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(54) VARIABLE DISPLACEMENT PUMP REGULATOR

(57) A regulator (10) includes a feedback lever (24) configured to move a compensating sleeve (32) in an axial direction of a compensating spool (30). The compensating spool (30) and the compensating sleeve (32) form: a self-pressure introduction chamber (34), into which a self-pressure Pd of a pump is introduced; and a pressure adjuster 51 configured to output a control pressure Pcl from the self-pressure introduction chamber (34) to a large-diameter pressure receiver (22) of a servo piston (21). The compensating sleeve (32) includes a stepped portion (42), to which a pressure in a direction in which a control spring (31) urges the compensating spool (30) is applied when the self-pressure Pd is introduced into the self-pressure introduction chamber (34).

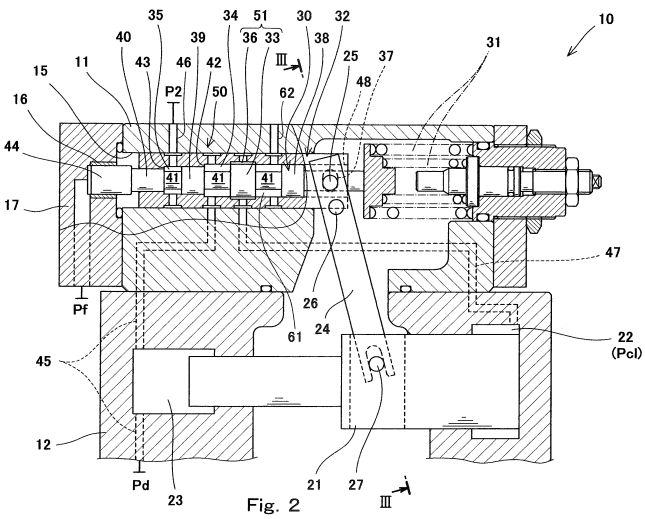

Fig. 2

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Description

Technical Field

[0001] The present invention relates to variable displacement pump regulators capable of controlling a delivery flow rate through tilting angle adjustment.

Background Art

[0002] Conventionally, power machinery such as hydraulic excavators, cranes, wheel loaders, and bulldozers (in the description and claims herein, such power machinery (heavy machinery) is collectively referred to as "work machines") has been used in, for example, civil engineering work and construction work. For example, in the case of a hydraulic excavator, various actuators are used to turn an upper turning unit of the hydraulic excavator and operate a bucket, an arm, a boom, etc. Hydraulic pressure is utilized to drive these actuators.

[0003] Hydraulic pumps supply hydraulic oil to these actuators at required flow rates in accordance with, for example, a turning speed in the case of the upper turning unit and an weight lifted by the bucket in the case of the arm or boom. In consideration of the number of actuators, required power, etc., a tandem hydraulic pump including a plurality of hydraulic pumps is used, for example.

[0004] Variable displacement pumps are used as these hydraulic pumps. The capacity of each variable displacement pump is controlled by a regulator. For example, in the case of using a pair of variable displacement pumps, regulators are provided for the respective variable displacement pumps. The regulators perform total horsepower control, in which the tilting angles of the respective pumps are controlled based on the delivery pressures of the respective pumps so that the sum of the horsepowers of both of the pumps will not exceed the horsepower of an engine that drives these pumps. A general regulator is configured such that: the position of the spool or sleeve of the regulator is controlled by a signal pressure, such as the self-pressure of a pump (i.e., the delivery pressure of a pump for which the regulator is provided); pressure oil having a pressure corresponding to the position of the spool or sleeve is led to a pressure adjuster; and the pressure adjuster outputs pressure oil having a control pressure. In this manner, capacity adjustment of a hydraulic pump is performed. Hydraulic pumps in which a servo piston is used as a part of a capacity adjusting mechanism are configured such that the capacity of a variable displacement pump is adjusted by positioning of the servo piston. Capacity adjustment of a variable displacement swash plate pump is performed by adjusting the tilting angle of the swash plate.

[0005] As one example of this kind of prior art, there is an automatic accelerator that includes: two variable displacement pumps driven by a prime mover; a plurality of actuators driven by pressure oil supplied from the variable displacement pumps; a first detector configured to detect command signals from an operation command unit, which commands the plurality of actuators to operate; a second detector configured to detect the load of the plurality of actuators; and an input unit configured to give a command indicating a reference target rotation speed of the prime mover. The automatic accelerator is configured to calculate a reference width of rotation speed correction, which decreases in accordance with a decrease in the reference target rotation speed, and correct a correction value of the reference target rotation speed by means of a correction value corrector (see Patent Literature 1, for example).

[0006] In the automatic accelerator, the capacities of the two variable displacement pumps are controlled by two regulators. To each regulator, the self-pressure of one variable displacement pump for which the regulator is provided and the counterpart-pressure of the other variable displacement pump are led. The spool position of each regulator is controlled by an operating piston, which is operated by these pressures.

Citation List

Patent Literature

[0007] PTL 1: Japanese Laid-Open Patent Application Publication No. H11-107322

Summary of Invention

Technical Problem

[0008] In such a variable displacement pump regulator as described above, the pressure adjuster is configured to output a control pressure in accordance with a signal pressure such as the delivery pressure (self-pressure) of the pump, which is applied to the spool, thereby causing the servo piston to move to change the pump capacity. When the servo piston moves, the sleeve of the regulator is also moved by a feedback lever which is in engagement with the servo piston, and thereby the tilting angle of the pump is controlled to be a target value. These controls, i.e., spool position control by means of the self-pressure and sleeve position control via the feedback lever, are always performed. Through these controls, the control pressure from the pressure adjuster of the regulator is led to the servo piston, and the delivery flow rate of the hydraulic pump is controlled in accordance with the load of an actuator of the work machine.

[0009] However, since an engagement portion where the feedback lever and the sleeve are engaged with each other is provided with an engagement pin, a contact portion of the engagement pin may be abraded as a result of long-term use.

[0010] In the case of the regulators of the above-described automatic accelerator, since the operating piston for controlling the position of the spool is provided separately from the spool, an axial urging force is not applied to the sleeve, and irrespective of the moving direction of the spool, a friction force and the like are applied to an engagement portion that contacts an engagement pin. Due to the applied force, the engagement portion becomes abraded non-directionally. The abrasion causes formation of a gap between the engagement pin and the portion that contacts the engagement pin. The gap causes a dead zone having a predetermined width relative to the control pressure of the regulator, the control pressure causing increase/decrease in the flow rate of the pump. Due to the dead zone, non-directional hysteresis occurs in horsepower control. The hysteresis increases as the abrasion of the engagement portion progresses due to long-term use.

[0011] Fig. 8 is a diagram showing horsepower control characteristics of a hydraulic pump, which are determined by a regulator, in the form of a relationship between a delivery pressure and a flow rate. A control line 101 is set to approximate an equivalent horsepower line 100, which is a design line. The control line 101 represents an example where two control springs are used. In this example, the number of control springs used in the horsepower control is changed in the middle of the control from 1 to 2, and thereby the slope of the control line 101 is varied such that the control line 101 approximates the equivalent horsepower line 100.

[0012] Accordingly, if an engagement portion where the sleeve and the pin are engaged with each other becomes abraded non-directionally as described above, then as shown in Fig. 8, hysteresis may occur on an actual control line 102 relative to the flow rate indicated by the target control line 101, and the control line may deviate to either the flow rate decreasing side or the flow rate increasing side.

[0013] Assume here that the flow rate relative to the delivery pressure exceeds the flow rate indicated by the equivalent horsepower line 100. For example, in the case of a hydraulic excavator, when a hydraulic excavator performs work that requires great horsepower such as earth-and-sand excavating and loading work, the delivery flow rate of its hydraulic pump may become excessively high, and consequently, there is a possibility that the horsepower of the hydraulic pump (in the case of including a plurality of hydraulic pumps, the sum of the horsepowers of all of the hydraulic pumps) exceeds the horsepower of the prime mover, causing a stall or unstable operation of the prime mover.

