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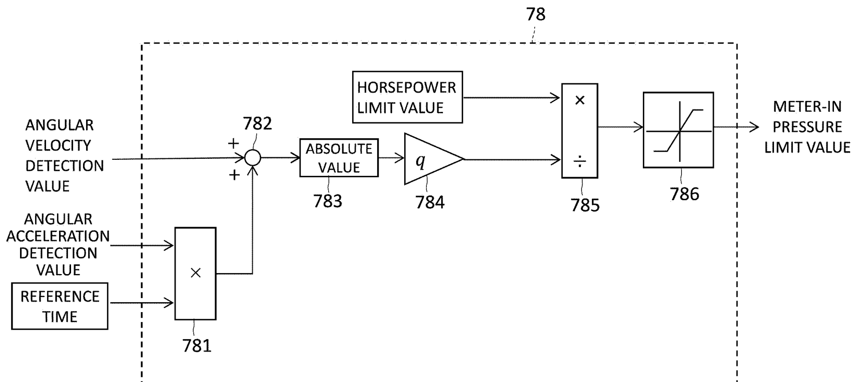
(54) **WORK MACHINE**

(57) The work machine includes a pressure adjustment device capable of adjusting the driving pressure of the swing hydraulic motor and a controller for controlling the pressure adjustment device. The controller calculates the target pressure of the swing hydraulic motor based on the speed deviation between the target speed of the swing hydraulic motor calculated based on the operation signal of the operating device and the actual driving speed detected by the speed sensor. Then, the controller limits the target pressure of the calculation result so that the estimated input horsepower to the swing hydraulic motor does not exceed the limit value when it is assumed that the driving pressure of the swing hydraulic motor reaches the target pressure after a predetermined time, and controls the pressure adjustment device based on the limited target pressure.

[FIG. 6]



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Description

Technical Field

[0001] The present invention relates to work machines, and more specifically, to work machines equipped with a swing body capable of swinging operation by a hydraulic motor.

Background Art

[0002] In hydraulic work machines such as hydraulic excavators, work devices composed of link members such as booms and arms are driven by hydraulic actuators such as hydraulic cylinders. Additionally, the swing body is rotationally driven relative to the travel body by a hydraulic motor, which is a hydraulic actuator. In work machines, the parts driven by hydraulic actuators generally have large inertial masses, and the control of acceleration and deceleration operations accounts for a significant proportion of the working time. The acceleration and deceleration of the work device and the upper swing body are determined by the driving cylinder thrust or motor torque. Therefore, to control the cylinder thrust or motor torque, it is required to accurately control the driving pressure of the hydraulic actuator to the target value. In general hydraulic excavators, the acceleration and deceleration of the work device and the upper swing body are adjusted by regulating the pressure in the hydraulic circuit using relief valves or bleed-off valves.

[0003] In contrast, Patent Document 1 describes a technique for controlling the pressure of the pressure oil supplied to the hydraulic motor (output torque of the hydraulic motor) by adjusting the capacity (flow rate) of the hydraulic pump so that the pressure of the hydraulic pump (swing pump) detected by the pressure detection device during the swing drive of the work machine becomes a predetermined target pressure. This technology allows for arbitrary adjustment of the output of the hydraulic motor.

[0004] Furthermore, in the work machine described in Patent Document 1, if there is a risk that the output of the hydraulic pump may exceed the maximum output of the engine, output control of the hydraulic pump is performed. Specifically, when the output of the hydraulic pump approaches the maximum output of the engine, the target pressure of the hydraulic pump is reduced to ensure the discharge flow rate of the hydraulic pump while achieving control (power limitation) that prevents the output of the hydraulic pump from exceeding the maximum output of the engine.

Prior Art Documents

Patent Documents

[0005] Patent Document 1: JP-2013-234683 A

Summary of the Invention

Problems to be Solved by the Invention

[0006] Hydraulic actuators in rotational systems, such as swing hydraulic motors, continue to accelerate if the torque is kept constant and no external force is applied, resulting in the required flow increasing over time. In such situations, as described in Patent Document 1, if the product of the discharge flow rate and pressure of the hydraulic pump suddenly decreases the target pressure of the hydraulic pump at the moment it reaches the vicinity of the engine's maximum output (limited horsepower) or maximum output, the torque of the swing hydraulic motor will sharply decrease, potentially causing discomfort to the operator. On the other hand, if the target pressure of the hydraulic pump is kept low in advance so as not to reach the engine's maximum output (limited horsepower), the engine's output (horsepower) cannot be fully utilized, especially at the start of the swing hydraulic motor.

[0007] The present invention has been made to solve the above problems, and its purpose is to provide a work machine capable of driving rotary hydraulic actuators within the range of power limitation with appropriate torque and acceleration.

Means for Solving the Problem

[0008] The present application includes multiple means for solving the above problems. One example is as follows. The work machine comprises a hydraulic pump that discharges pressure oil, a swing body capable of swing operation, a hydraulic actuator that swing-drives the swing body by supplying pressure oil from the hydraulic pump, an operating device that outputs an operation signal instructing an operation of the swing body, and a speed sensor that detects a driving speed of the hydraulic actuator. The work machine further comprises a pressure adjustment device capable of adjusting a driving pressure of the hydraulic actuator, and a controller that controls the pressure adjustment device. The controller is configured to calculate a target speed of the hydraulic actuator based on the operation signal from the operating device, calculate a target pressure of the hydraulic actuator based on a speed deviation, which is a difference between the calculated target speed and an actual driving speed of the hydraulic actuator detected by the speed sensor, limit the calculated target pressure so that an input horsepower to the hydraulic actuator does not exceed a limit value, the input horsepower being estimated when it is assumed that the driving pressure of the hydraulic actuator reaches the calculated target pressure after a predetermined time, and control the pressure adjustment device based on the limited target pressure.

Effect of the Invention

[0009] According to the present invention, the pressure adjustment device is controlled using the pre-limited target pressure of the hydraulic actuator, taking into account the estimated input horsepower to the hydraulic actuator at a future time after a predetermined period. Therefore, the hydraulic actuator can be driven with appropriate torque and acceleration within the range of power limitation. Other problems, configurations, and effects not mentioned above will be clarified by the description of the following embodiments.

Brief Description of the Drawings

[0010]

[FIG. 1]
FIG. 1 is an external view showing a hydraulic excavator as an embodiment of the work machine of the present invention.
[FIG. 2]
FIG. 2 is a hydraulic circuit diagram showing a hydraulic system mounted on an embodiment of the work machine of the present invention.
[FIG. 3]
FIG. 3 is a control block diagram of a controller constituting a part of an embodiment of the work machine shown in FIG. 2 of the present invention.
[FIG. 4]
FIG. 4 is a block diagram showing details of the target angular velocity calculation section, pump flow rate first target value calculation section, and target torque calculation section in the control block diagram of the controller shown in FIG. 3.
[FIG. 5]
FIG. 5 is a correspondence table showing details of the target meter-in pressure calculation section in the control block diagram of the controller shown in FIG. 3.
[FIG. 6]
FIG. 6 is a block diagram showing details of the meter-in pressure limit calculation section in the control block diagram of the controller shown in FIG. 3.
[FIG. 7]
FIG. 7 is a block diagram showing details of the pump flow rate second target value calculation section in the control block diagram of the controller shown in FIG. 3.
[FIG. 8]
FIG. 8 is a block diagram showing details of the angular velocity deviation ratio calculation section in the control block diagram of the controller shown in FIG. 3.
[FIG. 9]
FIG. 9 is a block diagram showing details of the pump flow rate control target value calculation section and bleed-off opening target value calculation section in the control block diagram of the controller shown in FIG. 3.
[FIG. 10]
FIG. 10 is a diagram showing the time waveform of simulation results regarding the behavior of the hydraulic pump and hydraulic motor during swing operation in a comparative example of the work machine of an embodiment of the present invention.
[FIG. 11]
FIG. 11 is a diagram showing the time waveform of simulation results regarding the behavior of the hydraulic pump and hydraulic motor during swing operation in an embodiment of the work machine of the present invention.

Modes for Carrying Out the Invention

[0011] Hereinafter, embodiments of the work machine of the present invention will be described with reference to the drawings. This embodiment is described using a hydraulic excavator as an example of a work machine.

[One Embodiment]

[0012] First, the schematic configuration of a hydraulic excavator as an embodiment of the work machine of the present invention will be described with reference to FIG. 1. FIG. 1 is an external view showing a hydraulic excavator as an embodiment of the work machine of the present invention. Here, the description is given using the direction as seen from the operator seated in the driver’s seat.

[0013] In FIG. 1, the hydraulic excavator includes a self-propelled lower travel body 1 and an upper swing body 2 mounted on the lower travel body 1 in a manner that allows it to swing. The lower travel body 1 and the upper swing body 2 together form the body of the hydraulic excavator. On the front side of the upper swing body 2, a front work device 3 for performing excavation work and the like is rotatably attached.

[0014] The lower travel body 1 has crawler-type travel devices 11 on both the left and right sides (only one side is shown in FIG. 1). The travel device 11 is driven by a travel hydraulic motor 12, which is a hydraulic actuator.

[0015] The upper swing body 2 is configured to be swing-driven around a swing axis line x relative to the lower travel body 1 by a swing device (not shown) including a swing hydraulic motor 33, which is a hydraulic actuator, and its reduction mechanism (see FIG. 2 described later). The upper swing body 2 has a cab 14 on its front side where the operator boards. In the cab 14, a joystick 56, which serves as an operating device described later (see FIG. 2 described later), is arranged. The upper swing body 2 houses a hydraulic pump 31 and various valves 34, 35, 36, 37, 39, 40, 41, 42 (see FIG. 2 described later), among others, as described later.

[0016] The front work device 3 is, for example, a multijointed work device configured by connecting multiple driven members in a manner that allows them to rotate in the vertical direction. The multiple driven members are composed of, for example, a boom 16, an arm 17, and a bucket 18 as a working tool. The boom 16, arm 17, and bucket 18 are each driven by hydraulic actuators, namely, a boom cylinder 21, an arm cylinder 22, and a bucket cylinder 23.

[0017] Next, the configuration of the hydraulic system mounted on an embodiment of the work machine of the present invention will be described with reference to FIG. 2. FIG. 2 is a hydraulic circuit diagram showing a hydraulic system mounted on an embodiment of the work machine of the present invention.

[0018] In FIG. 2, the hydraulic excavator includes a hydraulic system 30 that drives the lower travel body 1, upper swing body 2, and front work device 3 (see FIG. 1 for all) by hydraulic power. Note that in FIG. 2, only the hydraulic circuit related to the swing hydraulic motor 33 that swing-drives the upper swing body 2 is shown, and the hydraulic circuits related to the travel hydraulic motor 12 that drives the travel device 11 and the boom cylinder 21, arm cylinder 22, and bucket cylinder 23 that drive the front work device 3 are omitted.

[0019] The hydraulic system 30 includes a hydraulic pump 31 driven by a prime mover 32 (e.g., an electric motor or engine) to discharge pressure oil, and a swing hydraulic motor 33 that swing-drives the upper swing body 2 by the supply of pressure oil from the hydraulic pump 31. The hydraulic pump 31 is a variable displacement pump and has a regulator 31a for adjusting the pump volume. The regulator 31a functions as a flow adjustment device capable of adjusting the discharge flow rate of the hydraulic pump 31, and by adjusting the discharge flow rate of the hydraulic pump 31, it also functions as a pressure adjustment device capable of adjusting the drive pressure of the swing hydraulic motor 33. The regulator 31a adjusts the pump volume in response to control signals from the controller 60, for example. The swing hydraulic motor 33 has a pair of input/output ports, namely the first port 33a and the second port 33b. The swing hydraulic motor 33 is, for example, a fixed displacement hydraulic motor.

[0020] The pressure oil discharged from the hydraulic pump 31 is supplied to the swing hydraulic motor 33 via a load check valve 34 and a directional control valve 35. The load check valve 34 is provided on the discharge line 44 connecting the hydraulic pump 31 and the directional control valve 35. The load check valve 34 allows the flow of pressure oil from the hydraulic pump 31 to the directional control valve 35 while preventing the flow of pressure oil from the directional control valve 35 to the hydraulic pump 31. The directional control valve 35 controls the flow (direction and flow rate) of pressure oil supplied from the hydraulic pump 31 to the swing hydraulic motor 33. The position (stroke amount) of the directional control valve 35 is controlled in response to control signals (excitation current) from the controller 60.

