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FIG. 1

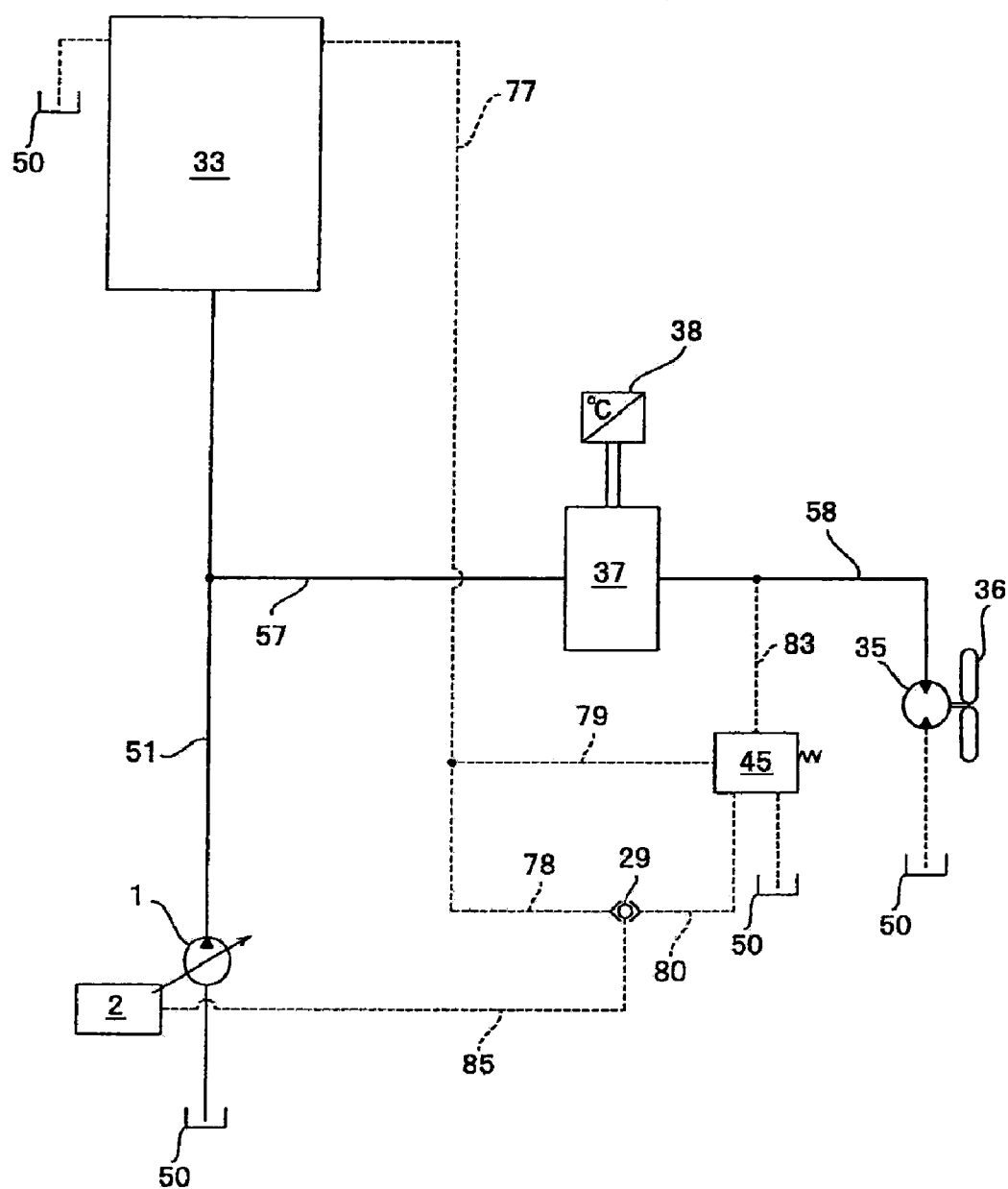


FIG. 2

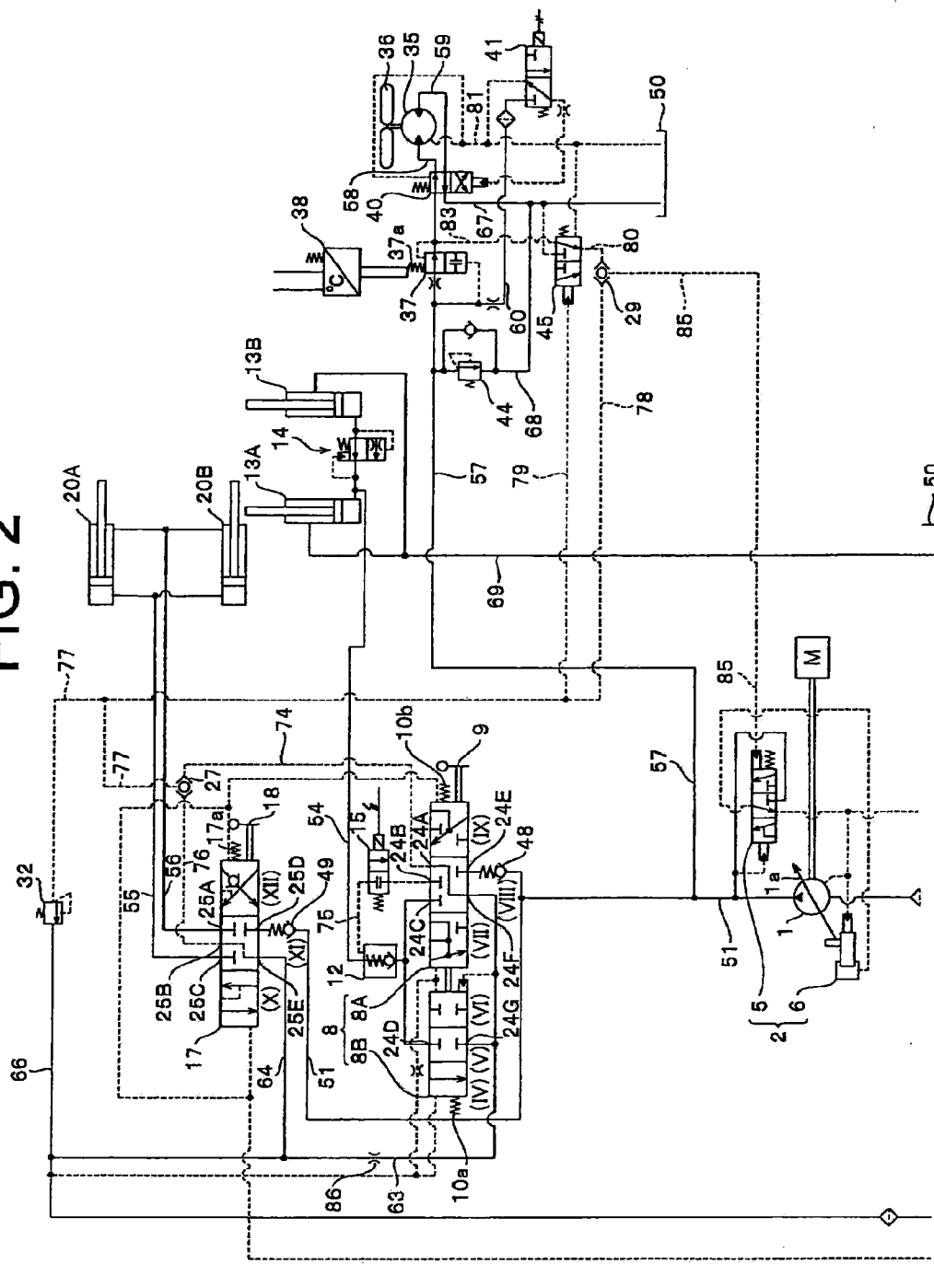


FIG. 3

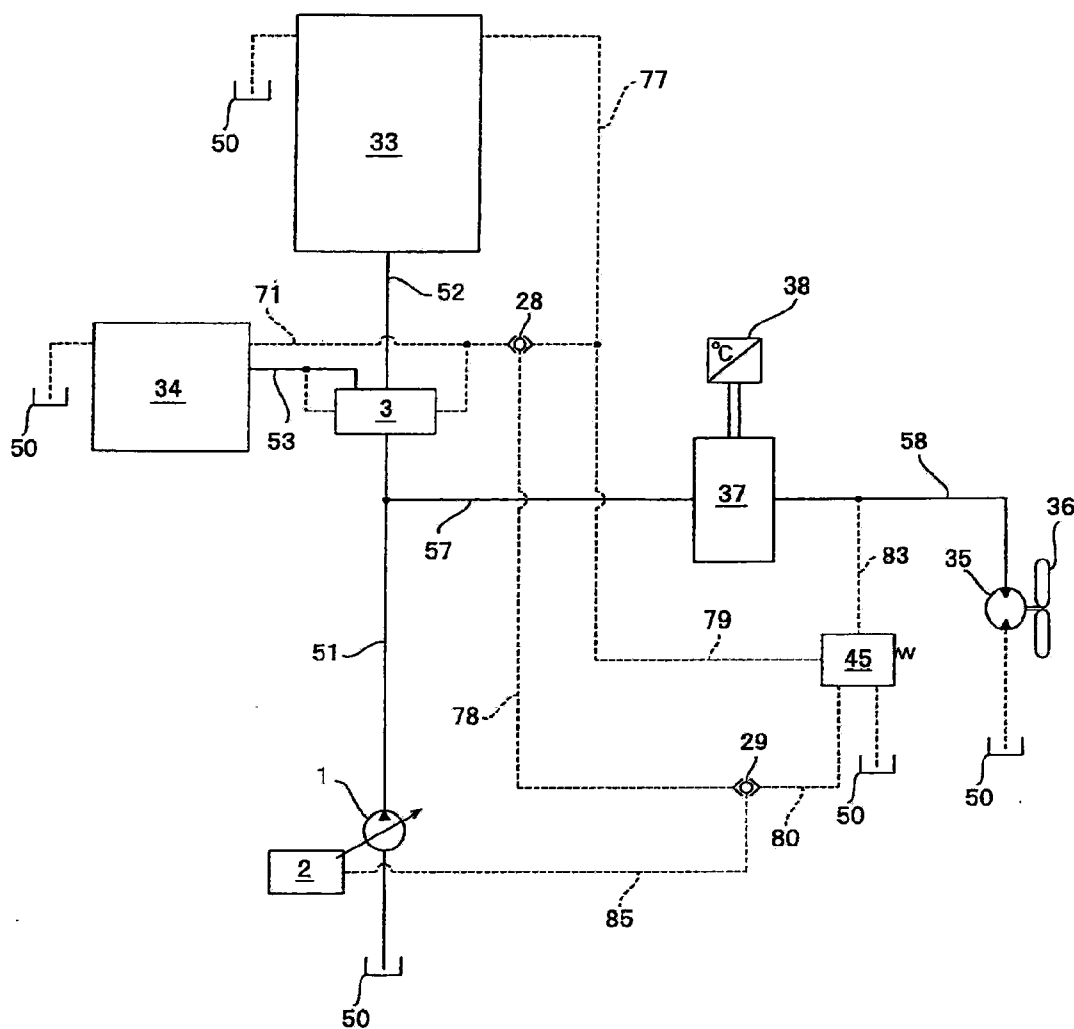


FIG. 4