[0014] Thus, in a case where hysteresis in the flow rate relative to the delivery pressure of the pump has increased due to abrasion of an engagement portion where the sleeve and the feedback lever are engaged with each other, even if the horsepower control function of the regulator is exerted, i.e., even if the delivery flow rate of the pump is caused to decrease similarly to the control line 101 for the purpose of reducing the horsepower of the pump, an increase in the delivery flow rate, the increase corresponding to the hysteresis, occurs. Therefore, it is highly possible that a stall of the prime mover occurs.

[0015] In view of the above, an object of the present invention is to provide a variable displacement pump regulator capable of stable horsepower control even if hysteresis in the flow rate of a hydraulic pump occurs in horsepower control characteristics of the hydraulic pump due to long-term use.

Solution to Problem

[0016] In order to achieve the above-described object, a variable displacement pump regulator according to the present invention includes: a servo piston configured to change a delivery flow rate of a variable displacement pump driven by a prime mover; a feedback lever for detecting a position of the servo piston; a compensating spool urged by a control spring in a direction toward one end of the compensating spool; and a compensating sleeve positioned surrounding the compensating spool and configured to move, by operation of the servo piston and the feedback lever, in an axial direction of the compensating spool via an engagement portion engaged with the feedback lever. The compensating spool and the compensating sleeve form: a delivery pressure introduction chamber, into which a delivery pressure of the pump is introduced; and a pressure adjuster configured to output a control pressure from the delivery pressure introduction chamber to a large-diameter pressure receiver of the servo piston. The compensating sleeve includes a stepped portion to which a pressure in the direction toward the one end of the compensating spool is applied when the delivery pressure is introduced into the delivery pressure introduction chamber.

[0017] In the above configuration, the compensating sleeve, which surrounds the compensating spool and which is configured to move in the axial direction, includes the stepped portion. Accordingly, the compensating sleeve is urged in the opposite direction to the control spring (the direction toward the one end of the compensating spool) by the delivery pressure introduced into the delivery pressure introduction chamber. As a result, a force in the opposite direction to the control spring is always applied to an engagement portion of the feedback lever engaged with the compensating sleeve. This causes abrasion to occur on a limited part of the engagement portion. Accordingly, even if the control spring side of the engagement portion between the feedback lever and the compensating sleeve becomes abraded due to long-term use, horsepower control characteristics of the pump cause a decrease in the delivery flow rate of the pump. Therefore, even if the engagement portion of the feedback lever becomes abraded, the horsepower of the pump will not exceed the horsepower of the prime mover (in some cases, partial horsepower of the prime mover corresponding to a single pump). This makes it possible to perform stable horsepower control and prevent a stall of the prime mover.

[0018] The stepped portion may be formed on an opposite side of the delivery pressure introduction chamber to the pressure adjuster, the stepped portion being formed by reducing an internal diameter of the compensating sleeve.

[0019] According to the above configuration, a pressure receiving area difference can be made between the pressure adjuster side of the compensating sleeve and the opposite side of the compensating sleeve so that the compensating sleeve will be urged in the opposite direction to the control spring (i.e., in the direction toward the one end of the compensating spool) by the delivery pressure introduced into the delivery pressure introduction chamber.

[0020] The delivery pressure introduction chamber may be a self-pressure introduction chamber, into which a self-pressure, which is the delivery pressure of the pump, is introduced. The compensating spool and the compensating sleeve may form a counterpart-pressure introduction chamber, into which a counterpart-pressure, which is a delivery pressure of another variable displacement pump, is introduced, the counterpart-pressure introduction chamber being formed at a position that is away from the self-pressure introduction chamber toward an opposite side to the pressure adjuster. In the description and claims herein, the term "self-pressure" refers to the delivery pressure of the regulator-side variable displacement pump in a configuration that includes a plurality of pumps, and the term "counterpart-pressure" refers to the delivery pressure of the other variable displacement pump in the configuration.

[0021] According to the above configuration, in the structure including the plurality of variable displacement pumps, the pump capacity can be changed by the regulator in accordance with the delivery pressures of the respective pumps so that the sum of the horsepower of the pump for which the regulator is provided and the horsepower of the other pump will not exceed the horsepower of the prime mover.

[0022] The counterpart-pressure introduction chamber may have a diameter less than that of the self-pressure introduction chamber. The compensating sleeve may include a stepped portion to which a pressure in the direction toward the one end of the compensating spool is applied when the counterpart-pressure is introduced into the counterpart-pressure introduction chamber.

[0023] According to the above configuration, also in the structure including the plurality of variable displacement pumps with a feature that the pump capacity is controlled in accordance with the self-pressure and the counterpart-pressure, an urging force in the opposite direction to the control spring (i.e., in the direction toward the one end of the compensating spool) can always be applied to the compensating sleeve. Specifically, an urging force toward the pump capacity decreasing side can always be applied to the engagement portion of the feedback lever also by the counterpart-pressure introduced into the counterpart-pressure introduction chamber.

Advantageous Effects of Invention

[0024] According to the present invention, even after long-term use, abrasion or the like occurs only on the pump capacity decreasing side. This makes it possible to realize a variable displacement pump regulator capable of stable horsepower control even after long-term use.

Brief Description of Drawings

[0025]

Fig. 1 is a hydraulic circuit diagram showing one embodiment of a variable displacement pump regulator according to the present invention.

Fig. 2 is a sectional view of the variable displacement pump regulator of Fig. 1 as viewed in the direction of arrows of line II-II of Fig. 3.

Fig. 3 is a sectional view as viewed in the direction of arrows of line III-III of Fig. 2.

Fig. 4 is a sectional view showing a state where a compensating spool of the variable displacement pump regulator of Fig. 2 has moved to the control spring side.

Fig. 5 is a sectional view showing a state where a compensating sleeve has moved as a result of a change in pump capacity, the change being made in the previous state shown in Fig. 4 where the compensating spool has moved to the control spring side.

Fig. 6 is a schematic diagram showing a relationship between a delivery pressure applied to the compensating sleeve shown in Fig. 2 and a force applied to an engagement portion of a feedback lever.

Fig. 7 is a horsepower control line diagram showing a change in control characteristics, which may occur when a change due to long-term use has occurred in the variable displacement pump regulator shown in Fig. 2.

Fig. 8 is a horsepower control line diagram showing a change in control characteristics, which may occur when a change due to long-term use has occurred in a conventional variable displacement pump regulator.

Description of Embodiments

[0026] Hereinafter, one embodiment of the present invention is described with reference to the drawings. In the embodiment described below, only one pump included in a swash-plate double pump such as a tandem pump is shown in the drawings; the delivery pressure of the shown pump is referred to as a "self-pressure Pd"; and the delivery pressure of the other pump is referred to as a "counterpart-pressure P2". The description below is given with reference to the drawings that only show components related to horsepower control of a variable displacement pump. Further, up-down and left-right directions referred to in the description and claims herein correspond to up-down and left-right directions in a sectional view of a regulator shown in Fig. 2.

[0027] As shown in Fig. 1, the delivery flow rate of a variable displacement pump 2 (hereinafter, simply referred to as a "pump 2") driven by a prime mover 1 is adjusted through control of a swash plate tilting angle by a regulator 10. The regulator 10 includes: a servo piston 21; a feedback lever 24 engaged with the servo piston 21; a compensating spool 30 urged by a control spring 31 in a direction toward one end of the compensating spool 30 (i.e., in one axial direction); and a compensating sleeve 32 positioned surrounding the compensating spool 30 (in other words, the compensating spool 30 is inserted in the compensating sleeve 32). As a result of axial movement of the compensating spool 30 and the compensating sleeve 32, a control pressure to be led to a large-diameter pressure receiver 22 of the servo piston 21 is adjusted.

[0028] A pressure adjuster 51 configured to control the position of the servo piston 21 and a load calculating unit for horsepower control are integrally provided between the compensating spool 30 and the compensating sleeve 32 in the present embodiment. The pressure adjuster 51 will be described in detail below.