[0021] A bleed-off valve 36 is provided on the line 47 branching from the discharge line 44 and connected to the hydraulic oil tank 38. The bleed-off valve 36 allows for the adjustment of the discharge pressure of the hydraulic pump 31 according to its degree of opening. That is, the bleed-off valve 36 functions as a pressure adjustment device capable of adjusting the drive pressure of the swing hydraulic motor 33 by releasing the pressure oil discharged from the hydraulic pump 31 to the hydraulic oil tank 38 according to its degree of opening. Additionally, the discharge port of the hydraulic pump 31 is connected to the hydraulic oil tank 38 via a main relief valve 37. The main relief valve 37 defines the upper limit of the discharge pressure of the hydraulic pump 31 and is configured to open when the discharge pressure of the hydraulic pump 31 exceeds the set pressure.

[0022] The first port 33a and the second port 33b of the swing hydraulic motor 33 are connected to the directional control valve 35 via the first line 45 and the second line 46, respectively. The first port 33a and the second port 33b of the swing hydraulic motor 33 are connected to the first swing relief valve 39 and the second swing relief valve 40 via the first line 45 and the second line 46, respectively. The first swing relief valve 39 and the second swing relief valve 40 open when the pressure in the first line 45 and the second line 46 exceeds the set pressure, respectively, thereby communicating the first line 45 and the second line 46 with the hydraulic oil tank 38, and they serve the overload prevention function of the swing hydraulic motor 33. Additionally, the first port 33a and the second port 33b of the swing hydraulic motor 33 are connected to the makeup first check valve 41 and the makeup second check valve 42 via the first line 45 and the second line 46, respectively. The makeup first check valve 41 prevents the flow of pressure oil from the first line 45 to the hydraulic oil tank 38 while allowing the flow of hydraulic oil from the hydraulic oil tank 38 to the first line 45. The makeup second check valve 42 prevents the flow of pressure oil from the second line 46 to the hydraulic oil tank 38 while allowing the flow of hydraulic oil from the hydraulic oil tank 38 to the second line 46. The makeup first check valve 41 and the makeup second check valve 42 serve the anti-void function of the swing hydraulic motor 33.

[0023] On the first line 45 and the second line 46, first pressure sensors 51a and 51b are provided to detect the pressure (drive pressure) on the first port 33a side and the second port 33b side of the swing hydraulic motor 33, respectively. The first pressure sensors 51a and 51b output pressure detection signals corresponding to the detected pressure on the first port 33a side and the second port 33b side (drive pressure) to the controller 60. On the discharge line 44, a second pressure sensor 52 is provided to detect the discharge pressure of the hydraulic pump 31. The second pressure sensor 52 outputs a discharge pressure detection signal corresponding to the detected discharge pressure to the controller 60. Additionally, a speed sensor 54 is installed on the swing hydraulic motor 33 to detect the actual angular velocity (drive speed) of the swing hydraulic motor 33. The speed sensor 54 outputs an angular velocity detection signal corresponding to the detected angular velocity to the controller 60.

[0024] The hydraulic system 30 further includes a joystick 56 as an operating device for instructing the swing operation of the upper swing body 2 or the drive of the swing hydraulic motor 33. The joystick 56 outputs an operation signal corresponding to its operation angle to the controller 60.

[0025] The controller 60 acquires the swing operation signal from the joystick 56, the angular velocity detection signal from the speed sensor 54 (the actual angular velocity of the swing hydraulic motor 33 detected by the speed sensor 54), the pressure detection signals from the first pressure sensors 51a and 51b (the pressure on the first port 33a side and the second port 33b side of the swing hydraulic motor 33 detected by the first pressure sensors 51a and 51b), and the discharge pressure detection signal from the second pressure sensor 52 (the discharge pressure of the hydraulic pump 31 detected by the second pressure sensor 52). The controller 60 performs predetermined calculations based on these operation signals and detection signals and outputs control signals corresponding to the calculation results to the regulator 31a of the hydraulic pump 31, the directional control valve 35, and the bleed-off valve 36. Details of the calculations will be described later. The controller 60 directly controls the pump volume (pump flow rate) of the hydraulic pump 31, the drive of the directional control valve 35, and the drive of the bleed-off valve 36, thereby ultimately controlling the drive pressure and motor flow rate of the swing hydraulic motor 33 (the swing operation of the upper swing body 2).

[0026] Next, an outline of the functions of the controller, which constitutes part of an embodiment of the work machine of the present invention, will be described with reference to FIG. 3. FIG. 3 is a control block diagram of the controller, which constitutes part of an embodiment of the work machine of the present invention shown in FIG. 2.

[0027] In FIG. 3, the controller 60 includes, as a hardware configuration, for example, a storage device 61 composed of RAM, ROM, etc., and a processing device 62 composed of a CPU, MPU, etc. The storage device 61 stores in advance the programs and various information necessary for controlling the pump volume (pump flow rate) of the hydraulic pump 31, the drive of the directional control valve 35, and the drive of the bleed-off valve 36. The processing device 62 reads programs and various information from the storage device 61 as appropriate and executes processing according to the programs to realize various functions. The controller 60 of this embodiment performs drive control of the swing hydraulic motor 33 by speed control using a target speed value or pressure control using a pressure target value that takes into account power limitations in advance, and it mainly includes the following control function sections.

[0028] The controller 60 includes a directional control valve control section 71 that captures the swing operation signal from the joystick 56 as an operating device and outputs a drive control signal to the directional control valve 35. The directional control valve control section 71 calculates the opening target value of the directional control valve 35 by referring to a first table (not shown) from the swing operation signal, and converts the calculated opening target value into a drive command value (command current value) by referring to a second table (not shown). The directional control valve control section 71 outputs the drive control signal of the drive command value to the directional control valve 35.

[0029] The controller 60 includes a target angular velocity calculation section 73 that captures the swing operation signal from the operating device 56 and outputs the target angular velocity ωt of the swing hydraulic motor 33. Details of the calculations of the target angular velocity calculation section 73 will be described later.

[0030] The controller 60 includes a pump flow rate first target value calculation section 74 that captures the target angular velocity ωt of the swing hydraulic motor 33, which is the calculation result of the target angular velocity calculation section 73, and outputs the pump flow rate first target value Qt1 of the hydraulic pump 31. The pump flow rate first target value Qt1 is derived from the target angular velocity ωt of the swing hydraulic motor 33 and is a target value used for speed control of the swing hydraulic motor 33. Details of the calculations of the pump flow rate first target value calculation section 74 will be described later.

[0031] Additionally, the controller 60 includes a target torque calculation section 76, a target meter-in pressure calculation section 77, a meter-in pressure limit calculation section 78, and a pump flow rate second target value calculation section 79. These series of calculation sections 76 to 79 derive the pump flow rate second target value Qt2 of the hydraulic pump 31, which is a target value used for pressure control of the swing hydraulic motor 33. The target torque calculation section 76 captures the target angular velocity ωt of the swing hydraulic motor 33, which is the calculation result of the target angular velocity calculation section 73, the actual angular velocity (drive speed) of the swing hydraulic motor 33 from the speed sensor 54, and the swing operation signal from the operating device 56, and outputs the target torque Tt of the swing hydraulic motor 33. The target meter-in pressure calculation section 77 captures the target torque Tt, which is the calculation result of the target torque calculation section 76, and outputs the target meter-in pressure Pt of the swing hydraulic motor 33. The meter-in pressure limit calculation section 78 captures the actual angular velocity (drive speed) and actual angular acceleration (drive acceleration) of the swing hydraulic motor 33 from the speed sensor 54, and outputs the meter-in pressure limit value PL of the swing hydraulic motor 33 to limit the input horsepower to the swing hydraulic motor 33 to below the horsepower limit value (horsepower limitation). The pump flow rate second target value calculation section 79 captures the target meter-in pressure Pt, which is the calculation result of the target meter-in pressure calculation section 77, the meter-in pressure limit value PL, which is the calculation result of the meter-in pressure limit calculation section 78, and the pressure (drive pressure) of the swing hydraulic motor 33 from the first pressure sensors 51a, 51b, and outputs the pump flow rate second target value Qt2. The details of the calculations of the target torque calculation section 76, the target meter-in pressure calculation section 77, the meter-in pressure limit calculation section 78, and the pump flow rate second target value calculation section 79 will be described later.

[0032] The controller 60 also includes an angular velocity deviation ratio calculation section 81 that captures the target angular velocity ωt of the swing hydraulic motor 33, which is the calculation result of the target angular velocity calculation section 73, and the actual angular velocity of the swing hydraulic motor 33 from the speed sensor 54, and outputs the angular velocity deviation ratio R. The angular velocity deviation ratio R is an indicator for switching the drive control of the swing hydraulic motor 33 to speed control or pressure control, as well as an indicator for switching the opening and closing of the bleed-off valve 36. The details of the calculations of the angular velocity deviation ratio calculation section 81 will be described later.

[0033] The controller 60 also captures the pump flow rate first target value Qt1, which is the calculation result of the pump flow rate first target value calculation section 74, the pump flow rate second target value Qt2, which is the calculation result of the pump flow rate second target value calculation section 79, and the angular velocity deviation ratio R, which is the calculation result of the angular velocity deviation ratio calculation section 81. Then, the controller 60 calculates the pump flow rate control target value Qc of the hydraulic pump 31 and includes a pump flow rate control target value calculation section 83 that outputs a control signal corresponding to the pump flow rate control target value Qc to the regulator 31a of the hydraulic pump 31. The pump flow rate control target value Qc is the final control target value of the discharge flow rate (pump capacity) of the hydraulic pump 31. The details of the calculations of the pump flow rate control target value calculation section 83 will be described later.

[0034] The controller 60 captures the angular velocity deviation ratio R, which is the calculation result of the angular velocity deviation ratio calculation section 81, calculates the opening target value Vt of the bleed-off valve 36, and includes a bleed-off opening target value calculation section 85 that outputs a control signal corresponding to the opening target value Vt to the bleed-off valve 36. The details of the calculations of the bleed-off opening target value calculation section 85 will be described later.

[0035] Next, an example of the details of the calculations of each function of the controller in one embodiment of the work machine of the present invention will be described with reference to FIGS. 4 to 10. FIG. 4 is a block diagram showing the details of the target angular velocity calculation section, the pump flow rate first target value calculation section, and the target torque calculation section in the control block diagram of the controller shown in FIG. 3.

[0036] The target angular velocity calculation section 73, as shown in FIG. 4, calculates the target angular velocity (target speed) of the swing hydraulic motor 33 by referring to the table 731 based on the swing operation signal input from the operating device 56. The swing operation signal is output, for example, in the range from -100 to +100. In the case of non-operation, it outputs 0; in the case of the maximum operation amount for left swing, it outputs -100; and in the case of the maximum operation amount for right swing, it outputs +100. In table 731, the angular velocity for left swing is negative, and the angular velocity for right swing is positive, depending on the sign of the swing operation signal. The target angular velocity calculation section 73 outputs the calculated target angular velocity to the pump flow rate first target value calculation section 74, the target torque calculation section 76, and the angular velocity deviation ratio calculation section 81 (see FIG. 8 described later).

[0037] In the pump flow rate first target value calculation section 74, the calculation section 741 takes the absolute value of the target angular velocity, which is the calculation result of the target angular velocity calculation section 73. Furthermore, the calculation section 741 calculates the pump flow rate first target value by multiplying the absolute value of the target angular velocity by the swing equivalent volume q (the volume required to rotate the upper swing body 2 at a unit angular velocity). The pump flow rate first target value is directly derived based on the target angular velocity of the swing hydraulic motor 33 and is a control value for performing speed control of the swing hydraulic motor 33. The pump flow rate first target value calculation section 74 outputs the calculated pump flow rate first target value to the pump flow rate control target value calculation section 83 (see FIG. 9 described later).