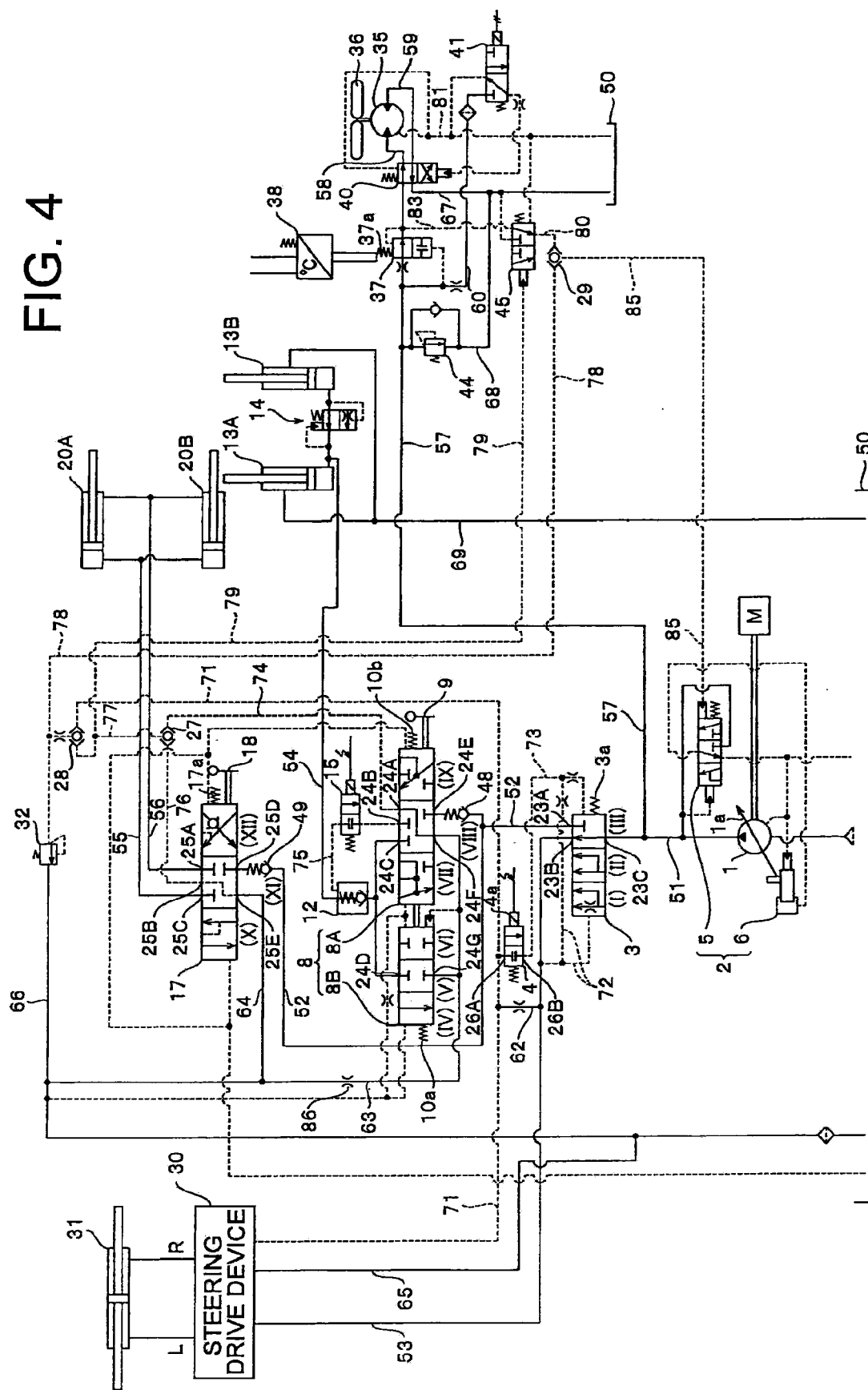


FIG. 6

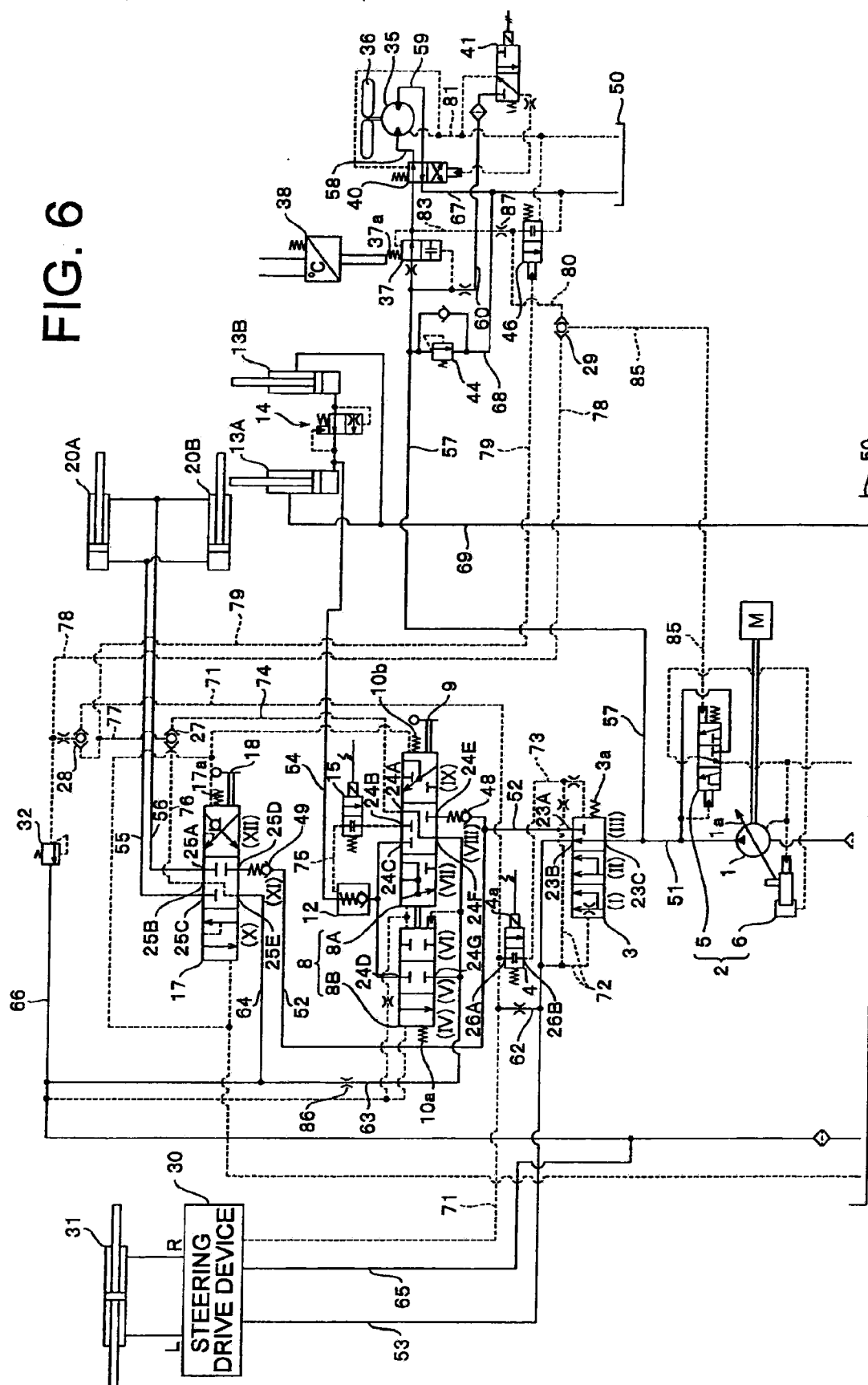
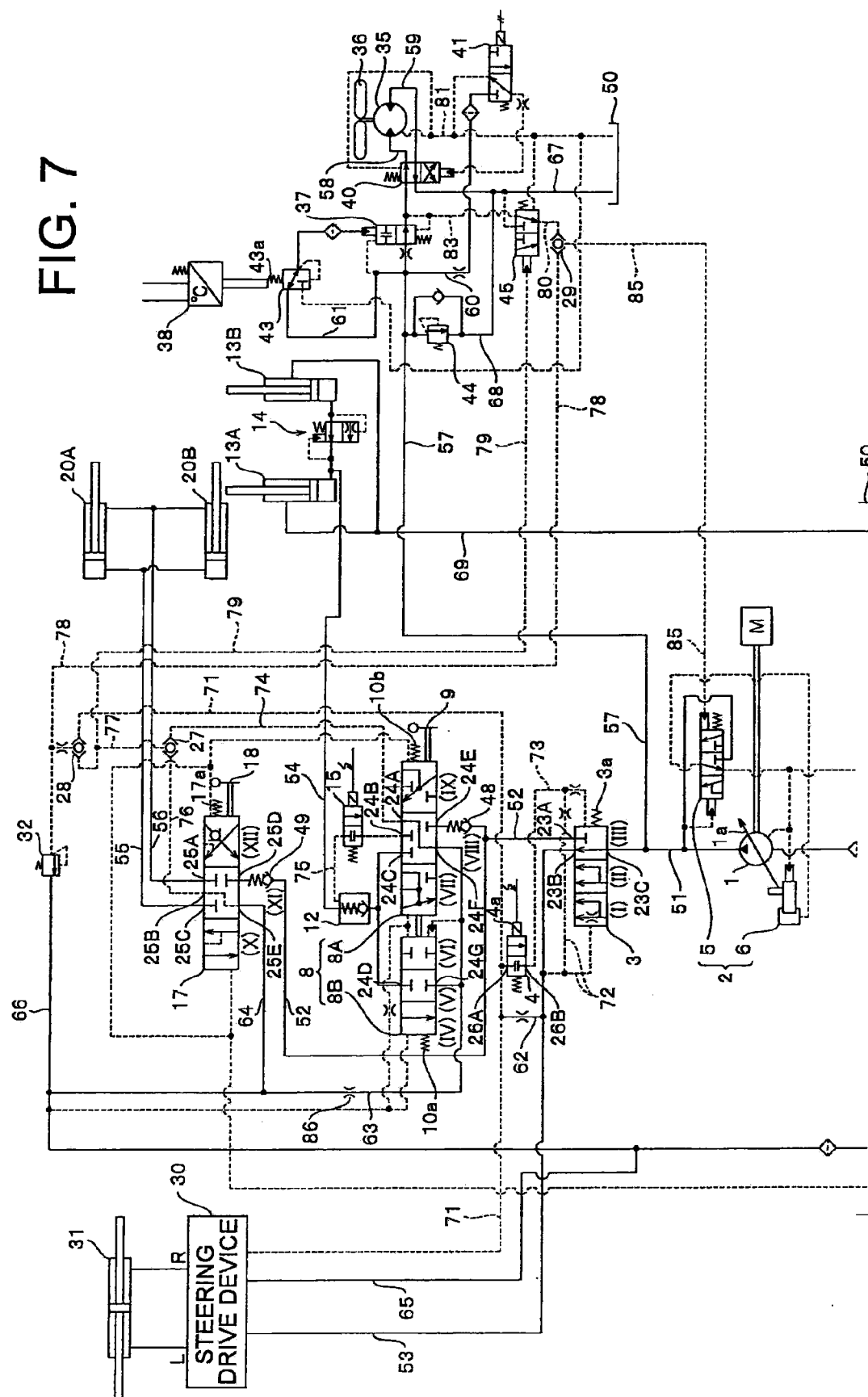


FIG. 7



8
G.
F/G

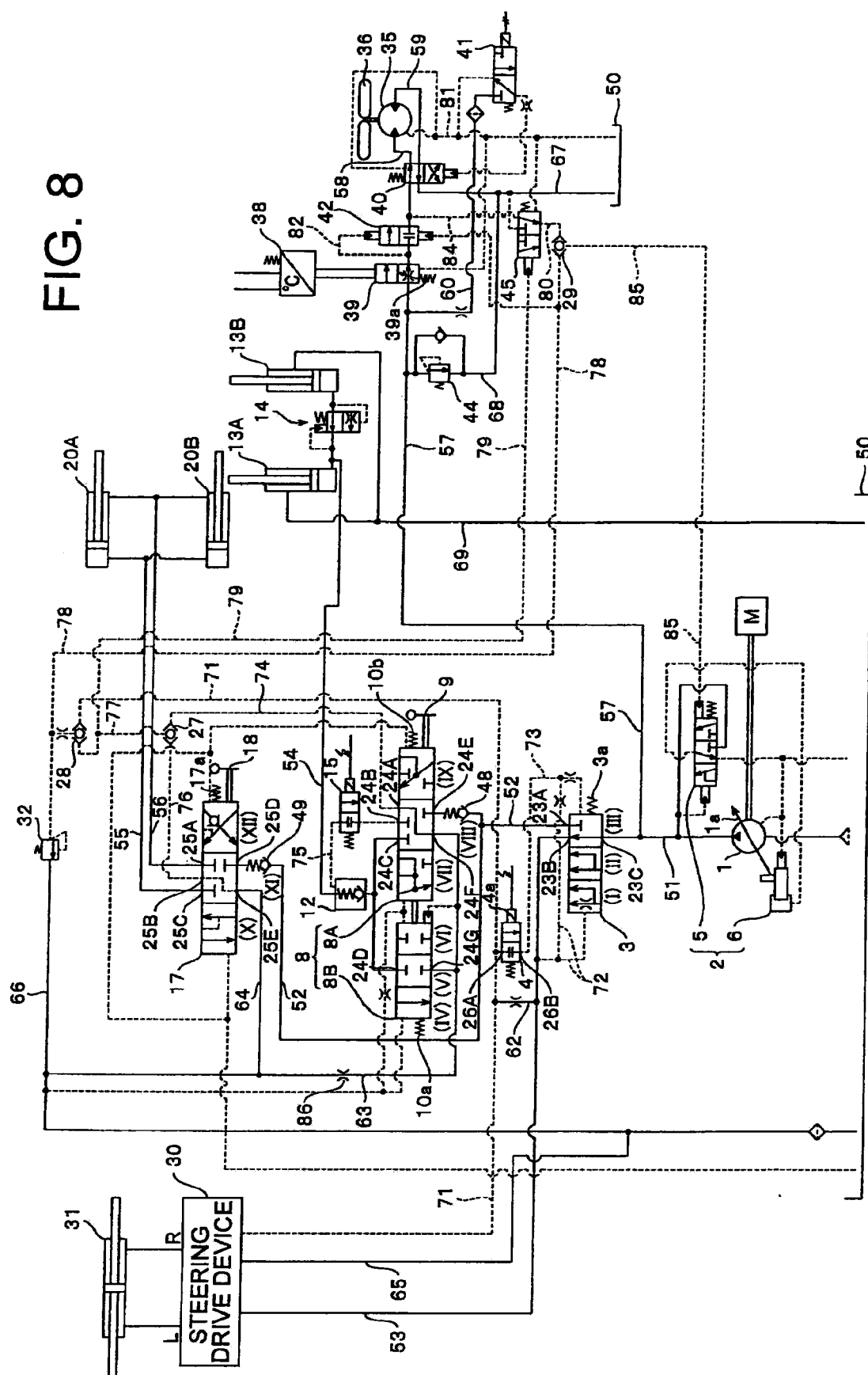
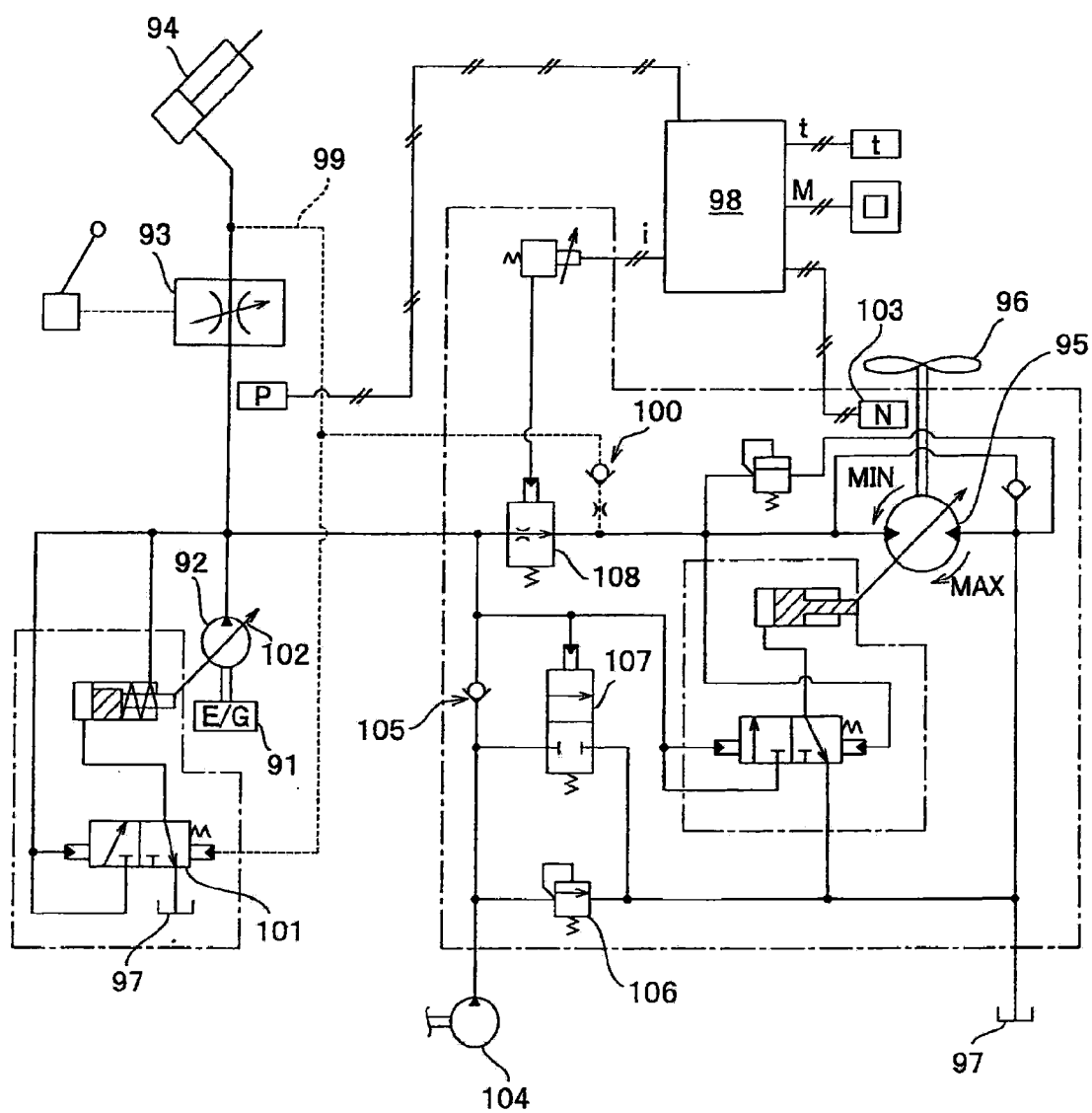