[0029] Since the double pump 2 is used in the present embodiment (only one pump included in the double pump 2 is shown in the drawings), a plurality of delivery pressure introduction champers 34 and 35 are provided between the compensating spool 30 and the compensating sleeve 32. In the present embodiment, the plurality of delivery pressure introduction champers 34 and 35 are configured as a self-pressure introduction chamber 34 into which the self-pressure Pd is introduced and a counterpart-pressure introduction chamber 35 into which the counterpart-pressure P2 is introduced. The counterpart-pressure introduction chamber 35 is provided at a position that is away from the self-pressure introduction chamber 34 in the opposite direction to the control spring 31 (i.e., in the aforementioned direction toward the one end of the compensating spool 30). As described below, the plurality of delivery pressure introduction champers 34 and 35 are provided with stepped portions 42 and 43, respectively. The counterpart-pressure introduction chamber 35 is formed to have a diameter less than that of the self-pressure introduction chamber 34.

[0030] It should be noted that, in the present embodiment, a horsepower setting pressure Pf is introduced into the compensating spool 30 at the opposite side to the control spring 31. The setting horsepower of the regulator 10 can be changed by changing the horsepower setting pressure Pf.

[0031] Hereinafter, the configuration of the variable displacement pump regulator 10 is described in more detail with reference to Figs. 2 and 3. In the present embodiment, a regulator casing 11 of the regulator 10 is fixed to a pump casing 12 by bolts 13 and 14 (see Fig. 3). The pump casing 12 includes the servo piston 21. The regulator casing 11 includes the compensating spool 30 and the compensating sleeve 32. The compensating sleeve 32, which surrounds the compensating spool 30, is configured to move axially.

[0032] As shown in Fig. 2, the control spring 31, which urges the compensating spool 30 in the direction toward the one end, is provided to the right (the other axial direction) of the compensating spool 30. The control spring 31 is displaced when the compensating spool 30 is moved to the right by delivery pressures that are introduced into the self-pressure introduction chamber 34 and the counterpart-pressure introduction chamber 35 of the compensating spool 30. The control spring 31 in the present embodiment is formed of two springs that are arranged coaxially. The position of the compensating spool 30 is determined in relation to the spring force of the control spring 31 and three pressures (Pd, P2, Pf) applied to the compensating spool 30. By forming the control spring 31 with two springs, a control line whose slope changes in the middle due to a flow rate change is made approximate to an equivalent horsepower line, as shown in Fig. 7 described below. It should be noted that one or more springs with linear deformation-load characteristics or non-linear deformation-load characteristics may be used as the control spring 31.

[0033] The compensating sleeve 32 is configured to move axially along a guide cylinder portion 15 provided in the regulator casing 11. One end of the feedback lever 24 is engaged with the compensating sleeve 32. The other end of the feedback lever 24 is engaged with the servo piston 21 via a control pin 27. The one end of the feedback lever 24 is provided with an engagement pin 25. An engagement groove 37 is formed in the side surface of the control-spring-side portion of the compensating sleeve 32. The engagement pin 25 of the feedback lever 24 is fitted in the engagement groove 37. The feedback lever 24 is supported on the regulator casing 11 by a supporting pin 26 provided on a middle portion of the feedback lever 24. The feedback lever 24 swings about the supporting pin 26 when the servo piston 21 moves. The swinging motion of the feedback lever 24 causes the compensating sleeve 32 to move axially. That is, the position of the compensating sleeve 32 is determined by the position of the servo piston 21.

[0034] The compensating spool 30 and the compensating sleeve 32 form the plurality of delivery pressure introduction champers 34 and 35. The compensating spool 30 includes: a large-diameter portion 38 serving as a guide portion; a pressure adjusting land 33 having the same diameter as that of the large-diameter portion 38; a medium-diameter portion 39 having a diameter less than that of the pressure adjusting land 33; and a small-diameter portion 40 having a diameter less than that of the medium-diameter portion 39. These portions are sequentially arranged in this order such that, among these portions, the large-diameter portion 38 is closest to the control spring 31 and the small-diameter portion 40 is farthest from the control spring 31. Shaft portions 41 connect between these portions, such that these portions are integrally connected to one another. An end portion of the compensating spool 30 in the opposite direction to the control spring 31 (i.e., in the left direction) is provided with a horsepower setting operating piston 44. The operating piston 44 is axially guided by a cylindrical guide 16 provided in a cover 17. It should be noted that the compensating spool 30 and the horsepower setting operating piston 44 may be either formed integrally or formed as separate components.

[0035] A guide cylinder portion 48 configured to axially guide the large-diameter portion 38 is formed on the control spring 31 side of the cylindrically formed inner surface of the compensating sleeve 32. A control pressure output chamber 36, which has a diameter greater than that of the guide portion 48 by a predetermined dimension, is provided at a predetermined position of the guide cylinder portion 48. The control pressure output chamber 36 is provided at a position corresponding to the pressure adjusting land 33 provided in the compensating spool 30. The control pressure output chamber 36 and the pressure adjusting land 33 form the pressure adjuster 51, which outputs a control pressure from the delivery pressure introduction chamber 34 to the large-diameter pressure receiver 22 of the servo piston 21. The self-pressure introduction chamber 34, which has the same diameter as that of the guide cylinder portion 48, is formed such that the self-pressure introduction chamber 34 is positioned in the opposite direction to the control spring 31 with respect to the control pressure output chamber 36. The counterpart-pressure introduction chamber 35, which has such an internal diameter as to allow the medium-diameter portion 39 to be guided axially, is formed such that the counterpart-pressure introduction chamber 35 is positioned in the opposite direction to the control spring 31 with respect the self-pressure introduction chamber 34. A portion of the compensating sleeve 32, the portion being positioned in the opposite direction to the control spring 31 with respect to the counterpart-pressure introduction chamber 35, is formed to have such an internal diameter as to allow the small-diameter portion 40 to be guided axially.

[0036] By thus forming the inner surface of the compensating sleeve 32 to have different internal diameters, a stepped portion 42 is formed on the medium-diameter portion 39 side (the opposite side to the pressure adjuster 51) of the self-pressure introduction chamber 34 formed between the pressure adjusting land 33 and the medium-diameter portion 39 of the compensating spool 30, and a stepped portion 43 is formed on the small-diameter portion 40 side of the counterpart-pressure introduction chamber 35 formed between the medium-diameter portion 39 and the small-diameter portion 40. The stepped portion 42 is formed on the medium-diameter portion side of the self-pressure introduction chamber 34 by the area difference between the area of the pressure adjusting land 33 and the area of the medium-diameter portion 39, the area difference being determined by the difference between the diameter of the pressure adjusting land 33 and the diameter of the medium-diameter portion 39. The stepped portion 43 is formed on the small-diameter portion side of the counterpart-pressure introduction chamber 35 by the area difference between the area of the medium-diameter portion 39 and the area of the small-diameter portion 40, the area difference being determined by the difference between the diameter of the medium-diameter portion 39 and the diameter of the small-diameter portion 40. Thus, the inner surface of the compensating sleeve 32 is stepped into three portions that have different areas from one another, such that the diameter of the counterpart-pressure introduction chamber 35 is less than the diameter of the self-pressure introduction chamber 34, and such that the diameter of the portion that axially guides the small-diameter portion 40 is less than the diameter of the counterpart-pressure introduction chamber 35.

[0037] An oil discharge chamber 61, which leads pressure oil from a control passage 47 described below to a tank passage 62 provided in the regulator casing 11, is formed between the large-diameter portion 38 and the pressure adjusting land 36 of the compensating spool 30. Communication passages 71 to 74 (reference signs 71 to 74 are shown only in Fig. 4) are provided in the compensating sleeve 32 for the purpose of allowing the oil discharge chamber 61, the control pressure output chamber 36, the self-pressure introduction chamber 34, and the counterpart-pressure introduction chamber 35 to communicate with the tank passage 62, the control passage 47, an introduction passage 45 described below, and an introduction passage 46 described below, respectively. Each of the communication passages 71 to 74 is constituted by an annular groove formed in the outer peripheral surface of the compensating sleeve 32 and a plurality of through-holes radially extending through the compensating sleeve 32.