[0038] In the target torque calculation section 76, first, the calculation section 761 calculates the angular velocity deviation by subtracting the actual angular velocity (angular velocity detection value) of the swing hydraulic motor 33 detected by the speed sensor 54 from the target angular velocity of the swing hydraulic motor 33, which is the calculation result of the target angular velocity calculation section 73. Next, the calculation unit 762 multiplies a predetermined proportional gain Kp by the angular velocity deviation, which is the calculation result of the calculation unit 761. Then, the target torque calculation section 76 calculates the target torque Tt of the swing hydraulic motor 33 by referring to the table 763 based on the output value of the calculation section 762. However, the torque limit value of the swing hydraulic motor 33 is set using the table 764 based on the swing operation signal, and the set torque limit value is input to the table 763, thereby setting the upper and lower limits of the input value to the table 763. The target torque calculation section 76 outputs the calculated target torque Tt to the target meter-in pressure calculation section 77 (see FIG. 5 described later).

[0039] FIG. 5 is a correspondence table showing the details of the target meter-in pressure calculation section in the control block diagram of the controller shown in FIG. 3. The target meter-in pressure calculation section 77 calculates the target meter-in pressure of the swing hydraulic motor 33 based on the swing operation signal from the operating device 56 and the target torque of the swing hydraulic motor 33, which is the calculation result of the target torque calculation section 76. In the swing hydraulic motor 33, the side where the pressure oil from the hydraulic pump 31 flows in is called the meter-in side, and the side where the pressure oil flows out is called the meter-out side. In this description, the first port 33a of the swing hydraulic motor 33 is the meter-in side during right swing, and the second port 33b is the meter-in side during left swing.

[0040] As shown in FIG. 5, if the operation signal is greater than the threshold th1 (indicating right swing) and the sign of the target torque Tt, which is the calculation result of the target torque calculation section 76, is positive (the direction of the torque is the same as the direction of the right swing), the calculation result (Tt/q) obtained by dividing the target torque Tt by the swing equivalent volume q becomes the target pressure value of the first port 33a, that is, the target meter-in pressure. At this time, the target pressure of the second port 33b is 0. Also, if the operation signal is greater than the threshold th1 and the sign of the target torque Tt, which is the calculation result, is negative (the direction of the torque is opposite to the direction of the right swing), the target pressure value of the first port 33a is 0, and the target pressure value of the second port 33b is (-Tt/q). The negative sign of the target pressure value of the second port 33b takes into account that the sign of the target torque Tt is negative.

[0041] On the other hand, if the operation signal is less than the threshold -th1 (indicating left swing) and the sign of the target torque Tt, which is the calculation result, is negative, the calculation result (-Tt/q) obtained by dividing the target torque Tt by the swing equivalent volume q becomes the target pressure value of the second port 33b, that is, the target meter-in pressure. The target pressure of the first port 33a is 0. Also, if the operation signal is less than the threshold -th1 (indicating right swing) and the sign of the target torque Tt, which is the calculation result, is positive, the target pressure value of the second port 33b is 0, and the target pressure value of the first port 33a is (Tt/q).

[0042] Note that if the operation signal is a deadband value from -th1 to th1, the meter-in side is the same port as the previous port. In this case, the target pressure values of the first port 33a and the second port 33b are as shown in FIG. 5.

[0043] In this way, the target meter-in pressure calculation section 77 determines which of the first port 33a and the second port 33b of the swing hydraulic motor 33 is the meter-in side based on the operation signal, and calculates the target pressure value (target meter-in pressure) on the meter-in side based on the target torque Tt of the swing hydraulic motor 33, which is the calculation result of the target torque calculation section 76.

[0044] FIG. 6 is a block diagram showing the details of the meter-in pressure limit calculation section 78 in the control block diagram of the controller shown in FIG. 3. In the meter-in pressure limit calculation section 78, first, the calculation section 781 multiplies the actual angular acceleration (angular acceleration detection value) of the swing hydraulic motor 33 detected by the speed sensor 54 by a reference time (e.g., 0.3 seconds), and the calculation section 782 adds the calculation result of the calculation section 781 to the actual angular velocity (angular velocity detection value) of the swing hydraulic motor 33 detected by the speed sensor 54. These calculations estimate the angular velocity of the swing hydraulic motor 33 at a future time after the reference time has elapsed from the current time during the calculation of the controller 60. It is also possible for the controller to perform differential calculations based on the time series of the actual angular velocity (angular velocity detection value) of the swing hydraulic motor 33 detected by the speed sensor 54, and to calculate the actual angular acceleration.

[0045] Next, the calculation section 783 takes the absolute value of the calculation result of the calculation section 782 (the estimated angular velocity of the swing hydraulic motor 33 after the reference time has elapsed from the current time), and the calculation section 784 multiplies the calculation result of the calculation section 783 by the swing equivalent volume q. This calculation estimates the drive flow rate of the swing hydraulic motor 33 at a future time after the reference time has elapsed from the current time.

[0046] Furthermore, the calculation section 785 divides the calculation result of the calculation section 784 (the estimated flow rate of the swing hydraulic motor 33 after the reference time has elapsed from the current time) by the input horsepower limit value to the swing hydraulic motor 33, and sets the upper and lower limits using the table 786 for the calculation result of the calculation section 785, thereby calculating the meter-in pressure limit value of the swing hydraulic motor 33 that does not exceed the horsepower limit.

[0047] The target meter-in pressure of the swing hydraulic motor 33, which is the calculation result of the target meter-in pressure calculation section 77, is limited by the meter-in pressure limit value, which is the calculation result of the meter-in pressure limit calculation section 78. As a result, considering the horsepower limit value, a horsepower limit target meter-in pressure with a restricted upper limit is calculated. In other words, these calculations limit the target meter-in pressure so that the estimated input horsepower to the swing hydraulic motor 33 does not exceed the horsepower limit value when it is assumed that the drive pressure of the swing hydraulic motor 33 reaches the target meter-in pressure at a future time after a reference time has elapsed from the current time (after a predetermined time). The horsepower limit target meter-in pressure, which is the calculation result, is input to the pump flow rate second target value calculation section 79 (see FIG. 7 described later).

[0048] FIG. 7 is a block diagram showing the details of the pump flow rate second target value calculation section 79 in the control block diagram of the controller shown in FIG. 3. In the pump flow rate second target value calculation section 79, first, the calculation section 791 calculates the pressure deviation by subtracting the actual meter-in pressure of the swing hydraulic motor 33 (the pressure detection value on the meter-in side of the swing hydraulic motor 33) detected by the first pressure sensors 51a, 51b from the input horsepower limit target meter-in pressure. Next, the pump flow rate second target value calculation section 79 multiplies the pressure deviation, which is the calculation result of the calculation section 791, by the proportional gain Kp2 in the calculation section 792, and then multiplies it by the integral gain Ki in the calculation section 794 after the integration processing in the calculation section 793. Then, the pump flow rate second target value calculation section 79 adds the calculation results of the calculation sections 793 and 794 in the calculation section 795, and limits the upper and lower bounds of the calculation result of the calculation section 795 using the table 796 to calculate the pump flow rate second target value of the hydraulic pump 31. The pump flow rate second target value is derived based on the pressure deviation, which is the difference between the horsepower limit target meter-in pressure and the actual meter-in pressure, and is a control value for performing pressure control of the swing hydraulic motor 33. The limitation by the table 796 restricts the pressure, for example, from 0 MPa to the set pressure of the main relief valve 37 of the hydraulic circuit. The pump flow rate second target value calculation section 79 outputs the pump flow rate second target value, which is the calculation result, to the pump flow rate control target value calculation section 83 (see FIG. 9 described later). Note that in this description, the pressure detection values of the first pressure sensors 51a, 51b are input to the calculation section 791 that calculates the pressure deviation, but if the discharge pressure of the hydraulic pump 31 can be considered an approximate value of the meter-in pressure of the swing hydraulic motor 33, it is also possible to input the pressure detection value of the second pressure sensor 52 to the calculation section 791.

[0049] FIG. 8 is a block diagram showing the details of the angular velocity deviation ratio calculation section 81 in the control block diagram of the controller shown in FIG. 3. In the angular velocity deviation ratio calculation section 81, the calculation section 811 calculates the angular velocity deviation by subtracting the actual angular velocity (angular velocity detection value) of the swing hydraulic motor 33 detected by the speed sensor 54 from the target angular velocity of the swing hydraulic motor 33, which is the calculation result of the target angular velocity calculation section 73. Next, after performing the processing of the calculation section 812 to prevent division by zero with respect to the target angular velocity of the swing hydraulic motor 33, which is the calculation result of the target angular velocity calculation section 73, the calculation section 813 divides the angular velocity deviation, which is the calculation result of the calculation section 811, by the target angular velocity of the swing hydraulic motor 33 after the processing of the calculation section 812, and the calculation section 814 takes the absolute value of the calculation result of the calculation section 813 to calculate the angular velocity deviation ratio. That is, the angular velocity deviation ratio indicates the ratio of the angular velocity deviation to the target angular velocity. The angular velocity deviation ratio calculation section 81 outputs the angular velocity deviation ratio, which is the calculation result, to the pump flow rate control target value calculation section 83 (see FIG. 9 described later).

[0050] FIG. 9 is a block diagram showing the details of the pump flow rate control target value calculation section 83 and the bleed-off opening target value calculation section 85 in the control block diagram of the controller shown in FIG. 3. The pump flow rate control target value calculation section 83, in summary, calculates the pump flow rate control target value of the hydraulic pump 31 by adding the pump flow rate first target value, which is the calculation result of the pump flow rate first target value calculation section 74, and the pump flow rate second target value, which is the calculation result of the pump flow rate second target value calculation section 79, according to the ratio determined based on the output value of the table 831.

[0051] Specifically, the table 831 outputs a value in the range from 0 to 1 based on the angular velocity deviation ratio, which is the calculation result of the angular velocity deviation ratio calculation section 81. In the table 831, for example, when the angular velocity deviation ratio is in a range smaller than the first threshold n1 (for example, a range of 0.2 or less), the output value is set to 0 or a value near 0. On the other hand, when the angular velocity deviation ratio is in a range larger than the second threshold n2, the output value is set to 1 or a value near 1.

[0052] The calculation section 832 multiplies the output value of the table 831 by the pump flow rate second target value for pressure control of the swing hydraulic motor 33. On the other hand, the calculation section 834 multiplies the calculation result of the calculation section 833, which subtracts the output value of the table 831 from 1, by the pump flow rate first target value for speed control of the swing hydraulic motor 33. Finally, the calculation section 835 calculates the pump flow rate control target value by adding the calculation results of the calculation sections 832 and 834.

[0053] In the range where the angular velocity deviation ratio is small, since the output value of the table 831 is set to approximately 0, the pump flow rate control target value becomes the pump flow rate first target value. On the other hand, in the range where the angular velocity deviation ratio is large, since the output value of the table 831 is set to approximately 1, the pump flow rate control target value becomes the pump flow rate second target value. That is, when the angular velocity deviation ratio is small, for example, when the angular velocity deviation is small and the target angular velocity is high, the controller 60 performs speed control on the swing hydraulic motor 33. In contrast, when the angular velocity deviation ratio is large, for example, when the angular velocity deviation is large and the target angular velocity is low, the controller 60 performs pressure control on the swing hydraulic motor 33. In this way, the angular velocity deviation ratio serves as an indicator for switching the control of the swing hydraulic motor 33 to speed control or pressure control.

[0054] The controller 60 calculates the target pump volume of the hydraulic pump 31 by dividing the pump flow rate control target value, which is the calculation result of the pump flow rate control target value calculation section 83, by the target prime mover speed. The controller 60 ultimately outputs a control signal corresponding to the target pump volume, which is the calculation result, to the regulator 31a of the hydraulic pump 31. As a result, the pump volume of the hydraulic pump 31 is controlled.