FIG. 9



FAN DRIVE SYSTEM

TECHNICAL FIELD

[0001] The present invention relates to a fan drive system which drives an engine cooling fan mounted in a forklift, a skid steer loader (SSL) and a crawler dump which are working apparatuses.

BACKGROUND ART

[0002] In a conventional working apparatus, an engine thereof is cooled using a water-cool cooling device. That is, the engine is cooled by circulating a coolant through a water jacket provided in an engine body. The coolant heated in the water jacket is led to a radiator and cooled, and the cooled coolant is again returned to the water jacket.

[0003] A cooling fan is disposed in front of a radiator, and a coolant passing through the radiator is cooled by wind generated by the cooling fan. Generally, the cooling fan is driven by an engine through a belt. Thus, the number of revolutions of the cooling fan corresponds to the number of revolutions of the engine.

[0004] If the cooling fan is driven directly by the engine, a disposition portion of the cooling fan is limited, and a freedom degree of layout of the radiator and other devices is limited. Therefore, there is employed a structure in which a hydraulic pump is used as a drive source instead of driving the cooling fan directly by the engine, and a hydraulic motor of the cooling fan is driven with a discharge flow rate from the hydraulic pump.

[0005] If the hydraulic pump for the cooling fan is used, the freedom degree of layout of the hydraulic motor, the cooling fan, the radiator and other devices is enhanced. However, since the hydraulic pump for the cooling fan must newly be disposed, the number of hydraulic pumps is increased as a whole.

[0006] In order to newly dispose the hydraulic pump, the layout structure becomes complicated, and a wider installation space is required. Further, there is a problem that the cost is increased and the number of parts is increased.

[0007] In order to solve this problem, there is proposed a fan drive system in which a variable displacement pump which supplies oil to an actuator is also used as a hydraulic pump for the hydraulic motor which drives the cooling fan.

[0008] As the fan drive system in which the variable displacement pump is also used as the hydraulic pump, there is proposed a drive device described in a patent document 1. The patent document 1 discloses a cooling fan device in which the variable displacement pump is also used as the hydraulic pump, and the hydraulic pump hydraulically drives the cooling fan. The drive device described in the patent document 1 is for a working machine such as a power shovel in which a working machine is frequently used during running, and a large displacement hydraulic pump is used so that oil can be simultaneously supplied to both a running device and the working machine.

[0009] There are many kinds of working apparatuses such as a working apparatus which frequently uses the working machine, and a working apparatus which is mainly used for running and in which the working machine is used only for short time during running. Examples of such working apparatuses which are mainly used for running are a forklift, a skid steer loader (SSL) and a crawler dump.

[0010] The present invention relates to a fan drive system for a working apparatus such as the forklift, but a cooling fan device described in the patent document 1 is not for the working apparatus such as the forklift. In the cooling fan device of the patent document 1, there is disclosed a structure in which a discharge flow rate from a hydraulic pump is supplied to an actuator and a hydraulic motor which drives the cooling fan. Hence, a drive device of the patent document 1 will be explained as a conventional example 1 of the present invention.

[0011] FIG. 9 is a hydraulic circuit diagram of the drive device of the patent document 1. As shown in FIG. 9, a main hydraulic pump 92 is constituted as a variable displacement pump which is driven by an engine 91. A discharge flow rate from the main hydraulic pump 92 is supplied to a working machine hydraulic cylinder 94 through an operation valve 93. A portion of the discharge flow rate from the main hydraulic pump 92 is supplied to the hydraulic motor 95 through a flow rate control valve 108. The hydraulic motor 95 drives a cooling fan 96.

[0012] The main hydraulic pump 92 is constituted as a load pressure sensitive type hydraulic pump. The main hydraulic pump 92 has a swash plate 102, and an angle of the swash plate 102 is controlled in accordance with a high-pressure-side load pressure among a load pressure in the hydraulic cylinder 94 and a load pressure in the hydraulic motor 95. The load pressure in the hydraulic cylinder 94 is taken out through a load sensing oil path 99 (LS oil path, hereinafter), and the high-pressure-side load pressure among the load pressure in the hydraulic motor 95 and the load pressure in the hydraulic cylinder 94 is led to the load sensing valve 101 by a check valve 100.

[0013] To complement a flow rate of oil to be supplied to the hydraulic motor 95, a fixed displacement hydraulic pump 104 is provided. A flow rate of oil supplied to the hydraulic motor 95 from the fixed displacement hydraulic pump 104 through a check valve 105 is controlled by a relief valve 106 and an unload valve 107.

[0014] If a pump pressure of the fixed displacement hydraulic pump 104 becomes excessively high, the relief valve 106 is operated to an open position side, and a discharge flow rate from the fixed displacement hydraulic pump 104 is discharged to a tank 97 through the relief valve 106. If a pump pressure of the main hydraulic pump 92 becomes high, the unload valve 107 is switched to an open position. With this, the discharge flow rate from the fixed displacement hydraulic pump 104 is discharged to the tank 97 through the unload valve 107.

[0015] A temperature of working oil in the tank 97 detected by a temperature sensor and the number of revolutions of the fan detected by a fan-revolution number sensor 103 are input to a controller 98. An opening area of the flow rate control valve 108 is controlled using these input detection signals. If the flow rate control valve 108 is controlled and the supply flow rate to the hydraulic motor 95 is controlled, an absorption torque of the hydraulic motor 95 can be controlled to a preset absorption torque.

[0016] With this, in the drive device for the cooling fan described in the patent document 1, even when an absorption torque of the hydraulic motor 95 is varied, variation of the number of revolutions of the cooling fan 96 can be suppressed, and the rotation of the cooling fan 96 can be stabilized. Even when a load of the hydraulic motor 95 is varied,

variation of the number of revolutions of the cooling fan **96** can be suppressed, and the rotation of the cooling fan **96** can be stabilized.

[0017] Patent document 1: Japanese Patent Laid-Open Publication No. 2000-161060

DISCLOSURE OF THE INVENTION

Problem to be Solved by the Invention

[0018] The drive device for the cooling fan described in the patent document 1 is for a working apparatus which frequently uses the working machine even during running like a power shovel. Therefore, a large displacement hydraulic pump is used as the main hydraulic pump **92** so that oil can simultaneously be supplied to all of a running device, the working machine and the hydraulic motor **95** for the cooling fan **96**. The fixed displacement hydraulic pump **104** is provided so that the flow rate can be complement when a flow rate of oil supplied to the hydraulic motor **95** which rotates the cooling fan **96** is reduced.

[0019] The present inventor thought that in the case of a working apparatus such as a forklift, a skid steer loader (SSL) and a crawler dump which is mainly used during running and its working machine is used only for a short time, it is unnecessary to use a large displacement hydraulic pump unlike the apparatus described in the patent document 1 as the maximum discharge flow rate which can be discharged from one hydraulic pump.

[0020] As a result of intensive research, the inventor found that it was only necessary that the hydraulic pump included a pump displacement capable of securing a discharge flow rate which could be supplied to the working machine that most required the flow rate. To provide a fan drive system which can adjust a pump displacement of a hydraulic pump in accordance with the maximum flow rate required for operating the working machine, and which can use a small variable displacement pump having a small pump displacement, the inventor created the following solving means.

[0021] (1) First, as a load pressure which controls the pump displacement of the hydraulic pump, a load pressure of the working machine and a load pressure of the hydraulic motor which drives the cooling fan are separated from each other, and the pump displacement of the hydraulic pump is controlled using one of the load pressures in accordance with conditions.

[0022] (2) When the working machine is operated, priority is placed on the load pressure of the working machine, and the pump displacement of the hydraulic pump is controlled in accordance with the load pressure of the working machine on which the priority is placed. (3) When the working machine is operated during running of the working apparatus, oil of sufficient flow rate can not be supplied to the hydraulic motor which drives the cooling fan, time during which the working machine is operated during running is short. Thus, even if the amount of wind to be supplied to the radiator is temporarily reduced, the temperature rise in the radiator can be suppressed to a low level.

[0023] (4) When the working machine is not operated, the pump displacement of the hydraulic pump is controlled in accordance with the load pressure of the hydraulic motor which drives the cooling fan. With this, even if the temperature in the radiator is temporarily increased, if the cooling fan normally rotates, the increased temperature can be decreased. (5) As the maximum pump displacement of the hydraulic

pump, even when time during which the working machine is operated is short, the pump displacement suitable for the maximum flow rate required for operating the working machine is employed. With this, the operation of the working machine can be stabilized. (6) Therefore, the pump displacement of the hydraulic pump can be reduced.

Means for Solving the Problem

[0024] The object of the present invention can be achieved by the inventions described in claims **1** to **5**.