[0038] The self-pressure Pd that has been introduced into a small-diameter pressure receiver 23 of the servo piston 21 is introduced from the pump 2 into the self-pressure introduction chamber 34 through the introduction passage 45 and the communication passage 73. The counterpart-pressure P2 is introduced from a counterpart pump, which is not shown, into the counterpart-pressure introduction chamber 35 through the introduction passage 46 and the communication passage 74.

[0039] Then, these delivery pressures cause the compensating spool 30 to move in such a direction as to contract the control spring 31 (i.e., move to the right) against the spring force of the control spring 31. This configuration serves as a calculating unit 50 in a horsepower controller of the regulator 10.

[0040] As a result of the compensating spool 30 moving in such a direction as to contract the control spring 31, the pressure adjusting land 33 moves axially. As a result, the self-pressure introduction chamber 34 and the control pressure output chamber 36 come into communication with each other, and the self-pressure Pd is introduced into the large-diameter pressure receiver 22 of the servo piston 21 through the communication passage 72 and the control passage 47 as a control pressure Pcl. Specifically, the movement of the pressure adjusting land 33 causes the area of opening between the self-pressure introduction chamber 34 and the control pressure output chamber 36 to change, and thereby the control pressure Pcl is adjusted. This configuration serves as the pressure adjuster 51 in the horsepower controller of the regulator 10, and the pressure adjuster 51 is incorporated in the calculating unit 50.

[0041] As described above, the pressure adjusting land 33 is incorporated in the calculating unit 50 of the horsepower controller, and thus an integrated configuration is formed. This makes it possible to realize a compact mechanism, in which the calculating unit 50 causes the compensating spool 30 to move to a predetermined position corresponding to the delivery pressures of the regulator-side pump and the counterpart pump, and the pressure adjusting land 33 leads the control pressure Pcl, which causes the servo piston 21 to move, to adjust the delivery flow rate of the pump 2.

[0042] Next, operations of the compensating spool 30 and the compensating sleeve 32 are described with reference to Figs. 4 and 5. When a force from the small-diameter pressure receiver 23 and a force from the large-diameter pressure receiver 22, which are applied to the servo piston 21, are in equilibrium, the relative positional relationship between the compensating spool 30 and the compensating sleeve 32 is as shown in Fig. 2. In this state, the control pressure output chamber 36 is blocked by the pressure adjusting land 33. Accordingly, the control pressure output chamber 36 is neither in communication with the self-pressure introduction chamber 34 nor in communication with the oil discharge chamber 61.

[0043] As shown in Fig. 4, in a case where the delivery pressure Pd is introduced into the self-pressure introduction chamber 34 of the variable displacement pump regulator 10, if the total pressure of the delivery pressure (self-pressure) Pd, the counterpart-pressure P2, and the setting pressure Pf is greater than the spring load of the control spring 31, then the compensating spool 30 moves toward the control spring 31. Accordingly, the pressure adjusting land 33 of the compensating spool 30 causes the self-pressure introduction chamber 34 and the control pressure output chamber 36 to come into communication with each other. As a result of the communication, the control pressure Pcl is introduced from the self-pressure introduction chamber 34 into the large-diameter pressure receiver 22 of the servo piston 21. Consequently, the servo piston 21 controls the tilting angle such that the delivery flow rate of the variable displacement pump 2 decreases.

[0044] The present embodiment gives an example of a double pump. Therefore, not only the self-pressure Pd introduced into the self-pressure introduction chamber 34 but also the counterpart-pressure P2 introduced into the counterpart-pressure introduction chamber 35 causes the compensating spool 30 to move. Owing to these pressures, the self-pressure flows from the self-pressure introduction chamber 34 to the control pressure output chamber 36, and the control pressure Pcl is introduced into the large-diameter pressure receiver 22 of the servo piston 21 through the control passage 47. Consequently, the delivery flow rate decreases in accordance with a required horsepower of either one of the pumps of the double pump.

[0045] Then, as shown in Fig. 5, when the servo piston 21 is moved to the left by the control pressure Pcl introduced into the large-diameter pressure receiver 22 of the servo piston 21, the feedback lever 24 swings about the supporting pin 26, thereby causing the compensating sleeve 32 to move. When the opening between the control pressure output chamber 36 and the self-pressure introduction chamber 34 is blocked owing to the movement of the compensating sleeve 32, the introduction of the control pressure Pcl from the self-pressure introduction chamber 34 into the large-diameter pressure receiver 22 is stopped. In this manner, the tilting angle of the pump 2 is adjusted by the servo piston 21 to be a tilting angle that corresponds to a required delivery flow rate.

[0046] When the self-pressure Pd introduced into the self-pressure introduction chamber 34 or the counterpart-pressure P2 introduced into the counterpart-pressure introduction chamber 35 decreases from the state shown in Fig. 2, the compensating spool 30 moves in a direction away from the control spring 31 (i.e., moves to the left), and thereby the control pressure output chamber 36 comes into communication with the oil discharge chamber 62. As a result, pressure oil is discharged from the large-diameter pressure receiver 22 to the tank passage 62 through the control passage 47, the communication passage 71, and the oil discharge chamber 61, and the servo piston 21 moves to the right while the compensating sleeve 32 moves to the left, so that that the opening between the control pressure output chamber 36 and the oil discharge chamber 61 is blocked.

[0047] In this manner, the tilting angle of the pump 2 is controlled, and when the delivery flow rate of the pump 2 has become a target flow rate, the positions of the compensating spool 30, the compensating sleeve 32, and the servo piston 21 are kept. Thus, the delivery flow rate control of the variable displacement pump 2 is always performed in accordance with the delivery pressure of each pump, which changes in accordance with, for example, work performed by the work machine.

[0048] Fig. 6 is a schematic diagram illustrating a relationship between a force applied to the engagement pin 25 of the feedback lever 24 and the delivery pressure Pd and counterpart-pressure P2 applied to the compensating sleeve 32. Owing to the area differences made by the stepped portions 42 and 43, the force F1 based on the self-pressure Pd that has been introduced into the self-pressure introduction chamber 34 and the force F2 based on the counterpart-pressure P2 that has been introduced into the counterpart-pressure introduction chamber 35 are applied to the stepped portions 42 and 43 of the compensating sleeve 32. Accordingly, whenever the delivery pressures Pd and P2 are applied, a load is always applied to a control spring side contact point 28 between the engagement pin 25 of the feedback lever 24 and the engagement groove 37 of the compensating sleeve 32 owing to these applied forces. To be more specific, since the compensating sleeve 32 is always urged to the left, a force that pushes the engagement pin 25 in a direction away from the control spring 31 is applied to the engagement pin 25, and a reaction force (i.e., a force in a direction toward the control spring 31) is applied to the engagement groove 37.

[0049] That is, owing to the self-pressure Pd and the counterpart-pressure P2 introduced into the self-pressure introduction chamber 34 and the counterpart-pressure introduction chamber 35, the compensating sleeve 32 is always in a state of being urged in a direction away from the control spring 31 (i.e., in the opposite direction to the control spring), and a load corresponding to the resultant force of the forces F1 and F2 is always applied to the joint.

[0050] Therefore, in a case where abrasion occurs on a portion where the engagement pin 25 of the feedback lever 24 and the engagement groove 37 to be engaged with the engagement pin 25 contact each other, the control spring side point 28 where the engagement pin 25 and the engagement groove 37 contact each other becomes abraded. Abrasion due to long-term use progresses only on the control spring side of the engagement pin 25 and the control spring side of the engagement groove 37. Accordingly, even if a change in the positional relationship between the compensating spool 30 and the compensating sleeve 32 due to the abrasion causes the angle of the feedback lever 24 to change, the servo piston 21 moves only in such a direction as to cause a decrease in the pump capacity. Therefore, even if a position of equilibrium at which, for example, the compensating spool 30 and the compensating sleeve 32 equilibrate with each other changes due to the abrasion, the horsepower control characteristics change only in such a manner as to cause a decrease in delivery flow rate in the horsepower control of the pump 2, i.e., cause a decrease in the horsepower of the pump.