[0055] The bleed-off opening target value calculation section 85 calculates the bleed-off opening target value by referring to the table 851 based on the angular velocity deviation ratio R, which is the calculation result of the angular velocity deviation ratio calculation section 81. The table 851 is set to make the opening of the bleed-off valve 36 as small as possible when the angular velocity deviation ratio R is smaller than the first threshold n1. This is to reduce the loss due to the outflow of pressure oil to the hydraulic oil tank 38 through the bleed-off valve 36. On the other hand, the table 851 is set to maintain the opening of the bleed-off valve 36 at a predetermined value when the angular velocity deviation ratio is larger than the second threshold n2. This is to enable the control of the pressure in the hydraulic circuit by the flow rate change of the hydraulic pump 31. Note that the first threshold n1 and the second threshold n2 may be the same value. The controller 60 outputs a control signal corresponding to the bleed-off opening target value, which is the calculation result of the bleed-off opening target value calculation section 85, to the bleed-off valve 36. As a result, the opening degree of the bleed-off valve 36 is controlled.

[0056] Next, the operation and effect of an embodiment of the work machine of the present invention will be described in comparison with the operation of a comparative example of the work machine. First, the behavior of the hydraulic pump and the swing hydraulic motor during swing operation in the work machine of the comparative example will be described using FIG. 10. FIG. 10 is a diagram showing the time waveform of simulation results regarding the behavior of the hydraulic pump and the swing hydraulic motor during swing operation in the work machine of the comparative example for an embodiment of the work machine of the present invention.

[0057] Graph (A) of FIG. 10 shows the time change of the swing operation signal. Graph (B) shows the time change of the discharge pressure of the hydraulic pump and the drive pressure of the swing hydraulic motor, and graph (C) shows the time change of the pump flow rate of the hydraulic pump and the motor flow rate of the swing hydraulic motor. Graph (D) shows the time change of the angular velocity of the swing hydraulic motor, and graph (E) shows the time change of the angular acceleration of the swing hydraulic motor. Graph (F) shows the time change of the output of the hydraulic pump.

[0058] In the work machine of the comparative example, when a swing operation is input, the output of the hydraulic pump is controlled to continue until the discharge pressure of the hydraulic pump reaches the set pressure of the main relief valve, until the output of the hydraulic pump reaches the horsepower limit. For example, as shown in graph (A), the discharge pressure of the hydraulic pump begins when the swing operation is at the horizontal axis 1.0, indicating time. Subsequently, when a constant manipulative variable (e.g., full manipulative variable) is input, the discharge pressure of the hydraulic pump rapidly increases as shown in graph (B), reaching the set pressure of the main relief valve, while the flow rate of the hydraulic pump gradually increases as shown in graph (C), in response to the swing operation. When the product of the hydraulic pump's flow rate and discharge pressure, i.e., the pump output, reaches the horsepower limit value (around the horizontal axis (time) 1.75 in graph (F)), the work machine executes control to reduce the target flow rate of the hydraulic pump to avoid exceeding the pump output.

[0059] At this time, since the swing hydraulic motor is accelerating as shown in graph (D), reducing the target flow rate of the hydraulic pump causes the discharge pressure of the hydraulic pump to drop sharply (refer to the first steep drop around the horizontal axis 1.75 in graph (B)), resulting in a sharp decrease in the angular acceleration of the swing hydraulic motor (refer to the first steep drop around the horizontal axis 1.75 in graph (E)). In response to this sharp decrease in discharge pressure, the hydraulic pump is controlled to increase the pump flow rate again. However, due to a certain time delay in acquiring pressure detection values from the pressure sensor and controlling the pump volume, the rapid increase and decrease in the discharge pressure of the hydraulic pump are repeated, causing control hunting (refer to the horizontal axis 1.75 and beyond in graph (B)). As a result, the rapid increase and decrease in the angular acceleration of the swing hydraulic motor are repeated (refer to the horizontal axis 1.75 and beyond in graph (E)), causing discomfort in the operator's handling feel. Furthermore, the repeated rapid increase and decrease in the discharge pressure of the hydraulic pump cause the pump output to temporarily exceed the horsepower limit value repeatedly (refer to the horizontal axis 1.75 and beyond in graph (F)), resulting in vibrational behavior of the machine body.

[0060] Next, the behavior of the hydraulic pump and swing hydraulic motor during swing operation in one embodiment of the work machine of the present invention is described using FIG. 11. FIG. 11 is a diagram showing the time waveform of simulation results regarding the behavior of the hydraulic pump and swing hydraulic motor in response to swing operation in one embodiment of the work machine of the present invention.

[0061] The items subject to time change shown in graphs (A) to (F) of FIG. 11 are the same as those shown in graphs (A) to (F) of FIG. 10. However, graph (B) also shows the time change of the target pressure of the hydraulic pump 31 (target pressure of the swing hydraulic motor 33). Additionally, graph (D) shows the time change of the target angular velocity of the swing hydraulic motor 33.

[0062] The controller 60 of this embodiment estimates the angular velocity of the swing hydraulic motor 33 at a future time after a reference time has elapsed from the current time during calculation when the angular velocity deviation ratio is large. Then, the controller 60 controls based on the estimated angular velocity to ensure that the input horsepower to the swing hydraulic motor 33 at the future time is limited to be below the horsepower limit value, using the target pressure of the swing hydraulic motor 33 (target discharge pressure of the hydraulic pump 31). Therefore, at the start of the swing hydraulic motor 33, where the angular velocity deviation ratio is large, the controller 60 sets a high target pressure for the hydraulic pump 31 (target pressure of the swing hydraulic motor 33 (target meter-in pressure)) as shown in graph (B) to increase the swing angular acceleration of the swing hydraulic motor 33 (refer to the horizontal axis (time) 1.0 vicinity in graph (E)), since the drive flow rate of the swing hydraulic motor 33 is low (refer to the horizontal axis (time) 1.0 vicinity where the swing operation starts in graph (C)). Subsequently, when the angular velocity of the swing hydraulic motor 33 increases, accelerating the swing hydraulic motor 33 with the initially high drive pressure causes the output of the hydraulic pump 31 (input horsepower to the swing hydraulic motor 33) to reach the horsepower limit. Therefore, the controller 60 pre-limits the pump target pressure (the product of the estimated flow rate and the pump target pressure) according to the estimated flow rate at a future time (a predetermined time later) after a certain time has elapsed from the current time, through the control calculation shown in FIG. 6. This prevents the output of the hydraulic pump 31 from exceeding the horsepower limit as shown in graph (F), while maintaining smooth and stable acceleration of the swing hydraulic motor 33 as shown in graph (E). Therefore, the work machine can achieve the desired swing acceleration without causing discomfort in the handling feel.

[0063] Additionally, the controller 60 of this embodiment controls based on the target speed of the swing hydraulic motor 33 when the angular velocity deviation ratio is small. For example, as shown in graph (D), when the actual angular velocity of the swing hydraulic motor 33 approaches the target angular velocity around the horizontal axis (time) 2.5, resulting in a small angular velocity deviation ratio, the pump flow rate control target value is switched from the second target value for pressure control to the first target value for speed control. Through this speed control, the controller 60 ultimately matches the actual angular velocity of the swing hydraulic motor 33 to the target angular velocity (refer to the horizontal axis 2.7 and beyond in graph (D)).

[0064] Additionally, the controller 60 of this embodiment controls the closing of the bleed-off valve 36 in response to switching the control target value of the hydraulic pump 31 from pressure control to speed control. By blocking the flow of pressure oil from the hydraulic pump 31 to the hydraulic oil tank 38 with the bleed-off valve 36, hydraulic losses can be reduced, enabling efficient swing operation of the work machine.

[0065] The hydraulic excavator according to one embodiment of the present invention includes a hydraulic pump 31 that discharges pressure oil, an upper swing body 2 (swing body) capable of swing operation, a swing hydraulic motor 33 (hydraulic actuator) that swing-drives the upper swing body 2 (swing body) by supplying pressure oil from the hydraulic pump 31, an operating device 56 that outputs an operation signal instructing the operation of the upper swing body 2 (swing body), and a speed sensor 54 that detects the drive speed of the swing hydraulic motor 33 (hydraulic actuator). Furthermore, the hydraulic excavator includes a regulator 31a and a bleed-off valve 36 as pressure adjustment devices capable of adjusting the drive pressure of the swing hydraulic motor 33 (hydraulic actuator), and a controller 60 that controls the regulator 31a and the bleed-off valve 36 (pressure adjustment devices). The controller 60 calculates the target speed of the swing hydraulic motor 33 (hydraulic actuator) based on the operation signal from the operating device 56, and calculates the target pressure of the swing hydraulic motor 33 (hydraulic actuator) based on the speed deviation, which is the difference between the calculated target speed and the actual drive speed of the swing hydraulic motor 33 (hydraulic actuator) detected by the speed sensor 54. Then, the controller 60 is configured to limit the calculated target pressure so that the estimated input horsepower to the swing hydraulic motor 33 (hydraulic actuator) does not exceed the limit value when it is assumed that the drive pressure of the swing hydraulic motor 33 (hydraulic actuator) reaches the calculated target pressure after a predetermined time, and to control the regulator 31a and the bleed-off valve 36 (pressure adjustment devices) based on the limited target pressure.

[0066] This configuration controls the regulator 31a and the bleed-off valve 36 (pressure adjustment devices) using the pre-limited target pressure of the swing hydraulic motor 33 (hydraulic actuator), considering the estimated input horsepower to the swing hydraulic motor 33 (hydraulic actuator) at a future time after a predetermined time, enabling the swing hydraulic motor 33 (hydraulic actuator) to be driven with appropriate torque and acceleration within the horsepower limit range.

[0067] Additionally, the controller 60 of the hydraulic excavator according to this embodiment is configured to estimate the input horsepower to the swing hydraulic motor 33 (hydraulic actuator) at a future time (a predetermined time later) by estimating the angular velocity (drive speed) of the swing hydraulic motor 33 (hydraulic actuator) at a future time (a predetermined time later) based on the actual angular velocity and actual angular acceleration of the swing hydraulic motor 33 (hydraulic actuator) obtained from the detection values of the speed sensor 54.

[0068] This configuration allows for easy and accurate estimation of the drive speed of the swing hydraulic motor 33 (hydraulic actuator) at a future time (a predetermined time later) by using the actual drive speed and actual drive acceleration of the swing hydraulic motor 33 (hydraulic actuator) obtained from the speed sensor 54. Therefore, it becomes possible to correctly limit the calculated target pressure so that the estimated input horsepower to the swing hydraulic motor 33 (hydraulic actuator), obtained from the integration of the estimated flow rate calculated from the estimated drive speed at a future time (a predetermined time later) and the target pressure of the swing hydraulic motor 33 (hydraulic actuator), does not exceed the horsepower limit value. As a result, the swing hydraulic motor 33 (hydraulic actuator) can be driven with smooth acceleration without exceeding the horsepower limit value.

[0069] Additionally, in this embodiment, the pressure adjustment device includes a regulator 31a as a flow adjustment device capable of adjusting the drive pressure of the swing hydraulic motor 33 (hydraulic actuator) by adjusting the discharge flow rate of the hydraulic pump 31. The controller 60 calculates the first target value of the pump flow rate of the hydraulic pump 31 based on the calculated target speed, and calculates the second target value of the pump flow rate of the hydraulic pump 31 based on the limited target pressure. Additionally, the controller 60 calculates the speed deviation ratio, which is the ratio of the speed deviation to the calculated target speed, and executes control of the regulator 31a (flow adjustment device) based on the first target value of the pump flow rate when the calculated speed deviation ratio is smaller than the first threshold n1. On the other hand, when the speed deviation ratio of the calculation result is greater than the second threshold n2, the controller 60 executes control of the regulator 31a (flow adjustment device) based on the second target value of the pump flow rate of the calculation result, as control of the pressure adjustment device based on the restricted target pressure.

[0070] According to this configuration, at the start of the swing hydraulic motor 33 with a large speed deviation ratio, the estimated flow rate of the swing hydraulic motor 33 (hydraulic actuator) is small, so the restricted target pressure can be set higher accordingly. Therefore, the acceleration of the swing hydraulic motor 33 (hydraulic actuator) can be increased. On the other hand, when the speed deviation ratio becomes smaller, that is, when the target speed of the swing hydraulic motor 33 (hydraulic actuator) approaches the driving speed, the regulator 31a (flow adjustment device) is controlled based on the first target value of the pump flow rate set based on the target speed, allowing for precise fine-tuning of the actual driving speed of the swing hydraulic motor 33 (hydraulic actuator).