[0025] That is, a first invention of the present application provides a fan drive system being characterized by comprising a load pressure sensitive variable displacement pump, a working machine circuit and a flow rate control valve to which a discharge flow rate from the variable displacement pump is supplied, a load pressure separation valve which is controlled by a maximum load pressure in the working machine circuit, a hydraulic motor which drives a cooling fan, a first discharge oil path which connects the variable displacement pump and the working machine circuit with each other, a second discharge oil path which is branched from the first discharge oil path and which is connected to the flow rate control valve with each other, a supply oil path which connects the flow rate control valve and the hydraulic motor with each other, a first pilot oil path which takes out the maximum load pressure in the working machine circuit; a second pilot oil path which takes out a load pressure that drives the hydraulic motor, and a shuttle valve which selects a high-pressure-side load pressure between the maximum load pressure in the first pilot oil path and the load pressure in the second pilot oil path, wherein

[0026] a pump displacement of the variable displacement pump is controlled in accordance with a differential pressure between the high-pressure-side load pressure selected by the shuttle valve and a pump pressure of the variable displacement pump, the load pressure separation valve is disposed in the second pilot oil path, the load pressure separation valve is controlled in accordance with a differential pressure between a pressing force by the maximum load pressure taken out by the first pilot oil path and a biasing force of a spring applied to the load pressure separation valve,

[0027] when the pressing force by the maximum load pressure is greater than the biasing force of the spring, the load pressure separation valve is switched from a position where the second pilot oil path is brought into communication with the shuttle valve to a position where the second pilot oil path is brought into communication with a tank, and a tank pressure is led to the shuttle valve, and when the pressing force by the maximum load pressure is smaller than the biasing force of the spring, the load pressure separation valve is switched from the position where the second pilot oil path is brought into communication with the tank to the position where the second pilot oil path is brought into communication with the shuttle valve, and the load pressure which drives the hydraulic motor is led to the shuttle valve.

[0028] According to a fan drive system of a second invention of the present application, in the structure of the above-described first invention, a throttle is disposed in the second pilot oil path instead of disposing the load pressure separation valve in the second pilot oil path, the second pilot oil path is branched into two oil paths downstream of the throttle, one of the branched oil paths is connected to the shuttle valve, the

load pressure separation valve is disposed at an intermediate portion of the other oil path, and the other oil path is connected to a tank,

[0029] the load pressure separation valve is controlled in accordance with a differential pressure between a pressing force by the maximum load pressure taken out from the first pilot oil path and a biasing force of a spring applied to the load pressure separation valve, when the pressing force by the maximum load pressure is greater than the biasing force of the spring, the load pressure separation valve is switched to a position where the other oil path is brought into communication with the tank, and a tank pressure is led to the shuttle valve, and

[0030] when the pressing force by the maximum load pressure is smaller than the biasing force of the spring, the load pressure separation valve is switched from the position where the other oil path is brought into communication with the tank to a position where a communication is shut off, and the load pressure which drives the hydraulic motor is led to the shuttle valve.

[0031] A third invention of the present application provides a fan drive system being characterized by comprising a load pressure sensitive variable displacement pump, a steering circuit, a working machine circuit and a flow rate control valve to which a discharge flow rate from the variable displacement pump is supplied, a priority valve which supplies the discharge flow rate from the variable displacement pump preferentially to the steering circuit using the steering circuit as a priority circuit with respect to the working machine circuit, a load pressure separation valve which is controlled by a maximum load pressure in the working machine circuit, a hydraulic motor which drives a cooling fan,

[0032] a third discharge oil path which connects the variable displacement pump and the priority valve with each other, a fourth discharge oil path which connects the priority valve and the steering circuit with each other, a fifth discharge oil path which connects the priority valve and the working machine circuit with each other, a sixth discharge oil path which is branched from the third discharge oil path and which is connected to the flow rate control valve, a supply oil path which connects the flow rate control valve and the hydraulic motor with each other,

[0033] a first pilot oil path which takes out the maximum load pressure in the working machine circuit, a second pilot oil path which takes out a load pressure for driving the hydraulic motor, a third pilot oil path which takes out a load pressure in the steering circuit,

[0034] a first shuttle valve which selects a high-pressure-side load pressure between the maximum load pressure in the first pilot oil path and the load pressure in the third pilot oil path, and a second shuttle valve which selects a high-pressure-side load pressure between the high-pressure-side load pressure selected by the first shuttle valve and the load pressure in the second pilot oil path, wherein

[0035] a pump displacement of the variable displacement pump is controlled in accordance with a differential pressure between the high-pressure-side load pressure selected by the second shuttle valve and a pump pressure of the variable displacement pump, the load pressure separation valve is disposed in the second pilot oil path, the load pressure separation valve is controlled in accordance with a differential pressure between a pressing force by the maximum load pressure taken out from the first pilot oil path and a biasing force of a spring applied to the load pressure separation valve,

[0036] when the pressing force by the maximum load pressure is greater than the biasing force of the spring, the load pressure separation valve is switched from a position where the second pilot oil path is brought into communication with the second shuttle valve to a position where the second pilot oil path is brought into communication with a tank, and a tank pressure is led to the second shuttle valve, and when the pressing force by the maximum load pressure is smaller than the biasing force of the spring, the load pressure separation valve is switched from the position where the second pilot oil path is brought into communication with the tank to the position where the second pilot oil path is brought into communication with the second shuttle valve, and the load pressure for driving the hydraulic motor is led to the second shuttle valve.

[0037] According to a fan drive system of a fourth invention of the present application, in the structure of the above-described third invention, a throttle is disposed in the second pilot oil path instead of disposing the load pressure separation valve in the second pilot oil path, the second pilot oil path is branched into two oil paths downstream of the throttle, one of the branched oil paths is connected to the second shuttle valve, the load pressure separation valve is disposed in an intermediate portion of the other oil path, and the other oil path is connected to a tank,

[0038] the load pressure separation valve is controlled in accordance with a differential pressure between a pressing force by the maximum load pressure taken out from the first pilot oil path and a biasing force of a spring applied to the load pressure separation valve, when the pressing force by the maximum load pressure is greater than the biasing force of the spring, the load pressure separation valve is switched to a position where the other oil path is brought into communication with the tank, and a tank pressure is led to the second shuttle valve, and

[0039] when the pressing force by the maximum load pressure is smaller than the biasing force of the spring, the load pressure separation valve is switched from the position where the other oil path is brought into communication with the tank to a position where a communication is shut off, and the load pressure for driving the hydraulic motor is led to the second shuttle valve.

[0040] According to a fan drive system of a fifth invention of the present invention, the control structure of the flow rate control valve of the first to fourth inventions is specified.

Effect of the Invention

[0041] When a load pressure of the working machine is generated in a working machine circuit, the displacement of the variable displacement pump can be controlled using the maximum load pressure among the load pressures of the working machine circuit. Further, it is possible to prevent the load pressure which controls the pump displacement of the variable displacement pump from varying during operation of the working machine, and the working machine can be operated in the stable state.

[0042] Further, the discharge flow rate from the variable displacement pump is not used for the operation of the working machine and it is possible to prevent the discharge flow rate from being consumed wastefully. Since the displacement

of the hydraulic pump can be a value required for operating the working machine, the displacement of the hydraulic pump can be reduced.

BRIEF DESCRIPTION OF THE DRAWINGS

- [0043] FIG. 1 is a simplified hydraulic circuit diagram (First embodiment);
- [0044] FIG. 2 is a hydraulic circuit diagram (First embodiment);
- [0045] FIG. 3 is a simplified hydraulic circuit diagram (Second embodiment);
- [0046] FIG. 4 is a hydraulic circuit diagram (Second embodiment);
- [0047] FIG. 5 is a hydraulic circuit diagram when a priority valve is not used (Third embodiment);
- [0048] FIG. 6 is a hydraulic circuit diagram when the priority valve is used (Third embodiment);
- [0049] FIG. 7 is a hydraulic circuit diagram (Fourth embodiment);
- [0050] FIG. 8 is a hydraulic circuit diagram (Fifth embodiment); and
- [0051] FIG. 9 is a hydraulic circuit (Conventional example 1).

EXPLANATION OF REFERENCE NUMERALS

- [0052] 1 Variable displacement pump
- [0053] 2 Displacement control device
- [0054] 3 Priority valve
- [0055] 4 Electromagnetic switching control valve
- [0056] 8 First direction switching valve
- [0057] 13A, 13B Lift cylinder
- [0058] 17 Second direction switching valve
- [0059] 20A, 20B Tilt cylinder
- [0060] 27 to 29 Shuttle valve
- [0061] 30 Steering drive device
- [0062] 31 Actuator
- [0063] 33 Working machine circuit
- [0064] 34 Steering circuit
- [0065] 35 Hydraulic motor
- [0066] 36 Cooling fan
- [0067] 37 Flow rate control valve
- [0068] 38 Thermo-module
- [0069] 39 Variable throttle valve
- [0070] 40 Normal/reverse rotation switching valve
- [0071] 41 Normal/reverse rotation solenoid valve
- [0072] 42 Pressure compensation valve
- [0073] 43 Decompression valve
- [0074] 45 Load pressure separation valve
- [0075] 46 Load pressure separation valve
- [0076] 92 Main hydraulic pump
- [0077] 95 Hydraulic motor
- [0078] 96 Cooling fan
- [0079] 98 Controller
- [0080] 101 LS valve
- [0081] 104 Fixed displacement hydraulic pump
- [0082] 106 Relief valve
- [0083] 107 Unload valve
- [0084] 108 Flow rate control valve

BEST MODE FOR CARRYING OUT THE INVENTION

[0085] Preferable embodiments of the present invention will be specifically explained below based on the accompa-

nying drawings. A structure of a fan drive system for a forklift of the invention is not limited to a hydraulic circuit structure having a fan drive system explained below, and other hydraulic circuit structures can be employed only if the technical idea of the invention is satisfied.

First Embodiment

[0086] A hydraulic circuit having a fan drive system according to the First embodiment of the present invention will be explained using FIGS. 1 and 2. FIG. 1 is a simplified hydraulic circuit diagram and FIG. 2 is a detailed hydraulic circuit diagram. First, the hydraulic circuit having the fan drive system according to the First embodiment of the invention will be briefly explained using FIG. 1 and then, the hydraulic circuit having the fan drive system according to the First embodiment of the invention will be explained using FIG. 2. Numbers of common members in FIGS. 1 and 2 will be explained using the same numbers of members.

[0087] As shown in FIG. 1, a discharge flow rate from a load pressure sensitive variable displacement pump 1 which is driven by an engine (not shown) is discharged to a discharge oil path 51 as a first discharge oil path. The discharge oil path 51 is branched into an oil path 57 as a second discharge oil path. The discharge oil path 51 is connected to a working machine circuit 33. The oil path 57 is connected to a flow rate control valve 37. A discharge flow rate from the variable displacement pump 1 controlled by the flow rate control valve 37 is supplied as an operation flow rate which drives a hydraulic motor 35 through an oil path 58 as a supply oil path.

[0088] The discharge flow rate from the variable displacement pump 1 is controlled by the displacement control device 2. The pump displacement of the variable displacement pump 1 is controlled by operating the displacement control device 2. The displacement control device 2 can be operated in accordance with a differential pressure between a pump pressure in the discharge oil path 51 and a high-pressure-side load pressure of the maximum load pressure in the working machine circuit 33 and a load pressure in the hydraulic motor 35.

[0089] The maximum load pressure in the working machine circuit 33 is taken out by a pilot oil path 77 as a first pilot oil path. The pilot oil path 77 is branched into a pilot oil path 78 connected to one side of the shuttle valve 29 and a pilot oil path 79 connected to the load pressure separation valve 45.

[0090] The load pressure of the hydraulic motor 35 is taken out by a pilot oil path 83 as a second pilot oil path. A load pressure separation valve 45 is disposed in the pilot oil path 83. The pilot oil path 83 is switched between connection to the pilot oil path 80 connected to the other side of the shuttle valve 29 and connection to a tank 50 by the load pressure separation valve 45.

[0091] The maximum load pressure in the working machine circuit 33 taken by the pilot oil path 77 is led to the load pressure separation valve 45 through the pilot oil path 79, and a spring force of a spring is applied to an end surface of the working machine circuit 33 opposite from an end surface to which the maximum load pressure is applied. When the load pressure is generated in the working machine circuit 33, the load pressure separation valve 45 is switched against the spring force of the spring, the pilot oil path 83 is connected to the tank 50, and the load pressure in the pilot oil path 80 is set to a tank pressure. When the load pressure is not generated in the working machine circuit 33, the load pressure separation

valve 45 is switched by the spring force of the spring, the pilot oil path 83 is connected to the pilot oil path 80, and the load pressure of the hydraulic motor 35 is set to the load pressure of the pilot oil path 80.

[0092] When a plurality of working machines are disposed in the working machine circuit 33, and a plurality of load pressures exist in the working machine circuit 33, the highest load pressure of the existing plurality of load pressures is taken out by the pilot oil path 77. Even if a plurality of working machines are provided in the working machine circuit 33, if a load pressure in one working machine exists in the working machine circuit 33, that load pressure is taken out by the pilot oil path 77.

[0093] The high-pressure-side load pressure taken out by the shuttle valve 29 is led to the displacement control device 2 through the pilot oil path 85. A pump pressure in the discharge oil path 51 is led to the displacement control device 2, the displacement control device 2 is operated in accordance with a differential pressure between the pump pressure and the high-pressure-side load pressure taken out by the shuttle valve 29, and the pump displacement of the variable displacement pump 1 is controlled.

[0094] An opening area of the flow rate control valve 37 is adjusted by a thermo-module 38 which is displaced in accordance with a coolant temperature cooled by a radiator (not shown). A structure of the thermo-module 38 will be explained later with reference to FIG. 2.

[0095] With this structure, when the load pressure is not generated in the working machine circuit 33, the pump displacement of the variable displacement pump 1 can be controlled in accordance with the load pressure of the hydraulic motor 35. When the load pressure is generated in the working machine circuit 33, the pump displacement of the variable displacement pump 1 can be controlled in accordance with the maximum load pressure of the working machine circuit 33.

