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(54) **PUMP CONTROL METHOD AND PUMP CONTROL DEVICE**

(57) Variable delivery main pumps (11) driven by an engine (13) have a capacity control means (35) for changing the discharge rates of the pumps. Power shift pressure output from a proportional control solenoid valve (46) based on signals from a controller (45) is controlled in accordance with the engine speed detected by an engine speed detector (44) and pump delivery pressure of a main pump (11) detected by a pump discharge pressure detector (42). The power shift pressure is in-

troduced into a regulator control valve (39) of the capacity control means (35), and an angle of a swash plate (36) of the main pump (11) is controlled by a regulator (37) so that the pump discharge pressure - discharge rate characteristic is shifted to be optimized. With the configuration as above, the invention provides a pump control device that is capable of reducing the degree of change in the engine speed occurring due to a sudden change in load pressure.

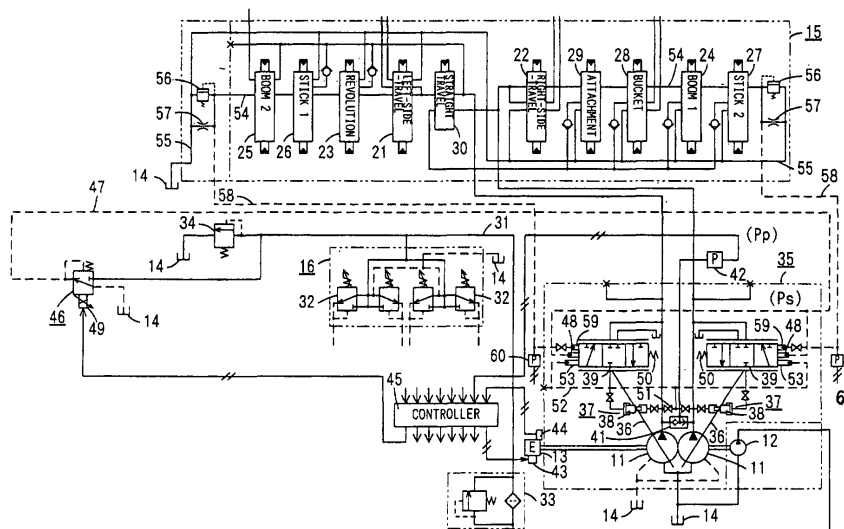


FIG. 1

Description

TECHNICAL FIELD

[0001] The present invention relates to a pump control method for controlling a variable capacity pump. The invention also relates to a pump control device used for such a method.

BACKGROUND OF THE INVENTION

[0002] Examples of conventional control devices for controlling variable capacity pumps of a construction machine, such as a hydraulic excavator, is disclosed in Japanese Patent Provisional Publication No. 1998-220359. The device disclosed in said patent publication functions in such a manner as to set a pump absorbing torque which previously continues stepwise against an engine speed and is crossed at an engine rated point by receiving a signal from a rotation speed sensor for detecting an engine speed, calculate the pump absorbing torque corresponding to the increase and decrease of the engine speed, set the pump absorbing torque to a predetermined value on the basis of the calculated results, and output a command to a regulator control valve so as to adjust the regulators of the variable capacity pump.

[0003] As described above, the conventional control calls for detecting an engine speed to compute a pump absorbing torque and adjust a regulator of a variable capacity pump. Therefore, it presents a problem in that a sudden change in load pressure causes a substantial change in the engine speed.

[0004] In order to solve this problem, an object of the present invention is to provide a pump control method and a pump control device that enable the reduction in the degree of change in the engine speed occurring as a result of a sudden change in load pressure.

DISCLOSURE OF THE INVENTION

[0005] A pump control method according to the present invention calls for detecting a rotation speed of an engine, detecting a pump discharge pressure from a variable capacity pump that is adapted to be driven by said engine and has a capacity control means for changing and thereby controlling the pump discharge rate, and guiding power shift pressure that corresponds to both the engine speed and the pump discharge pressure to the capacity control means, thereby shifting the pump discharge pressure - discharge rate characteristic of the variable capacity pump to the most appropriate value. By detecting a pump discharge pressure from a variable capacity pump and using the detected value as a control signal for controlling the capacity control means, the method described above reduces the degree of change in the engine speed occurring as a result of a sudden change in load pressure, and thereby enables the stable

pump control.

[0006] A pump control method according to another feature of the present invention calls for detecting a rotation speed of an engine, detecting a pump discharge pressure from a variable capacity pump that is adapted to be driven by said engine and has a capacity control means for changing and thereby controlling the pump discharge rate, guiding negative control pressure, which is a pressure generated when the control valve for controlling the working fluid fed from the variable capacity pump to the load is either at the neutral position or in the course of minute operation, to the capacity control means so as to minimize the pump discharge rate, guiding the pump discharge pressure to the capacity control means so as to control at a constant level the pump power fed from the engine to the variable capacity pump, and guiding the power shift pressure corresponding to the engine speed and the pump discharge pressure to the capacity control means, thereby shifting the pump discharge pressure - discharge rate characteristic of the variable capacity pump to the most appropriate value. As a result of the feature described above, the method is effective in controlling the pump discharge rate to a minimum by guiding negative control pressure, which is generated when the control valve is either at the neutral position or in the course of minute operation, to the capacity control means. When the negative control pressure decreases, the pump power fed from the engine to the variable capacity pump is controlled at a constant level with the pump discharge pressure being guided to the capacity control means. Furthermore, by guiding the power shift pressure corresponding to the engine speed and the pump discharge pressure to the capacity control means so as to shift the pump discharge pressure - discharge rate characteristic to the most appropriate value, the method described above reduces the degree of change in the engine speed occurring as a result of a sudden change in load pressure, and, therefore, has the ability to control the capacity control means of the variable capacity pump at the optimal condition according to various circumstances.

[0007] A pump control device according to the present invention includes a capacity control means for changing and thereby controlling the discharge rate of a variable capacity pump adapted to be driven by an engine, an engine speed detecting means for detecting a rotation speed of the engine, a pump discharge pressure detecting means for detecting a discharge pressure from the variable capacity pump, and a power shift control means for guiding power shift pressure, which corresponds to the engine speed and the pump discharge pressure, to the capacity control means so as to shift the pump discharge pressure - discharge rate characteristic of the variable capacity pump to the most appropriate value. By detecting a pump discharge pressure from the variable capacity pump with the pump discharge pressure detecting means and including the detected value in the control signals that are input into the

power shift control means and used for controlling the capacity control means, the invention provides a superior pump control device which is capable of reducing the degree of change in the engine speed occurring as a result of a sudden change in load pressure and thereby ensures the stable pump control without necessitating a substantial change in the existing hardware or the system and merely requiring a modification for including pump discharge pressure in control signals.