[0071] In this embodiment, the pressure adjustment device further includes a bleed-off valve 36 that releases the pressure oil discharged from the hydraulic pump 31 to the hydraulic oil tank 38. When the controller 60 executes the control of the regulator 31a (flow adjustment device) based on the first target value of the pump flow rate from the calculation result, it simultaneously controls the bleed-off valve 36 to a closed state, whereas when executing control based on the second target value of the pump flow rate, it simultaneously controls the bleed-off valve 36 to maintain a predetermined opening degree.

[0072] According to this configuration, by closing the bleed-off valve 36 during the execution of speed control based on the target speed of the swing hydraulic motor 33 (hydraulic actuator), it is possible to block the outflow of pressure oil from the hydraulic pump 31 to the hydraulic oil tank 38 via the bleed-off valve 36, thereby reducing energy loss. On the other hand, by opening the bleed-off valve 36 during the execution of pressure control based on the target pressure of the swing hydraulic motor 33 (hydraulic actuator), the control of the driving pressure of the swing hydraulic motor 33 (hydraulic actuator) can be easily executed by increasing or decreasing the flow rate of the hydraulic pump 31.

[Other Embodiments]

[0073] In the above-described embodiment, an example of applying the present invention to a hydraulic excavator was shown, but the present invention can be widely applied to various work machines equipped with a rotatable swing body.

[0074] Furthermore, the present invention is not limited to the above-described embodiment and includes various modifications. The above-described embodiments have been detailed to clearly explain the present invention and are not necessarily limited to including all the configurations described. For example, it is possible to replace part of the configuration of one embodiment with the configuration of another embodiment, and it is also possible to add the configuration of another embodiment to the configuration of one embodiment. Additionally, it is possible to add, delete, or replace parts of the configuration of each embodiment with other configurations.

[0075] For example, in the above-described embodiment, an example configuration was shown using a regulator 31a as a flow adjustment device capable of adjusting the discharge flow rate of the hydraulic pump 31, and a bleed-off valve 36 to release the pressure oil discharged from the hydraulic pump 31 to the hydraulic oil tank 38, as a pressure adjustment device capable of adjusting the driving pressure of the swing hydraulic motor 33. However, if the hydraulic pump is of a fixed displacement type, it is also possible to adjust the driving pressure of the swing hydraulic motor 33 by controlling the discharge flow rate of the hydraulic pump by changing the rotational speed of the prime mover 32. That is, the prime mover 32 functions as a flow adjustment device capable of adjusting the discharge flow rate of the hydraulic pump by adjusting its rotational speed, and by adjusting the discharge flow rate of the hydraulic pump, it functions as a pressure adjustment device capable of adjusting the driving pressure of the swing hydraulic motor 33.

Description of Reference Characters

[0076]

2: Upper swing body (swing body)
31: Hydraulic pump
31a: Regulator (pressure adjustment device; flow adjustment device)
32: Prime mover (pressure adjustment device; flow adjustment device)
33: Swing hydraulic motor (hydraulic actuator)
36: Bleed-off valve (pressure adjustment device)
38: Hydraulic oil tank
54: Speed sensor
56: Joystick (operating device)
60: Controller

Claims

1. The work machine comprising:

a hydraulic pump that discharges pressure oil;
a swing body capable of swing operation;
a hydraulic actuator that swing-drives the swing body by supplying pressure oil from the hydraulic pump;
an operating device that outputs an operation signal instructing an operation of the swing body; and
a speed sensor that detects a driving speed of the hydraulic actuator, wherein
the work machine further comprises,
a pressure adjustment device capable of adjusting a driving pressure of the hydraulic actuator, and
a controller that controls the pressure adjustment device; and
the controller is configured to:

calculate a target speed of the hydraulic actuator based on the operation signal from the operating device,
calculate a target pressure of the hydraulic actuator based on a speed deviation, which is a difference between the calculated target speed and an actual driving speed of the hydraulic actuator detected by the speed sensor,
limit the calculated target pressure so that an input horsepower to the hydraulic actuator does not exceed a limit value, the input horsepower being estimated when it is assumed that the driving pressure of the hydraulic actuator reaches the calculated target pressure after a predetermined time, and
control the pressure adjustment device based on the limited target pressure.

2. The work machine according to claim 1, wherein the controller estimates the input horsepower to the hydraulic actuator after the predetermined time by estimating the driving speed of the hydraulic actuator after the predetermined time based on the actual driving speed and an actual driving acceleration of the hydraulic actuator obtained from a detection value of the speed sensor.

3. The work machine according to claim 1, wherein

the pressure adjustment device includes a flow rate adjustment device capable of adjusting the driving pressure of the hydraulic actuator by adjusting a discharge flow rate of the hydraulic pump,
the controller calculates a first target value of the pump flow rate of the hydraulic pump based on the calculated target speed and calculates a second target value of the pump flow rate of the hydraulic pump based on the limited target pressure,
the controller calculates a speed deviation ratio, which is a ratio of the speed deviation to the calculated target speed, and if the calculated speed deviation ratio is smaller than a first threshold, executes controlling the flow rate ad-

justment device based on the first target value of the pump flow rate, and

the controller, if the calculated speed deviation ratio is greater than a second threshold, executes controlling the flow rate adjustment device based on the second target value of the pump flow rate as the control of the pressure adjustment device based on the limited target pressure.

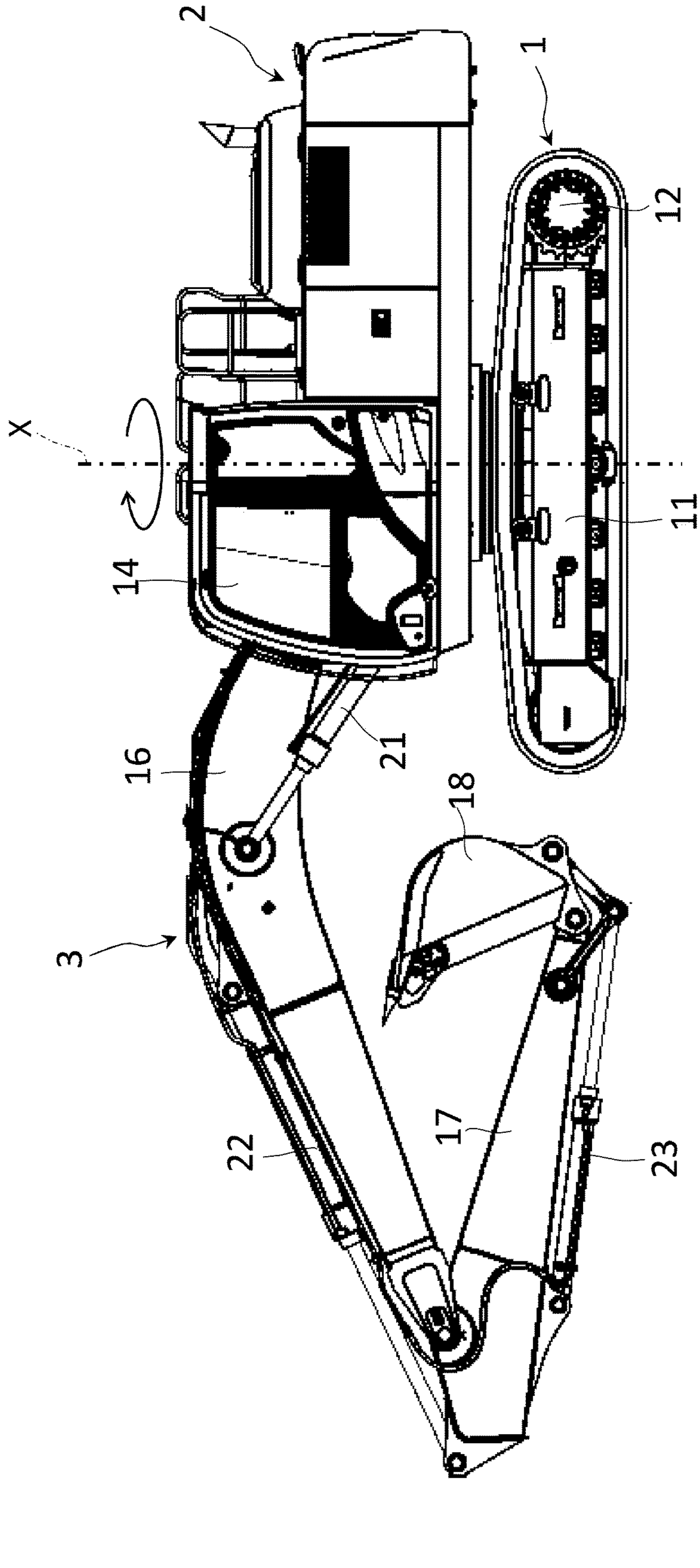
4. The work machine according to claim 3, wherein

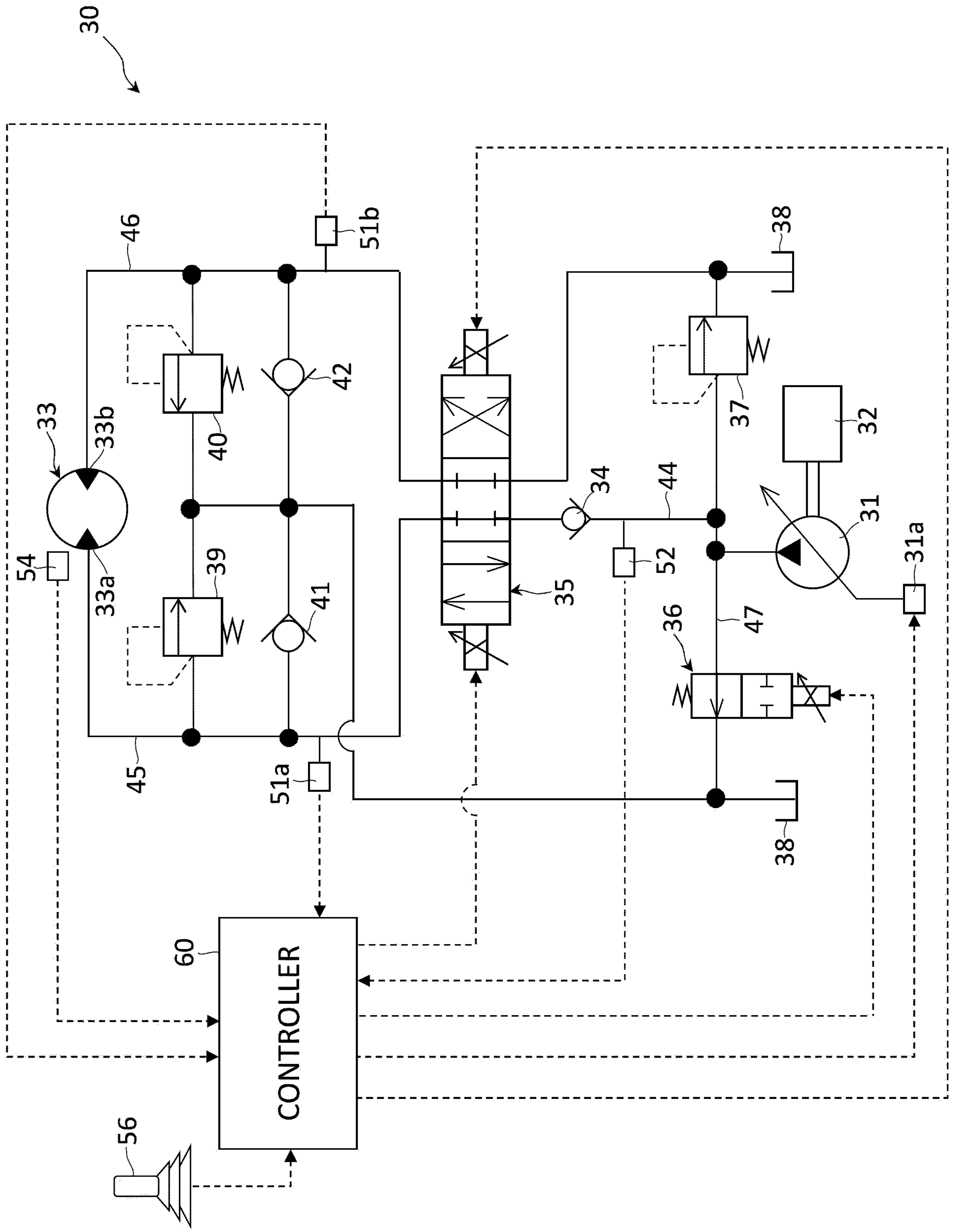
the pressure adjustment device further includes a bleed-off valve that releases the pressure oil discharged from the hydraulic pump to a hydraulic oil tank;

the controller simultaneously controls the bleed-off valve to a closed state in a case executing controlling the flow rate adjustment device based on the first target value of the pump flow rate; and

the controller simultaneously controls the bleed-off valve to be maintained at a predetermined opening degree in a case executing controlling the flow rate adjustment device based on the second target value of the pump flow rate.

