to said control valve by pressures of said third pressure receiving portion and said fourth pressure receiving portion.

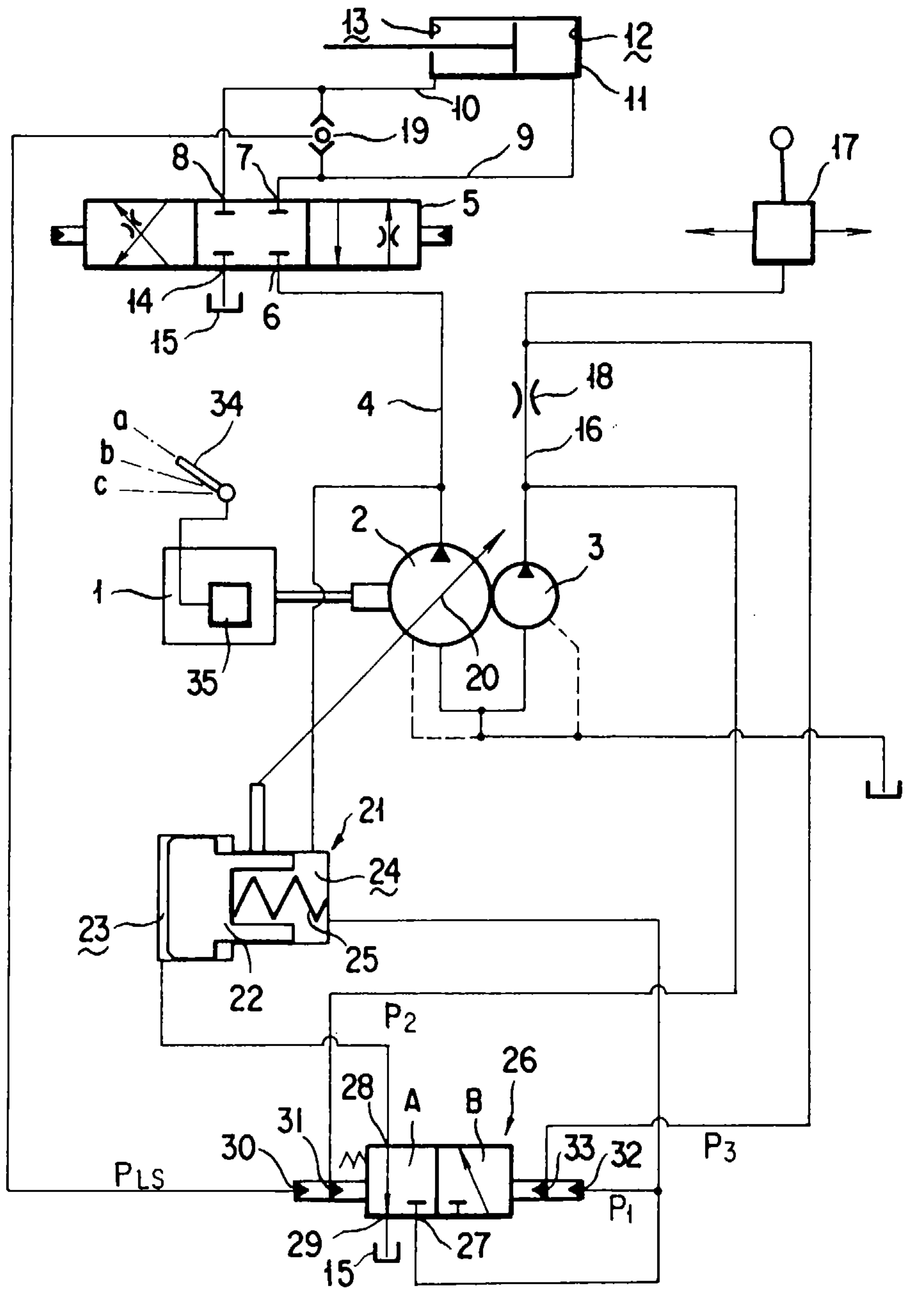
7. A displacement control system for a variable displacement hydraulic pump as set forth in any one of claims 2, 4 and 6, wherein said control valve comprises