[0008] A pump control device according to yet another feature of the present invention includes a capacity control means for changing and thereby controlling the discharge rate of a variable capacity pump adapted to be driven by an engine, an engine speed detecting means for detecting a rotation speed of the engine, a pump discharge pressure detecting means for detecting a discharge pressure from the variable capacity pump, and a power shift control means for guiding power shift pressure, which corresponds to the engine speed and the pump discharge pressure, to the capacity control means so as to shift the pump discharge pressure - discharge rate characteristic of the variable capacity pump to the most appropriate value, a constant power control means for guiding pump discharge pressure to the capacity control means so as to control at a constant level the pump power fed from the engine to the variable capacity pump, and a negative control means for keeping the pump discharge rate at a minimum by guiding negative control pressure to the capacity control means, said negative control pressure being generated when the control valve for controlling the working fluid fed from the variable capacity pump to the load is either at the neutral position or in the course of minute operation. With the configuration as above, the pump discharge rate can be kept at a minimum by guiding negative control pressure, which is generated when the control valve is either at the neutral position or in the course of minute operation, to the capacity control means by the negative control means. When the negative control pressure decreases, the constant power control means guides the pump discharge pressure to the capacity control means and controls at a constant level the pump power fed from the engine to the variable capacity pump. Furthermore, the power shift control means guides the power shift pressure corresponding to the engine speed and the pump discharge pressure to the capacity control means so as to shift the pump discharge pressure - discharge rate characteristic to the most appropriate value, thereby reducing the degree of change in the engine speed occurring as a result of a sudden change in load pressure, and consequently controlling the capacity control means of the variable capacity pump at the optimal condition according to various circumstances.

[0009] According to yet another feature of the present invention, the capacity control means of the pump control device includes a swash plate for adjusting the pump discharge rate, and a mechanical regulator of a fluid pressure actuator type adapted to control the angle of

the swash plate, wherein said mechanical regulator is provided with a piston adapted to function by receiving spring force applied in such a direction as to increase the angle of the swash plate, and a regulator control valve which is of a pilot-operated type and adapted to control the piston in such a direction as to reduce the angle of the swash plate by using the fluid pressure applied against said spring force. With the configuration as above, the stroke of the piston of the mechanical regulator can be controlled by a regulator control valve of a pilot-operated type by guiding signal pressure from the negative control means, the constant power control means and the power shift control means to said regulator control valve. Therefore, the invention provides a superior pump control device without necessitating a substantial change in the existing hardware, and using the existing mechanical regulators unchanged.

[0010] According to yet another feature of the present invention, the power shift control means of the pump control device has a controller for computing the power shift pressure, which is in accordance with the engine speed and the pump discharge pressure, and an electromagnetic proportional action valve for controlling pilot pressure based on electric signals output from said controller, said pilot pressure being input to the regulator control valve of the capacity control means. As the regulator control valve is controlled as desired by means of an electromagnetic proportional action valve which functions according to electric signals from the controller, the device according to the invention is capable of controlling the pump discharge pressure - discharge rate characteristics at an optimal condition regardless of whether the regulator is a conventional mechanical regulator.

BRIEF DESCRIPTION OF THE DRAWINGS

[0011]

Fig. 1 is a hydraulic circuit diagram of a pump control device according to an embodiment of the present invention; Fig 2 is a block diagram showing the function of a controller of said pump control device; Fig. 3 is a characteristic diagram showing engine speed - output power characteristics regarding an underspeed control torque computing portion of said controller; Fig. 4 is a block diagram showing the function of a main pump allowable torque computing portion of said controller; Fig. 5 is a block diagram showing the function of a main pump control portion of said controller; Fig. 6 is a characteristic diagram showing pump discharge pressure - flow rate characteristics to explain conversion between a torque and a first break point computed by said main pump control portion; and Fig 7 is a characteristic diagram showing pump discharge pressure - pump discharge rate characteristics.

PREFERRED EMBODIMENT OF THE INVENTION

[0012] Next, an embodiment of the present invention is explained hereunder, referring to relevant drawings.

[0013] Fig. 1 shows a hydraulic circuit of a hydraulic excavator. The hydraulic excavator (not shown) has a lower structure and a revolving super structure, which is mounted on the lower structure and provided with a pair of main pumps 11, a pilot pump 12, an engine 12 for driving the main pumps 11 and the pilot pump 12, and a tank 14 for containing the working fluid for the hydraulic circuit. The main pumps 11 serve as the variable capacity pumps referred to in the claims and the disclosure of the invention and function to feed hydraulic fluid to right and left hydraulic traveling motors, a hydraulic rotating motor and hydraulic cylinders of a front attachment.

[0014] The hydraulic excavator is also provided with a hydraulic circuit for controlling actions of the right and left hydraulic traveling motors, the hydraulic rotating motor, and the hydraulic cylinders of the front attachment.

[0015] The hydraulic circuit includes a control valve 15, a pilot valve, and a hydraulic line, which is a pipe arrangement for connecting these valves. The control valve 15 controls the directions and the flow rates of the hydraulic fluid fed from the two main pumps 11 to the right and left hydraulic traveling motors, the hydraulic rotating motor and the hydraulic cylinders of the front attachment. The pilot valve (hereinafter called the remote control valve 16) functions to remotely control the control valve 15 by using pilot hydraulic pressure fed from the pilot pump 12.

[0016] The control valve 15 has various stems, i.e. spools, for controlling the directions and the flow rates of the hydraulic fluid: a left-side-travel control stem 21 for controlling the left hydraulic traveling motor, a right-side-travel control stem 22 for controlling the right hydraulic traveling motor, a revolution control stem 23 for controlling the hydraulic rotating motor, a first boom control stem 24 and a second boom control stem 25 for controlling the hydraulic boom cylinders, a first stick control stem 26 and a second stick control stem 27 for controlling the hydraulic stick cylinders, a bucket control stem 28 for controlling the hydraulic bucket cylinder, and an attachment control stem 29 for controlling an attachment that is attached to the front end of the stick in the place of the bucket. These stems are evenly arranged in the two groups respectively associated with the two main pumps. The control valve 15 also includes a straight-travel control stem 30 for driving the hydraulic excavator straight by feeding the hydraulic fluid from just one of the main pumps 11 to only the left-side-travel control stem 21 and the right-side-travel control stem 22 so as to drive the right and left hydraulic traveling motors at an equal speed.

[0017] The aforementioned remote control valve 16, only a part of which is shown in the drawing, has pressure reducing valves 32, which are adapted to be man-

ually operated by an operator in the cab of the hydraulic operator with operation levers so as to reduce the pilot pressure fed from the pilot pump 12 through a pilot pressure line 31. The pilot pressure controlled by the pressure reducing valves 32 is then fed to pressure chambers provided at the ends of the respective stems of the control valve 15. A filter unit 33 and a relief valve 34, which functions to maintain the pilot pressure at a set level, are disposed in the pilot pressure line 31.