[FIG. 1]




30
46
51b
38
42
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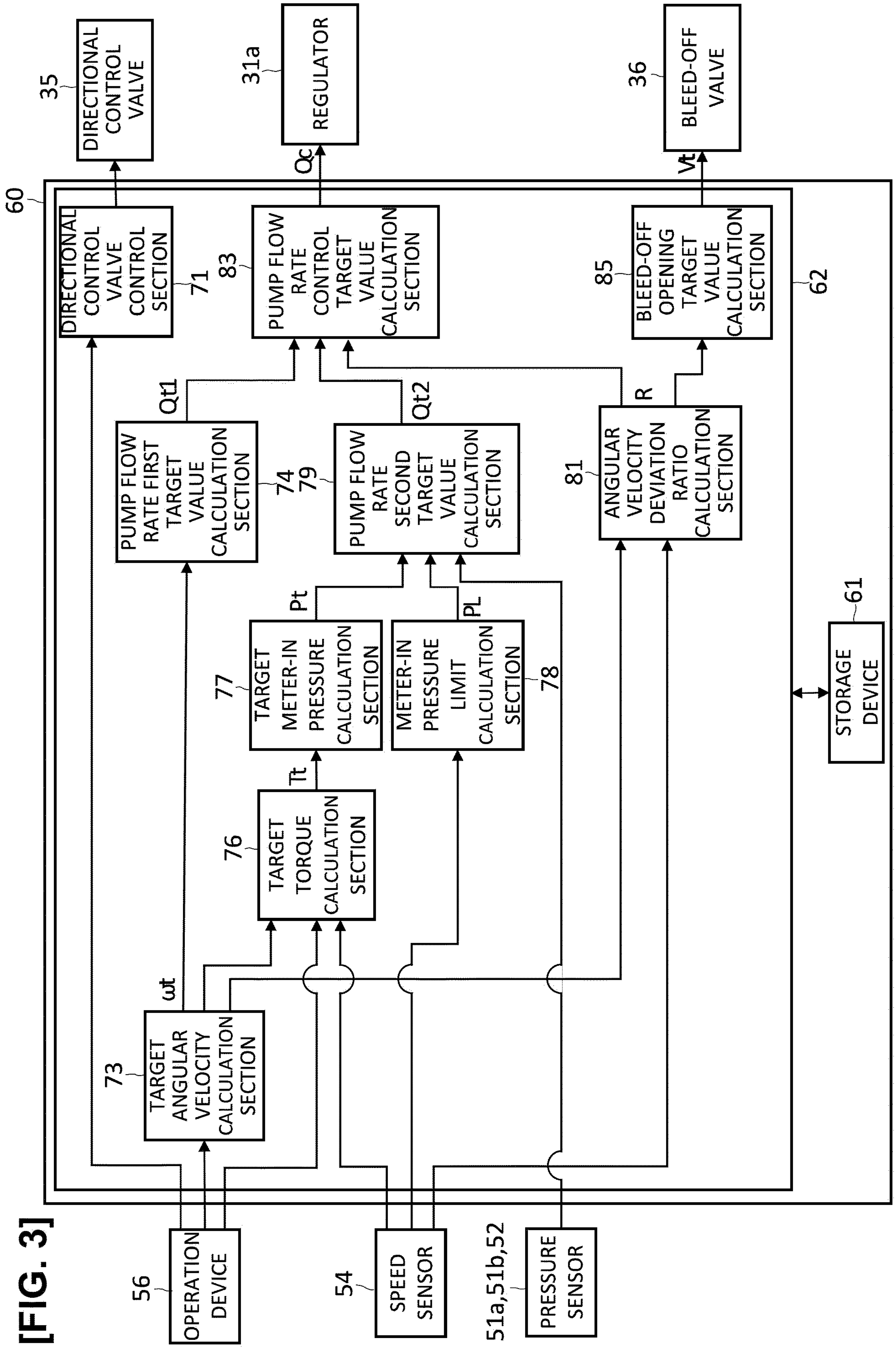
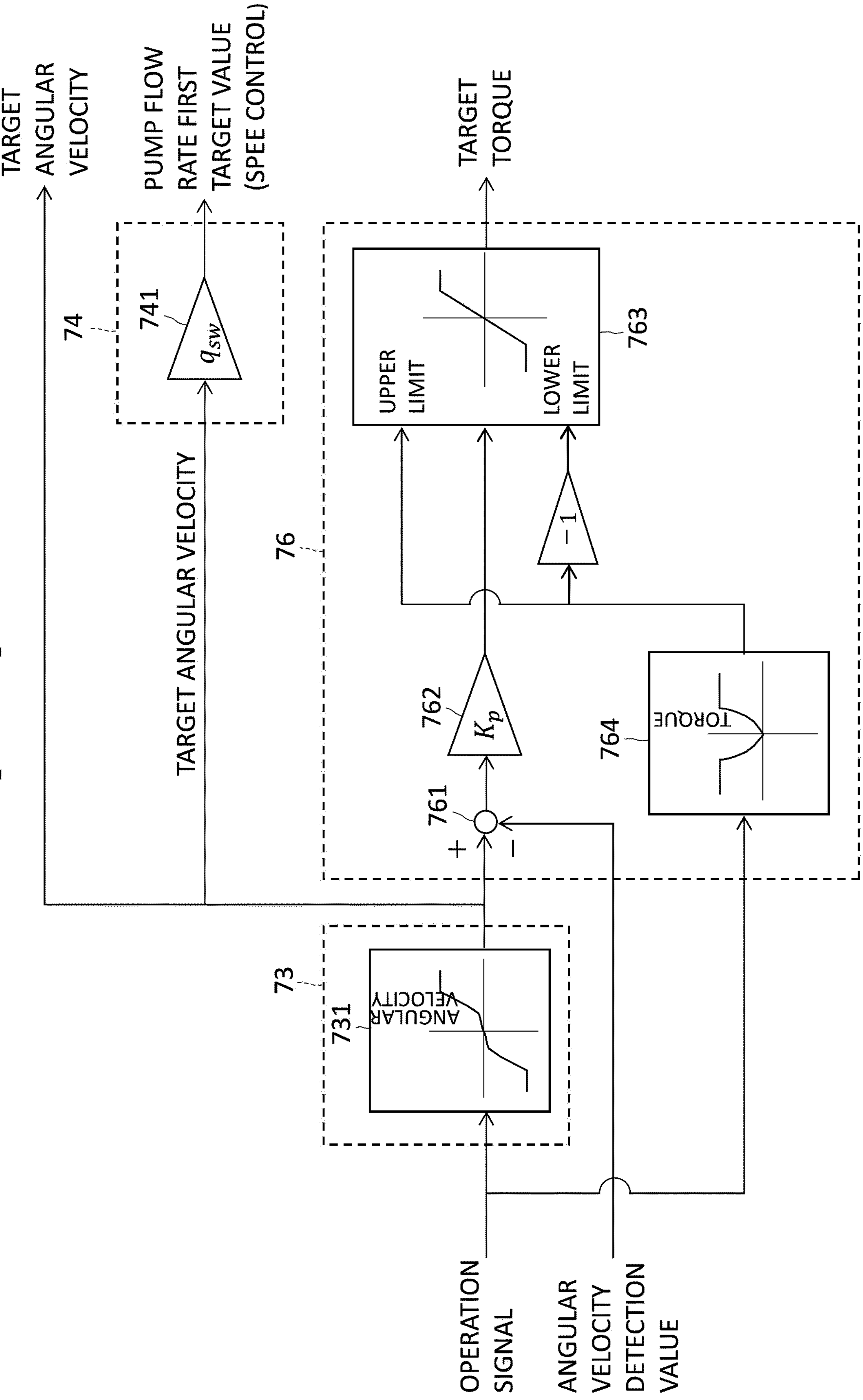
33b
40
37
32
34
44
31
54
33a
39
41
52
31a
35
36
47
45
51a
38
60
CONTROLLER
56

[FIG. 2]

[FIG. 3]


35
DIRECTIONAL CONTROL VALVE
31a
REGULATOR
36
BLEED-OFF VALVE
60
Qc
Vt
DIRECTIONAL CONTROL VALVE CONTROL SECTION
71
83
PUMP FLOW RATE CONTROL TARGET VALUE CALCULATION SECTION
85
BLEED-OFF OPENING TARGET VALUE CALCULATION SECTION
62
Qt1
PUMP FLOW RATE FIRST TARGET VALUE CALCULATION SECTION
74
79
PUMP FLOW RATE SECOND TARGET VALUE CALCULATION SECTION
Qt2
R
81
ANGULAR VELOCITY DEVIATION RATIO CALCULATION SECTION
Pt
PL
77
TARGET METER-IN PRESSURE CALCULATION SECTION
METER-IN PRESSURE LIMIT CALCULATION SECTION
78
61
STORAGE DEVICE
Tt
76
TARGET TORQUE CALCULATION SECTION
ωt
73
TARGET ANGULAR VELOCITY CALCULATION SECTION
56
OPERATION DEVICE
54
SPEED SENSOR
51a,51b,52
PRESSURE SENSOR

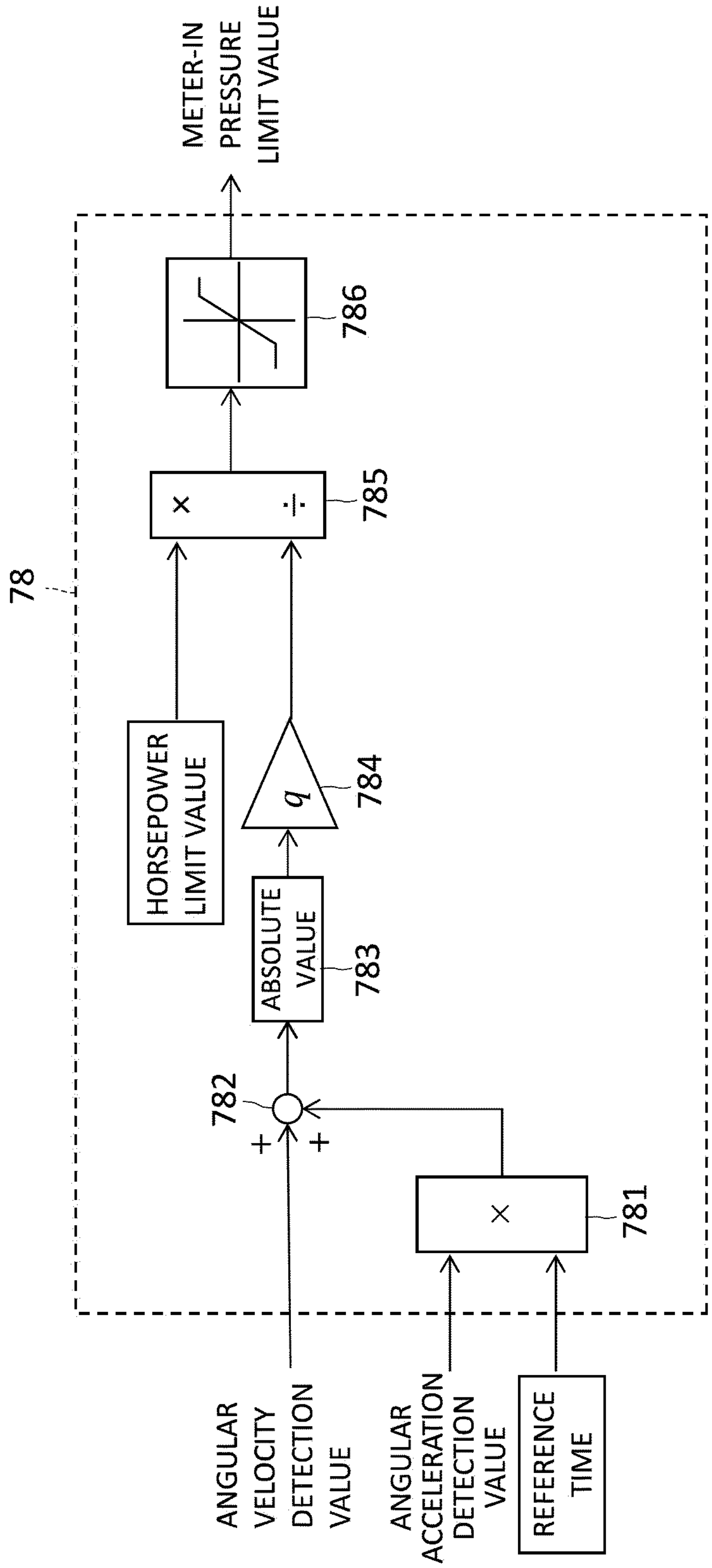
[FIG. 4]