[0051] Fig. 7 is a diagram showing horsepower characteristics of the variable displacement pump regulator 10 in the form of a relationship between a delivery pressure and a flow rate. A control line 101 is set to approximate an equivalent horsepower line 100, which is a design line. According to the above-described variable displacement pump regulator 10, in relation to the control line 101, even if abrasion of the engagement pin (engagement portion) 25 of the feedback lever 24 occurs and thereby a change occurs in the control characteristics, the characteristics change only in such a manner as to cause a decrease in delivery flow rate as indicated by a control line 102.

[0052] That is, the control line 101, which is initially set to approximate the equivalent horsepower line 100, changes after long-term use such that the control line 101 shifts to the flow rate decreasing side as indicated by the control line 102. This makes it possible to realize the variable displacement pump regulator 10, which is capable of stable operation even after long-term use.

[0053] As described above, according to the variable displacement pump regulator 10, the calculating unit 50 and the pressure adjuster 51 are configured in an integrated manner. The self-pressure introduction chamber 34 and the counterpart-pressure introduction chamber 35 are provided with the stepped portions 42 and 43, such that the compensating sleeve 32 has the aforementioned area differences, and the delivery pressures Pd and P2 cause the forces F1 and F2, which are in the opposite direction to the control spring, to be applied to the respective areas. Accordingly, whenever the delivery pressures are applied to the self-pressure introduction chamber 34 and the counterpart-pressure introduction chamber 35, the forces F1 and F2 in the opposite direction to the control spring are always applied to the compensating sleeve 32.

[0054] Therefore, the engagement pin (engagement portion) 25 of the feedback lever 24 controlling the position of the compensating sleeve 32 always makes a contact at the control spring side point 28, and abrasion due to long-term use only occurs on the control spring side point 28 where the engagement pin 25 and the engagement groove 37 contact each other.

[0055] In other words, the compensating sleeve 32 whose position is controlled by the feedback lever 24 in accordance with the movement of the servo piston 21 has a feature that even if abrasion due to long-term use occurs on a part of the engagement pin 25, the part being the engagement portion where the compensating sleeve 32 is engaged with the feedback lever 24, the abrasion occurs directionally. The abrasion always occurs such that the feedback lever 24 inclines in such a direction as to cause the servo piston 21 to reduce the delivery flow rate of the pump 2.

[0056] Therefore, even if the portion where the engagement pin 25 and the engagement groove 37 contact each other becomes abraded due to long-term use, the feedback lever 24 inclines always only in a direction that causes a change in pump tilting to the capacity decreasing side. This makes it possible to realize the variable displacement pump regulator 10, which is capable of stable horsepower control even after long-term use.

[0057] Moreover, in the present embodiment, since the self-pressure Pd and the counterpart-pressure P2 of the double pump are introduced into the calculating unit 50, the servo piston 21 can be driven in accordance with the self-pressure Pd and the counterpart-pressure P2, and thereby the delivery flow rate of the pump 2 can be controlled so that the target power will not be exceeded.

[0058] The present embodiment has been described above by taking the variable displacement pump regulator 10 for the double pump as an example. However, the configuration of the pump is not limited to the above-described embodiment. The pump may have a different configuration. For example, the pump may be configured as a single pump.

[0059] In the above-described embodiment, the stepped portions 42 and 43 are formed, and thereby the areas of the two delivery pressure introduction chambers 34 and 35 are made different from each other. However, as an alternative, the regulator 10 may include only the self-pressure introduction chamber 34. As another alternative, the regulator 10 may include more than two delivery pressure introduction chambers, and moreover, required pressures for the flow rate control may be applied thereto.

[0060] The above-described embodiment has given non-limiting examples. Various modifications can be made to the embodiment without departing from the spirit of the present invention. Thus, the present invention is not limited to the above-described embodiment.

Industrial Applicability

[0061] The variable displacement pump regulator according to the present invention is applicable to work machines that are used in civil engineering work, construction work, etc., such as hydraulic excavators, cranes, wheel loaders, and bulldozers.

Reference Signs List

[0062]

1 prime mover
2 variable displacement pump
10 variable displacement pump regulator
11 regulator casing
21 servo piston
22 large-diameter pressure receiver
23 small-diameter pressure receiver
24 feedback lever
25 engagement pin (engagement portion)
30 compensating spool
31 control spring
32 compensating sleeve
33 pressure adjusting land
34 self-pressure introduction chamber (delivery pressure introduction chamber)
35 counterpart-pressure introduction chamber (delivery pressure introduction chamber)
36 control pressure output chamber
37 engagement groove
42 stepped portion
43 stepped portion
51 pressure adjuster
101 control line
102 control line (hysteresis)
Pd self-pressure (delivery pressure)
P2 counterpart-pressure (delivery pressure)
Pcl control pressure

Claims

1. A variable displacement pump regulator comprising:

a servo piston configured to change a delivery flow rate of a variable displacement pump driven by a prime mover;
a feedback lever for detecting a position of the servo piston;
a compensating spool urged by a control spring in a direction toward one end of the compensating spool; and
a compensating sleeve positioned surrounding the compensating spool and configured to move, by operation of the servo piston and the feedback lever, in an axial direction of the compensating spool via an engagement portion engaged with the feedback lever, wherein
the compensating spool and the compensating sleeve form:

a delivery pressure introduction chamber, into which a delivery pressure of the pump is introduced; and
a pressure adjuster configured to output a control pressure from the delivery pressure introduction chamber to a large-diameter pressure receiver of the servo piston, and

the compensating sleeve includes a stepped portion to which a pressure in the direction toward the one end of the compensating spool is applied when the delivery pressure is introduced into the delivery pressure introduction chamber.

2. The variable displacement pump regulator according to claim 1, wherein
the stepped portion is formed on an opposite side of the delivery pressure introduction chamber to the pressure adjuster, the stepped portion being formed by reducing an internal diameter of the compensating sleeve.

3. The variable displacement pump regulator according to claim 1 or 2, wherein
the delivery pressure introduction chamber is a self-pressure introduction chamber, into which a self-pressure, which is the delivery pressure of the pump, is introduced, and
the compensating spool and the compensating sleeve form a counterpart-pressure introduction chamber, into which a counterpart-pressure, which is a delivery pressure of another variable displacement pump, is introduced, the counterpart-pressure introduction chamber being formed at a position that

is away from the self-pressure introduction chamber toward an opposite side to the pressure adjuster.

4. The variable displacement pump regulator according to claim 3, wherein
the counterpart-pressure introduction chamber has a diameter less than that of the self-pressure introduction chamber, and
the compensating sleeve includes a stepped portion to which a pressure in the direction toward the one end of the compensating spool is applied when the counterpart-pressure is introduced into the counterpart-pressure introduction chamber.