[0018] Each one of the two main pumps 11 mentioned above includes a capacity control means 35 that is adapted to change the discharge rate of the pump and has a swash plate 36 and a mechanical regulator 37 of a fluid pressure actuator type. Each swash plate 36 is adapted to control the displacement volume of the corresponding pump in accordance with the angle of the swash plate, thereby controlling the discharge rate of the pump. Each mechanical regulator 37 functions to control the angle of the swash plate 36 associated therewith.

[0019] Each mechanical regulator 37 has a piston 38 and a pilot-operated regulator control valve 39. Each piston 38 functions by receiving spring force applied in such a direction as to increase the angle of the corresponding swash plate 36. Each regulator control valve 39 is integrated in a regulator main body that includes the piston 38 associated therewith and adapted to control the piston 38 in such a direction as to reduce the angle of the swash plate 36 by using the fluid pressure against the aforementioned spring force.

[0020] One end of each piston 38 receives the pump discharge pressure of the associated main pump 11, while the controlled pressure, i.e. the pump discharge pressure controlled by the associated regulator control valve 39, and the spring force are applied to the other end of the piston 38. Each swash plate 36 is provided with a swash plate position detector (not shown) for detecting its angle, i.e. the position of the swash plate.

[0021] A pump discharge pressure detector 42, which serves as a pump discharge pressure detecting means for detecting the higher of the discharge pressures of the two main pumps 11, is connected to the discharge lines of the main pumps 11 via a shuttle valve 41 for retrieving said higher discharge pressure.

[0022] The engine 13 has a governor 43, which is adapted to control the rotation speed of the engine (hereinafter referred to as "engine rotation speed" or "engine speed"), and an engine speed detector 44 serving as an engine rotation speed detecting means for detecting the engine speed. The target rotation speed of the engine 13 is set by using an axel dial, which serves as a rotation speed setting means, or an underspeed setting means for setting an underspeed with respect to the rated engine speed.

[0023] The hydraulic circuit also includes a controller 45 and a proportional control solenoid valve 46 that together function as a power shift control means for shifting the main pumps 11 so as to maintain the optimal

pump discharge pressure - discharge rate characteristics by guiding power shift pressure P_s to the capacity control means 35 for the main pumps 11, said power shift pressure P_s being in accordance with the engine speed and the pump discharge pressure respectively detected by the engine speed detector 44 and the pump discharge pressure detector 42 mentioned above. The controller 45 is adapted to compute the power shift pressure P_s based on the engine speed and the pump discharge pressure. The proportional control solenoid valve 46, which serves as an electromagnetic proportional action valve, controls the power shift pressure P_s , i.e. pilot pressure to be input to the regulator control valves 39 in accordance with electric signals output from the controller 45.

[0024] The aforementioned pilot pressure line 31 is connected to a primary port of the proportional control solenoid valve 46, while its secondary port communicates through a power shift pressure line 47 with pilot pressure inlets 48 of the two regulator control valves 39 of the capacity control means 35.

[0025] Based on electric signals from the controller 45 to its solenoid 49, the proportional control solenoid valve 46 performs proportional control of the pilot pressure, which is controlled at a constant level by the aforementioned relief valve 34, and guides the pilot pressure to the pilot pressure inlets 48 of the regulator control valves 39, thereby displacing the regulator control valves 39 against the force of springs 50. By thus displacing the regulator control valves 39, the proportional control solenoid valve 46 precisely controls the strokes of the pistons 38 of the mechanical regulators 37 so as to rotate the swash plates 36 to desired angles.

[0026] A line 52 is drawn from the discharge line of each main pump 11 through a valve 51 and communicates with another pilot pressure inlet 53 of each respective regulator control valve 39 of the capacity control means 35. Each line 52 serves as a constant power control means for guiding the pump discharge pressure of the associated main pump 11 to the corresponding regulator control valve 39 of the capacity control means 35 and thereby controlling the pump power, which is fed from the engine 13 to the main pump 11, at a constant level.

[0027] The hydraulic circuit also includes a negative control means for controlling the pump discharge rate at the minimum level by guiding to the capacity control means 35 a negative control pressure generated when the control valve 15 is either at the neutral position or in the course of minute operation, said control valve 15 having the function of controlling the working fluid fed from the main pumps 11 to the various loads, i.e. the hydraulic actuators. To be more specific, center bypass lines 54 respectively associated with the aforementioned two groups are provided, each center bypass line 54 being so formed as to communicate with the tank 14 via the stems 21 - 30 associated therewith when these stems 21 - 30 of the control valve 15 are either at the

neutral position or in the course of minute operation. A relief valve 56 and a throttle 57 are disposed at the border between each center bypass line 54 and a tank line 55 that communicates with the tank 14. A negative control line 58 is drawn out from the upstream side of the throttle 57 and communicates with yet another pilot pressure inlet 59 of the regulator control valve 39 associated therewith. A pressure detector 60 for detecting negative control pressure is disposed in each negative control line 58 and connected to the aforementioned controller 45.

[0028] As described above, the mechanical regulators 37 according to the embodiment use the existing components as they are to guide signal pressure from the negative control means, the constant power control means and the power shift control means to the regulator control valves 39 of a pilot-operated type, and control the pistons 38 of the mechanical regulators 37 with said regulator control valves 39, thereby controlling the angles of the swash plates 36 so that the displacement volumes of the pumps are controlled based on the angles of the swash plates 36.

[0029] In addition to the aforementioned pump discharge pressure detector 42, engine speed detector 44 and swash plate position detectors, other sensors, such as operation-degree detectors for detecting degree of operation of the operation levers, flow rate detectors for detecting pump discharge rates of the main pumps 11, load pressure detector for detecting load pressure of the hydraulic actuators, etc., may be provided when it is necessary. Output from these detectors is input as control data signals into the controller 45.

[0030] Next, the configuration of the controller 45 is explained hereunder.

[0031] As shown in Fig. 2, the pump control system of the controller 45 can principally be divided into three modules: a hydraulic circuit control section 61, an engine speed control section 62 and a main pump control section 63.

[0032] Based on temperature signals TEMP indicating the temperature of the hydraulic fluid, detection signals SWim from implement switches adapted to detect whether joysticks for revolving action and the front attachment of the hydraulic excavator have been operated and detection signals SWtr from travel switches adapted to detect whether travel levers for travel of the hydraulic excavator have been operated, the hydraulic circuit control section 61 computes the required flow rate Q required by the hydraulic circuit.