a valve body, a sleeve bore formed in said valve body and having a pump pressure supply port, a flow out port and a control port, a sleeve disposed within said sleeve bore and having ports respectively being communicating with said pump pressure supply port, said flow out port and said control port, a spool slidably disposed within said sleeve and defining said first pressure receiving portion and said second pressure receiving portion having equal pressure receiving diameter, and said first auxiliary pressure receiving portion and said second auxiliary pressure receiving portion having equal pressure receiving diameter, and an additional spring biasing said spool in one direction;

by a differential pressure between the pressure of said first pressure receiving portion and the pressure of said second pressure receiving portion and a resilient force of said additional spring, said spool being shifted to a position for establishing a communication one of between said pump pressure supply port and said control port and between said control port and said flow out port and said flow out port, and shutting off the other,

by a differential pressure between the pressure of said first auxiliary pressure receiving portion and the pressure of said second auxiliary pressure receiving portion, said spool being biased in a direction against said additional spring.

FIG. 1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PLS
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FIG. 2

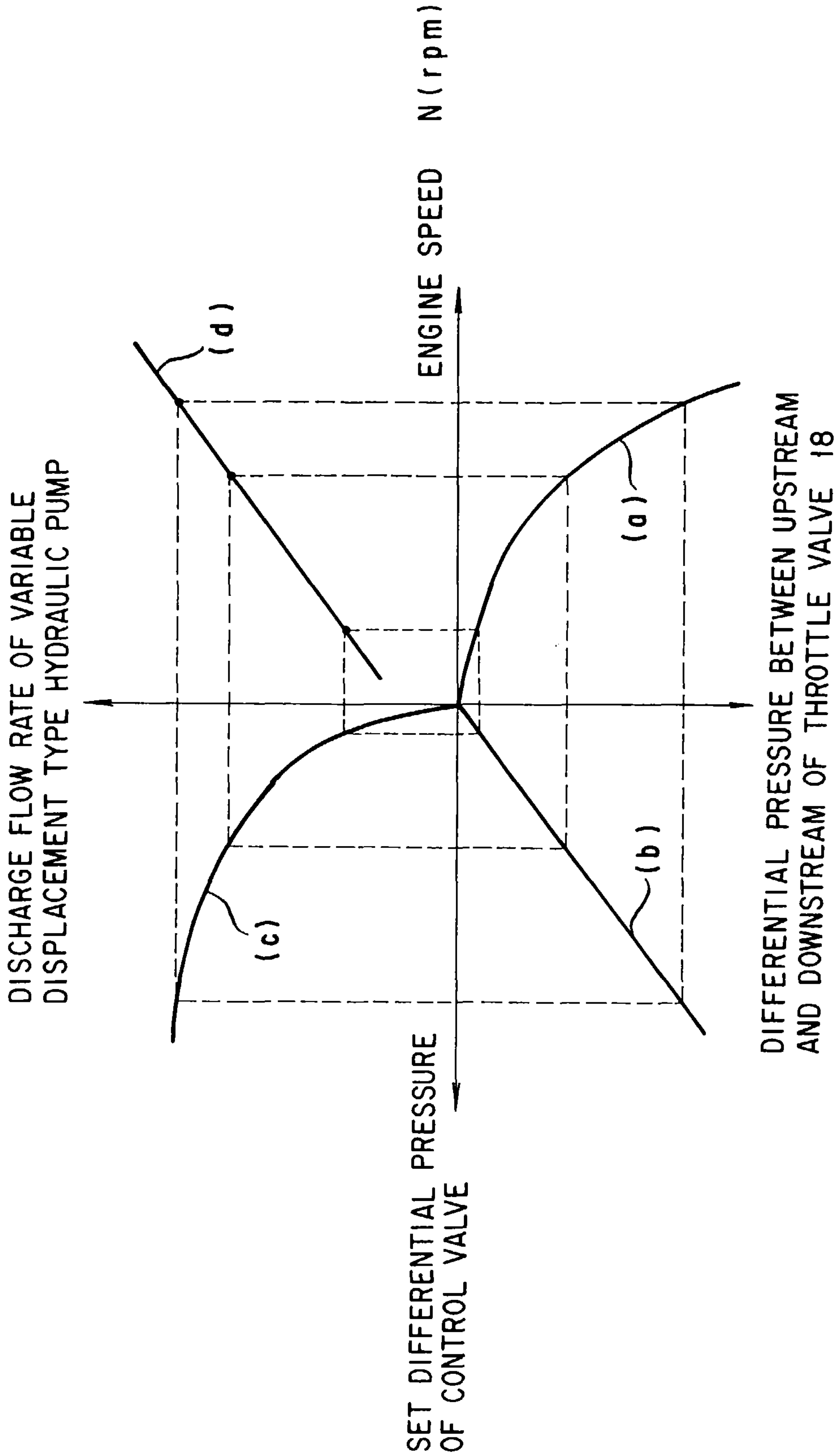
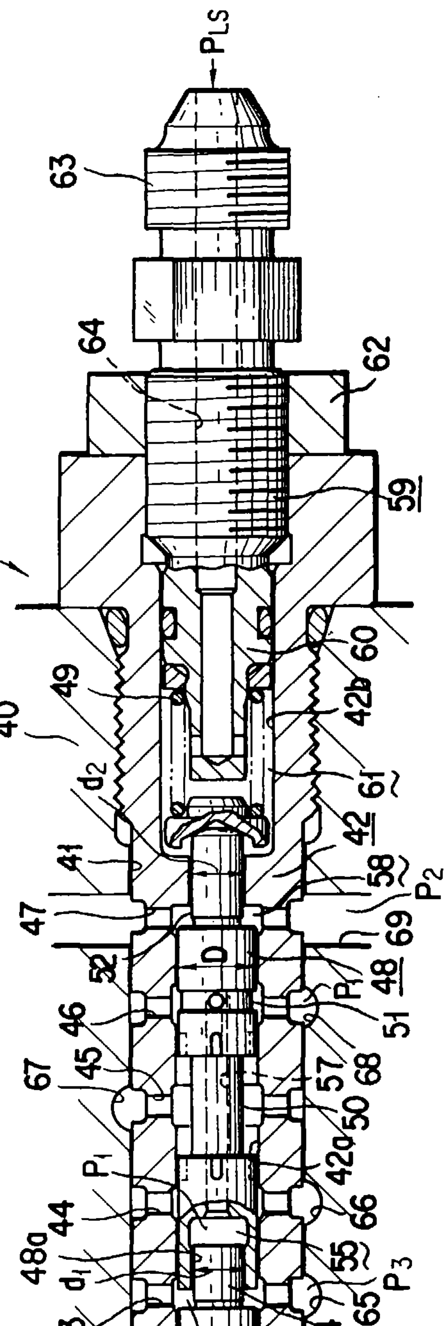