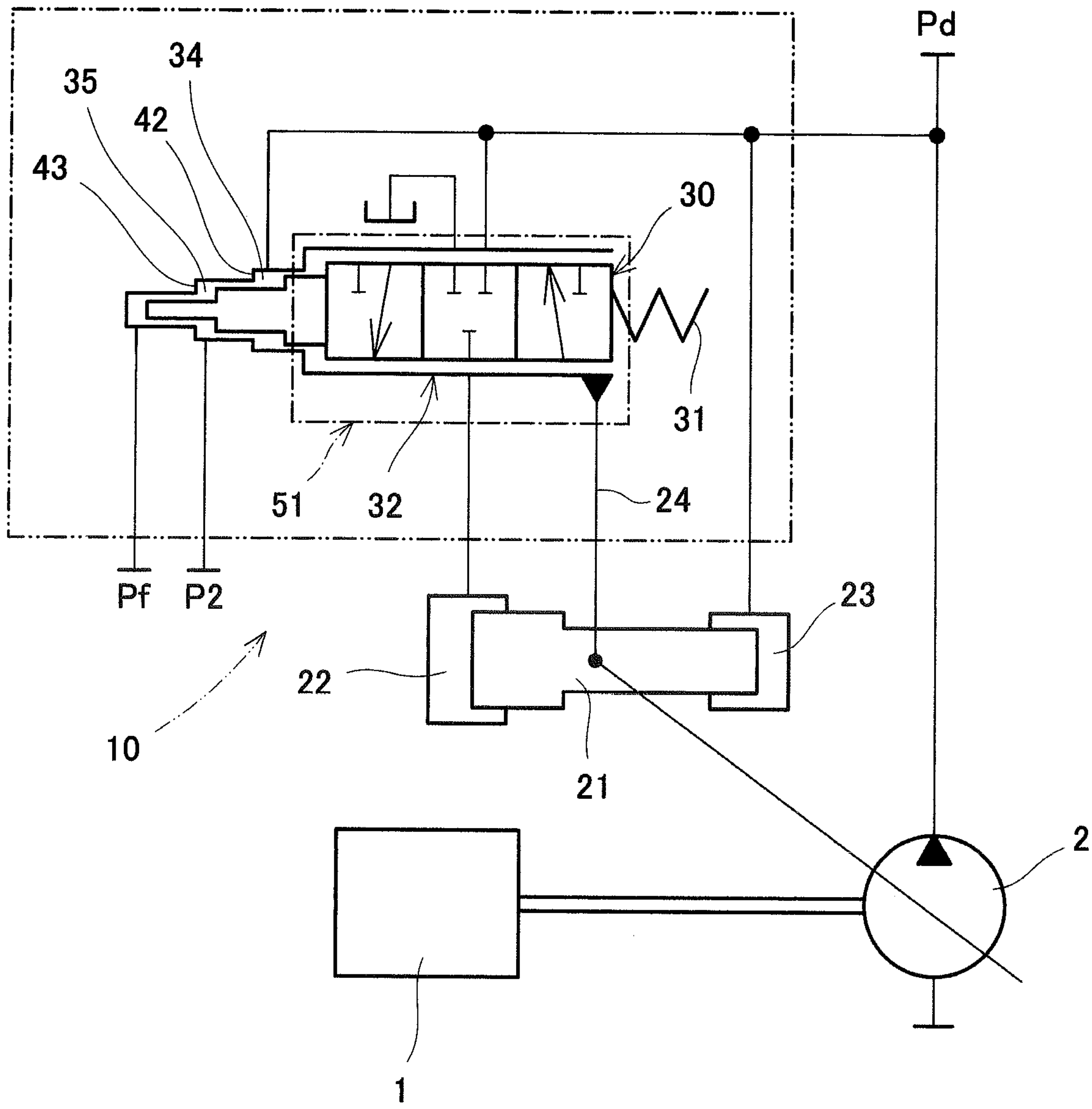

Fig. 1

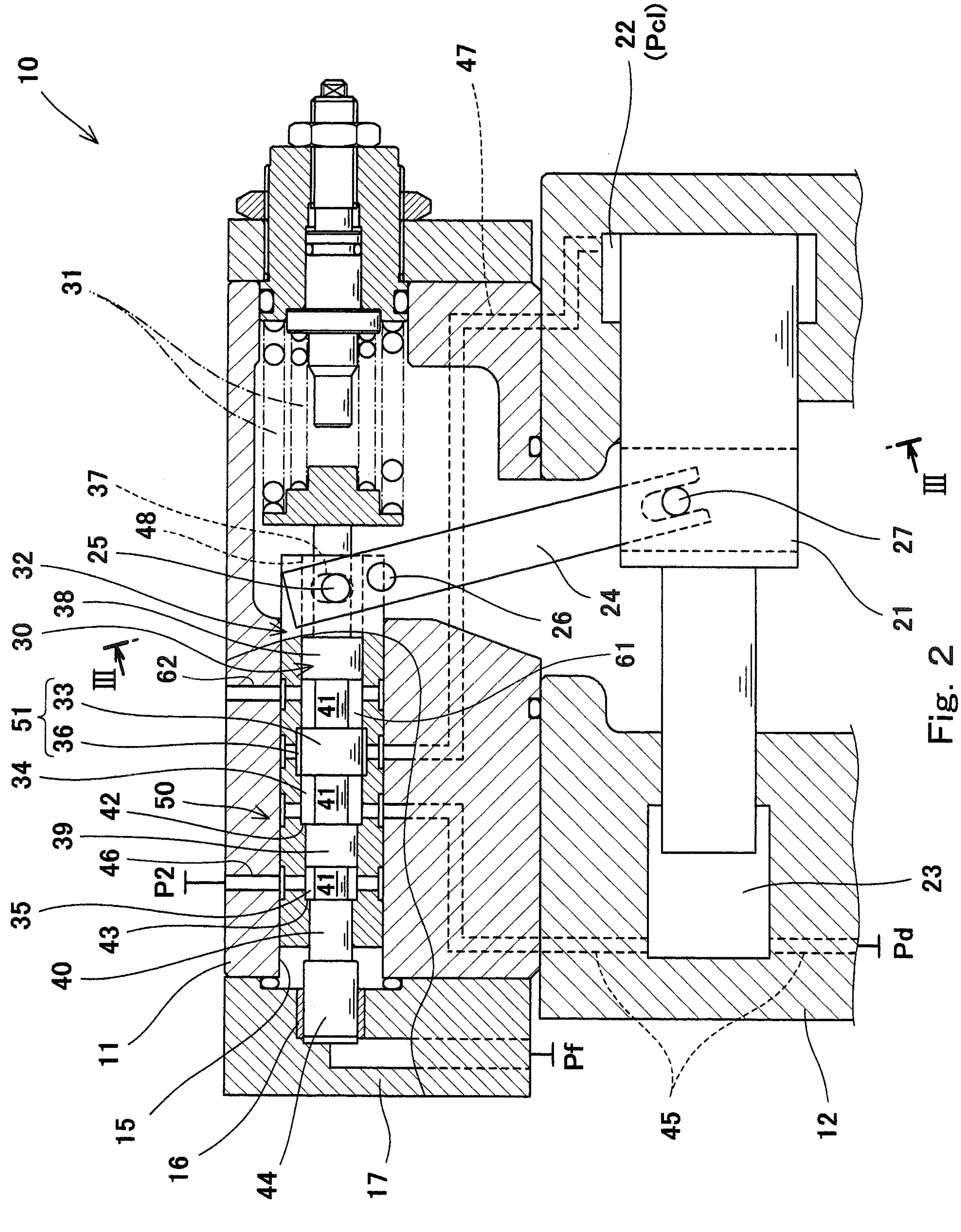
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Fig. 2