[0033] The engine speed control section 62 includes a control status determining portion 64, an underspeed control torque computing portion 65, an engine stall prevention torque computing portion 66 and a main pump allowable torque computing portion 67. Based on various factors including the power mode PM, the work mode WM, the set engine speed N_{ac} set by an axel dial, the engine speed N detected by the engine speed detector 44 and the pump discharge pressure P_p detected

by the pump discharge pressure detector 42 after selection of the higher discharge pressure by the shuttle valve 41, the engine speed control section 62 determines the main pump allowable torque $T_{mpallow}$, i.e. the magnitude of the torque that can be used by the main pumps 11.

[0034] The control status determining portion 64 is connected to the aforementioned computing portions 65 to 67 and has the ability of determining control statuses based on various signals input from the axel dial, the engine speed sensor, etc., when the engine speed is controlled. Examples of control statuses determined by the control statuses determining portion 64 include whether or not to activate the engine stall prevention function or the underspeed control function, which is the ability to maintain the engine speed at the proximity of the rated engine speed.

[0035] As shown in Fig. 3, the rated engine speed or the rated power is a discontinuity point on the border between the governor range, where the engine output characteristics are controlled by the governor, and the lagging range, where the engine output characteristics are not controlled by the governor. Therefore, in order to ensure stable driving, the underspeed control torque computing portion 65 is adapted to set an underspeed N_{us} and compute an underspeed control torque to make the engine speed that is less than the rated speed by the degree of this underspeed N_{us} as the target speed, in other words to move the target rotation speed in the lagging range shown in Fig. 3 leftward by the distance equivalent to the underspeed N_{us} . The underspeed control torque computing portion 65 uses as input signals a set rotation speed, which has been set by means of the axel dial, and an engine rotation speed detected by the engine speed detector 44.

[0036] Basically, the underspeed control torque computing portion 65 performs underspeed control of the engine 13 by controlling the regulators 37 of the main pumps 11 so as to control the pump absorbing torque, i.e. the load torque which the main pumps 11 absorb in the form of (pump discharge pressure) X (pump discharge rate) from the engine output torque.

[0037] The aforementioned engine stall prevention torque computing portion 66 computes an engine stall prevention torque based on an engine rotation speed detected by the engine speed detector 44 so as to prevent engine stalling caused by a load.

[0038] The main pump allowable torque computing portion 67 functions to determine a main pump allowable torque $T_{mpallow}$, i.e. the magnitude of the torque that can be used by the two main pumps 11, based on an acceleration torque set by the axel dial, an underspeed control torque output from the underspeed control torque computing portion 65 and an engine stall prevention torque output from the engine stall prevention torque computing portion 66.

[0039] As shown in Fig. 4, the main pump allowable torque computing portion 67 computes a static torque

T_s by adding an engine stall prevention torque T_{as} computed by the engine stall prevention torque computing portion 66 to a standard torque T_t by means of an adder 71, computes a dynamic torque T_d by adding an acceleration torque to said static torque T_s by means of an adder 72, computes a main pump allowable torque $T_{mpallow}$ by adding an underspeed control torque T_{us} computed by the aforementioned underspeed control torque computing portion 65 to said dynamic torque T_d by means of an adder 73, and outputs the main pump allowable torque $T_{mpallow}$ to the aforementioned main pump control section 63.

[0040] As shown in Fig. 5, the main pump control section 63 converts the required flow rate Q (%), which has been output from the aforementioned hydraulic circuit control section 61, to the main pump required torque T_{mpreq} (%) by a torque converter 74 based on the first bent portion pressure, i.e. the pump discharge pressure at the first break point in the pump discharge pressure - flow rate characteristic diagram of Fig. 6.

[0041] A main pump required torque T_{mpreq} is a torque that is determined in accordance with the condition of the load applied to a main pump 11 and required by the pump, while a main pump allowable torque $T_{mpallow}$ is a torque that is allowed in accordance with the condition of the load applied to the engine 13. Of the main pump required torque T_{mpreq} and the main pump allowable torque $T_{mpallow}$, the smaller of the aforementioned torques is the pump absorbing torque that is actually used by the main pump 11.

[0042] The main pump allowable torque $T_{mpallow}$ output from the engine speed control section 62 and the main pump required (requiring?) torque T_{mpreq} are input into a torque selecting means 75, which then chooses the smaller torque (in other words finds the pump absorbing torque). The pump absorbing torque is converted into a first break point pressure by a converter 76. By a converter 77, the first break point pressure is converted into a power shift pressure P_s , which is then converted by a converter 78 into an input signal (a control current) I_{ps} for power shifting, a signal which is required by the solenoid 49 of the proportional control solenoid valve 46 to output the power shift pressure P_s .

[0043] Therefore, as shown in Fig. 2, when a power shift input signal I_{ps} is input into the solenoid 49 of the proportional control solenoid valve 46, the proportional control solenoid valve 46 outputs the power shift pressure P_s , which serves as the regulator control pressure signal computed, and the corresponding regulator control valve 39 is controlled by said power shift pressure P_s . As a result of control of the regulator control valve 39, the swash plate 36 of the corresponding main pump 11 is controlled.

[0044] Conventional pump control is performed by detecting an engine speed. As described above, however, the control method according to the present invention also calls for detecting a pump discharge pressure P_p (or an actuator load pressure) and inputting the detected

pressure into the control means, thereby enabling the control of the power required by the pump while using a mechanical regulator 37 of a conventional two-stage spring type.

[0045] To be more precise, by inclusion of a pump discharge pressure detector 42 explained above, the control means according to the invention is capable of reducing the difference between a target control torque and an actual torque by detecting a pump discharge pressure P_p and controlling the power shift pressure P_s so as to optimize the P-Q characteristic of each mechanical regulator 37 with respect to a regulator driving unit of a conventional two-stage spring type, which is used for controlling a swash plate intended for controlling each pump of a hydraulic excavator.

[0046] In other words, as shown in Fig. 7, the constant power control means controls the regulators 37 so that the relationship between the pump discharge pressure P_p and the pump discharge rate Q of each pump (said relationship is hereinafter referred to as the pump P-Q characteristic) changes on a specific curve representing a constant pump power. Meanwhile, the controller 45 of the power shift control means computes the pump absorbing torque, which is the torque to be corrected, based on the difference in the rotation speed between the target engine speed and the actual engine speed, and outputs the corresponding power shift electric signal I_{ps} , thereby controlling the proportional control solenoid valve 46 and consequently controlling the power shift pressure P_s to shift the pump P-Q characteristic from the specific constant pump power curve to another curve. As a result, the bent of the spring 50 used by each mechanical regulator 37 is corrected so that the aforementioned difference in the engine speed is consequently adjusted. When increasing the pump power, the break point is moved to a constant pump power curve located towards the upper right corner as viewed in Fig. 7.

[0047] Next, the function of the embodiment shown in Fig. 1 is explained hereunder based on the control details described above.