TARGET ANGULAR VELOCITY
PUMP FLOW RATE FIRST TARGET VALUE (SPEE CONTROL)
TARGET TORQUE
74
741
q_{sw}
TARGET ANGULAR VELOCITY
76
UPPER LIMIT
LOWER LIMIT
763
−1
762
K_p
764
TORQUE
761
+
−
73
731
ANGULAR VELOCITY
OPERATION SIGNAL
ANGULAR VELOCITY DETECTION VALUE

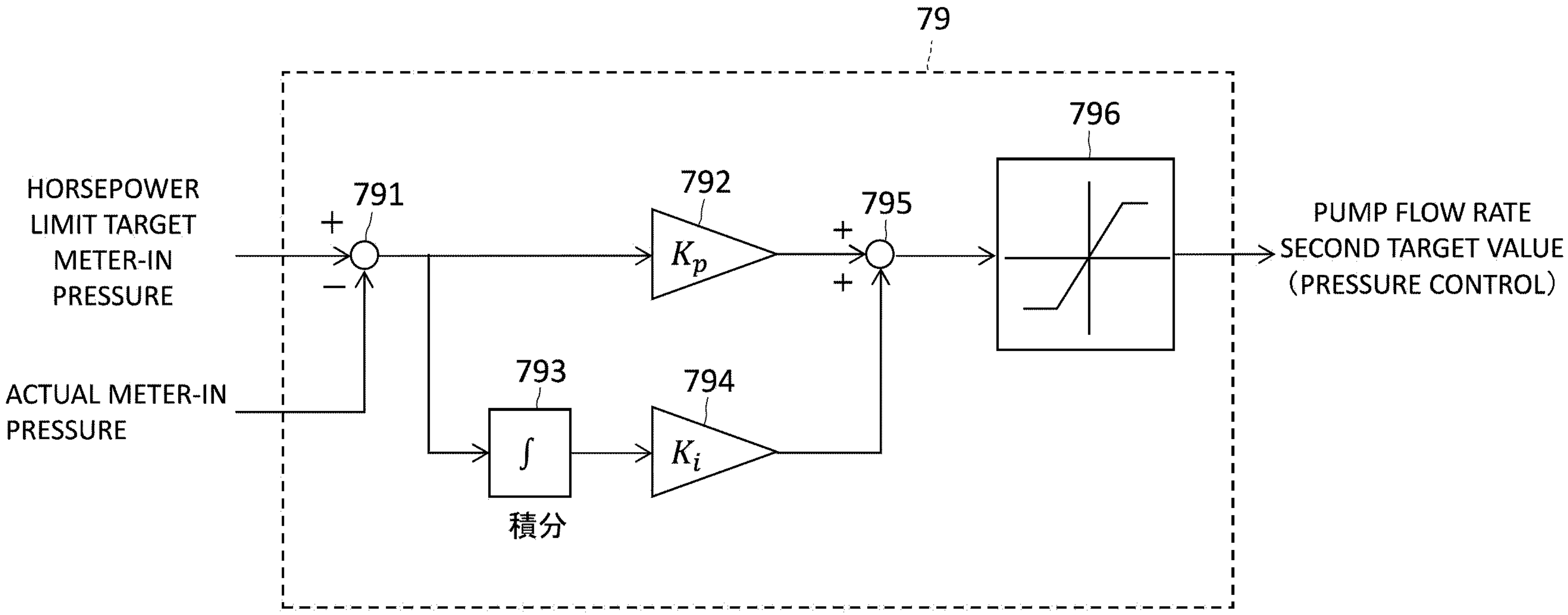
[FIG. 5]

SWING DIRECTION	OPERATION SIGNAL	SIGN OF TARGET TORQUE	TARGET PRESSURE VALUE OF FIRST PORT	TARGET PRESSURE VALUE OF SECOND PORT	METER-IN SIDE
RIGHT	>th1	+ (RIGHTWARD)	Tt/q	0	FIRST PORT
RIGHT	>th1	- (LEFTWARD)	0	-Tt/q	FIRST PORT
	-th1 < O < th1	+	Tt/q	0	(SAME AS LAST TIME)
	-th1 < O < th1	-	0	-Tt/q	(SAME AS LAST TIME)
LEFT	<-th1	+ (RIGHTWARD)	Tt/q	0	SECOND PORT
LEFT	<-th1	- (LEFTWARD)	0	-Tt/q	SECOND PORT

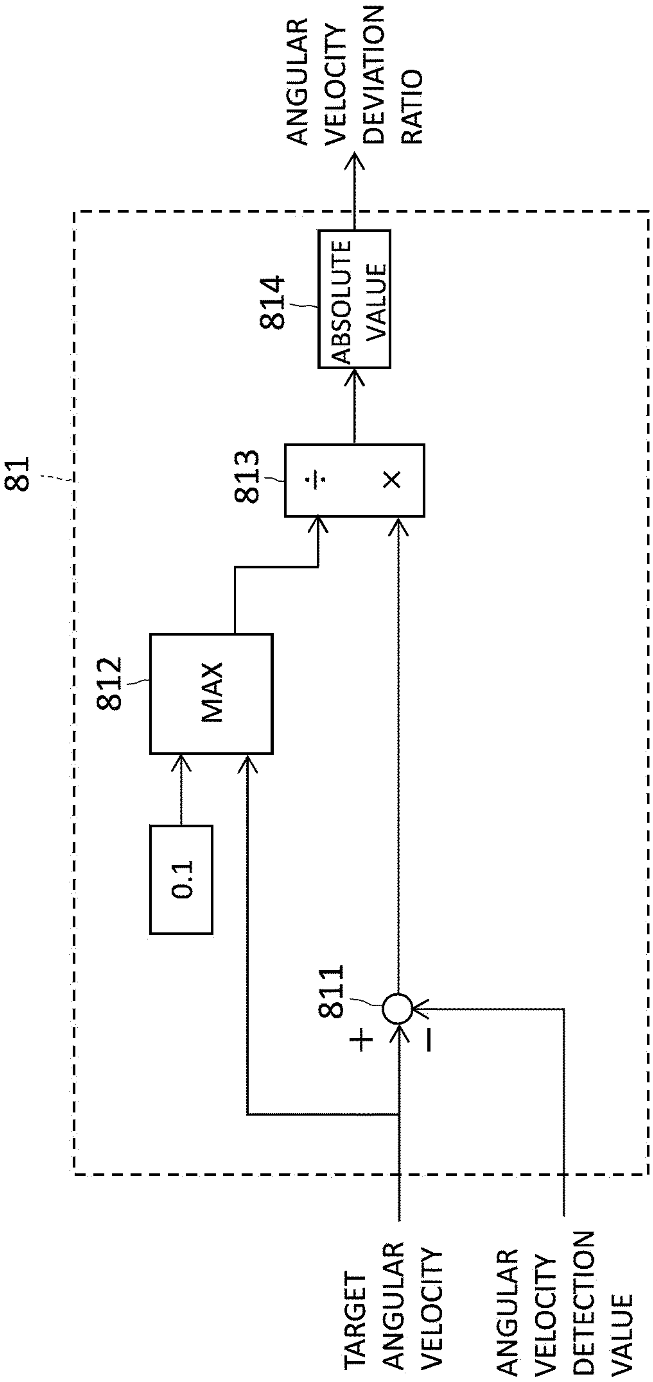
[FIG. 6]


METER-IN PRESSURE LIMIT VALUE
786
785
78
HORSEPOWER LIMIT VALUE
784
ABSOLUTE VALUE
783
782
781
ANGULAR VELOCITY DETECTION VALUE
ANGULAR ACCELERATION DETECTION VALUE
REFERENCE TIME

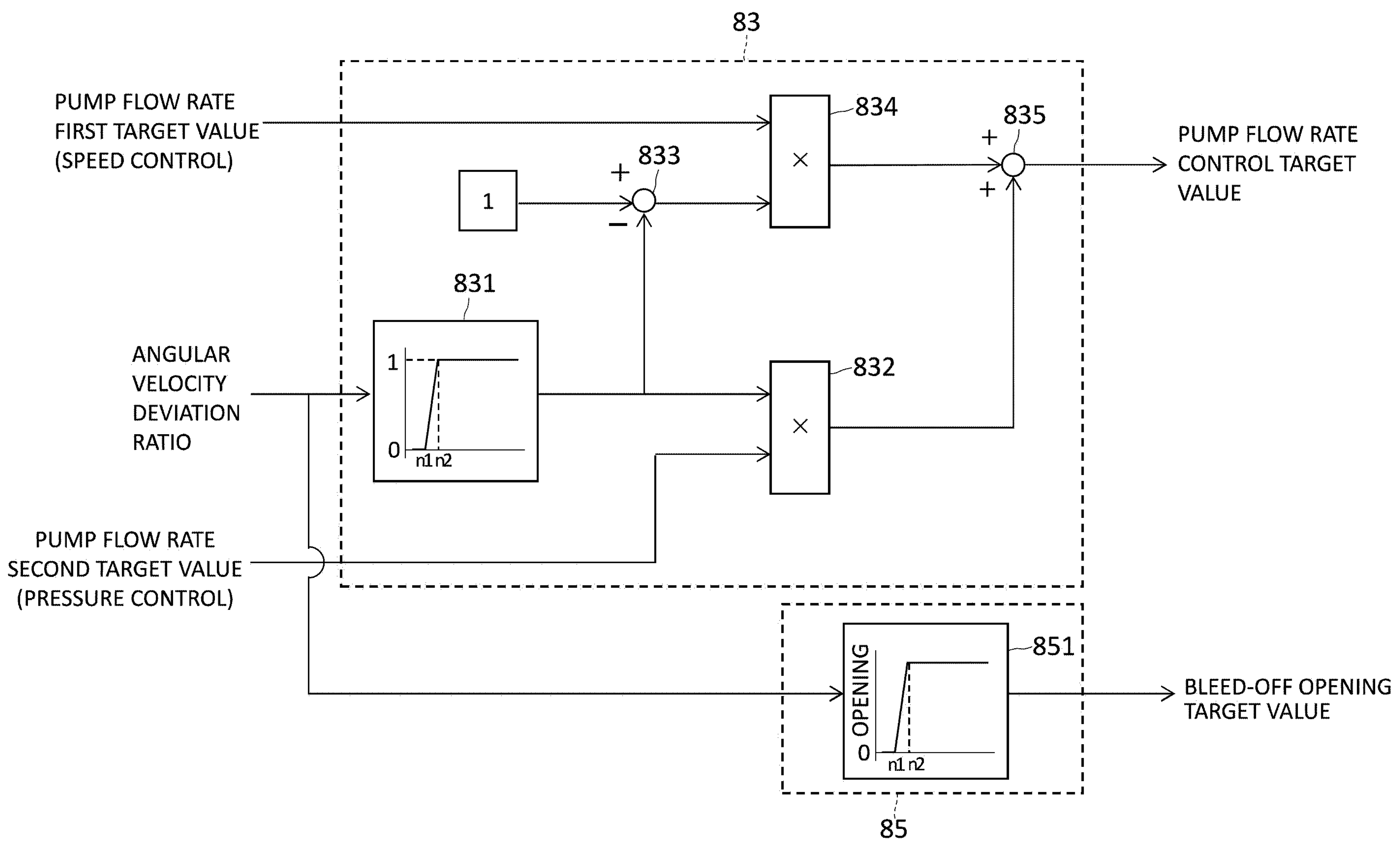
[FIG. 7]