[0096] Next, the hydraulic circuit having the fan drive system according to the First embodiment will be explained in detail using FIG. 2. Numbers of common members in FIGS. 1 and 2 will be explained using the same numbers of members. As shown in FIG. 2, a discharge flow rate from the load pressure sensitive variable displacement pump 1 driven by an engine M is supplied to the discharge oil path 51.

[0097] The discharge oil path 51 is connected to a pump port 24E of a first direction switching valve 8 through a check valve 48, and is also connected to a pump port 25D of a second direction switching valve 17 through a check valve 49.

[0098] A direction control valve 8 is connected to bottom sides of a pair of lift cylinders 13A and 13B through an oil path 54. A pilot check valve 12 is disposed in the oil path 54, and the pilot check valve 12 is controlled by an electromagnetic switching valve 15. Oil returning from head sides of the pair of lift cylinders 13A and 13B is discharged into a tank 50 through a drain oil path 69, and oil returning from bottom sides of the pair of lift cylinders 13A and 13B is discharged and controlled through the oil path 54.

[0099] A direction switching valve 17 is connected to a pair of tilt cylinders 20A and 20B through oil paths 55 and 56.

[0100] Load pressure on the bottom sides of the pair of lift cylinders 13A and 13B are taken out by a pilot oil path 74, and are led to one end side of a shuttle valve 27. Load pressures in the pair of tilt cylinders 20A and 20B are taken out by a pilot oil path 76, and are led to the other end side of the shuttle valve 27.

[0101] A high-pressure-side load pressure selected by the shuttle valve 27 is taken out by the pilot oil path 77, and is led to one end side of the shuttle valve 29 through the pilot oil path 78, and is led to the load pressure separation valve 45 through the pilot oil path 79.

[0102] The first direction switching valve 8 can be switched by operation of an operating lever 9 and is constituted as a switching valve having seven ports 24A to 24G. The first direction switching valve 8 has a spool structure divided into two spools, i.e., a first spool 8A and a second spool 8B. By operating the operating lever 9, the first spool 8A can be switched from a position VII to a position IX. The second spool 8B follows the movement of the first spool 8A by a biasing force of the spring 10a and can be switched from the position IV to a position VI. The spring 10b is constituted as a neutral spring which returns the first direction switching valve 8 to neutral V and VIII positions.

[0103] The ports 24C and 24D pass the oil path 54 through a check valve 12 and are connected to bottom sides of the lift cylinders 13A and 13B. The port 24A is constituted as a port which detects a load pressure on the bottom sides of the lift cylinders 13A and 13B which move a fork (not shown) upward and downward. The port 24A supplies the load pressure on the bottom sides of the lift cylinders 13A and 13B to the shuttle valve 27 through the pilot oil path 74. The port 24B is connected to an oil pressure chamber of the pilot check valve 12 through the electromagnetic switching valve 15.

[0104] The pump port 24E is constituted as a port to which a discharge flow rate of the variable displacement pump 1 which passed an oil path 52 is supplied through the check valve 48. The tank ports 24F and 24G are constituted as ports through which flow rates discharged from the bottom sides of the lift cylinders 13A and 13B are discharged to the tank 50 through a drain oil path 63. A throttle 86 is disposed in the drain oil path 63, and a pressure upstream of the throttle 86 is applied to the second spool 8B as a pilot pressure. The second spool 8B is controlled independently from the first spool 8A in accordance with a differential pressure between the pressure upstream of the throttle 86 and the tank pressure.

[0105] The position VIII of the first direction switching valve 8 is a neutral position of the first direction switching valve 8, and when the first direction switching valve 8 is in the neutral position, the second spool 8B is switched to the position V which is a neutral position. If the first direction switching valve 8 is switched to the position IX by the operation of the operating lever 9, oil from the oil path 52 can pass through the check valve 12 from the port 24C and can be supplied to the bottom sides of the lift cylinders 13A and 13B through the oil path 54. At that time, the second spool 8B is switched to the position VI by a pressing force from the first direction switching valve 8.

[0106] With this, a fork (not shown) can be moved upward. At that time, since the second spool 8B is switched to the position VI by the operation of the first spool 8A, the port 24D is shut off. Oil on the head sides of the lift cylinders 13A and 13B is discharged into the tank 50 through the drain oil path 69.

[0107] If the first direction switching valve 8 is switched to the position VII by the operation of the operating lever 9, oil from the oil path 52 is shut off, and supply of oil to the bottom sides of the lift cylinders 13A and 13B is cut. At that time, if the electromagnetic switching valve 15 is controlled and the pilot oil path 75 is brought into communication, oil discharged from the bottom sides of the lift cylinders 13A and

13B can pass through the tank port 24F from the port 24C and can be discharged to the tank 50 through the drain oil path 63.

[0108] At that time, the second spool 8B follows the switching of the first direction switching valve 8 toward the position VII by the biasing force of the spring 10a and is switched to the position IV. At the same time, oil discharged from the bottom sides of the lift cylinders 13A and 13B pass through the tank port 24G from the port 24D and can be discharged to the tank 50 through the drain oil path 63. With this, the fork (not shown) can be moved downward.

[0109] At that time, the second spool 8B is switched to the position V in accordance with a differential pressure between the tank pressure and a pressure in the drain oil path 63 upstream of the throttle 86. That is, the flow rate of oil discharged from the port 24D is controlled in accordance with the differential pressure between the tank pressure and the pressure in the drain oil path 63 upstream of the throttle 86 applied to the second spool 8B.

[0110] The total discharge flow rate of returning oil discharged from the bottom sides of the lift cylinders 13A and 13B is a sum flow rate of the discharge flow rate by the second spool 8B and the discharge flow rate by the first spool 8A.

[0111] With this, the first spool 8A can be provided with flow rate control characteristics with respect to the port 24D, and downwardly moving speeds of the lift cylinders 13A and 13B can be controlled by the first direction switching valve 8.

[0112] FIG. 2 shows an example in which the first direction switching valve 8 has a flow rate control valve function for controlling the downwardly moving speeds of the lift cylinders 13A and 13B, but the flow rate control valve can be disposed in the oil path 54. By disposing the flow rate control valve in the oil path 54, it is possible to prevent the downwardly moving speed of the fork from increasing excessively when the fork is moved downward in a state that the load pressures of the lift cylinders 13A and 13B are high.

[0113] The check valve 12 connected to the bottom sides of the lift cylinders 13A and 13B through the oil path 54 is controlled by the electromagnetic switching valve 15. The electromagnetic switching valve 15 operates as a safety device, and when a driver sits on a driver's seat, a solenoid provided in the electromagnetic switching valve 15 operates and the electromagnetic switching valve 15 can be switched to the communication position. When the driver does not sit on the driver's seat, the solenoid does not operate, and the electromagnetic switching valve 15 is switched to the shut off position by a biasing force of a spring.

[0114] When the direction control valve 8 is in the switching position (IV), (VII) and the electromagnetic switching valve 15 is in the shut off position, even if an attempt is made to lower the lift cylinders 13A and 13B, since the pressure on the side of the lift cylinders 13A and 13B of the pilot check valve 12, i.e., the pressure in the pilot oil path 75 is not reduced, the pilot check valve 12 does not open.

[0115] Thus, the oil returning from the lift cylinders 13A and 13B is stopped in the pilot check valve 12.

[0116] When the direction control valve 8 is in the switching position (IV), (VII) and the electromagnetic switching valve 15 is in the communication position, the pressure in the pilot oil path 75 passes through the tank port 24F from the port 24B and becomes equal to a pressure communicated with the tank 50. With this, the pilot check valve 12 can be in the communication state. That is, oil returning from the bottom sides of the lift cylinders 13A and 13B is returned to the actuator ports 24C and 24D through the pilot check valve 12.

[0117] A lowering safety valve 14 is disposed between the lift cylinder 13A and the lift cylinder 13B. The lowering safety valve 14 has a function for preventing a pressure on the bottom side of the lift cylinder 13A from reducing abruptly even when the oil path 54 or the like is damaged. With this, it is possible to prevent the fork from moving downward abruptly by the damage of the oil path 54.

[0118] The second direction switching valve 17 can be switched between three position by the operation of an operating lever 18 and is constituted as a switching valve having five ports 25A to 25E. A spring 17a applied to the second direction switching valve 17 is constituted as a neutral spring which returns the second direction switching valve 17 to the neutral position XI.

[0119] The port 25A is connected to the head sides of the tilt cylinders 20A and 20B through an oil path 56, and the port 25C is connected to the bottom sides of the tilt cylinders 20A and 20B through an oil path 55.

[0120] The port 25B is constituted as a port for detecting the load pressures of the tilt cylinders 20A and 20B, and is connected to the shuttle valve 27 through a pilot oil path 76. The pump port 25D is constituted as a port to which the discharge flow rate of the variable displacement pump 1 passed through the oil path 52 is supplied through the check valve 49. The tank port 25E is constituted as a port through which oil discharged from the tilt cylinders 20A and 20B is discharged to the tank 50 through a drain oil path 64.

[0121] The second direction switching valve 17 is provided with a mechanism which prevents oil on the head sides of the tilt cylinders 20A and 20B from flowing into the tank even if the spool is operated and switched to the position XII when the engine is stopped, i.e., when oil does not flow to the oil path 52.

[0122] The supply of oil to the tilt cylinders 20A and 20B can be controlled by supplying oil supplied from the oil path 52 to the tilt cylinders 20A and 20B through the port 25A or 25C from the pump port 25D in the second direction switching valve 17. Oil discharged from the tilt cylinders 20A and 20B can be returned to the tank 50 from the drain oil path 64 through the oil path 55 or the oil path 56.

[0123] Oil discharged from the variable displacement pump 1 to the discharge oil path 51 passes through the oil path 57 branched from the discharge oil path 51 and is supplied to the hydraulic motor 35 which drives a cooling fan 36. The flow rate control valve 37 which controls the flow rate of oil supplied to the hydraulic motor 35 is disposed in the oil path 57. To prevent the pressure in the oil path 57 from exceeding a predetermined value, a relief valve 44 is disposed in a drain oil path 68 branched from the oil path 57.

[0124] An opening area of the flow rate control valve 37 is controlled by a differential pressure between an upstream pressure and a downstream pressure of the flow rate control valve 37 and by the spring force of a spring 37a. The spring force of the spring 37a is adjusted by the thermo-module 38 which is displaced in accordance with the coolant temperature cooled by the radiator (not shown).

[0125] When the coolant temperature is high, the thermo-module 38 operates such as to increase the spring force of the spring 37a, and when the coolant temperature is low, the thermo-module 38 operates such as to reduce the spring force of the spring 37a. Therefore, when the coolant temperature is high, the opening area of the flow rate control valve 37 is increased and the flow rate of oil to be supplied to the hydraulic motor 35 can be increased. With this, it is possible to rotate

the cooling fan 36 at high speed, the amount of air supplied to the radiator can be increased and the coolant temperature can be decreased.

[0126] When the coolant temperature is low, the thermo-module 38 operates such as to weaken the spring force of the spring 37a, the opening area of the flow rate control valve 37 is reduced and the flow rate of oil supplied to the hydraulic motor 35 is reduced. With this, the rotation of the cooling fan 36 is decelerated, the amount of air supplied to the radiator is reduced and the coolant temperature can be increased.

[0127] A normal/reverse rotation switching valve 40 which controls the rotation direction of the hydraulic motor 35 is disposed between the flow rate control valve 37 and the hydraulic motor 35. By switching the normal/reverse rotation switching valve 40, it is possible to select the oil path 58 or the oil path 59 connected to the hydraulic motor 35, and to supply oil from the oil path 57 to the selected oil path 58 or oil path 59. At that time, the oil discharged from the hydraulic motor 35 is discharged to the drain oil path 67 through the oil path 59 or the oil path 58.

[0128] The switching of the normal/reverse rotation switching valve 40 is controlled by the operation of a normal/reverse rotation solenoid valve 41. The normal/reverse rotation solenoid valve 41 selects a tank pressure in a drain oil path 81 connected to the hydraulic motor 35 and a pressure in the oil path 60 branched from the oil path 57, and applies one of the pressures to the normal/reverse rotation switching valve 40. The pressure applied to the normal/reverse rotation switching valve 40 is set to the tank pressure or the pressure in the oil path 60. With this, the normal/reverse rotation switching valve 40 is switched between a position where the hydraulic motor 35 is normally rotated and a position where the hydraulic motor 35 is reversely rotated.

[0129] A hydraulic pressure downstream of the flow rate control valve 37 is taken out by the pilot oil path 83 as a load pressure applied to the hydraulic motor 35. The pilot oil path 83 is connected to the load pressure separation valve 45. The load pressure separation valve 45 is constituted as a two-position and three-port valve. As the spring applied to the load pressure separation valve 45, it is possible to use a spring having strength of 0.5 MPa.

[0130] When a working machine is used as a spring applied to the load pressure separation valve 45, it is possible to use a spring having such a strength that the load pressure separation valve 45 is switched by the maximum load pressure of the working machine circuit immediately. By using the spring having such a strength, if the working machine is used, the load pressure separation valve 45 can immediately be switched against a biasing force of a spring applied to the load pressure separation valve 45.

[0131] With this, during operation of the working machine, the pump displacement of the variable displacement pump 1 is controlled by a load pressure of the operated working machine.