[0048] When each one of the stems 21 to 30 of the control valve 15 associated with one of the aforementioned two groups is either at the neutral position or in the course of minute operation, the negative control means guides the negative control pressure that has been generated at the upstream side of the throttle 57 of the center bypass line 54 associated with said group, through the negative control line 58 to the pilot pressure inlet 59 of the regulator control valve 39, and the regulator 37 controls the swash plate 36 to minimize the pump discharge rate.

[0049] When the negative control pressure is reduced by changing the position of any stem 21 to 30, the regulator control valve 39 is controlled by the pump discharge pressure P_p guided through the line 52 of the constant power control means to the pilot pressure inlet 53 of the regulator control valve 39 so that the regulator

37 controls the angle of the swash plate 36 to maintain the constant pump power (or the pump absorbing torque) fed from the engine 13 to the main pump 11. In other words, with a change in the pump discharge pressure P_p shown in Fig. 7, the angle of the swash plate 36 associated therewith is controlled by the regulator 37 so that the pump discharge rate Q changes along a single constant pump power curve.

[0050] The power shift pressure P_s according to the engine speed N and the pump discharge pressure P_p , which have respectively been detected as above, is computed by the controller 45, and the proportional control solenoid valve 46 of the power shift control means is controlled based on the control signals resulting from said computation. The power shift pressure P_s , i.e. the pilot pressure reduced by the proportional control solenoid valve 46, is guided to the pilot pressure inlets 48 of the regulator control valves 39, and the regulator 37 controls the swash plate 36 so as to maintain the optimal pump discharge pressure - discharge rate characteristics. In other words, the pump power is shifted from a constant pump power curve to another constant pump power curve as viewed in Fig. 7.

[0051] Conventional constant power control is performed by detecting and feeding back an engine speed N . As described above, however, the control device according to the present invention is adapted to also detect a pump discharge pressure P_p output from a main pump 11 and include the detected pump discharge pressure P_p in control data signals for controlling the capacity control means 35. Therefore, the control device according to the present invention is capable of reducing the change in the rotation speed of the engine 13, which change occurs as a sudden change in load pressure.

[0052] While holding down cost increases, the invention offers a good control device that does not necessitate an expensive control means or a substantial change in the system and merely require an input of a detected pump discharge pressure P_p as a control data signal by using the existing hardware as it is.

[0053] The application of a control method or a control device according to the invention is not limited to control of swash plate pumps, and the range of their application includes any other pumps having similar structures, including pumps of a bent axis type.

POSSIBLE INDUSTRIAL APPLICATION

[0054] A control method and a control device according to the invention are intended to reduce the degree of change in the engine speed resulting from a sudden change in load pressure in a variable capacity pump and are applicable to not only a construction machine but also work machines of other types and stationary industrial machines, provided that the machine calls for driving a variable capacity pump by using an engine.

Claims**1.** A pump control method that calls for:

detecting a rotation speed of an engine; 5
 detecting a pump discharge pressure from a
 variable capacity pump that is adapted to be
 driven by said engine and has a capacity con-
 trol means for changing and thereby controlling
 the pump discharge rate; and 10
 guiding power shift pressure that corresponds
 to both the engine speed and the pump dis-
 charge pressure to the capacity control means,
 thereby shifting the pump discharge pressure -
 discharge rate characteristic of the variable ca- 15
 pacity pump to the most appropriate value.

2. A pump control method that calls for:

detecting a rotation speed of an engine; 20
 detecting a pump discharge pressure from a
 variable capacity pump that is adapted to be
 driven by said engine and has a capacity con-
 trol means for changing and thereby controlling
 the pump discharge rate; 25
 guiding negative control pressure to the capac-
 ity control means so as to minimize the pump
 discharge rate, said negative control pressure
 being a pressure generated when the control
 valve for controlling the working fluid fed from 30
 the variable capacity pump to the load is either
 at the neutral position or in the course of minute
 operation;
 guiding the pump discharge pressure to the ca- 35
 pacity control means so as to control at a con-
 stant level the pump power fed from the engine
 to the variable capacity pump; and
 guiding power shift pressure that corresponds 40
 to both the engine speed and the pump dis-
 charge pressure to the capacity control means,
 thereby shifting the pump discharge pressure -
 discharge rate characteristic of the variable ca-
 pacity pump to the most appropriate value.

3. A pump control device including: 45

a capacity control means for changing and
 thereby controlling the discharge rate of a var-
 iable capacity pump adapted to be driven by an
 engine; 50
 an engine speed detecting means for detecting
 a rotation speed of the engine;
 a pump discharge pressure detecting means
 for detecting a discharge pressure from the var-
 iable capacity pump; and 55
 a power shift control means for guiding power
 shift pressure, which corresponds to the engine
 speed and the pump discharge pressure, to the

capacity control means so as to shift the pump
 discharge pressure - discharge rate character-
 istic of the variable capacity pump to the most
 appropriate value.

4. A pump control device including:

a capacity control means for changing and
 thereby controlling the discharge rate of a var-
 iable capacity pump adapted to be driven by an
 engine;
 an engine speed detecting means for detecting
 a rotation speed of the engine;
 a pump discharge pressure detecting means
 for detecting a discharge pressure from the var-
 iable capacity pump;
 a power shift control means for guiding power
 shift pressure, which corresponds to the engine
 speed and the pump discharge pressure, to the
 capacity control means so as to shift the pump
 discharge pressure - discharge rate character-
 istic of the variable capacity pump to the most
 appropriate value;
 a constant power control means for guiding
 pump discharge pressure to the capacity con-
 trol means so as to control at a constant level
 the pump power fed from the engine to the var-
 iable capacity pump; and
 a negative control means for keeping the pump
 discharge rate at a minimum by guiding nega-
 tive control pressure to the capacity control
 means, said negative control pressure being
 generated when the control valve for controlling
 the working fluid fed from the variable capacity
 pump to the load is either at the neutral position
 or in the course of minute operation.

5. A working fluid supply control device as claimed in
 claim 3 or claim 4, wherein:

the capacity control means includes:

a swash plate for adjusting the pump dis-
 charge rate, and
 a mechanical regulator of a fluid pressure
 actuator type adapted to control the angle
 of the swash plate,

said mechanical regulator provided with:

a piston adapted to function by receiving
 spring force applied in such a direction as
 to increase the angle of the swash plate,
 and
 a regulator control valve which is of a pilot-
 operated type and adapted to control the
 piston in such a direction as to reduce the
 angle of the swash plate by using the fluid

pressure applied against said spring force.

6. A working fluid supply control device as claimed in claim 5, wherein:

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the power shift control means includes:

a controller for computing the power shift pressure, which is in accordance with the engine speed and the pump discharge pressure, and

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an electromagnetic proportional action valve for controlling pilot pressure based on electric signals output from said controller, said pilot pressure being input to the regulator control valve of the capacity control means.

