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(71) Applicant: **Komatsu Ltd.**
Minato-ku
Tokyo 107-8414 (JP)

(72) Inventors:
• **SATO, Etsuro**
Hiratsuka-shi
Kanagawa 254-8567 (JP)

• **TACHIBANA, Hideaki**
Hiratsuka-shi
Kanagawa 254-8567 (JP)
• **HISADA, Youhei**
Oyama-shi
Tochigi 323-8558 (JP)
• **OHTA, Kazuo**
Hiratsuka-shi
Kanagawa 254-8567 (JP)

(74) Representative: **Goddard, Heinz J.**
Forrester & Boehmert
Pettenkoferstrasse 20-22
80336 München (DE)

(54) **VARIABLE VALVE DEVICE AND CONTROL METHOD THEREOF**

(57) Provided are a variable valve device and a control method thereof for actually measuring a variable performance of valve response with changes in temperature environment and accurately correcting valve opening and closing timing of the variable valve based on measurement results. The variable valve device has a hydraulic actuator that implements discharge of operating oil in a pressure chamber of a cylinder by a closing movement of an intake valve, and a hydraulic control valve that controls supply and discharge of the operating oil to and from the hydraulic actuator by opening and closing, and prevents outflow of the operating oil from the pressure chamber during closing, thereby preventing the closing movement of the intake valve. After a signal that closes the hydraulic control valve in a current engine control cycle is output at a prescribed crank angle, the crank angle of a crankshaft, when surge pressure in the pressure chamber has exceeded a prescribed threshold value resulting from closing of the hydraulic control valve, is determined as an operation start crank angle of the hydraulic control valve, and at least either an output crank angle or a waveform of the signal that closes the hydraulic control valve is corrected based on the operation start crank angle, and moreover, the corrected signal is output to the hydraulic control valve in a next control cycle.

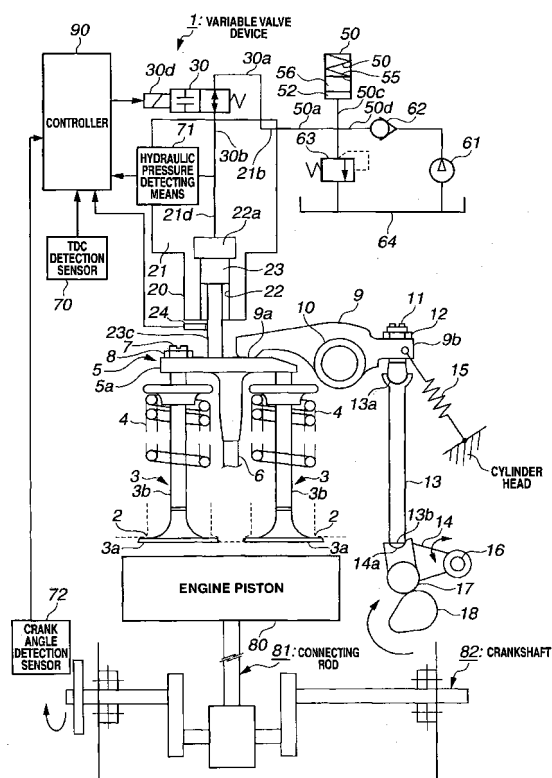


FIG.1

Description

TECHNICAL FIELD

[0001] The present invention relates to a variable valve device and a control method thereof.

BACKGROUND ART

[0002] Engines installed in vehicles have conventionally been provided with a variable valve device that adjusts intake air volume and exhaust air volume by controlling the timing at which intake valves and exhaust valves open and close corresponding to the operating status of the engine.

[0003] Variable valve devices consist of those using an electric motor or hydraulic pressure as a drive source, and hydraulically driven variable valve devices that are driven by hydraulic pressure, responsiveness of hydraulic pressure control may decrease depending on the temperature environment, such as during cold weather or during engine starting, thereby resulting in a decrease in control accuracy of the variable valve device.

[0004] A decrease in control accuracy of the variable valve device can cause a change in the opening and closing timing of intake valves resulting in a change in compression ratio which in turn has an effect on combustion.

[0005] Patent Document 1 discloses technology relating to a variable valve control system and control method that determines the response performance of a valve actuator, for which the response performance thereof varies according to the viscosity of an operating oil, based on the result of measuring the viscosity of the operating oil, and controls the valve actuator by correcting opening and closing timing of a variable valve provided with the valve actuator based on the determined response performance.

[0006] In addition, Patent Document 2 discloses technology relating to an engine valve control system and control method that controls the activation timing of a variable valve corresponding to the temperature environment of an engine, such as engine temperature, operating oil temperature or fluid temperature.

PRIOR ART DOCUMENTS

Patent Documents

[0007] Patent Document 1: U.S. Patent No. 7178491

Patent Document 2: U.S. Patent No. 7059282

DISCLOSURE OF THE INVENTION

Problems to be Solved by the Invention

[0008] The technologies indicated in the above-mentioned Patent Documents 1 and 2 relate to indirectly estimating an activation timing of a variable valve based on a measured result of a temperature environment such as viscosity of operating oil or an engine temperature of operating oil temperature, and since they do not measure actual opening and closing timing of an intake valve or exhaust valve according to a change in the viscosity of operating oil or change in a temperature environment, they have the shortcoming of being unable to accurately correct opening and closing timing of the variable valve.

[0009] With the foregoing in view, an object of the present invention is to provide a variable valve device and a control method thereof that actually measure the response performance of a variable valve according to a change in the temperature environment, and accurately correct the opening and closing timing of the variable valve based on the result of that measurement.

Means to Solve the Problems

[0010] In order to achieve the above-mentioned object, a first invention is a hydraulically driven variable valve device that has a valve train that opens or closes an intake port by moving an intake valve of an engine, and varies a crank angle at which the intake port is fully closed by the valve train, and this valve device has: a hydraulic actuator that is activated by closing movement of the intake valve and discharges, using a closing movement of the intake valve, operating oil in a pressure chamber provided in a cylinder; a hydraulic control valve that controls supply and discharge of operating oil to and from the hydraulic actuator by opening and closing and prevents outflow of the operating oil from a pressure chamber of the hydraulic actuator when closing is implemented, thereby preventing the closing movement of the intake valve; a crank angle detection sensor that detects a crank angle indicating an angle of rotation of the engine crankshaft;

a TDC detection sensor that detects that each cylinder of the engine has reached a top dead center (TDC); hydraulic pressure detecting means that detectd hydraulic pressure of the pressure chamber of the hydraulic actuator; and a controller that stops, during implementation of the closing movement of the intake valve, the closing movement of the intake valve and outputs a signal that closes the hydraulic control valve in order to maintain an open state of the intake port at a prescribed opening for a prescribed amount of time when a prescribed crank angle is determined to have been reached based on a detection signal from the TDC detection sensor and a detection signal from the crank angle detection sensor, wherein, after outputting a signal that closes the hydraulic control valve at the prescribed crank angle in the current control cycle, the controller monitors hydraulic surge pressure of the pressure chamber resulting from closing of the hydraulic control valve in use of a detection signal of the hydraulic pressure detecting means, determines an operation start crank angle of the hydraulic control valve based on the surge pressure, corrects at least either an output crank angle or waveform of a signal that closes the hydraulic control valve based on the operation start crank angle, and outputs the corrected signal to the hydraulic control valve in a next control cycle.

[0011] In addition, a second invention relates to the first invention, wherein the controller corrects at least either an output crank angle or waveform of a signal that closes the hydraulic control valve so that the crank angle, at which the intake port is fully closed when the closing movement of the intake valve is completed, is within a prescribed range after the signal that closes the hydraulic control valve is switched off.

[0012] In addition, a third invention is a control method for a variable valve device that varies a crank angle at which an intake port is fully closed by moving an intake valve of an engine by a valve train, opening or closing the intake port, and activating a hydraulic actuator by moving the intake valve to prevent outflow of operating oil from a pressure chamber provided in a cylinder of the hydraulic actuator, the method comprising: implementing discharge of the operating oil in the pressure chamber of the hydraulic actuator by activating the hydraulic actuator by closing movement of the intake valve in the current control cycle; closing a hydraulic control valve that controls supply and discharge of operating oil to and from the hydraulic actuator when a crank angle reaches a prescribed crank angle during closing movement of the intake valve after each cylinder of an engine reaches a top dead center, preventing outflow of the operating oil from the pressure chamber, thereby preventing closing movement of the intake valve; stopping the closing movement of the intake valve by closing the hydraulic control valve for a prescribed amount of time, maintaining an open state of the intake port at a prescribed opening for a prescribed amount of time; monitoring surge pressure of hydraulic pressure of the pressure chamber resulting from prevention of outflow of the operating oil from the pressure chamber, determining timing of the start of actual variable valve actuation (VVA) based on the surge pressure; correcting at least either an output crank angle or a waveform of a signal that closes the hydraulic control valve based on the actual VVA starting timing, and outputting the corrected signal to the hydraulic control valve in a next control cycle.

[0013] In addition, a fourth invention is characterized by the method according to the third invention, comprising: comparing the actual VVA starting timing with a preset prescribed VVA starting timing; and carrying out, when the actual VVA starting timing comes after the prescribed VVA starting timing, at least either correction in which an output crank angle of the signal that closes the hydraulic control valve of the next control cycle is made to be earlier than the current output crank angle, or correction in which an output value of the signal that closes the hydraulic control valve of the next control cycle is made to be larger than the current output value, while on the other hand, carrying out, when the actual VVA starting timing comes before the prescribed VVA starting timing, at least either correction in which the output crank angle of the signal that closes the hydraulic control valve of the next control cycle is made to be later than the current output crank angle, or correction in which the output value of the signal that closes the hydraulic control valve of the next control cycle is made to be smaller than the current output value.