[0132] Therefore, it is possible to prevent a load pressure which controls the pump displacement of the variable displacement pump 1 during operation of the working machine from changing from the load pressure of the working machine to the load pressure of the hydraulic motor 35, and the operability of the working machine can be stabilized.

[0133] By switching the load pressure separation valve 45, the load pressure supplied from the pilot oil path 80 to the shuttle valve 29 can be set to the load pressure of the hydraulic motor 35 or the tank pressure. One of the pressures selected

by switching the load pressure separation valve 45 can be led to the shuttle valve 29 as load pressure selecting means through the pilot oil path 80.

[0134] To control the switching of the load pressure separation valve 45, the high-pressure-side load pressure selected by the shuttle valve 27 is taken out by the pilot oil path 77 and led to the load pressure separation valve 45 through the pilot oil path 79 branched from the pilot oil path 77. When the lift cylinders 13A and 13B or the tilt cylinders 20A and 20B are operated, the maximum load pressure of the working machine applied to the working machine circuit is applied to the load pressure separation valve 45 through the pilot oil path 79.

[0135] At that time, the load pressure separation valve 45 is switched against the biasing force of the spring, the pilot oil path 83 is connected to the tank 50, and the load pressure separation valve 45 is switched to a position where the load pressure in the pilot oil path 80 is set to the tank pressure. When the load pressure of the working machine is not generated in the working machine circuit, the load pressure separation valve 45 is switched to a position where the load pressure in the pilot oil path 83 is supplied to the shuttle valve 29.

[0136] Of the load pressure on the bottom sides of the lift cylinders 13A and 13B and the load pressure in the tilt cylinders 20A and 20B, high-pressure-side one of the load pressures is selected by the shuttle valve 27 and output to the pilot oil path 77. That is, the high-pressure-side load pressure selected by the shuttle valve 27 is applied as a load pressure of the working machine applied to the working machine circuit.

[0137] The high-pressure-side load pressure which is taken out by the pilot oil path 77 and selected by the shuttle valve 27 is led to the shuttle valve 29 through the pilot oil path 78 branched from the pilot oil path 77. A relief valve 32 is disposed in the pilot oil path 78 on the output side of the shuttle valve 27 so that the high-pressure-side load pressure selected by the shuttle valve 27 does not exceed a predetermined value. The relief valve 32 is connected to the tank 50 through a drain oil path 66.

[0138] That is, the high-pressure-side load pressure selected by the shuttle valve 27 becomes the maximum load pressure of the working machine circuit, and is supplied to the shuttle valve 29. In the shuttle valve 29, of the maximum load pressure of the working machine circuit and the load pressure selected by the load pressure separation valve 45, high-pressure-side one of the load pressures is output to the pilot oil path 85. The load pressure selected by the load pressure separation valve 45 is the load pressure of the hydraulic motor 35 or the tank pressure. The high-pressure-side load pressure selected by the shuttle valve 29 is led to the displacement control device 2 which controls the displacement of the variable displacement pump 1 through the pilot oil path 85.

[0139] The displacement control device 2 includes a switching valve 5 constituted as a three-position and three-port switching valve, and a drive cylinder 6 which controls a swash plate angle of a swash plate 1a of the variable displacement pump 1. The switching valve 5 is switched in accordance with a differential pressure between the high-pressure-side load pressure selected by the shuttle valve 29 and the pump pressure from the variable displacement pump 1. The drive cylinder 6 which controls the swash plate angle of the swash plate 1a of the variable displacement pump 1 can be operated by the switching operation of the switching valve 5.

[0140] That is, when the high-pressure-side load pressure selected by the shuttle valve 29, the spring pressure of the

spring applied to the switching valve 5 and the pump pressure from the variable displacement pump 1 are balanced, the switching valve 5 is in the neutral position shown in FIG. 2, and the drive cylinder 6 maintains the swash plate angle of the swash plate 1a in the current state.

[0141] If a differential pressure between the high-pressure-side load pressure selected by the shuttle valve 29 and the pump pressure from the variable displacement pump 1 is increased, the switching valve 5 is switched to the left position in FIG. 2, the pump displacement of the variable displacement pump 1 is increased and the discharge flow rate from the variable displacement pump 1 is increased. If the differential pressure between the high-pressure-side load pressure selected by the shuttle valve 29 and the pump pressure from the variable displacement pump 1 becomes small, the switching valve 5 is switched to the right position in FIG. 2, the discharge displacement of the variable displacement pump 1 is reduced, and the discharge flow rate from the variable displacement pump 1 is reduced.

[0142] With this, the displacement control device 2 is operated in accordance with a differential pressure between the high-pressure-side load pressure selected by the shuttle valve 29 and the pump pressure in the oil path 51, and the discharge flow rate of the variable displacement pump 1 is controlled in accordance with the differential pressure.

[0143] That is, the lift cylinders 13A and 13B or the tilt cylinders 20A and 20B are operated, and when the load pressure is generated in the lift cylinders 13A and 13B or the tilt cylinders 20A and 20B, the load pressure separation valve 45 is switched to the shut off position by the load pressure. With this, the pilot oil path 80 is brought into communication with the tank 50, and the pressure applied to the shuttle valve 29 through the pilot oil path 80 becomes the tank pressure.

[0144] When the load pressure of the working machine is generated in the working machine circuit, the load pressure of the hydraulic motor 35 which drives the cooling fan 36 can be handled as a tank pressure state forcibly. Therefore, the maximum load pressure in the working machine circuit and the load pressure in the hydraulic motor 35 handled as the tank pressure can be compared with each other in the shuttle valve 29.

[0145] With this, the discharge amount discharged from the variable displacement pump 1 is controlled based on the maximum load pressure of the working machine circuit. In other words, when the load pressure is generated in the lift cylinders 13A and 13B or the tilt cylinders 20A and 20B, the discharge flow rate from the variable displacement pump 1 is not controlled by the load pressure of the hydraulic motor 35.

[0146] Even if the load pressure in the hydraulic motor 35 is higher than the load pressure of the lift cylinders 13A and 13B or the load pressure of the tilt cylinders 20A and 20B, the pump displacement of the variable displacement pump 1 is not controlled by the load pressure of the hydraulic motor 35 but is controlled based on the maximum load pressure of the working machine circuit when the load pressure is generated in the working machine circuit. Therefore, when the load pressure is generated in the working machine circuit, the working machine can be operated stably.

[0147] When the load pressure of the working machine is generated in the working machine circuit, even if the fork is vertically moved in a no load state during running, the pump displacement of the variable displacement pump 1 is con-

trolled such that it becomes equal to the pump displacement in accordance with the maximum load pressure in the working machine circuit.

[0148] In this case, even if the load pressure of the hydraulic motor 35 which drives the cooling fan 36 is higher than the maximum load pressure in the working machine circuit, the swash plate angle of the variable displacement pump 1 is not controlled in accordance with the load pressure of the hydraulic motor 35 which drives the cooling fan 36, but is controlled in accordance with the maximum load pressure in the working machine circuit. With this, a flow rate suitable for the maximum load pressure in the working machine circuit is discharged from the variable displacement pump 1, and the working machine can be operated stably.

[0149] When the fork is vertically moved with no load during running, a flow rate required for rotating the cooling fan 36 and obtaining a sufficient wind amount is not supplied to the hydraulic motor 35 which drives the cooling fan 36. However, since the operation time during which the fork is vertically moved with no load is short, even if the wind amount supplied to the radiator is temporarily reduced, it is possible to suppress the temperature rise in the radiator to a low level.

[0150] Further, after the operating time during which the fork is vertically moved with no load during running is completed, the pump displacement of the variable displacement pump 1 is controlled in accordance with the load pressure of the hydraulic motor 35 which drives the cooling fan 36.

Second Embodiment

[0151] A hydraulic circuit having a fan drive system according to the Second embodiment of the present invention will be explained using FIGS. 3 and 4. The Second embodiment has a steering drive circuit in addition to the hydraulic circuit of the First embodiment. Maximum one of load pressures of a steering drive device 30, the lift cylinders 13A and 13B and the tilt cylinders 20A and 20B is used as a load pressure in the pilot oil path 78 as a first pilot oil path which is led to the shuttle valve 29. In this structure, the Second embodiment is different from the First embodiment, but other structure is the same as that of the First embodiment.

[0152] Therefore, members of the Second embodiment which are the same as those of the First embodiment are designated with the same reference numerals, and explanation thereof will be omitted, and a structure which is different from that of the First embodiment will be explained mainly. The pilot oil path 78 connected to the shuttle valve 29 is the pilot oil path connected to the pilot oil path 77 in the First embodiment, but the pilot oil path 78 of the Second embodiment is constituted as a pilot oil path which takes out a high-pressure-side load pressure selected by a shuttle valve 28.

[0153] FIG. 3 shows a simplified hydraulic circuit diagram of the Second embodiment like FIG. 1, and FIG. 4 is a detailed hydraulic circuit diagram of the Second embodiment like FIG. 2.

[0154] As shown in FIG. 3, oil discharged from a load pressure sensitive variable displacement pump 1 which is driven by an engine (not shown) passes through a discharge oil path 51 as a third discharge oil path, and is supplied to a load pressure sensitive priority valve 3. Oil which is output from the priority valve 3 is used as working oil which operates a working machine and steering.

[0155] That is, the oil output from the priority valve 3 is supplied to a steering circuit 34 through an oil path 53 as a fourth discharge oil path, and is supplied to the working machine circuit 33 through the oil path 52 as a fifth discharge oil path.

[0156] Further, oil flowing through the oil path 57 as a sixth discharge oil path branched from the discharge oil path 51 upstream of the priority valve 3 passes through the oil path 58 as the supply oil path through the flow rate control valve 37, and is used as working oil which drives the hydraulic motor 35.

[0157] A discharge flow rate from the variable displacement pump 1 is controlled by the displacement control device 2, and the operation of the displacement control device 2 can be controlled in accordance with a differential pressure between the pump pressure and the highest load pressure among the maximum load pressure in the working machine circuit 33, the maximum load pressure in the steering circuit 34 and the load pressure of the hydraulic motor 35.

[0158] The maximum load pressure in the working machine circuit 33 is taken out by the pilot oil path 77 as a first pilot oil path, and the load pressure in the steering circuit 34 is taken out by the pilot oil path 71 as a third pilot oil path. The pilot oil path 77 and the pilot oil path 71 are connected to the shuttle valve 28 which selects the high-pressure-side load pressure. The high-pressure-side load pressure among the maximum load pressure in the working machine circuit 33 and the load pressure in the steering circuit 34 is selected by the shuttle valve 28 as the first shuttle valve, and is taken out by the pilot oil path 78.

[0159] The high-pressure-side load pressure selected by the shuttle valve 28 taken out in the pilot oil path 78 is led to the shuttle valve 29 as a second shuttle valve. The maximum load pressure of the working machine circuit 33 taken out by the pilot oil path 77 is led to the load pressure separation valve 45 through the pilot oil path 79.

[0160] The load pressure separation valve 45 is controlled in accordance with a differential pressure between a load pressure in the pilot oil path 79 and a spring force of a spring applied to the load pressure separation valve 45. That is, when a load pressure is generated in the working machine circuit 33, the load pressure separation valve 45 sets the load pressure in the pilot oil path 80 to the tank pressure, and when the load pressure is not generated in the working machine circuit 33, the load pressure separation valve 45 sets the load pressure in the pilot oil path 80 to the load pressure of the hydraulic motor 35.

[0161] With this structure, the discharge displacement of the variable displacement pump 1 is operated in accordance with a differential pressure between the load pressure in the pilot oil path 85 selected by the shuttle valve 29 and the pump pressure in the discharge oil path 51. Further, when the load pressure is generated in the working machine circuit 33, the discharge displacement of the variable displacement pump 1 is controlled in accordance with a differential pressure between the pump pressure and the high-pressure-side load pressure among the maximum load pressure of the working machine circuit 33 and the load pressure of the steering circuit 34.

[0162] Next, the hydraulic circuit having the fan drive system will be explained in detail using FIG. 4. In FIG. 4 and FIGS. 1 to 3, the same constituent members are designated with the same reference numerals. As shown in FIG. 4, a discharge flow rate from the load pressure sensitive variable

displacement pump 1 driven by the engine M is supplied to the load pressure sensitive priority valve 3 through the oil path 51. The priority valve 3 is constituted as a three-position and three-port switching valve.

[0163] A pump port 23C of the priority valve 3 is connected to the variable displacement pump 1 through the discharge oil path 51. A port 23A passes through the oil path 52 and is connected to a port 24E of the first direction switching valve 8 through the check valve 48, and is connected to a port 25D of the second direction switching valve 17 through the check valve 49. The port 23B is connected to the steering drive device 30 through the oil path 53.

[0164] The steering drive device 30 can operate a steering operating actuator 31. A discharge flow rate from the steering drive device 30 can be discharged into the tank 50 through the drain oil path 65.

[0165] A position of the priority valve 3 is switched in accordance with a differential pressure between a hydraulic pressure of the oil path 53 which supplies oil to the steering drive device 30 and a load pressure in the actuator 31 taken out from the pilot oil path 71 through the electromagnetic switching control valve 4.