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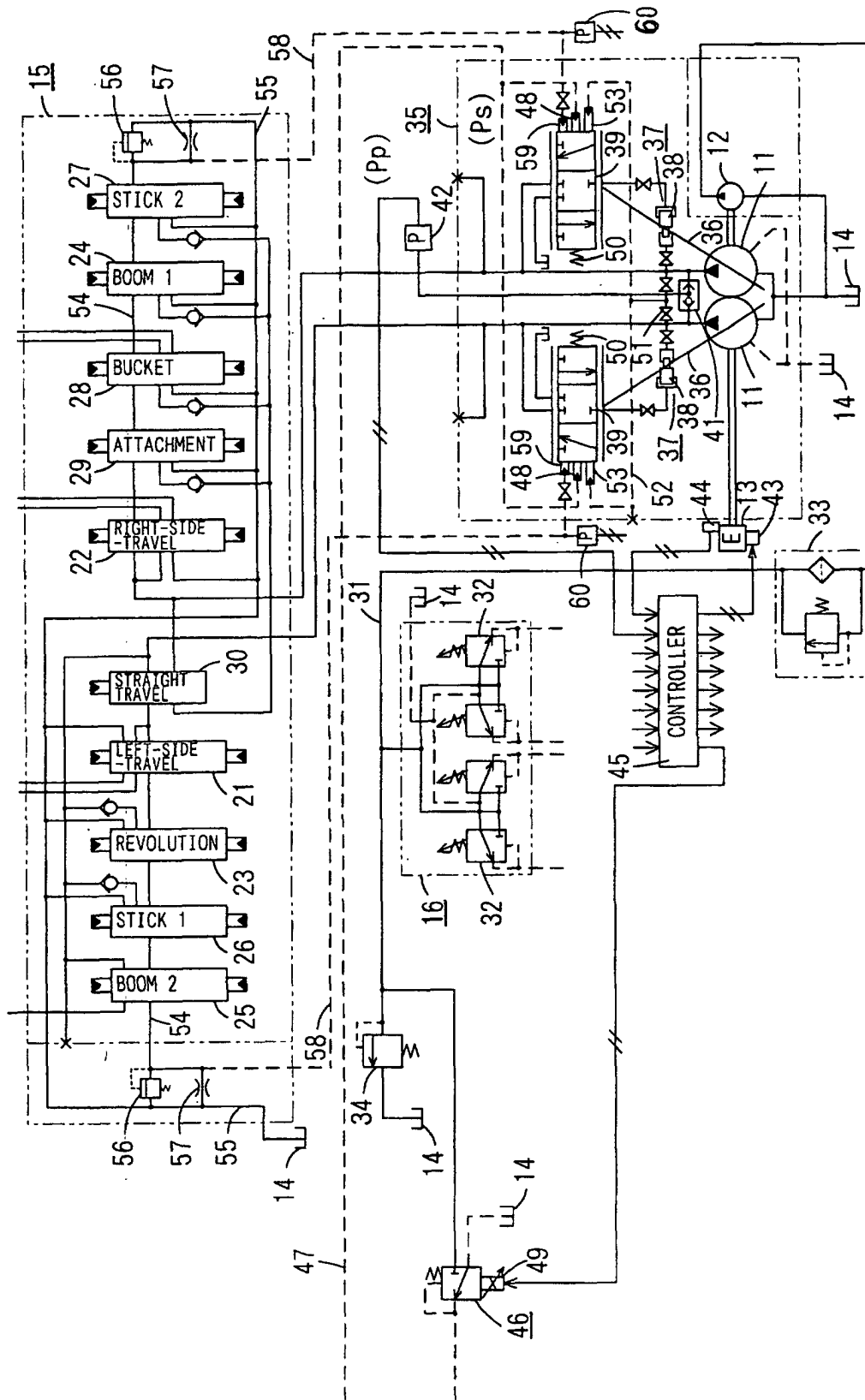