[0014] In addition, a fifth invention is characterized by the method according to the third invention, the method comprising: comparing the actual VVA starting timing with the preset prescribed VVA starting timing; and carrying out, when the actual VVA starting time comes after the prescribed VVA starting timing, at least either correction in which, an output time of the signal that closes the hydraulic control valve of the next control cycle is made to be longer than the current output time and an output crank angle is made to be earlier than the current output crank angle, or correction in which the output time of the signal that closes the hydraulic control valve of the next control cycle is made to be longer than the current output time and an output value is made to be larger than the current output value, and carrying out, when the actual VVA starting timing is comes before the prescribed VVA starting timing, at least either correction in which the output time of the signal that closes the hydraulic control valve of the next control cycle is made to be shorter than the current output time and the output crank angle is made to be later than the current output crank angle, or correction in which the output time of the signal that closes the hydraulic control valve of the next control cycle is made to be shorter than the current output time and the output value is made to be smaller than the current output value.

(Effect of the Invention)

[0015] According to the present invention, since response performance from the time an activating command is issued to a hydraulically driven variable valve until the time an operation corresponding to that activating command is actually

executed by the variable valve is measured, and opening and closing timing of the variable valve is corrected so as to be within a prescribed timing based on that response performance, opening and closing timing of the variable valve can be controlled more accurately to a prescribed timing regardless of the temperature environment of operating oil.

[0016] According to the present invention, since the execution of an operation by a variable valve corresponding to an operating command issued to the variable valve is detected based on a result of measuring hydraulic pressure of a hydraulic actuator of the variable valve, actual operation of the variable valve can be measured more accurately, and response performance of the variable valve can be more accurately detected based on the result of that measurement.

BEST MODE FOR CARRYING OUT THE INVENTION

[0017] The following provides a detailed explanation of an embodiment of the variable valve device and control method thereof according to the present invention with reference to the drawings.

[0018] FIG 1 is a conceptual drawing showing a variable valve device 1 according to the present invention, and in the present embodiment, although the variable valve device 1 is explained on the premise of being applied to a four-cycle diesel engine, the variable valve device according to the present invention is not limited to the present embodiment.

[0019] As shown in FIG. 1, the variable valve device 1 is composed of a controller for controlling the variable valve device (to be simply referred to as a controller) 90, a gap sensor 24, a top dead center (TDC) detection sensor 70, hydraulic pressure detecting means 71, a crank angle detection sensor 72 and a diesel engine to which the variable valve device 1 is applied.

[0020] The controller 90 carries out control according to the present invention based on measurement results of various signals from each of the above-mentioned sensors.

[0021] The gap sensor 24 is connected to the controller 90, is arranged to the side of a rod portion 23c of a piston 23 to be subsequently described, measures a gap between the rod portion 23c and the gap sensor 24, and outputs that measurement signal in the form of a valve lift signal (see FIG 10D) to the controller 90.

[0022] The TDC detection sensor 70 is connected to the controller 90, detects that an engine piston 80 of an intake stroke is located at top dead center for each cylinder of the diesel engine, and outputs a TDC detection sensor signal (see FIG. 10A) to the controller 90.

[0023] The hydraulic pressure detecting means 71 is connected to the controller 90, measures hydraulic pressure of a pressure chamber of a hydraulic actuator 20 to be subsequently described, and outputs that measurement signal in the form of a hydraulic pressure signal (see FIG. 10E) to the controller 90.

[0024] The crank angle detection sensor 72 measures an angle of rotation of a crankshaft 82 of the diesel engine (to be simply referred to as crank angle), and outputs the measurement signal in the form of an engine speed detection signal (pulse output of the number of pulses corresponding to the crank angle, see FIG. 10B) to the controller 90.

[0025] More specifically, a pulse signal is output for each prescribed angle of rotation in manner of a crank angle of 0 degrees, 30 degrees, 60 degrees and the like up to 360 degrees, and the controller 90 detects the crank angle by measuring this output pulse signal.

[0026] The diesel engine has a cylinder block and a cylinder head, and a cylindrical cylinder not shown is provided in the cylinder block that enables the engine piston 80 to slide in the vertical direction. In addition, a pair of intake ports 2 that penetrate to the outside of the cylinder and a pair of exhaust ports not shown are provided in the cylinder head.

[0027] An intake valve 3 that moves so as to close or open the intake port 2 (vertical direction in FIG. 1) is respectively arranged in the pair of intake ports 2, and an exhaust valve (not shown) that moves to close or open the exhaust port is respectively provided in the pair of exhaust ports.

[0028] Each intake valve 3 and each exhaust valve is a poppet valve having an umbrella shape, and has a valve portion (umbrella-shaped portion) 3a that respectively closes or opens the intake port 2 and the exhaust port, and a stem (rod-like portion) 3b that slides within the cylinder head.

[0029] A valve spring 4 is respectively attached to the stem 3b of the pair of intake valves 3 respectively inserted into each intake port 2, and each valve spring 4 urges in a direction in which the valve portion 3a of each intake valve 3 closes the intake port 2.

[0030] A crosshead 5, having a T-shape when viewed from the side and which pushes on the ends of the stems 3b of the pair of intake valve 3, is provided above the cylinder head. The crosshead 5 is guided by a shaft 6 provided parallel to the direction of motion of each intake valve 3, and is able to ascend and descend in the direction of motion of each intake valve 3 (vertical direction in FIG. 1).

[0031] Thus, when the crosshead 5 is lowered, the crosshead 5 pushes on the ends of the stems 3b of the pair of intake valves 3, and moves in a direction in which each intake valve 3 opens the intake ports 2 in opposition to the urging force of each valve spring 4.

[0032] In addition, adjustment screw 7 that adjusts so that the intake valves 3 and the crosshead 5 are pushed together is provided on one arm 5a of the crosshead 5. The adjustment screw 7 is able to be screwed into the crosshead 5, and is able to adjust the gap between the crosshead 5 and one of the intake valves 3 of the pair of intake valves 3.

[0033] For example, the above-mentioned gap is adjusted so that the pair of intake valves 3 simultaneously open and close each intake port 2.

[0034] A lock nut 8 is screwed onto the adjustment screw 7, and the adjustment screw 7 can be prevented from loosening by tightening the lock nut 8 to the crosshead 5 after adjusting the adjustment screw 7.

[0035] A rocker arm 9 is provided above the crosshead 5 (in FIG. 1). The rocker arm 9 is able to pivot by using a rocker shaft 10 as an axis thereof, and is composed so that one end (left end in FIG. 1) serves as a pushing portion 9a that pushes on the crosshead 5, while the other end serves as an operating portion 9b.

[0036] The pushing portion 9a of the rocker arm 9 is arranged to be able to push on roughly the center of the crosshead 5, and when the rocker arm 9 pivots in the counter-clockwise direction (in FIG. 1), the pushing portion 9a of the rocker arm 9 pushes on the crosshead 5 and the intake valves 3 move downward and open the intake ports 2.

[0037] On the other hand, when the rocker arm 9 pivots clockwise (in FIG. 1), the intake valves 3 move upward due to the urging force of the valve springs 4, and close the intake ports 2 while raising the crosshead 5.

[0038] An adjustment screw 11 that adjusts a gap between the pushing portion 9a and the crosshead 5 is screwed into the operating portion 9b of the rocker arm 9. The adjustment screw 11 has a semi-spherical shape on one end thereof, while external threads are formed on the other end.

[0039] In addition, a lock nut 12 is screwed into the other end of the adjustment screw 11 that is screwed into the operating portion 9b of the rocker arm 9, and the adjustment screw 11 can be prevented from loosening by tightening the lock unit 12 to the rocker arm 9.

[0040] One end of the adjustment screw 11 having a semi-spherical shape is housed in one end of a push rod 13.

[0041] A semi-spherical indentation 13a is formed in one end of the push rod 13, and is able to house the semi-spherically shaped end of the adjustment screw 11.

[0042] The push rod 13 causes the rocker arm 9 to pivot in the counter-clockwise direction (in FIG. 1). The other end 13b of the push rod 13 is housed in a push rod housing portion 14a provided in the top of an arm portion of a tappet arm 14.

[0043] A return spring 15 is suspended between the operating portion 9b of the rocker arm 9 and the cylinder head. The return spring 15 urges the rocker arm 9 in the clockwise direction (in FIG. 1), and is able to sustain one end of the adjustment screw 11 in a state of being housed in the indentation 13a of the push rod 13.

[0044] Furthermore, the return spring 15 may also be a helical torsion coil spring wound around the rocker shaft 10 provided the rocker arm 9 urges in the clockwise direction (in FIG. 1).

[0045] In this case, one end of the coil spring is fixed to the rocker arm 9 while the other end is fixed to the cylinder head.

[0046] The tappet arm 14 is attached so as to be able to rotate by using a tappet shaft 16 as a shaft, and when the tappet arm 14 rotates in the clockwise direction (in FIG. 1), the tappet arm 14 pushes up on the push rod 13 causing the rocker arm 9 to swivel in the counter-clockwise direction (in FIG. 1).

[0047] In addition, a roller follower 17 is rotatably attached beneath the arm portion of the tappet arm 14, and a cam 18 that makes rolling contact with the roller follower 17 is rotatably provided beneath the roller follower 17.