[0166] The priority valve 3 can be switched between three positions, from a position I to a position III. In the position III, supply of the discharge flow rate from the variable displacement pump 1 to the oil path 52 which is a supply oil path to the lift cylinders 13A and 13B and the tilt cylinders 20A and 20B is stopped, and a discharge flow rate from the variable displacement pump 1 can be supplied to the oil path 53 which is a supply oil path to the precedence steering drive device 30.

[0167] In the position II, the discharge flow rate from the variable displacement pump 1 can be supplied to the oil path 52 and the oil path 53. In the position I, the discharge flow rate from the variable displacement pump 1 can be supplied to the oil path 52, and the discharge flow rate from the variable displacement pump 1 can be supplied to the oil path 53 through a throttle.

[0168] An output pressure which is output from the priority valve 3 to the oil path 53 can be taken out by the pilot oil path 72. The oil path 53 is in communication with the pilot oil path 72 through an oil path 62 and a throttle disposed in the oil path 62.

[0169] A portion of the pilot oil path 72 merges with the pilot oil path 73 branched from the pilot oil path 71, and is connected to one end of the priority valve 3 through a throttle. Oil from the pilot oil path 72 and the pilot oil path 73 becomes a first detection pressure and is applied to the priority valve 3 together with a biasing force of a spring 3a. The first detection pressure and the biasing force of the spring 3a become a first operation pressure and can switch the priority valve 3 to the position III. The other of the pilot oil path 72 is led to the other end of the priority valve 3 where the spring 3a is not disposed, and functions as a second detection pressure which switches the priority valve 3 to the position I.

[0170] The electromagnetic switching control valve 4 is disposed in the pilot oil path 73, and if a solenoid 4a of the electromagnetic switching control valve 4 is energized or de-energized, the electromagnetic switching control valve 4 can be switched between an open valve state and a close valve state. In FIG. 4, the solenoid 4a is brought into the de-energize state and the electromagnetic switching control valve 4 closes the pilot oil path 73.

[0171] When the electromagnetic switching control valve 4 is opened, a pressure in the pilot oil path 73 becomes equal to the load pressure in the pilot oil path 71.

[0172] A pressure in the pilot oil path 71 and the biasing force of the spring 3a apply as a first operation pressure which switches the priority valve 3 to the position III. If a differential pressure between the second operation pressure and the first operation pressure exceeds a preset differential pressure for driving the steering operation actuator 31, the priority valve 3 switched from the position III to the position II or the position I by the second operation pressure in accordance with the differential pressure.

[0173] With this, a flow rate required for driving the actuator 31 of the steering drive device 30 can always be output to the oil path 53. Of the discharge flow rate from the variable displacement pump 1, an excess of the flow rate required for driving the actuator 31 can be supplied from the oil path 52 to the lift cylinders 13A and 13B and/or the tilt cylinders 20A and 20B.

[0174] If the electromagnetic switching control valve 4 is switched from this state to the close valve state, i.e., the shut off state of the pilot oil path 73 and the pilot oil path 71, the first detection pressure and the second detection pressure becomes equal to the pilot pressure in the oil path 53, and they becomes equal to each other.

[0175] With this, the priority valve 3 is switched to the position III by the biasing force of the spring 3a, and the switched position III state is maintained. Therefore, the priority valve 3 stops supply of oil to the lift cylinders 13A and 13B and/or the tilt cylinders 20A and 20B. That is, the priority valve 3 supplies oil only to the steering drive device 30 having a higher priority, and a state where the priority valve 3 is in communication only with the oil path 53 is maintained.

[0176] The switching of the electromagnetic switching control valve 4 can be controlled by a sitting-state confirming switch disposed on a driver's seat. That is, when the sitting-state confirming switch detects that a driver sits on the driver's seat, the solenoid 4a of the electromagnetic switching control valve 4 is energized, and the electromagnetic switching control valve 4 maintains communication state.

[0177] With this, as a pressure led to the priority valve 3 by the pilot oil path 73, it is possible to utilize a load pressure of the steering drive device 30 in the pilot oil path 71, and the priority valve 3 is controlled in accordance with a differential pressure between the load pressure of the steering drive device 30 and the pump pressure supplied to the steering drive device 30.

[0178] When the sitting-state confirming switch detects that a driver moves away from the driver's seat, the solenoid 4a of the electromagnetic switching control valve 4 is de-energized as shown in FIG. 4, and the pilot oil path 71 and the pilot oil path 73 are shut off. With this, pressures led to the priority valve 3 through the pilot oil path 72 are substantially equal to each other.

[0179] Therefore, the priority valve 3 is switched to the position III by the biasing force of the spring 3a and this position III state is maintained. At that time, supply of oil to the lift cylinders 13A and 13B and/or the tilt cylinders 20A and 20B is stopped. That is, in a state that a driver moves away from a driver's seat, it is possible to establish a state where the working machine can not be operated.

[0180] A switching control valve which brings the pilot oil path 71 and the pilot oil path 73 into the communication state and shut off state is not limited to the electromagnetic switch-

ing control valve, and other switching control valve can be used. The switching control valve may be brought into the communication state by an ON signal, or the switching control valve may be brought into the communication state by an OFF signal.

[0181] A control signal which controls the electromagnetic switching control valve 4 is not limited to a signal by the sitting-state confirming switch disposed on the driver's seat, and it is possible to use a detection signal by other detection switch as the control signal for the electromagnetic switching control valve, and to control of the electromagnetic switching control valve 4 using other control signal.

[0182] The load pressure of the lift cylinders 13A and 13B detected by the pilot oil path 71 and the load pressure of the tilt cylinders 20A and 20B detected by the pilot oil path 76 are led to the shuttle valve 27. The high-pressure-side load pressure selected by the shuttle valve 27 is led to the shuttle valve 28 through the pilot oil path 77. The load pressure in the pilot oil path 77 is led to the load pressure separation valve 45 as the maximum load pressure in the working machine circuit 33 (see FIG. 1).

[0183] The load pressure of the steering drive device 30 detected by the pilot oil path 71 is led to the shuttle valve 28. A high-pressure-side load pressure among the maximum load pressure in the working machine circuit and the load pressure in the steering drive device 30 is selected by the shuttle valve 28, and this is output to the pilot oil path 78. The pilot oil path 78 is connected to the shuttle valve 29. The shuttle valve 29 outputs a higher one of the high-pressure-side load pressure selected by the shuttle valve 28 and the load pressure in the pilot oil path 80 to the pilot oil path 85.

[0184] As the load pressure in the pilot oil path 80, the load pressure separation valve 45 is controlled. With this, the load pressure in the hydraulic motor 35 or the tank pressure is selected. The pump displacement of the variable displacement pump 1 is controlled in accordance with a differential pressure between the pump pressure and the high-pressure-side load pressure selected by the shuttle valve 29.

[0185] With this, when the load pressure is generated in the working machine circuit, the pump displacement of the variable displacement pump 1 is controlled in accordance with a differential pressure between the pump pressure and the high-pressure-side load pressure among the maximum load pressure in the working machine circuit and the load pressure in the steering drive device 30. When the load pressure is not generated in the working machine circuit, the pump displacement of the variable displacement pump 1 is controlled in accordance with a differential pressure between the pump pressure and the high-pressure-side load pressure selected by the shuttle valve 29 among the load pressure in the steering drive device 30 and the load pressure in the hydraulic motor 35.

Third Embodiment

[0186] FIGS. 5 and 6 show a hydraulic circuit having a fan drive system according to the Third embodiment of the present invention. A circuit structure for leading a load pressure to the pilot oil path 80 in the Third embodiment is different from those of the First and Second embodiments. Other structure is similar to the circuit structure shown in FIG. 2 that is a circuit structure of the First embodiment in FIG. 5, and other structure is similar to the circuit structure shown in FIG. 4 that is a circuit structure of the Second embodiment in FIG. 6.

[0187] Concerning FIG. 5 where the priority valve 3 is not disposed, the same structure as that of the First embodiment and the same members of the Third embodiment which are the same as those of the First embodiment are designated with the same reference numerals, and explanation thereof will be omitted. Concerning FIG. 6 where the priority valve 3 is used, the same structure as that of the Second embodiment and the same members of the Third embodiment which are the same as those of the First embodiment are designated with the same reference numerals, and explanation thereof will be omitted.

[0188] As shown in FIGS. 5 and 6, a structure of a load pressure separation valve 46 of the Third embodiment is different from that of the load pressure separation valve 45 in the embodiments 1 and 2. That is, as shown in FIGS. 5 and 6, the load pressure separation valve 46 of the Third embodiment is constituted as a two-position and two-port switching valve. A throttle 87 is disposed in the pilot oil path 83 connected downstream of the flow rate control valve 37.

[0189] The pilot oil path 83 is branched into two downstream of the throttle 87, and one of the branched path, i.e., a pilot oil path 80 is connected to the shuttle valve 29. The other branched pilot oil path is connected to the tank 50 through the load pressure separation valve 46 disposed at an intermediate portion.

[0190] When a load pressure is generated in the lift cylinders 13A and 13B or the tilt cylinders 20A and 20B, the load pressure in the working machine circuit is applied to the load pressure separation valve 46 through the pilot oil path 79. With this, the load pressure separation valve 46 is switched to a switching position where the pilot oil path 83 is brought into communication with the tank 50. At that time, a pressure in the pilot oil path 80 supplied to the shuttle valve 29 becomes a tank pressure, and the tank pressure is supplied as the load pressure supplied to the shuttle valve 29.

[0191] That is, the tank pressure is supplied as a load pressure supplied from the pilot oil path 80 to the shuttle valve 29. Since the throttle 87 is provided upstream of the pilot oil path 80, the load pressure of the hydraulic motor 35 is held by the throttle 87.

[0192] Therefore, at that time, in the hydraulic circuit shown in FIG. 5, a high-pressure-side load pressure selected by the shuttle valve 27 is output to the pilot oil path 85 from the shuttle valve 29. In the hydraulic circuit shown in FIG. 6, a high-pressure-side load pressure selected by the shuttle valve 28 is output to the pilot oil path 85 from the shuttle valve 29.

[0193] That is, a high-pressure-side load pressure which is output from the shuttle valve 29 to the pilot oil path 85 is the maximum load pressure of the working machine circuit in the case of the hydraulic circuit shown in FIG. 5, and is a high-pressure-side load pressure among the maximum load pressure of the working machine circuit and the load pressure in the steering drive device 30 in the case of the hydraulic circuit shown in FIG. 6. The pump displacement of the variable displacement pump 1 is controlled in accordance with a differential pressure between the high-pressure-side load pressure which is output to the pilot oil path 85 and the pump pressure of the variable displacement pump 1.

[0194] When a load pressure of the working machine in the lift cylinders 13A and 13B or the tilt cylinders 20A and 20B is not generated, the load pressure separation valve 46 is switched to the switching position where the communication between the pilot oil path 83 and the tank 50 is cut. At that time, in the hydraulic circuit shown in FIG. 5, the load pres-

sure of the hydraulic motor 35 is output from the shuttle valve 29 to the pilot oil path 85. In the hydraulic circuit shown in FIG. 6, a high-pressure-side load pressure among the high-pressure-side load pressure selected by the shuttle valve 28 and a load pressure in the hydraulic motor 35 is output to the pilot oil path 85. The displacement control device 2 is controlled in accordance with a differential pressure between the high-pressure-side load pressure which is output to the pilot oil path 85 and the pump pressure of the variable displacement pump 1.

Fourth Embodiment

[0195] FIG. 7 shows a hydraulic circuit having a fan drive system according to the Fourth embodiment of the present invention. In the Fourth embodiment, the same structure as that of the Second embodiment are designated with the same reference numerals, and explanation thereof will be omitted. In the Fourth embodiment, a control structure for controlling a flow rate of oil supplied to the hydraulic motor 35 is different from the control structure for controlling a flow rate of oil supplied to the hydraulic motor 35 in the Second embodiment.

[0196] Other structure is the same as that of the Second embodiment. Thus, in the Fourth embodiment, concerning the same structure as that of the Second embodiment, the same members are designated with the same reference numerals and explanation thereof will be omitted.

[0197] As a structure of the load pressure separation valve 45 in the Fourth embodiment, the structure of the load pressure separation valve 45 explained in the Second embodiment is used, but the structure using the throttle 87 and the load pressure separation valve 46 explained in the Third embodiment can also be used as the structure of the load pressure separation valve 45 in the Fourth embodiment. A control structure for controlling the flow rate of oil supplied to the hydraulic motor 35 in the Fourth embodiment can be applied as a control structure for controlling the flow rate of oil supplied to the hydraulic motor 35 shown in FIGS. 2 and 5.

[0198] An oil path 61 which is branched off from the oil path 57 upstream of the flow rate control valve 37 is connected to a decompression valve 43. The decompression valve 43 is controlled by a thermo-module 38. When a coolant temperature is high, the thermo-module 38 increases a spring force of a spring 43a, and when the coolant temperature is low, the thermo-module 38 reduces the spring force of the spring 43a.

[0199] Therefore, when the coolant temperature is high, high pressure oil is applied to the flow rate control valve 37 from the decompression valve 43. With this, the flow rate control valve 37 is controlled such that its opening area is increased, and the flow rate of oil supplied to the hydraulic motor 35 can be increased. Therefore, the cooling fan 36 can be rotated at high speed, the amount of wind supplied to the radiator can be increased and the coolant temperature can be decreased.