FIG. 1

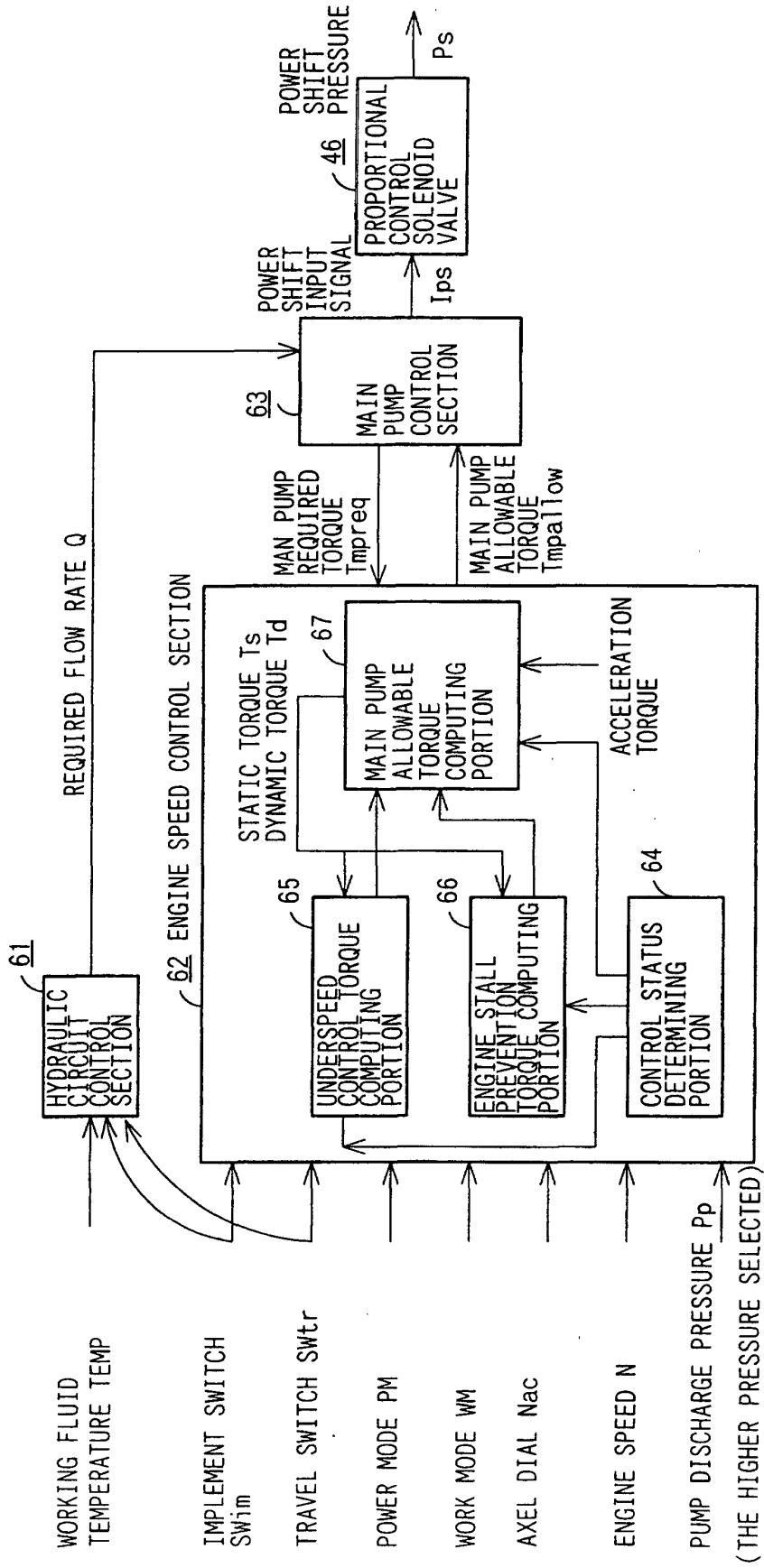


FIG. 2

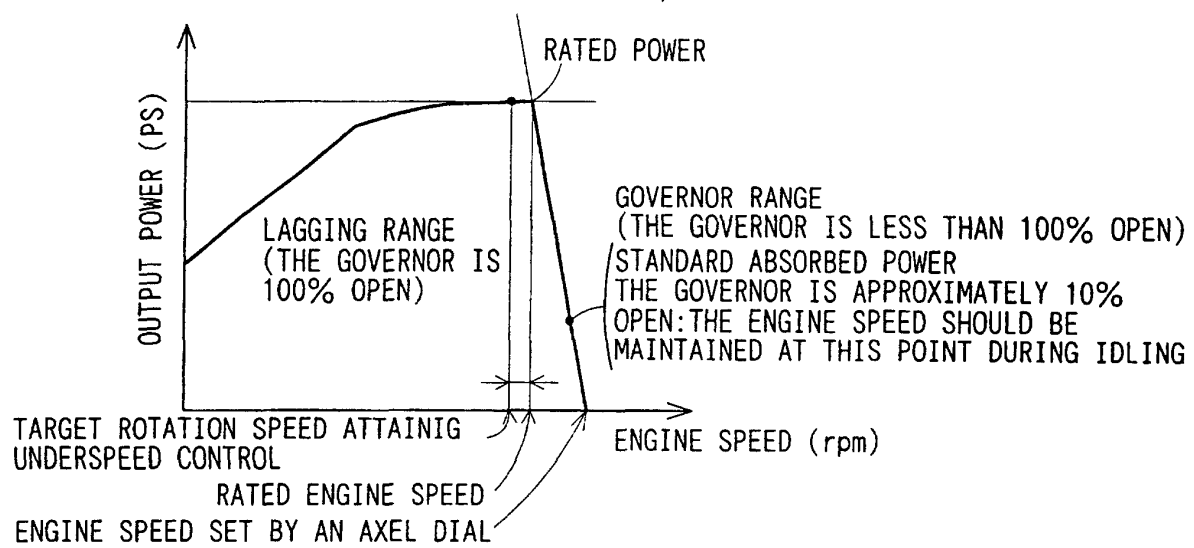


FIG. 3

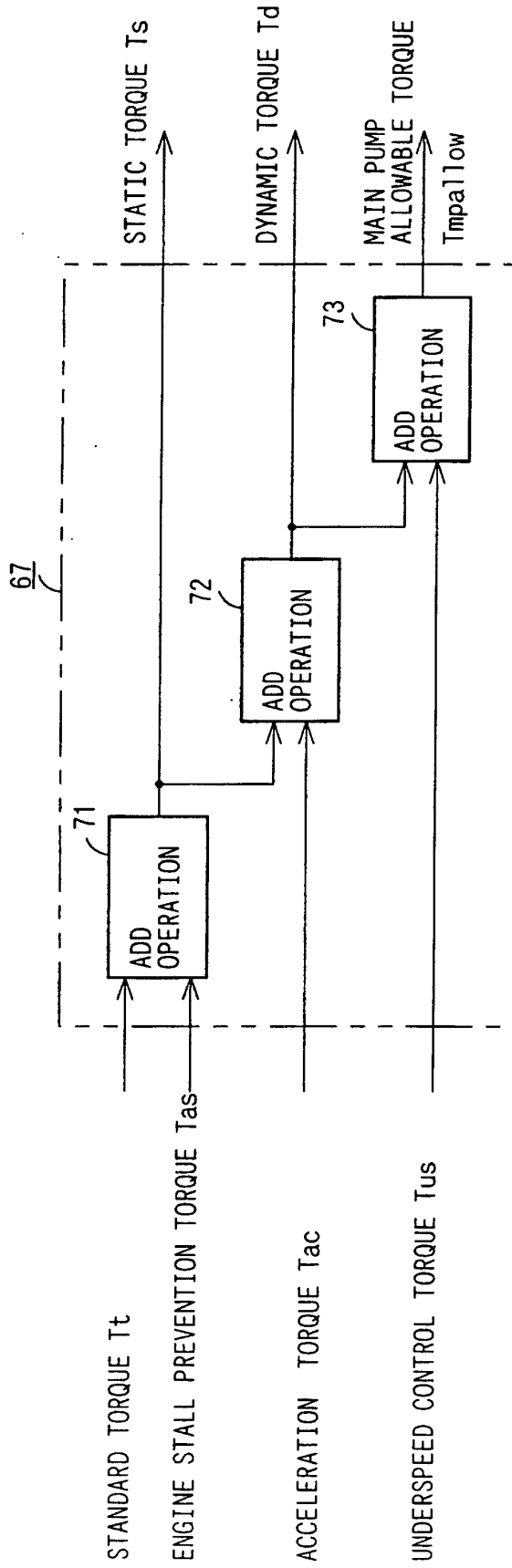


FIG. 4

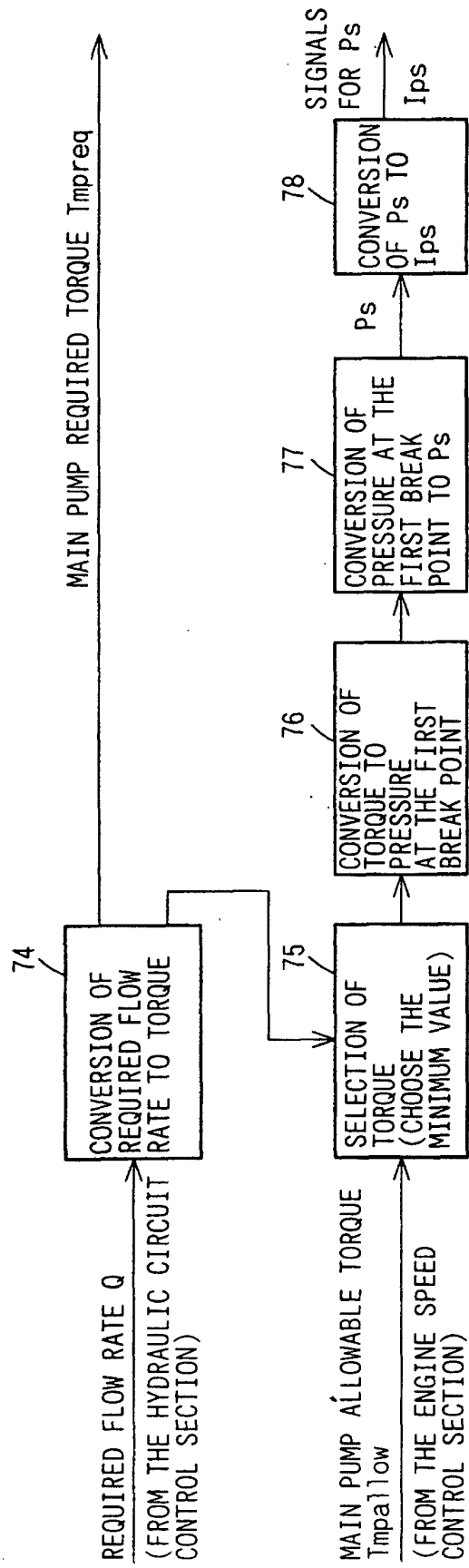


FIG. 5

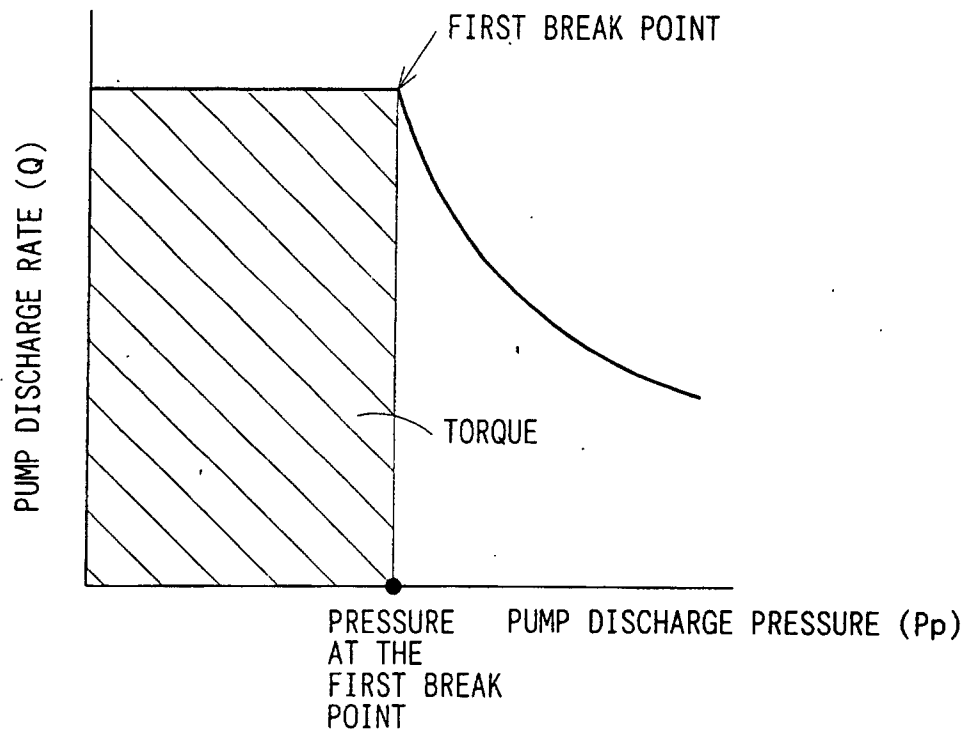


FIG. 6

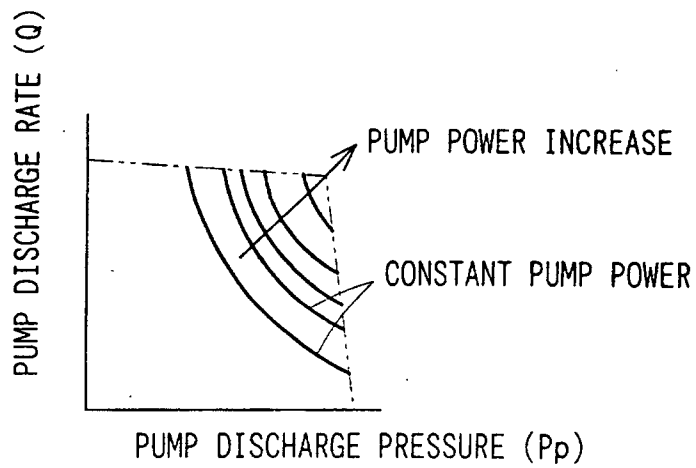


FIG. 7

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP00/07152

A. CLASSIFICATION OF SUBJECT MATTER Int. Cl. ⁷ F04B49/06		
According to International Patent Classification (IPC) or to both national classification and IPC		
B. FIELDS SEARCHED		
Minimum documentation searched (classification system followed by classification symbols) Int. Cl. ⁷ F04B49/06		
Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched Jitsuyo Shinan Koho 1926-1996 Toroku Jitsuyo Shinan Koho 1994-2000 Kokai Jitsuyo Shinan Koho 1971-2000 Jitsuyo Shinan Toroku Koho 1996-2000		
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.
X A Y	JP, 11-125187, A (Shin Caterpillar Mitsubishi Ltd.), 11 May, 1999 (11.05.99) (Family: none)	1, 3 2, 4 5, 6
Y	JP, 4-91380, A (Yuken Kogyo K.K.), 24 March, 1992 (24.03.92) (Family: none)	5
Y	JP, 62-101892, A (Kawasaki Heavy Industries, Ltd.), 12 May, 1987 (12.05.87) (Family: none)	6
Y	JP, 10-220359, A (Komatsu Ltd.), 18 August, 1998 (18.08.98) (Family: none)	6
☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.		
* "A" "E" "L" "O" "P"	Special categories of cited documents: "A" document defining the general state of the art which is not considered to be of particular relevance "E" earlier document but published on or after the international filing date "L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified) "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 22 December, 2000 (22.12.00)		Date of mailing of the international search report 16 January, 2001 (16.01.01)
Name and mailing address of the ISA/ Japanese Patent Office		Authorized officer
Facsimile No.		Telephone No.

Form PCT/ISA/210 (second sheet) (July 1992)