[0048] The cam 18 rotates in coordination with the engine crankshaft 82, and is able to move the intake valves 3 in the direction in which the intake ports 2 open through the tappet arm 14, the push rod 13, the rocker arm 9 and the crosshead 5.

[0049] Thus, timing of opening of the intake ports 2 and the amount of valve lift of the intake valves 3 is controlled according to the outer shape (cam profile) of the cam 18.

[0050] Furthermore, for the sake of convenience in explaining the present embodiment, movement by the intake valves 3 in the direction that opens the intake ports 2 is referred to as "opening movement", movement in the direction that closes the intake ports 2 is referred to as "closing movement", an amount corresponding to a distance from the position of the intake valves 3 when the intake ports 2 are fully closed to the position of the intake valves 3 when the intake ports 2 are fully open is referred to as "valve lift", and "valve lift" is indicated with a positive value corresponding to the amount thereof or with a value of zero (0) when the intake ports 2 are fully closed.

[0051] The hydraulic actuator 20 is provided above the crosshead 5. The hydraulic actuator 20 is arranged so that the end of the rod portion 23c of the piston 23 is able to contact the crosshead 5 and move in response to operation of the crosshead 5.

[0052] The hydraulic actuator 20 is composed so that the end of the rod portion 23c pushes on the crosshead 5 at a prescribed timing and the intake valves 3 are able to sustain the intake ports 2 in an open state at a prescribed opening regardless of operation of the cam 18, the tappet arm 14, the push rod 13 and the rocker arm 9.

[0053] The hydraulic actuator 20 applied in the present embodiment is of a single action type, a cylinder 22 is integrally formed in a block 21, and is composed to be able to house and mount a hydraulic control valve 30 that controls the flow of operating oil.

[0054] The hydraulic control valve 30 is, for example, a two-port solenoid valve having an input port 30a and an output port 30b.

[0055] A supply/discharge conduit 21d that connects the cylinder 22 and the output port 30b of the hydraulic control valve 30, and a first conduit 21b that connects an output port 50a of an accumulator 50 and the input port 30a of the

hydraulic control valve 30, are formed in the block 21.

[0056] A cylindrical pressure chamber 22a is formed in the cylinder 22 of the hydraulic actuator 20, one end of the pressure chamber 22a is open to allow insertion of the piston 23, and is composed so as to be closed by the piston 23.

[0057] In addition, the other end of the pressure chamber 22a is connected to the output port 30b of the hydraulic control valve 30 through the supply/discharge conduit 21d.

[0058] The piston 23 is slidably housed in the axial direction (vertical direction in FIG. 1) in the cylinder 22. The piston 23 has the rod portion 23c that advances to the outside of the cylinder 22.

[0059] The rod-portion 23c has a tapered shape that gradually becomes narrower moving from the base to the tip, and is able to push on the crosshead 5.

[0060] As was previously described, the gap sensor 24 connected to the controller 90 is provided to the side of the rod portion 23c of the piston 23, and a gap between the rod portion 23c and the gap sensor 24 is measured by this gap sensor 24.

[0061] This gap sensor 24 is able to measure the gap between itself and the rod portion 23c by measuring eddy current, for example. Since the rod portion 23c has a tapered shape, in the case the rod portion 23c advances from the cylinder 22, the resulting decrease in the above-mentioned gap is measured, while in the case the rod portion 23c retracts into the cylinder 22, the resulting increase in the gap is measured.

[0062] The controller 90 is able to monitor operation of the rod portion 23c by monitoring the gap between the gap sensor 24 and the rod portion 23c measured by the gap sensor 24, and is therefore able to detect operation (amount of movement) of the intake valves 3 from operation of the rod portion 23c that contacts the crosshead 5 and operates together with the crosshead 5.

[0063] The hydraulic control valve 30 is attached to the block 21. The hydraulic control valve 30 is a two-port solenoid valve having the input port 30a and the output port 30b as previously described.

[0064] The input port 30a and the output port 30b are in communication when the hydraulic pressure valve 30 is in the normal state (when a solenoid 30d is not magnetized), while the communicating state of the input port 30a and the output port 30b is interrupted when the solenoid 30d is magnetized.

[0065] Thus, the hydraulic control valve 30 is able to switch between a state that allows operating oil to be supplied and a state in which the supply of operating oil is interrupted by magnetization and demagnetization of the hydraulic control valve 30.

[0066] More specifically, when operating oil is supplied to the supply/discharge conduit 21d via the first conduit 21b and the hydraulic control valve 30, the operating oil is supplied to the pressure chamber 22a, the supplied operating oil acts on the piston 23, the piston 23 is pushed out from the cylinder 22, and the rod portion 23c lowers.

[0067] Subsequently, when the solenoid 30d of the hydraulic control valve 30 is magnetized, the communicating state between the input port 30a and the output port 30b is interrupted, and even if the rod portion 23c is pushed up towards the cylinder 22 (upward in FIG. 1) while in this state, operating oil is sealed in the pressure chamber 22a, and operation of the piston 23 is obstructed and stopped by this sealed operating oil.

[0068] Subsequently, when the solenoid 30d of the hydraulic control valve 30 is demagnetized, the communicating state between the input port 30a and the output port 30b is restored, and when the rod portion 23c of the piston 23 is pushed up towards the cylinder 22 while in this state, the piston 23 rises and operating oil flows out from the supply/discharge conduit 21d.

[0069] The operating oil that has flowed out from the supply/discharge conduit 21d gradually flows outside the hydraulic actuator 20 via the output port 30b and input port 30a of the hydraulic control valve 30 as well as the first conduit 21b, the piston 23 is housed in the pressure chamber 22a of the cylinder 22, and the series of operations of the hydraulic actuator 20 ends.

[0070] The solenoid 30d of the hydraulic control valve 30 is connected to the controller 90, and the timing and duration of magnetization of the hydraulic control valve 30 are controlled by the controller 90.

[0071] Furthermore, the controller 90 is able to arbitrarily control the hydraulic control valve 30 is millisecond (1/1000 second) units.

[0072] The previously described hydraulic pressure detecting means 71 connected to the controller 90 is provided in the pressure chamber 22a of the hydraulic actuator 20, and hydraulic pressure of the pressure chamber 22a is measured by this hydraulic pressure detecting means 71.

[0073] The controller 90 is able to detect surge pressure when operating oil is sealed in the pressure chamber 22a by monitoring hydraulic pressure of the pressure chamber 22a measured by the hydraulic pressure detecting means 71.

[0074] The output port 50a of the accumulator 50 is connected to the first conduit 21b of the block 21.

[0075] The accumulator 50 serves as a pressure storing means that accumulates hydraulic pressure, and the accumulator 50 in the present embodiment is a mechanical accumulator.

[0076] The accumulator 50 has the output port 50a, an input conduit 50c that communicates with the output port 50a, and an input port 50d that communicates with the input conduit 50c, and a pressure storage unit 52 is provided in the input conduit 50c.

[0077] The pressure storage unit 52 has a cylinder 55 formed in the body of the accumulator 50. The cylinder 55 communicates with the input conduit 50c, and enables operating oil supplied from the input port 50d and operating oil supplied from the output port 50a to flow therein.

[0078] A plunger 56, which slides in the axial direction of the cylinder 55, and a compression spring 57, which urges the plunger 56 towards a bottom wall of the cylinder 55 (downward in FIG. 1), are provided within the cylinder 55.

[0079] Thus, even if low-pressure operating oil is supplied from the input port 50d of the accumulator 50 and the operating oil pushes on the plunger 56, the plunger 56 is unable to overcome the urging force of the compression spring 57, and operating oil flows out from the output port 50a.

[0080] On the other hand, if operating oil at a pressure higher than the operating oil that has flowed out from the cylinder 22 of the hydraulic actuator 20, and higher than the operating oil that is supplied from the input port 50d, is supplied from the output port 50a of the accumulator 50, the operating oil pushes on the plunger 56, and the plunger 56 moves in opposition to the urging force of the compression spring 57. At this time, operating oil is stored (accumulated) in the pressure storage unit 52.

[0081] The hydraulic actuator 20, the hydraulic control valve 30 and the accumulator 50 form a hydraulic circuit 60 like that shown in FIG. 2 (hydraulic pressure circuit surrounded by the dotted line). Low-pressure operating oil can be supplied to the hydraulic circuit 60 via the input port 50d of the accumulator 50 from a lubrication unit 61 which is ancillary to the engine and which supplies lubricating oil to the engine.

[0082] In addition, a check valve 62 is arranged between the lubrication unit 61 and the hydraulic circuit 60, operating oil is supplied from the lubrication unit 61 to the hydraulic circuit 60 only in the case hydraulic pressure of the hydraulic circuit 60 is less than hydraulic pressure of the lubrication unit 61, and operating oil is prevented from flowing into the lubrication unit 61 from the side of the hydraulic circuit 60 by this check valve 62.

[0083] In addition, a relief valve 63 is provided between the check valve 62 and the hydraulic circuit 60. The relief valve 63 discharges operating oil of the hydraulic circuit 60 into an engine oil pan 64 in the case the pressure of the hydraulic circuit 60 has become higher than a preset pressure.

[0084] Due to the presence of the hydraulic circuit 60 formed in this manner, when the engine is started, low-pressure operating oil is supplied from the lubrication unit 61 ancillary to the engine via the check valve 62 to the accumulator 50, the hydraulic control valve 30 and the hydraulic actuator 20 in that order, and the operating oil is filled into the hydraulic control valve 30 and the hydraulic actuator 20.