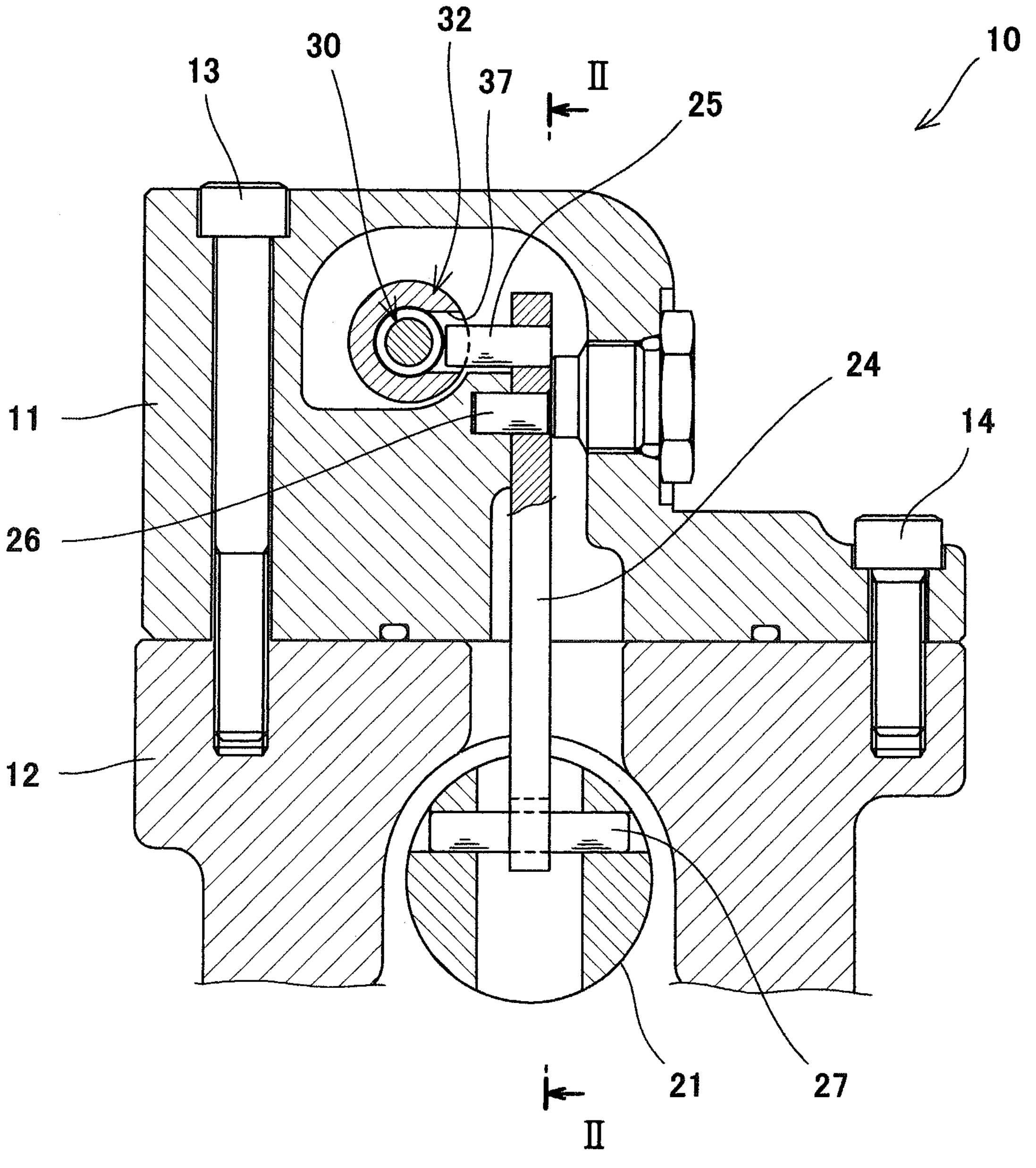

Fig. 3

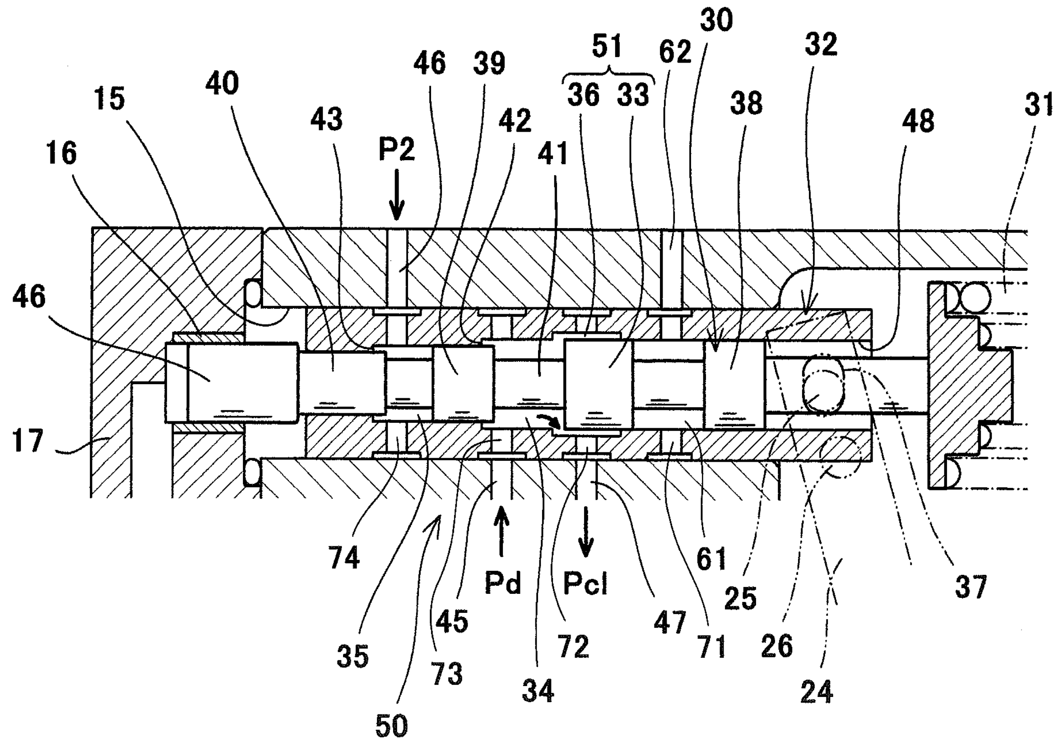

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Fig. 4

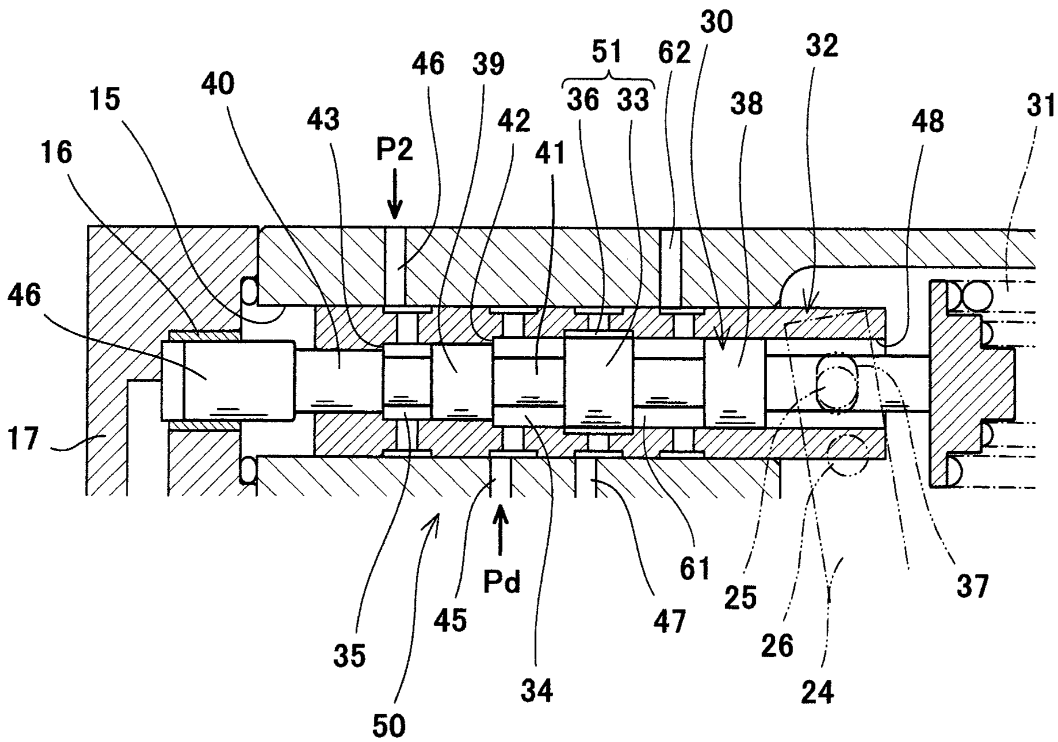

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Fig. 5

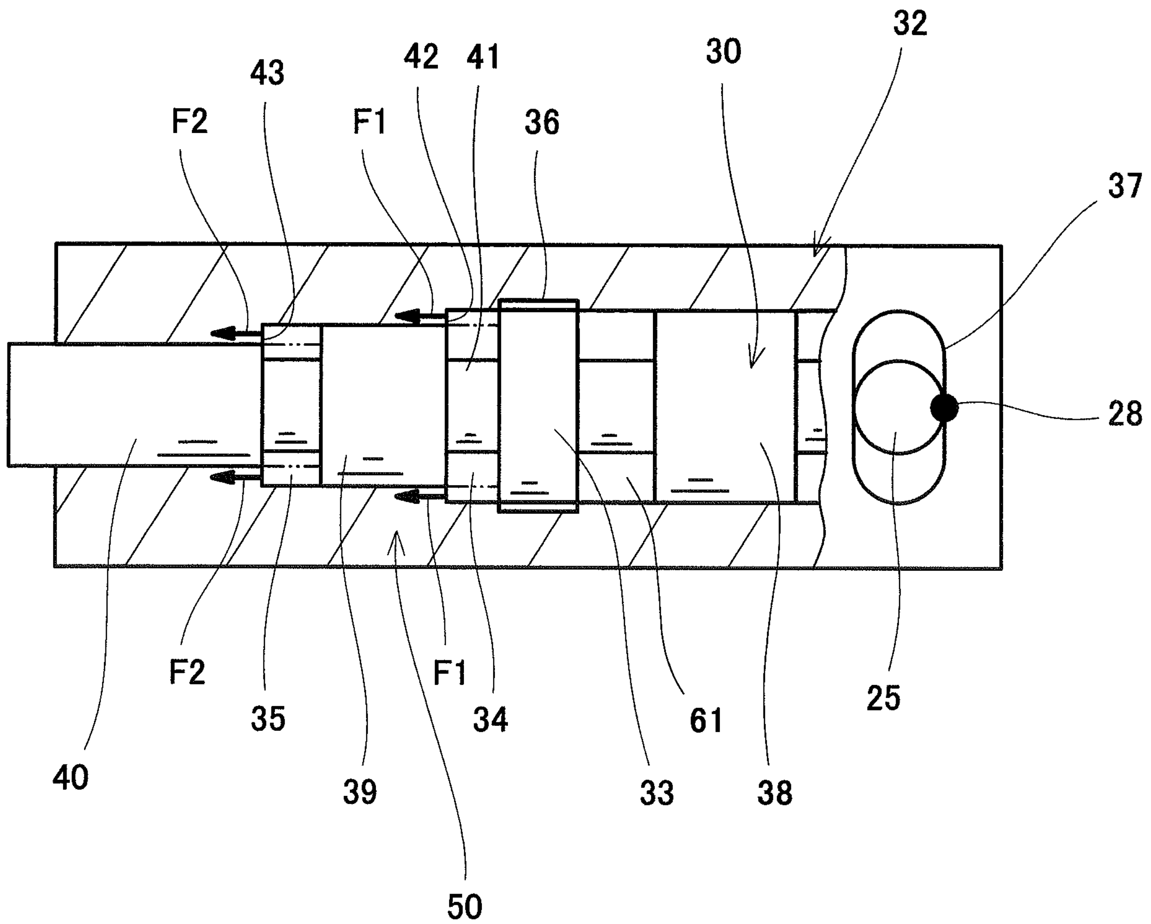
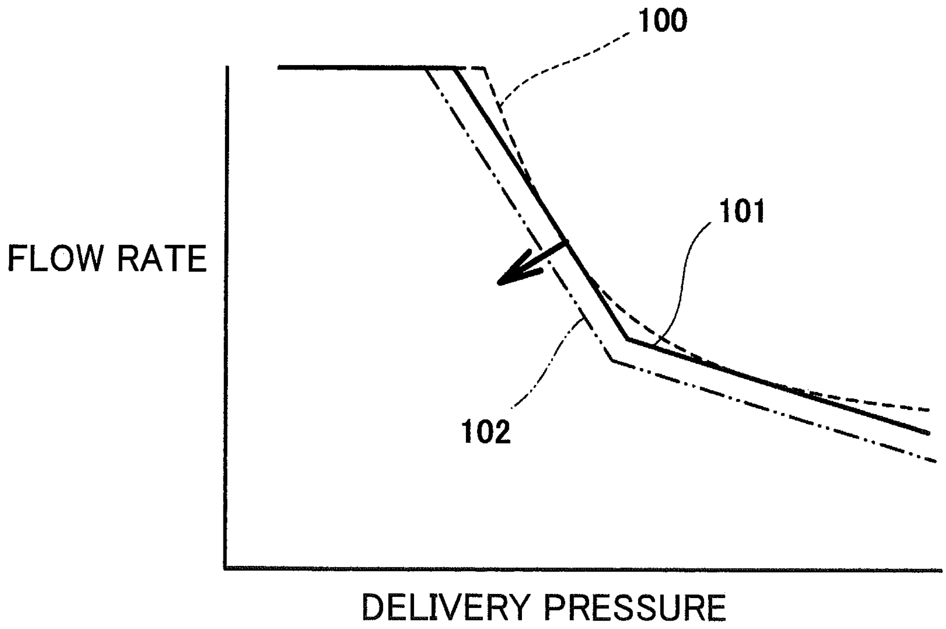


Fig. 6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FLOW RATE
102
DELIVERY PRESSURE

Fig. 7

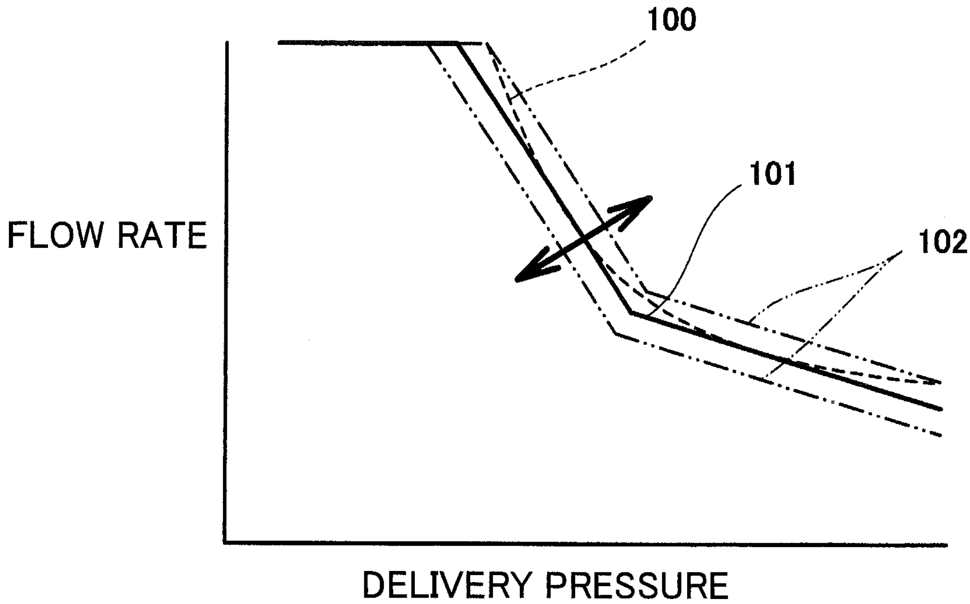
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Fig. 8

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2013/006898

A. CLASSIFICATION OF SUBJECT MATTER
F04B49/00(2006.01)i, *F04B1/26*(2006.01)i

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)
F04B49/00, F04B1/26

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2014
Kokai Jitsuyo Shinan Koho	1971-2014	Toroku Jitsuyo Shinan Koho	1994-2014

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.
A	JP 2008-175062 A (Nachi-Fujikoshi Corp.), 31 July 2008 (31.07.2008), paragraph [0022]; fig. 1, 4 (Family: none)	1-4
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 116564/1979(Laid-open No. 34102/1981) (Toshiba Machine Co., Ltd.), 03 April 1981 (03.04.1981), fig. 1 (Family: none)	1-4

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

* Special categories of cited documents:
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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
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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search 13 February, 2014 (13.02.14)	Date of mailing of the international search report 25 February, 2014 (25.02.14)
Name and mailing address of the ISA/ Japanese Patent Office Facsimile No.	Authorized officer Telephone No.

Form PCT/ISA/210 (second sheet) (July 2009)

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

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