DISCHARGE FLOW RATE OF VARIABLE
DISPLACEMENT TYPE HYDRAULIC PUMP
ENGINE SPEED N(rpm)
DIFFERENTIAL PRESSURE BETWEEN UPSTREAM
AND DOWNSTREAM OF THROTTLE VALVE 18
SET DIFFERENTIAL PRESSURE
OF CONTROL VALVE
(a)
(b)
(c)
(d)

FIG. 3



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D
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68
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FIG. 4

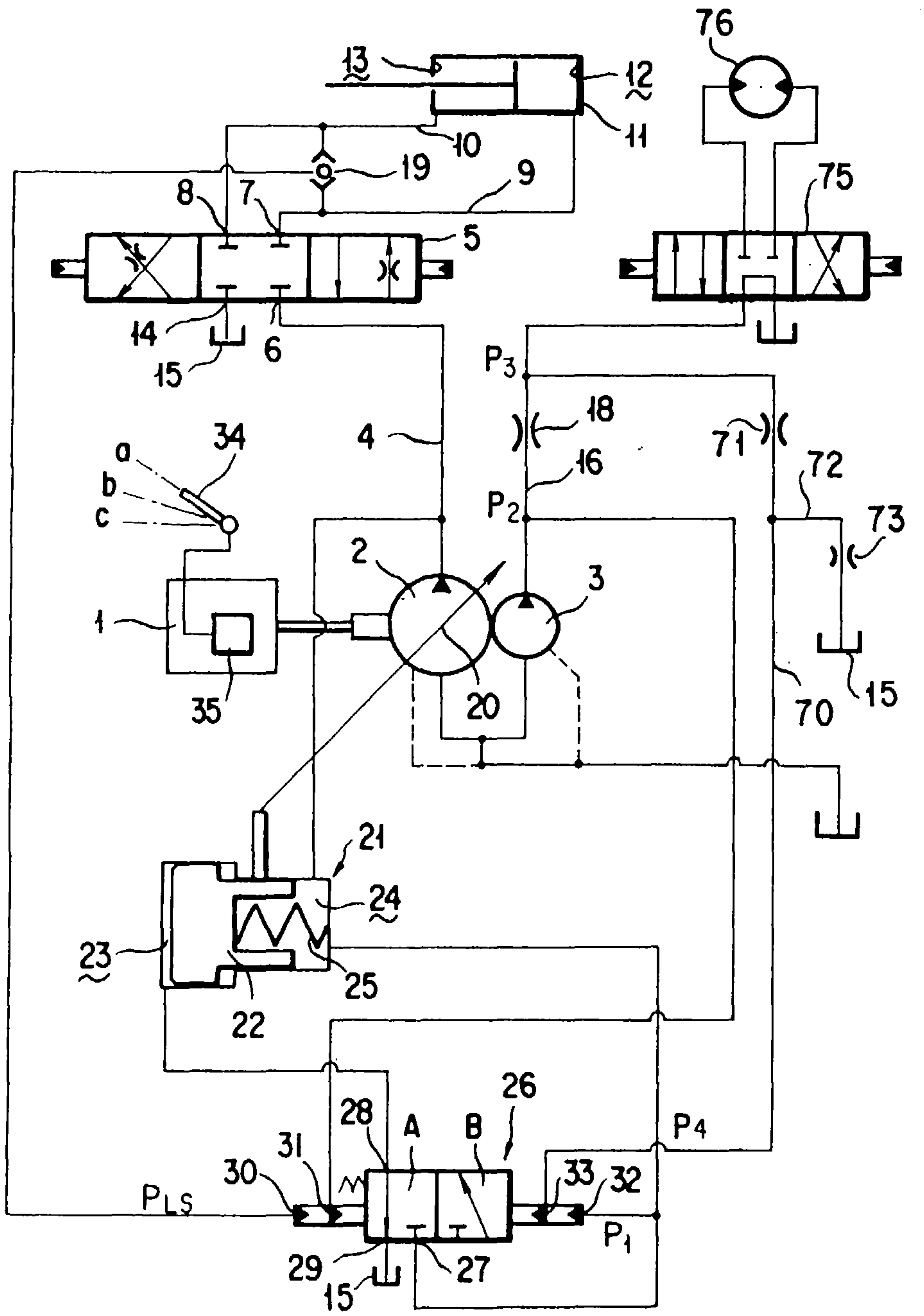
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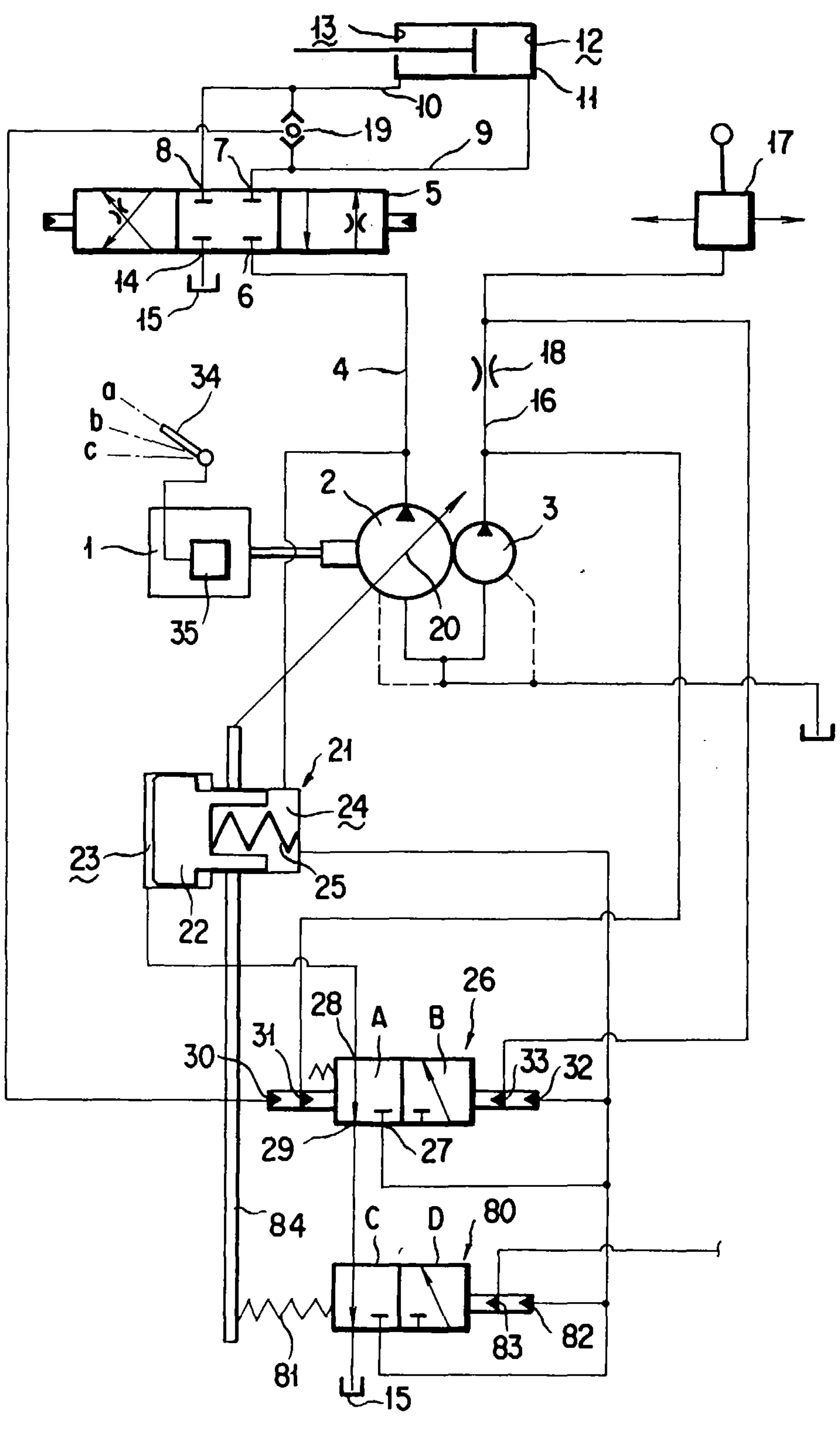
FIG. 5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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP96/01005

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl^6 F15B11/00, 11/02

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int. Cl^6 F15B11/00, 11/02

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Jitsuyo Shinan Koho 1926 - 1996
Kokai Jitsuyo Shinan Koho 1971 - 1996

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
	JP, 5-99126, A (Komatsu Ltd.), April 20, 1993 (20. 04. 93)(Family: none)	
X	Fig. 2	1, 2, 7
X	Figs. 2 and 3	5, 6
Y	Fig. 2	3, 4
	JP, 6-159309, A (Hitachi Construction Machinery Co., Ltd.), June 7, 1994 (07. 06. 94)(Family: none)	
Y	Fig. 1	3, 4

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:
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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
July 8, 1996 (08. 07. 96)	July 23, 1996 (23. 07. 96)
Name and mailing address of the ISA/ Japanese Patent Office	Authorized officer
Facsimile No.	Telephone No.

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