[0085] Accompanying engine operation, motive power is transmitted to the cam 18, the tappet arm 14, the push rod 13, the rocker arm 9 and the crosshead 5 in that order in coordination with the engine piston 80 and the crankshaft 82, and during the engine intake stroke, the intake ports 2 are opened and closed by opening movement or closing movement of the intake valves 3, while during the engine compression stroke, explosion stroke and exhaust stroke, the intake ports 2 are closed by closing movement of the intake valves 3.

[0086] The following provides an explanation of operation of the variable valve device 1 configured in this manner with reference to FIGS. 3 to 9.

[0087] Furthermore, FIGS. 3 to 8 are schematic diagrams illustrating the action of the variable valve device 1, while FIG. 9 is a drawing indicating the relationship between cam rotation angle and valve lift during the engine intake stroke.

[0088] As shown in FIG. 3, in the compression stroke, explosion stroke and exhaust stroke of an engine to which the variable valve device 1 is applied, the intake valves 3 fully close the intake ports 2 due to urging force of the valve springs 4, the relationship between the rotation angle of the cam 18 and the valve lift at this time is as shown in the closing region of FIG. 9, and the value of valve lift of the intake valves 3 is zero regardless of the rotation angle of the cam 18.

[0089] In addition, as shown in FIG. 4, when the engine intake stroke is started, motive power is transmitted from the cam 18 rotating in coordination with the crankshaft 82 to the tappet arm 14, the push rod 13, the rocker arm 9 and the crosshead 5 in that order, and the intake valves 3 lower (opening movement) to gradually open the intake ports 2.

[0090] At this time, operating oil is supplied to the hydraulic control valve 30 and the hydraulic actuator 20 in that order from the lubrication unit 61, the rod portion 23c of the piston 23 gradually advances downward while contacting the crosshead 5, and operating oil accumulated in the accumulator 50 is gradually supplied to the pressure chamber 22a of the cylinder 22 of the hydraulic actuator 20.

[0091] In addition, the relationship between the rotation angle of the cam 18 and valve lift at this time is as shown in the opening action region of FIG. 9, and the valve lift of the intake valves 3 gradually increases accompanying an increase in the rotation angle of the cam 18.

[0092] Furthermore, since the force at which the rod portion 23c advances downward does not have sufficient force to push the intake valves 3 downward in opposition to the urging force of the valve springs 4 of the intake valves 3, the intake valves 3 are not pushed downward as a result of the rod portion 23c pushing on the crosshead 5 regardless of the operation of the rocker arm 9.

[0093] In addition, in the case operating oil has not accumulated in the accumulator 50, operating oil is gradually supplied from the lubrication unit 61 ancillary to the engine to the hydraulic circuit 60 via the check valve 62.

[0094] As shown in FIG. 9, when the valve lift of the intake valves 3 reaches a maximum (location of point P in the

drawing) accompanying opening movement of the intake valves 3, the intake ports 2 are fully open as shown in FIG. 5, after which the crosshead 5, the rocker arm 9, the push rod 13 and the tappet arm 14 operate by following the movement of the cam 18 as shown in FIG. 6, and the intake valves 3 move upward (closing movement) to gradually close the intake ports 2 due to the urging force of the valve springs 4 of the intake valves 3 and the return spring 15.

[0095] The relationship between the rotation angle of the cam 18 and valve lift is indicated by the closing action region A of FIG. 9 (curve extending from point P to point R through point Q), and valve lift gradually decreases accompanying an increase in the rotation angle of the cam 18. At this time, as a result of the rod portion 23c of the piston 23 gradually being housed in the cylinder 22, operating oil of the pressure chamber 22a of the cylinder 22 is accumulated in the accumulator 50.

[0096] Thus, the hydraulic actuator 20 has the function of a piston pump, and operating oil is accumulated in the accumulator 50 via the hydraulic actuator 20 and the hydraulic control valve 30.

[0097] As shown in the closing action region A of FIG. 9, when the valve lift reaches a minimum (location of point R in the drawing), namely when the valve lift becomes zero, the intake valves 3 fully close the intake ports 2 as shown in FIG. 7.

[0098] Within the closing action region A described above, when the hydraulic control valve 30 is magnetized at a prescribed crank angle (cam rotation angle) in the closing action region A, as shown in FIG. 8 the communicating state between the input port 30a and the output port 30b of the hydraulic control valve 30 is interrupted, and the hydraulic control valve 30 changes from an operating oil supply/discharge state to an operating oil interrupted state.

[0099] Whereupon, operating oil is sealed in the pressure chamber 22a of the cylinder 22, and operation of the piston 23 is obstructed and stopped by operating oil sealed in the pressure chamber 22a.

[0100] Whereupon, the rod portion 23c of the stopped piston 23 pushes on the crosshead 5, and as shown in FIG. 8, the intake valves 3 sustain the open state of the intake ports 2 at a prescribed opening, and closing timing of the intake ports 2 during the engine intake stroke is delayed.

[0101] The relationship between the rotation angle of the cam 18 and valve lift at this time is indicated by the closing delay region of FIG. 9, and during a prescribed time when the hydraulic control valve 30 is magnetized, the valve lift of the intake valves 3 remains constant even if the rotation angle of the cam 18 increases.

[0102] On the other hand, even in the case the rod portion 23c of the stopped piston 23 pushes on the crosshead 5 and the intake valves 3 sustain the intake ports 2 in an open state, the rocker arm 9 presses against the push rod 13 and is controlled by the outer shape (cam profile) of the cam 18. Thus, the push rod 13 does not come off the rocker arm 9, and the rocker arm 9 operates so as to form a gap between the crosshead 5 and the rocker arm 9.

[0103] When the hydraulic control valve 30 is demagnetized after a preset amount of time has elapsed, the hydraulic control valve 30 is switched to a state in which the input port 30a and the output port 30b are communicating, and the hydraulic control valve 30 changes from the operating oil interrupted state to the operating oil supply/discharge state.

[0104] Whereupon, the intake valves 3 gradually close the intake ports 2 due to the urging force of the valve springs 4 of the intake valves 3, and at this time, the crosshead 5 rises and pushes the rod portion 23c of the piston 23 up, the piston 23 is gradually again housed in the cylinder 22, and operating oil in the pressure chamber 22a of the cylinder 22 is accumulated in the accumulator 50 via the hydraulic control valve 30.

[0105] The relationship between the rotation angle of the cam 18 and valve lift at this time is shown in the closing action region B of FIG. 9, and valve lift gradually decreases accompanying an increase in the rotation angle of the cam 18.

[0106] When the valve lift reaches a minimum (location of point S in the drawing), namely when the valve lift becomes 0, as shown in the closing action region B, the intake ports 2 are fully closed by the intake valves 3 as shown in the previously described FIG. 7.

[0107] In this manner, as a result of magnetizing the hydraulic control valve 30 at a prescribed timing during the intake stroke, the variable valve device 1 is able to carry out control in which rising movement of the rod portion 23c of the piston 23 and closing movement of the intake valves 3 are stopped, and the timing at which the intake ports 2 are closed by the intake valves 3 is delayed (to be referred to as "intake closing delay control" for the sake of convenience in the explanation), and by carrying out this control corresponding to the operating status of the engine, intake volume can be adjusted to that which is suitable for the engine operating status.

[0108] However, if the temperature of the operating oil that drives the hydraulic control valve 30 changes, the operating speed of the hydraulic control valve 30 may change, and this change in operating speed may cause a shift in the opening and closing timing of the variable valve, thereby resulting in variations in the timing at which the intake valves 3 fully close the intake ports 2 (to be referred to as the "closing end" for the sake of convenience in the explanation).

[0109] Variations in this closing end timing may result in a change in compression ratio that may have an effect on combustion.

[0110] Therefore, the intake closing delay control of the variable valve device 1 of the present invention is configured so as to suppress variations in the closing end timing of the intake valves 3 even if there is a change in response performance indicating operating speed of the hydraulic control valve 30 and the like caused by the temperature environment.

[0111] FIGS. 10A-10E are explanatory drawings indicating an example of a measurement signal measured with each sensor accompanying operation of the variable valve device 1 described above.

[0112] Furthermore, FIG. 10A is a drawing indicating an example of a TDC detection sensor signal 100 measured by the TDC detection sensor 70, FIG. 10B is a drawing indicating an example of an engine speed detection signal (pulse signal) 110 measured by the crank angle detection sensor 72, FIG. 10C is a drawing indicating an example of an intake closing delay control signal (to be referred to as a "VAA starting signal") 120 generated and output by the controller 90, FIG. 10D is a drawing indicating an example of a valve lift signal 130 measured by the gap sensor 24, and FIG. 10E is a drawing indicating an example of a hydraulic pressure signal 140 measured by the hydraulic pressure detecting means 71.

[0113] Control of the variable valve device 1 is carried out by the controller 90, and in order to carry out closing delay control during the intake step, the controller 90 monitors the TDC detection sensor signal 100 output from the TDC detection sensor 70, the engine speed detection signal (pulse signal of a number of pulses corresponding to the crank angle) 110 output from the crank angle detection sensor 72, the valve lift signal 130 output from the gap sensor 24, and the hydraulic pressure signal 140 output from the hydraulic pressure detecting means 71 once the engine piston 80 is located at top dead center during the intake stroke for each cylinder of the engine, and when the TDC detection sensor signal 100 is detected for each cylinder, together starting to count the number of pulses of the engine speed detection signal 110, engine speed is calculated based on the engine speed detection signal 110.

[0114] Operation of the intake valves 3 during this period is measured by the gap sensor 24 as the closing region and opening action region of the valve lift signal 130, and hydraulic - pressure of the pressure chamber 22 is measured by the hydraulic pressure detecting means 71 as the operating oil supply/discharge region A of the hydraulic pressure signal 140.