[0200] When the coolant temperature is low, the thermo-module 38 decreases the spring force of the spring 43a, and decompressed oil is applied to the flow rate control valve 37 from the decompression valve 43. The flow rate control valve 37 is controlled such that its opening area is reduced, and the flow rate of oil supplied to the hydraulic motor 35 is reduced. With this, the rotation of the cooling fan 36 is decelerated, the

amount of wind supplied to the radiator is reduced and the coolant temperature can be increased.

Fifth Embodiment

[0201] FIG. 8 shows a hydraulic circuit having a fan drive system according to the Fifth embodiment of the present invention. In the Fifth embodiment, the same members as those of the Second embodiment are designated with the same reference numerals and explanation thereof will be omitted. In the Fifth embodiment, as a control structure for controlling a flow rate of oil supplied to the hydraulic motor 35, a variable throttle valve 39 is used. A pressure compensation valve 42 is disposed downstream of the variable throttle valve 39, and a pilot oil path 84 as a second pilot oil path which detects a hydraulic pump downstream of the pressure compensation valve 42 is connected to the load pressure separation valve 45. These structures of the Fifth embodiment are different from those of the Second embodiment.

[0202] Other structure is the same as that of the First embodiment. Therefore, in the Fourth embodiment, the same members as those of the Second embodiment are designated with the same reference numerals and explanation thereof will be omitted.

[0203] As a structure of a load pressure separation valve of the Fifth embodiment, an example in which the structure of the load pressure separation valve 45 explained in the Second embodiment is indicated, but the structure using the throttle 87 and the load pressure separation valve 46 explained in the Third embodiment can be used instead of the load pressure separation valve 45 in the Fifth embodiment. A control structure for controlling a flow rate of oil supplied to the hydraulic motor 35 in the Fifth embodiment can be applied instead of the control structure for controlling the flow rate of oil supplied to the hydraulic motor 35 shown in FIGS. 2 and 5.

[0204] The variable throttle valve 39 is controlled by the thermo-module 38. When the coolant temperature is high, the thermo-module 38 controls the variable throttle valve 39 to the entirely communication state against a spring force of a spring 39a. When the coolant temperature is low, the variable throttle amount of the variable throttle valve 39 is controlled in accordance with the coolant temperature detected by the thermo-module 38.

[0205] Therefore, when the coolant temperature is high, a large amount of oil of the oil path 57 can be supplied to the pressure compensation valve 42 from the variable throttle valve 39. When the coolant temperature is low, the oil of the oil path 57 can be supplied to the pressure compensation valve 42 from the variable throttle valve 39 in the throttled state.

[0206] The pressure compensation valve 42 is switched and controlled in accordance with a differential pressure between a hydraulic pump downstream of the variable throttle valve 39 taken out by a pilot oil path 82 and the highest one of a pressure in the pilot oil path 78, i.e., load pressures in the lift cylinders 13A and 13B, the load pressures in the tilt cylinders 20A and 20B and a load pressure in the steering drive device 30.

[0207] When a pressure in the pilot oil path 78 is higher than a pressure in the oil path 57, the pressure compensation valve 42 stops the supply of oil in the oil path 57 to the hydraulic motor 35. That is, when a pressure in the pilot oil path 78 is higher than a pump pressure in the variable displacement pump 1, all of discharge flow rates from the variable displacement pump 1 can be supplied to the lift cylinders

13A and 13B and the tilt cylinders 20A and 20B which are working machines, and the operation of the hydraulic motor 35 is temporarily stopped. With this, a large amount of oil can be used in the working machine.

[0208] When the pressure in the pilot oil path 78 becomes substantially equal to the pressure in the oil path 57 or when the pressure in the oil path 57 becomes higher than the pressure in the pilot oil path 78, the pressure compensation valve 42 can supply oil of the oil path 57 to the hydraulic motor 35.

INDUSTRIAL APPLICABILITY

[0209] The technical idea of the present invention can be applied to a hydraulic pressure system which needs control a load sensing type variable displacement pump while taking a load pressure in a hydraulic motor which drives a cooling fan into account.

1. A fan drive system comprising:

- a load pressure sensitive variable displacement pump;
 - a work machine circuit and a flow rate control valve to which a discharge flow rate from the variable displacement pump is supplied;
 - a load pressure separation valve which is controlled by a maximum load pressure in the working machine circuit;
 - a hydraulic motor which drives a cooling fan;
 - a first discharge oil path which connects the variable displacement pump and the working machine circuit with each other;
 - a second discharge oil path which is branched from the first discharge oil path and which is connected to the flow rate control valve with each other;
 - a supply oil path which connects the flow rate control valve and the hydraulic motor with each other;
 - a first pilot oil path which takes out the maximum load pressure in the working machine circuit;
 - a second pilot oil path which takes out a load pressure that drives the hydraulic motor; and
 - a shuttle valve which selects a high-pressure-side load pressure between the maximum load pressure in the first pilot oil path and the load pressure in the second pilot oil path,
- wherein a pump displacement of the variable displacement pump is controlled in accordance with a differential pressure between the high-pressure-side load pressure selected by the shuttle valve and a pump pressure of the variable displacement pump,
- wherein, the load pressure separation valve is disposed in the second pilot oil path,
- wherein the load pressure separation valve is controlled in accordance with a differential pressure between a pressing force by the maximum load pressure taken out by the first pilot oil path and a biasing force of a spring applied to the load pressure separation valve,
- wherein, when the pressing force by the maximum load pressure is greater than the biasing force of the spring, the load pressure separation valve is switched from a position where the second pilot oil path is brought into communication with the shuttle valve to a position where the second pilot oil path is brought into communication with a tank, and a tank pressure is led to the shuttle valve, and
- wherein, when the pressing force by the maximum load pressure is smaller than the biasing force of the spring, the load pressure separation valve is switched from the position where the second pilot oil path is brought into

communication with the tank to the position where the second pilot oil path is brought into communication with the shuttle valve, and the load pressure which drives the hydraulic motor is led to the shuttle valve.

2. A fan drive system comprising:

- a load pressure sensitive variable displacement pump;
 - a working machine circuit and a flow rate control valve to which a discharge flow rate from the variable displacement pump is supplied;
 - a load pressure separation valve which is controlled by a maximum load pressure in the working machine circuit;
 - a hydraulic motor which drives a cooling fan;
 - a first discharge oil path which connects the variable displacement pump and the working machine circuit with each other;
 - a second discharge oil path which is branched from the first discharge oil path and which is connected to the flow rate control valve with each other;
 - a supply oil path which connects the flow rate control valve and the hydraulic motor with each other;
 - a first pilot oil path which takes out the maximum load pressure in the working machine circuit;
 - a second pilot oil path which takes out a load pressure that drives the hydraulic motor; and
 - a throttle disposed in the second pilot oil path; and
 - a shuttle valve which selects a high-pressure-side load pressure between the maximum load pressure in the first pilot oil path and the load pressure in the second pilot oil path,
- wherein a pump displacement of the variable displacement pump is controlled in accordance with a differential pressure between the high-pressure-side load pressure selected by the shuttle valve and a pump pressure of the variable displacement pump,
- wherein, the second pilot oil path is branched into two oil paths downstream of the throttle, one of the branched oil paths is connected to the shuttle valve,
- wherein, the load pressure separation valve is disposed at an intermediate portion of the other oil path, and the other oil path is connected to a tank,
- wherein, the load pressure separation valve is controlled in accordance with a differential pressure between a pressing force by the maximum load pressure taken out from the first pilot oil path and a biasing force of a spring applied to the load pressure separation valve,
- wherein, when the pressing force by the maximum load pressure is greater than the biasing force of the spring, the load pressure separation valve is switched to a position where the other oil path is brought into communication with the tank, and a tank pressure is led to the shuttle valve, and
- wherein, when the pressing force by the maximum load pressure is smaller than the biasing force of the spring, the load pressure separation valve is switched from the position where the other oil path is brought into communication with the tank to a position where a communication is shut off, and the load pressure which drives the hydraulic motor is led to the shuttle valve.

3. A fan drive system comprising:

- a load pressure sensitive variable displacement pump;
- a steering circuit, a working machine circuit and a flow rate control valve to which a discharge flow rate from the variable displacement pump is supplied;

a priority valve which supplies the discharge flow rate from the variable displacement pump preferentially to the steering circuit using the steering circuit as a priority circuit with respect to the working machine circuit;

- a load pressure separation valve which is controlled by a maximum load pressure in the working machine circuit;
- a hydraulic motor which drives a cooling fan;
- a third discharge oil path which connects the variable displacement pump and the priority valve with each other;
- a fourth discharge oil path which connects the priority valve and the steering circuit with each other;
- a fifth discharge oil path which connects the priority valve and the working machine circuit with each other;
- a sixth discharge oil path which is branched from the third discharge oil path and which is connected to the flow rate control valve;
- a supply oil path which connects the flow rate control valve and the hydraulic motor with each other;
- a first pilot oil path which takes out the maximum load pressure in the working machine circuit;
- a second pilot oil path which takes out a load pressure for driving the hydraulic motor;
- a third pilot oil path which takes out a load pressure in the steering circuit;

a first shuttle valve which selects a high-pressure-side load pressure between the maximum load pressure in the first pilot oil path and the load pressure in the third pilot oil path; and

a second shuttle valve which selects a high-pressure-side load pressure between the high-pressure-side load pressure selected by the first shuttle valve and the load pressure in the second pilot oil path,

wherein a pump displacement of the variable displacement pump is controlled in accordance with a differential pressure between the high-pressure-side load pressure selected by the second shuttle valve and a pump pressure of the variable displacement pump,

wherein, the load pressure separation valve is disposed in the second pilot oil path,

wherein, the load pressure separation valve is controlled in accordance with a differential pressure between a pressing force by the maximum load pressure taken out from the first pilot oil path and a biasing force of a spring applied to the load pressure separation valve,

wherein, when the pressing force by the maximum load pressure is greater than the biasing force of the spring, the load pressure separation valve is switched from a position where the second pilot oil path is brought into communication with the second shuttle valve to a position where the second pilot oil path is brought into communication with a tank, and a tank pressure is led to the second shuttle valve, and

wherein, when the pressing force by the maximum load pressure is smaller than the biasing force of the spring, the load pressure separation valve is switched from the position where the second pilot oil path is brought into communication with the tank to the position where the second pilot oil path is brought into communication with the second shuttle valve, and the load pressure for driving the hydraulic motor is led to the second shuttle valve.

4. A fan drive system comprising:

- a load pressure sensitive variable displacement pump;
- a steering circuit, a working machine circuit and a flow

rate control valve to which a discharge flow rate from the variable displacement pump is supplied;

a priority valve which supplies the discharge flow rate from the variable displacement pump preferentially to the steering circuit using the steering circuit as a priority circuit with respect to the working machine circuit;

a load pressure separation valve which is controlled by a maximum load pressure in the working machine circuit;

a hydraulic motor which drives a cooling fan;

a third discharge oil path which connects the variable displacement pump and the priority valve with each other;

a fourth discharge oil path which connects the priority valve and the steering circuit with each other;

a fifth discharge oil path which connects the priority valve and the working machine circuit with each other;

a sixth discharge oil path which is branched from the third discharge oil path and which is connected to the flow rate control valve;

a supply oil path which connects the flow rate control valve and the hydraulic motor with each other;

a first pilot oil path which takes out the maximum load pressure in the working machine circuit;

a second pilot oil path which takes out a load pressure for driving the hydraulic motor;

a third pilot oil path which takes out a load pressure in the steering circuit;

a throttle disposed in the second pilot oil path;

a first shuttle valve which selects a high-pressure-side load pressure between the maximum load pressure in the first pilot oil path and the load pressure in the third pilot oil path; and

a second shuttle valve which selects a high-pressure-side load pressure between the high-pressure-side load pressure selected by the first shuttle valve and the load pressure in the second pilot oil path,

wherein a pump displacement of the variable displacement pump is controlled in accordance with a differential pressure between the high-pressure-side load pressure

selected by the second shuttle valve and a pump pressure of the variable displacement pump,

wherein, the second pilot oil path is branched into two oil paths downstream of the throttle, one of the branched oil paths is connected to the second shuttle valve,

wherein, the load pressure separation valve is disposed in an intermediate portion of the other oil path, and the other oil path is connected to a tank,

wherein, the load pressure separation valve is controlled in accordance with a differential pressure between a pressing force by the maximum load pressure taken out from the first pilot oil path and a biasing force of a spring applied to the load pressure separation valve,

wherein, when the pressing force by the maximum load pressure is greater than the biasing force of the spring, the load pressure separation valve is switched to a position where the other oil path is brought into communication with the tank, and a tank pressure is led to the second shuttle valve, and

wherein, when the pressing force by the maximum load pressure is smaller than the biasing force of the spring, the load pressure separation valve is switched from the position where the other oil path is brought into communication with the tank to a position where a communication is shut off, and the load pressure for driving the hydraulic motor is led to the second shuttle valve.

5. The fan drive system according to claim 1, wherein the flow rate control valve is controlled in accordance with a temperature of a coolant.

6. The fan drive system according to claim 2, wherein the flow rate control valve is controlled in accordance with a temperature of a coolant.

7. The fan drive system according to claim 3, wherein the flow rate control valve is controlled in accordance with a temperature of a coolant.

8. The fan drive system according to claim 4, wherein the flow rate control valve is controlled in accordance with a temperature of a coolant.

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