79
HORSEPOWER LIMIT TARGET METER-IN PRESSURE
ACTUAL METER-IN PRESSURE
791
+
−
792
K_p
793
∫
積分
794
K_i
795
+
+
796
PUMP FLOW RATE SECOND TARGET VALUE (PRESSURE CONTROL)

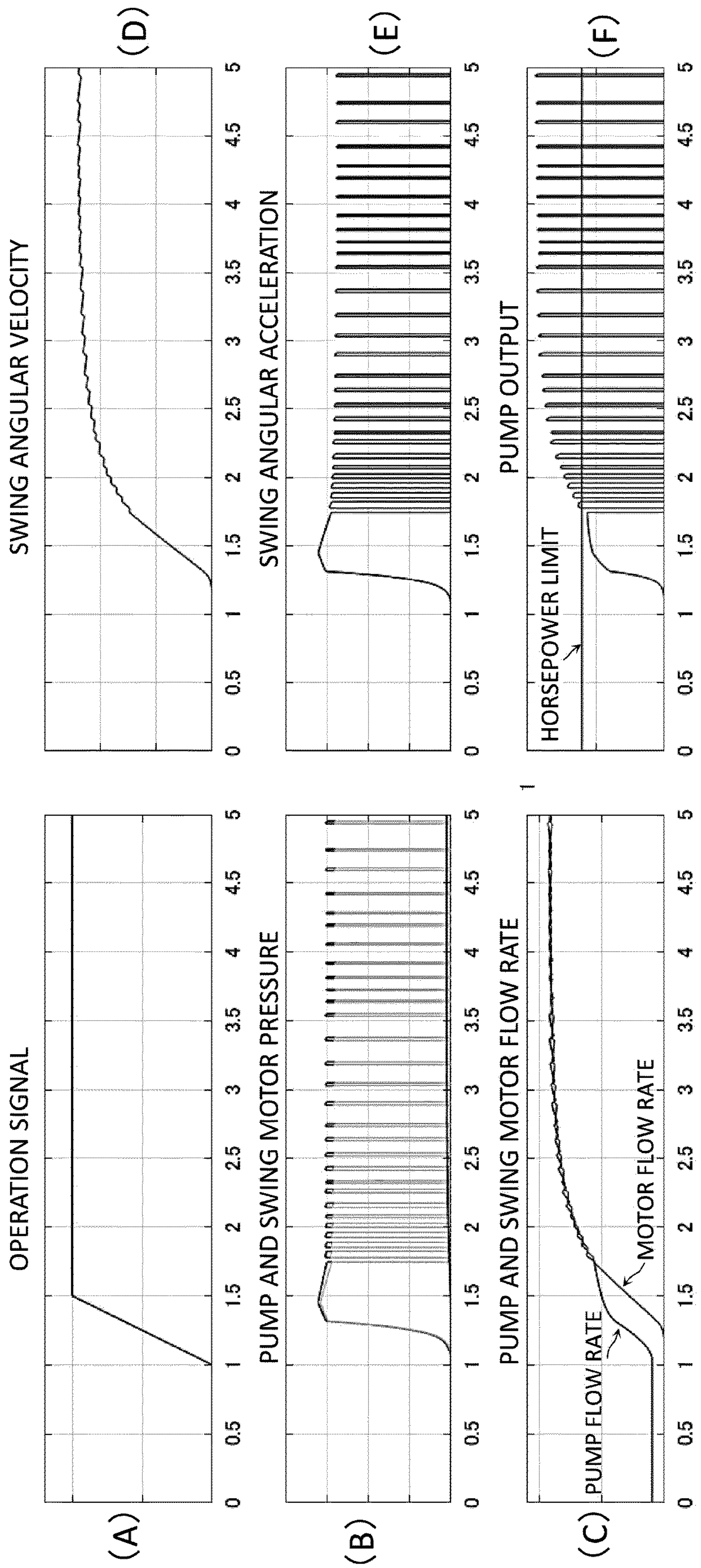
[FIG. 8]


81
811
812
813
814
0.1
MAX
÷
×
ABSOLUTE VALUE
+
−
TARGET ANGULAR VELOCITY
ANGULAR VELOCITY DETECTION VALUE
ANGULAR VELOCITY DEVIATION RATIO

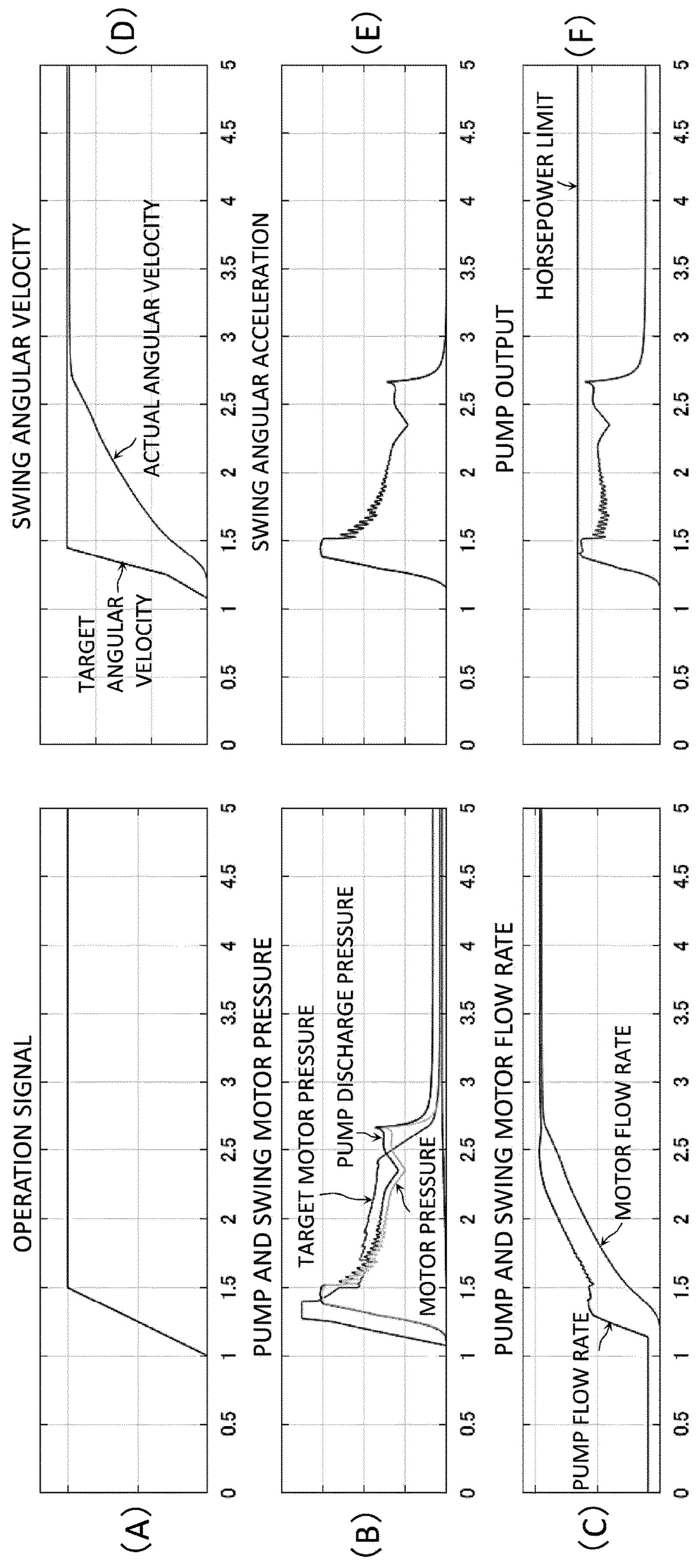
[FIG. 9]


83
PUMP FLOW RATE
FIRST TARGET VALUE
(SPEED CONTROL)
1
+
833
−
834
×
835
+
+
PUMP FLOW RATE
CONTROL TARGET
VALUE
831
1
0
n1 n2
ANGULAR
VELOCITY
DEVIATION
RATIO
832
×
PUMP FLOW RATE
SECOND TARGET VALUE
(PRESSURE CONTROL)
851
OPENING
0
n1 n2
BLEED-OFF OPENING
TARGET VALUE
85

[FIG. 10]


(A)
OPERATION SIGNAL
(B)
PUMP AND SWING MOTOR PRESSURE
(C)
PUMP AND SWING MOTOR FLOW RATE
PUMP FLOW RATE
MOTOR FLOW RATE
(D)
SWING ANGULAR VELOCITY
(E)
SWING ANGULAR ACCELERATION
(F)
PUMP OUTPUT
HORSEPOWER LIMIT

[FIG. 11]


(A)
OPERATION SIGNAL
(B)
PUMP AND SWING MOTOR PRESSURE
TARGET MOTOR PRESSURE
PUMP DISCHARGE PRESSURE
MOTOR PRESSURE
(C)
PUMP AND SWING MOTOR FLOW RATE
PUMP FLOW RATE
MOTOR FLOW RATE
(D)
SWING ANGULAR VELOCITY
TARGET ANGULAR VELOCITY
ACTUAL ANGULAR VELOCITY
(E)
SWING ANGULAR ACCELERATION
(F)
PUMP OUTPUT
HORSEPOWER LIMIT
0 0.5 1 1.5 2 2.5 3 3.5 4 4.5 5

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2023/035674

A. CLASSIFICATION OF SUBJECT MATTER

F15B 11/00(2006.01)i; ***E02F 9/20***(2006.01)i; ***F15B 11/028***(2006.01)i
FI: F15B11/00 F; E02F9/20 M; E02F9/20 Q; F15B11/028 G

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F15B11/00; E02F9/20; F15B11/028

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

Published examined utility model applications of Japan 1922-1996
Published unexamined utility model applications of Japan 1971-2023
Registered utility model specifications of Japan 1996-2023
Published registered utility model applications of Japan 1994-2023

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y A	JP 2016-169570 A (SUMITOMO HEAVY INDUSTRIES) 23 September 2016 (2016-09-23) paragraphs [0010]-[0086], fig. 1-8	1-2 3-4
Y	JP 09-242708 A (KOBE STEEL LTD) 16 September 1997 (1997-09-16) paragraphs [0002], [0078]-[0086], fig. 1	1-2
Y	JP 2012-097890 A (HITACHI CONSTR MACH CO LTD) 24 May 2012 (2012-05-24) paragraph [0049], fig. 2	1-2
A	JP 04-041395 A (KOBE STEEL LTD) 12 February 1992 (1992-02-12) entire text, all drawings	1-4
A	US 2014/0033689 A1 (OPDENBOSCH, Patrick) 06 February 2014 (2014-02-06) entire text, all drawings	1-4

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

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"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
27 November 2023	**12 December 2023**

Name and mailing address of the ISA/JP	Authorized officer
Japan Patent Office (ISA/JP) **3-4-3 Kasumigaseki, Chiyoda-ku, Tokyo 100-8915** **Japan**	Telephone No.

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INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/JP2023/035674

Patent document cited in search report	Publication date (day/month/year)	Patent family member(s)	Publication date (day/month/year)
JP 2016-169570 A	23 September 2016	CN 105971051 A	
JP 09-242708 A	16 September 1997	(Family: none)	
JP 2012-097890 A	24 May 2012	US 2013/0227936 A1 paragraph [0060], fig. 2 WO 2012/060298 A1 EP 2636906 A1 CN 103201522 A KR 10-2013-0143580 A	
JP 04-041395 A	12 February 1992	(Family: none)	
US 2014/0033689 A1	06 February 2014	WO 2014/022152 A1 CN 104520594 A	

Form PCT/ISA/210 (patent family annex) (January 2015)

REFERENCES CITED IN THE DESCRIPTION

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- JP 2013234683 A **[0005]**