[0115] When the counted number of pulses reaches a preset number of pulses (to be referred to as the "VVA starting set pulses") 111 (see FIG. 10B), the controller 90 determines that the time for output of the VVA starting signal that magnetizes the hydraulic control valve 30 (to be referred to as the "magnetization command timing (Pcom)" for the sake of convenience in the explanation) has been reached, generates the VVA starting signal 120 (see FIG. 10C) and outputs to the hydraulic control valve 30 to magnetize the hydraulic control valve 30.

[0116] Furthermore, the VVA starting set pulses 111 is set so that the counted number of pulses of the engine speed detection signal 110 reaches the VVA starting set pulses 111 during closing movement of the intake valves 3.

[0117] When the VVA starting signal 120 is output to the hydraulic control valve 30, a magnetization current corresponding to a voltage value V_c of the VVA starting signal 120 is output to the hydraulic control valve 30 and the hydraulic control valve 30 is magnetized during the time the VVA starting signal is set to ON, while output of the magnetization current to the hydraulic control valve 30 is stopped and the hydraulic control valve 30 is demagnetized during the time it is set to OFF.

[0118] Thus, the hydraulic control valve 30 is magnetized by the VVA starting signal 120 only for a preset fixed time period (to be referred to as "VVA holding time" for the sake of convenience of the explanation) TW from the magnetization command timing (Pcom).

[0119] When the hydraulic control valve 30 is magnetized, since the hydraulic control valve 30 changes from the operating oil supply/discharge state to the operating oil interrupted state as previously described, operating oil is sealed in the pressure chamber 22a of the cylinder 22, and the rod portion 23c of the piston 23 stops, during the VVA holding time TW, the rod portion 23c stops, closing movement of the intake valves 3 also stop, the intake ports 2 are sustained in an open state at a prescribed opening, and closing timing of the intake ports 2 is delayed.

[0120] Operation of the intake valves during this time is measured as the closing delay region of the valve lift signal 130 (see FIG. 10D), and hydraulic pressure of the pressure chamber 22a is measured as the operating oil interruption range of the hydraulic pressure signal 140 (see FIG. 10E).

[0121] In the operating oil interruption region of the hydraulic pressure signal 140, since the rod portion 23c stops after having risen by contacting the crosshead 5, the rising force of the crosshead 5 pushes on the stopped rod portion 23c, and that pressure is measured as a sudden increase in hydraulic pressure (surge pressure) in the pressure chamber 22a, after which it is measured as an oscillating hydraulic pressure signal until the hydraulic control valve 30 is demagnetized.

[0122] When the VVA starting signal 120 is set to OFF after the VVA starting signal 120 has been set to ON and the VVA holding time TW has elapsed, output of the magnetization current to the pressure control valve 30 is stopped, and the pressure control valve 30 is demagnetized.

[0123] When the hydraulic control valve 30 has been demagnetized, the hydraulic control valve 30 changes from the operating oil interrupted state to the operating oil supply/discharge state as previously described, and the intake valves 3 carry out closing movement due to the urging force of the valve springs 4 of the intake valves 3, thereby gradually closing the intake ports 2 after which the intake ports 2 are fully closed.

[0124] Operation of the intake valves 3 during this time is measured as the closing operation region B of the valve lift signal 130 (see FIG. 10D), while hydraulic pressure of the pressure chamber 22a is measured as the operating oil supply/

discharge region B of the hydraulic pressure signal 140 (see FIG. 10E).

[0125] However, response performance of the hydraulic control valve 30 may change according to the temperature of the operating oil as previously described.

[0126] FIGS. 11A-11C are explanatory drawings illustrating changes in response performance of the hydraulic control valve 30 caused by changes in the temperature of the operating oil and variations in the closing end timing of the intake valves.

[0127] Furthermore, FIG 11A is a drawing indicating an example of the VVA starting signal 120, FIG. 11B is a drawing indicating an example of the valve lift signal 131 during output of the VVA starting signal 120 in a certain operating oil temperature environment of the operating oil, and FIG. 11C is a drawing indicating an example of a valve lift signal 132 during output of the VVA starting signal 120 in another operating oil temperature environment.

[0128] As shown in FIGS. 11B and 11C, after the VVA starting signal 120 has been set to ON, the closing movement of the intake valves 3 stop, the amount of time until the intake valves 3 change to a state in which the intake ports 2 are sustained in an open state at a prescribed opening (to be referred to as the "VVA response performance (ΔP)" for the sake of convenience of the explanation) and the timing of the closing end of the intake valves 3 accompanying the VVA response performance (ΔP) (to be referred to as the "closing end timing (S)" for the sake of convenience of the explanation) are such that the VVA response performance (ΔP) becomes $\Delta P1$ the closing end timing (S) of the intake valves 3 at that time becomes S 1 in the case the operating oil is in a certain temperature environment, while in the case the operating oil is in a different temperature environment, the VVA response performance (ΔP) becomes $\Delta P2$ and the closing end timing (S) of the intake valves 3 at that time becomes S2.

[0129] In this manner, the closing end timing (S) of the intake valves 3 may change according to the temperature environment of the operating oil.

[0130] This change in the closing end timing (S) of the intake valves 3 may cause a change in combustion ratio and have an effect on combustion as previously described.

[0131] Therefore, in order to suppress variations in the closing end timing (S) of the intake valves 3, the controller 90 of the variable valve device 1 detects the timing at which control that sustains the open state of the intake ports 2 is actually implemented after setting the VVA starting signal 120 to ON based on the measurement result of the hydraulic pressure signal detected by the hydraulic pressure detecting means 71 of the pressure chamber 22a, and controls so that the detected timing is within the timing of a prescribed range C1.

[0132] More specifically, after the VVA starting signal 120 has been set to ON, the controller 90 detects a crank angle when hydraulic pressure of the pressure chamber 22a has exceeded a preset prescribed value (threshold) D1, and carries out the intake closing delay control so that variations in the closing end timing (S) of the intake valves 3 are suppressed, by correcting the timing at which the VVA starting signal is switched ON and OFF or by correcting the output waveform of the VVA starting signal, so that the detected crank angle is within a prescribed range C1 centering on a preset prescribed VVA operation start crank angle.

[0133] The following provides an explanation of this intake closing delay control according to the present invention with reference to FIGS. 12 to 16.

[0134] FIGS. 12 and 13 are flow charts showing the control sequence of the intake closing delay control, while FIGS. 14A-14E to 16A-16E are explanatory drawings showing examples of measurement signals measured with each sensor in the control sequence of the intake closing delay control.

[0135] As shown in FIGS. 12 and 14A-14E, in order to determine the VVA starting signal that magnetizes the hydraulic control valve 30 of the variable device 1 and the output timing thereof (magnetization command timing (Pcom)), the controller 90 monitors the TDC detection sensor signal 100 output from the TDC detection sensor 70 (see FIG. 14A), the engine speed detection signal 110 output from the crank angle detection sensor 72 (see FIG. 10B), a valve lift signal 133 output from the gap sensor 24 (refer to the valve lift signal indicated at points O, P, Q1 and S1 in FIG. 14D), and a hydraulic pressure signal 143 output from the hydraulic pressure detecting means 71 (refer to the hydraulic signal indicated with the broken line in FIG. 14E), and together with detecting the TDC detection signal 100 (S100) and calculating engine speed based on the engine speed detection signal, begins counting the number of pulses of the engine speed detection signal 110 (S101, S102).

[0136] When the counted number of pulses has reached a preset number of pulses (VVA starting set pulses 111) (see FIG. 10B) (YES in S103), the controller 90 determines that the above-mentioned magnetization command timing (Pcom) has been reached, and together with storing this magnetization command timing (Pcom) information in a storage device not shown, magnetizes the hydraulic control valve 30 by outputting an on command of a VVA starting signal 123 (see FIG 14B) to the hydraulic control valve 30 (S104).

[0137] The information of the magnetization command timing (Pcom) is indicated with, for example, the amount of time from the time the TDC detection sensor signal 100 is detected until the VVA starting signal 123 is set from OFF to ON (or may be indicated with a crank angle based on the number of pulses of the engine speed detection signal 110).

[0138] When the VVA starting signal 123 is output to the hydraulic control valve 30, the hydraulic control valve 30 is magnetized for the VVA holding time TW from the time of the magnetization command timing (Pcom) based on the VVA

starting signal 123.

[0139] Furthermore, in S103 in the case the counted number of pulses has not reached the

[0140] VVA starting set pulses 111 in S103 (NO in S103), the controller 90 waits until the number of pulses of the engine speed detection signal 110 reaches the VVA starting set pulses 111.

[0141] In S104, when the hydraulic control valve is magnetized, operating oil is sealed in the pressure chamber 22a of the cylinder 22, rising movement of the rod portion 23c of the piston 23 and closing movement of the intake valves 23 stop, the intake ports 2 are sustained in an open state at a prescribed opening, and operation of the intake valves 3 during that time is measured by the gap sensor 24 as the valve lift signal 133 (see FIG. 14D).

[0142] In addition, at that time, pushing force corresponding to the rising force of the crosshead 5 is applied to the rod portion 23c, hydraulic pressure of the pressure chamber 22a increases suddenly (surge pressure) due to this pressure, and is measured by the hydraulic pressure detecting means 71 as a surge pressure signal M1 of the hydraulic pressure signal 143 as shown in FIG. 14E.

[0143] A time (or crank angle) PM1 during which the surge pressure signal M1 of the hydraulic pressure signal 143 is measured indicates that closing movement of the intake valves 3 has actually stopped and that sustaining the intake ports 2 in an open state at a prescribed opening has begun.

[0144] Next, in S105, when the monitored hydraulic pressure signal 143 is detected to have exceeded a preset threshold D1, namely when the surge pressure signal M1 of the hydraulic pressure signal 143 is detected, the time (or crank angle) at that time is detected as the time (or crank angle) from the time of detection of the TDC detection sensor signal 100 by treating as the rise time (or crank angle) PM1 of the surge pressure signal M1 (to be referred to as the "VVA operation start timing (Pac)" for the sake of convenience of the explanation).

[0145] In S106, after waiting until the WA holding time Tw has elapsed, the VVA starting signal is set to ON in S107 when the WA holding time Tw has elapsed (YES).

[0146] Next, in S108, the WA response performance (ΔP) of the current control cycle is calculated according to the following equation (1) as the VVA response performance ($\Delta P1$) based on the actual determined VVA operation start timing (Pac) and a preset prescribed standard VVA operation start timing (Pacs) (S108).

[0147]

$$\text{VVA response performance } (\Delta P) = \text{VVA operation start timing}$$

$$(\text{Pac}) - \text{standard VVA operation start timing (Pacs)} \quad (1)$$

[0148] Here, VVA response performance (ΔP , $\Delta P1$) uses the standard VVA operation start timing (Pacs) as a standard for assessing responsiveness, and thus, may be a positive or negative value depending on the magnitude of the actual VVA operation start timing (Pac). In addition, the duration of control from the time of TDC detection to the next time of TDC detection is referred to as a "control cycle".

[0149] When the VVA response performance ($\Delta P1$) has been calculated, a determination is made as to whether or not that VVA response performance ($\Delta P1$) is within the preset prescribed range C1 (C1: range of $0 \leq |\Delta P| < \alpha$, wherein, $0 < \alpha$) (S109), and in the case it is within the range C1 (YES in S109), control in which a VVA starting signal of the same waveform as the VVA starting signal 123 of the current cycle is output to the hydraulic control valve 30 at the same output timing as the VVA starting signal 123 (magnetization command timing (Pcom)), is carried out during control for each cylinder of the next control cycle (S110).

[0150] Namely, if the VVA response performance ($\Delta P1$) is within the prescribed range C1, since the closing end timing (S) of the intake valves 3 accompanying that VVA response performance ($\Delta P1$) is treated as being within an allowed range of variation of closing end timing, correction of the magnetization command timing (Pcom) and the waveform of the VVA starting signal are not required for the next control cycle.

[0151] Furthermore, the prescribed range C1 is a value that indicates a range over which the magnitude of the VVA response performance (ΔP) is allowed, and indicates a value that indicates a range that does not have an effect on engine combustion if a variation in closing end timing is within this range.

[0152] On the other hand, in the case the calculated VVA response performance ($\Delta P1$) is not within the prescribed range C1 (NO in S109), processing proceeds to S111. As shown in FIG. 13, in S111, a determination is made as to whether or not the magnitude of the crank angle corresponding to the absolute value of the VVA response performance ($\Delta P1$) is within the range of less than 30%, 30% to less than 60% or 60% or more of a control range of the intake stroke (crank angle range able to be controlled by the variable valve device during the intake stroke). In the case the magnitude of the crank angle corresponding to the absolute value of the VVA response performance ($\Delta P1$) is less than 30% of the control range (YES in S112), the output timing and waveform of a VVA starting signal 124 are set for each cylinder of the next control cycle so that the waveform of the VVA starting signal 123 is corrected to the waveform of the VVA

starting signal 124 (see FIG. 14C) and output to the hydraulic control valve 30 in the next control cycle at the same output timing (magnetization command timing (Pcom)) of the VVA starting signal 123 of the current control cycle (S 113).

[0153] More specifically, the output timing of the VVA starting signal 124 output to the hydraulic control valve 30 in the next control cycle is made to be the same as the magnetization command timing (Pcom) of the VVA starting signal 123 of the current control cycle, and a voltage obtained by adding a prescribed adjustment voltage ΔV to an output voltage V_c of the VVA starting signal 123 is used for the output voltage of the VVA starting signal 124 to correct the closing end timing of the intake valves 3 so be within a prescribed timing range by improving the response performance of the hydraulic control valve 30.

[0154] Here, the absolute value of the adjustment voltage ΔV is an empirical value, and its sign is the same as the sign of the VVA response performance (ΔP , $\Delta P1$). Thus, when the VVA response performance (ΔP , $\Delta P1$) is positive, the output value of the VVA starting signal 124 becomes $V_c + |\Delta V|$, and when the VVA response performance (ΔP , $\Delta P1$) is negative, the output voltage of the VVA starting signal 124 becomes $V_c - |\Delta V|$.

[0155] Furthermore, the magnitudes of the output voltages of the VVA starting signals 123 and 124 are set to prescribed ranges that enable the hydraulic control valve 30 to operate normally in opposition to hydraulic pressure within an allowed operating temperature range.

[0156] As a result of correcting in this manner, in the next control cycle, a surge pressure signal M2 that exceeds the prescribed threshold value D1 is detected as a measurement value of a hydraulic pressure signal 144 (see FIG. 14E), and on the basis of the rise time (or crank angle) PM2 of that surge pressure signal M2 (VVA operation start timing (Pac2)) and the standard VVA operation start timing (Pacs), the VVA response performance (ΔP) calculated according to the equation (1) is expected to be improved from $\Delta P1$ in the case of having output the VVA starting signal 123 of the current control cycle to ΔPt ($= \text{Pac2} - \text{Pacs}$, $|\Delta Pt| < |\Delta P1|$) in the case of having output the VVA starting signal 124.

[0157] As a result, the closing end timing (S) of the intake valves 3 changes from S1 to St (see FIG. 14D), and is adjusted to a constant closing end timing (crank angle) St corresponding to the operating status of the engine, such as engine speed or engine load.

[0158] In addition, in the case the magnitude of the crank angle corresponding to the absolute value of the calculated VVA response performance ($\Delta P1$) is 30% to less than 60% of the control range of the intake stroke (NO in S 112, YES in S114), as shown with a VVA starting signal 126 of FIG. 15C, the output timing and waveform of the VVA starting signal 126 are set for each cylinder in the next control cycle so that the waveform of the VVA starting signal 123 is corrected to the waveform of the VVA starting signal 126 (see FIG. 15C) and output to the hydraulic control valve 30 by correcting the output timing of the VVA starting signal 126 to a timing (Pcom1) that is earlier by $\Delta P2$ than the magnetization command timing (Pcom) of the VVA starting signal 123 (see FIG. 15B) of the current control cycle (S115).

[0159] Here, $\Delta P2$ is determined by multiplying a prescribed coefficient K1 ($0 < K1 < 1$) by the VVA response performance $\Delta P1$ determined in the current control cycle, and the magnitude of the coefficient K1 is an empirical value. Thus, $\Delta P2$ has the same sign as $\Delta P1$.

[0160] More specifically, the output timing of the VVA starting signal 126 output to the hydraulic control valve 30 in the next control cycle is set to a timing (Pcom1) that is earlier by $\Delta P2$ than the magnetization command timing (Pcom) of the VVA starting signal 123 of the current control cycle, and the output voltage of the VVA starting signal 126 is made to be a voltage determined by adding a prescribed adjustment voltage ΔV to the output voltage V_c of the VVA starting signal 123 to correct the closing end timing of the intake valves 3 to be within a prescribed timing range by improving response performance of the hydraulic control valve 30.

[0161] Here, the "timing that is earlier by $\Delta P2$ " refers to timing that is earlier by $\Delta P2$ when $\Delta P2$ is positive or delayed by $|\Delta P2|$ when $\Delta P2$ is negative. In addition, the output timing of the VVA starting signal 126 determined as previously described is set to be within a prescribed controllable range so that control of the variable valve device can be carried out normally.

[0162] Furthermore, the sign of the adjustment voltage ΔV is determined according to the sign of the VVA response performance (ΔP , $\Delta P1$), the manner in which the output voltage of the VVA starting signal is calculated is the same as previously described.

[0163] As a result of correcting in this manner, in the next control cycle, a surge pressure signal M4 that exceeds the prescribed threshold value D1 is detected as a measurement value of a hydraulic pressure signal 146 (see FIG. 15E), and on the basis of the rise time (or crank angle) PM4 of that surge pressure signal M4 (VVA operation start timing (Pac4)) and the standard VVA operation start timing (Pacs), the VVA response performance (ΔP) calculated according to the equation (1) is expected to be improved from $\Delta P1$ in the case of having output the VVA starting signal 123 of the current control cycle to ΔPt ($= \text{Pac4} - \text{Pacs}$, $|\Delta Pt| < |\Delta P1|$) in the case of having output the VVA starting signal 126.

[0164] As a result, the closing end timing (S) of the intake valves 3 changes from S2 to St (see FIG. 15D), and is adjusted to a constant closing end timing (crank angle) St corresponding to the operating status of the engine, such as engine speed or engine load.

[0165] In addition, in the case the magnitude of the crank angle corresponding to the absolute value of the calculated VVA response performance ($\Delta P1$) is 60% or more of the control range of the intake stroke (NO in S114), as shown as

a VVA starting signal 128 of FIG. 16C, the output timing and waveform of the VVA starting signal 128 are set for each cylinder in the next control cycle so that the VVA starting signal 128 (see FIG. 16C) of the same waveform as the VVA starting signal 123 is output to the hydraulic control valve 30 by correcting the output timing of the VVA starting signal 128 to a timing that is earlier by $\Delta P3$ than the magnetization command timing (Pcom) of the VVA starting signal 123 (see FIG. 16B) of the current control cycle (S 116).

[0166] Here, $\Delta P3$ is determined by multiplying a prescribed coefficient $K2$ ($0 < K2 < 1$) by the VVA response performance $\Delta P1$ determined in the current control cycle, the magnitude of the coefficient $K2$ is an empirical value, and $K2 > K1$. Thus, $\Delta P3$ has the same sign as $\Delta P1$ and $|\Delta P3| > |\Delta P2|$.

[0167] More specifically, the output timing of the VVA starting signal 128 output to the hydraulic control valve 30 in the next control cycle is set to a timing (Pcom2) that is earlier by $\Delta P3$ than the magnetization command timing (Pcom) of the VVA starting signal 123 of the current control cycle, and the output voltage of the VVA starting signal 128 is set to the same voltage value as the output voltage Vc of the VVA starting signal 123 to correct the closing end timing of the intake valves 3 so as to be within a prescribed timing range.

[0168] Here, the "timing that is earlier by $\Delta P3$ " refers to timing that is earlier by $\Delta P3$ when $\Delta P3$ is positive or delayed by $|\Delta P3|$ when $\Delta P3$ is negative. In addition, as was previously described, the output timing of the VVA starting signal 128 determined as previously described is set to be within a prescribed controllable range so that control of the variable valve device can be carried out normally.

[0169] As a result of correcting in this manner, in the next control cycle, a surge pressure signal M6 that exceeds the prescribed threshold value D1 is detected as a measurement value of a hydraulic pressure signal 148 (see FIG. 16E), and on the basis of the rise time (or crank angle) PM6 of that surge pressure signal M6 (VVA operation start timing (Pac6)) and the standard VVA operation start timing (Pacs), the VVA response performance (ΔP) calculated according to the equation (1) is improved from $\Delta P1$ in the case of having output the VVA starting signal 123 of the current control cycle to ΔPt ($= Pac6 - Pacs$, $|\Delta Pt| < |\Delta P1|$) in the case of having output the VVA starting signal 128.

[0170] As a result, the closing end timing (S) of the intake valves 3 changes from $S3$ to St (see FIG. 16D), and is adjusted to a constant closing end timing (crank angle) St corresponding to the operating status of the engine, such as engine speed or engine load.

[0171] Although the explanation thus far indicated a method for correcting a VVA starting signal during intake closing delay control by using as examples the case of changing the output timing (magnetization command timing (Pcom)) of a VVA starting signal of this control and the case of changing the output waveform by increasing or decreasing the output voltage thereof, other correction may be carried out in the manner described below.

[0172] As shown in FIG. 17A, another correction method may be employed in which the waveform of a VVA starting signal to be corrected for the next control cycle may be corrected for a VVA starting signal of the current control cycle with a waveform of a VVA starting signal 129a that has the same voltage but for which the VVA holding time TW has been extended by ΔT (increasing or decreasing the VVA holding time TW by $|\Delta T|$). Here, ΔT is an adjustment amount set corresponding to ΔP , and the sign thereof is the same as that of ΔP . Thus, when $\Delta P > 0$, the VVA holding time TW is set to be extended by ΔT and when $\Delta P < 0$, the VVA holding time TW is set to be shortened by $|\Delta T|$.

[0173] In this case, in the case the VVA response performance (ΔP) has changed due to a temperature change of the operating oil and activation of commencement of sustaining the open state of the intake ports 2 has been delayed, since control for sustaining the open state of the intake ports at a large opening is inherently controlled so that the open state of the intake ports 2 is sustained in a state in which the opening becomes smaller, correction is made so as to compensate for the change in intake volume by extending the VVA holding time TW by ΔT , and the output timing of the VVA starting signal 129a (magnetization command timing (Pcom)) is changed to output earlier so that the closing end timing of the intake valves 3 is within a prescribed range.

[0174] In addition, as shown in FIG. 17B, a waveform of a VVA starting signal to be corrected for the next control cycle may be corrected for a VVA starting signal of the current control cycle with a waveform a VVA starting signal 129b for which the magnitude of the voltage has been increased or decreased and for which the VVA holding time TW has been extended by ΔT (increasing or decreasing the VVA holding time TW by $|\Delta T|$). Here, ΔT is an adjustment amount set in the same manner as previously described, and the manner of setting by increasing or decreasing the VVA holding time TW by $|\Delta T|$ corresponding to the sign of ΔP is the same as previously described.

[0175] In this case, in the case the VVA response performance (ΔP) has changed due to a temperature change of the operating oil and activation of commencement of sustaining the open state of the intake ports 2 has been delayed, since control for sustaining the open state of the intake ports at a large opening is inherently controlled so that the open state of the intake ports 2 is sustained in a state in which the opening becomes smaller, correction is made so as to control the end closing timing of the intake valves 3 so as to be within a prescribed range and compensate for the change in intake volume by extending the VVA holding time TW by ΔT by improving the response performance of the hydraulic control valve 30 by determining the output voltage of the VVA starting signal 129b of the next control cycle by adding the prescribed adjustment voltage ΔV to the output voltage Vc of the VVA starting signal of the current control cycle.

(Industrial Applicability)

[0176] As has been described above, the variable valve device according to the present invention is useful for an engine valve device that varies operation of engine valves, and in particular, can be applied to a technology for engine valves of a diesel engine.

BRIEF DESCRIPTION OF THE DRAWINGS

[0177]

FIG. 1 is a conceptual drawing showing a variable valve device 1 according to the present invention;
 FIG. 2 is a drawing showing an example of a hydraulic circuit 60;
 FIG. 3 is a schematic diagram showing the action of the variable valve device 1;
 FIG. 4 is a schematic diagram showing the action of the variable valve device 1;
 FIG. 5 is a schematic diagram showing the action of the variable valve device 1;
 FIG. 6 is a schematic diagram showing the action of the variable valve device 1;
 FIG. 7 is a schematic diagram showing the action of the variable valve device 1;
 FIG. 8 is a schematic diagram showing the action of the variable valve device 1;
 FIG. 9 is a drawing indicating the relationship between cam rotation angle and valve lift during an intake stroke;
 FIGS. 10A-10E are explanatory drawings showing an example of a measurement signal measured with each sensor accompanying activation of the variable valve device 1;
 FIGS. 11A-11C are explanatory drawings illustrating changes in response performance of a hydraulic control valve 30 caused by changes in temperature of operating oil and variations in closing end timing of intake valves;
 FIG. 12 is a flow chart showing a control sequence of intake closing delay control;
 FIG. 13 is a flow chart showing a control sequence of intake closing delay control;
 FIGS. 14A-14E are explanatory drawings showing an example of a measurement signal measured with each sensor in a control sequence of intake closing delay control;
 FIGS. 15A-15E are explanatory drawings of a control method for suppressing variations in closing end timing;
 FIGS. 16A-16E are explanatory drawings of a control method for suppressing variations in closing end timing; and,
 FIGS. 17A-17B are drawings showing examples of corrected VVA starting signals.

Claims

1. A hydraulically driven variable valve device that has a valve train that opens or closes an intake port by moving an intake valve of an engine, and varies a crank angle at which the intake port is fully closed by the valve train, the valve device comprising:
 - a hydraulic actuator that is activated by movement of the intake valve and discharges, using a closing movement of the intake valve, operating oil in a pressure chamber provided in a cylinder;
 - a hydraulic control valve that controls supply and discharge of operating oil to and from the hydraulic actuator by opening and closing, and prevents outflow of the operating oil from a pressure chamber of the hydraulic actuator when closing is implemented, thereby preventing the closing movement of the intake valve;
 - a crank angle detection sensor that detects a crank angle indicating an angle of rotation of the engine crankshaft;
 - a TDC detection sensor that detects that each cylinder of the engine has reached a top dead center (TDC);
 - hydraulic pressure detecting means that detects hydraulic pressure of the pressure chamber of the hydraulic actuator; and
 - a controller that stops, during implementation of the closing movement of the intake valve, the closing movement of the intake valve and outputs a signal that closes the hydraulic control valve in order to maintain an open state of the intake port at a prescribed opening for a prescribed amount of time when a prescribed crank angle is determined to have been reached based on a detection signal from the TDC detection sensor and a detection signal from the crank angle detection sensor, wherein
 - after outputting a signal that closes the hydraulic control valve at the prescribed crank angle in a current control cycle, the controller monitors hydraulic surge pressure of the pressure chamber resulting from closing of the hydraulic control valve in use of a detection signal of the hydraulic pressure detecting means, determines an operation start crank angle of the hydraulic control valve based on the surge pressure, corrects at least either an output crank angle or waveform of a signal that closes the hydraulic control valve based on the operation start crank angle, and outputs the corrected signal to the hydraulic control valve in a next control cycle.

2. The variable valve device according to claim 1, wherein the controller corrects at least either the output crank angle or waveform of the signal that closes the hydraulic control valve so that the crank angle, at which the intake port is fully closed when the closing movement of the intake valve is completed, is within a prescribed range after the signal that closes the hydraulic control valve is switched off.

3. A control method for a variable valve device that varies a crank angle at which an intake port is fully closed by opening or closing the intake port by moving an intake valve of an engine by a valve train, and activating a hydraulic actuator by moving the intake valve, and moreover preventing outflow of operating oil from a pressure chamber provided in a cylinder of the hydraulic actuator by a variable valve device, the method comprising:

in a current control cycle,
implementing discharge of the operating oil in the pressure chamber of the hydraulic actuator by activating the hydraulic actuator by closing movement of the intake valve;
closing a hydraulic control valve that controls supply and discharge of the operating oil to and from the hydraulic actuator when a crank angle reaches a prescribed angle during closing movement of the intake valve after each cylinder of an engine reaches a top dead center, and preventing outflow of the operating oil from the pressure chamber, thereby preventing closing movement of the intake valve;
stopping the closing movement of the intake valve by closing the hydraulic control valve for a prescribed amount of time, and maintaining an open state of the intake port at a prescribed opening for a prescribed amount of time;
monitoring surge pressure of hydraulic pressure of the pressure chamber resulting from prevention of outflow of the operating oil from the pressure chamber, and determining timing of the start of actual variable valve actuation (VVA) based on the surge pressure,
correcting at least either an output crank angle or a waveform of a signal that closes the hydraulic control valve based on the actual VVA starting timing; and
outputting the corrected signal to the hydraulic control valve in a next control cycle.

4. The control method for a variable valve device according to claim 3, comprising:

comparing the actual VVA starting timing with a preset prescribed VVA starting timing; and
carrying out, when the actual VVA starting timing comes after the prescribed VVA starting timing, at least either correction in which an output crank angle of the signal that closes the hydraulic control valve of the next control cycle is made to be earlier than the current output crank angle, or correction in which an output value of the signal that closes the hydraulic control valve of the next control cycle is made to be larger than the current output value, while on the other hand,
carrying out, when the actual VVA starting timing comes before the prescribed VVA starting timing, at least either correction in which the output crank angle of the signal that closes the hydraulic control valve of the next control cycle is made to be later than the current output crank angle, or correction in which the output value of the signal that closes the hydraulic control valve of the next control cycle is made to be smaller than the current output value, is carried out.

5. The control method for a variable valve device according to claim 3, comprising:

comparing the actual VVA starting timing with the preset prescribed VVA starting timing; and
carrying out, when the actual VVA starting time comes after the prescribed VVA starting timing, at least either correction in which, an output time of the signal that closes the hydraulic control valve of the next control cycle is made to be longer than the current output time and an output crank angle is made to be earlier than the current output crank angle, or correction in which the output time of the signal that closes the hydraulic control valve of the next control cycle is made to be longer than the current output time and an output value is made to be larger than the current output value, while on the other hand,
carrying out, when the actual VVA starting timing comes before the prescribed VVA starting timing, at least either correction in which the output time of the signal that closes the hydraulic control valve of the next control cycle is made to be shorter than the current output time and the output crank angle is made to be later than the current output crank angle, or correction in which the output time of the signal that closes the hydraulic control valve of the next control cycle is made to be shorter than the current output time and the output value is made to be smaller than the current output value.

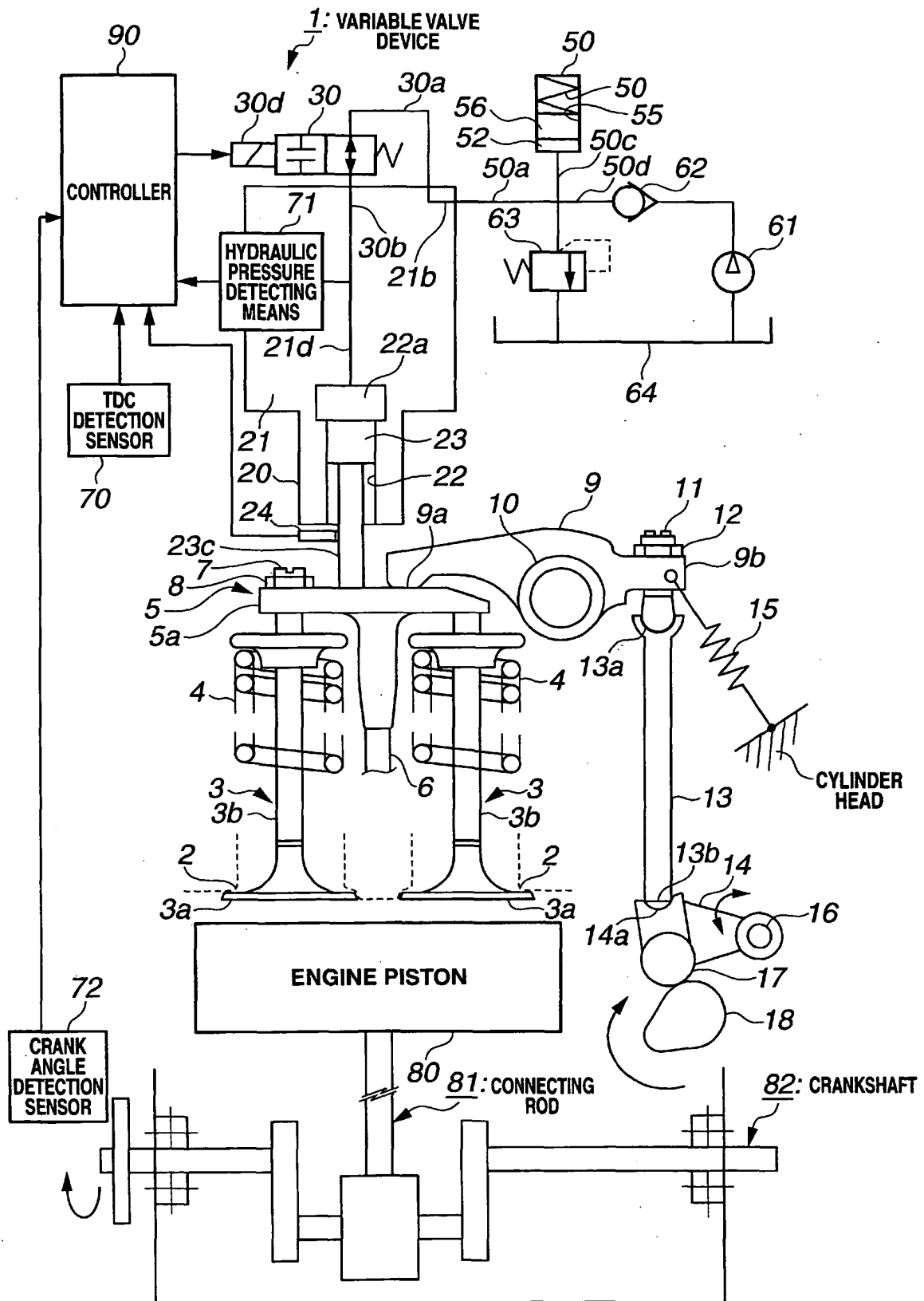


FIG.1

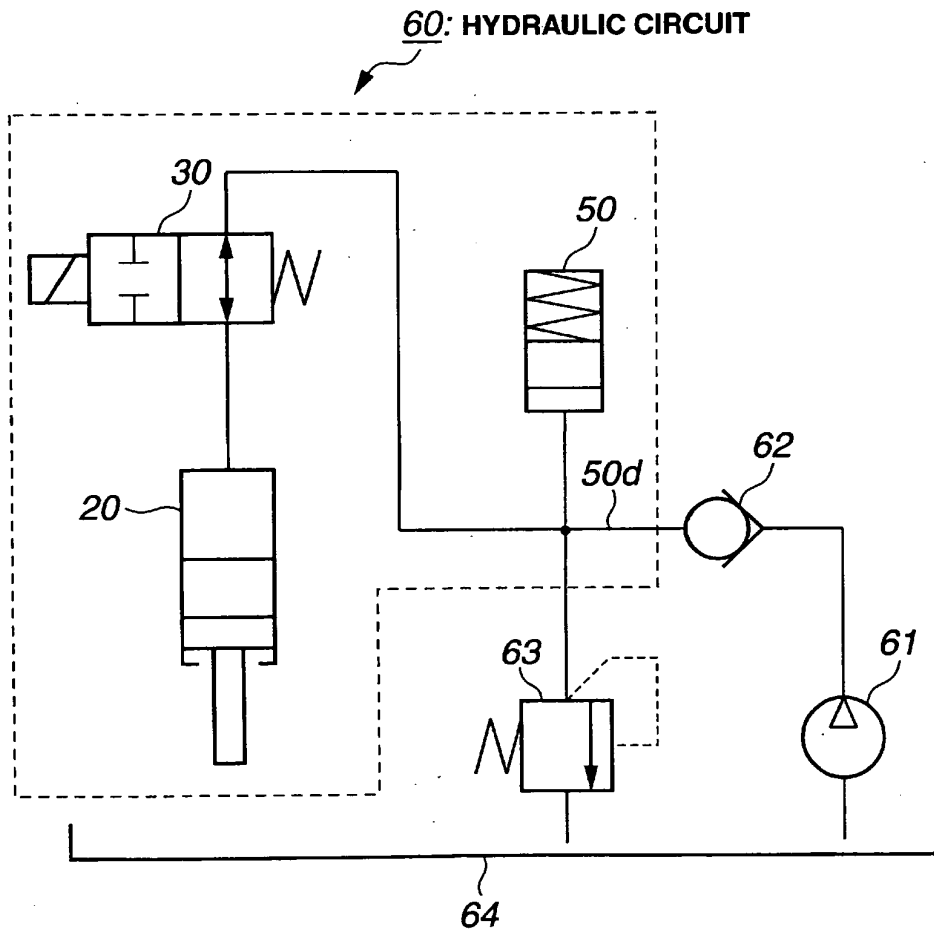


FIG.2

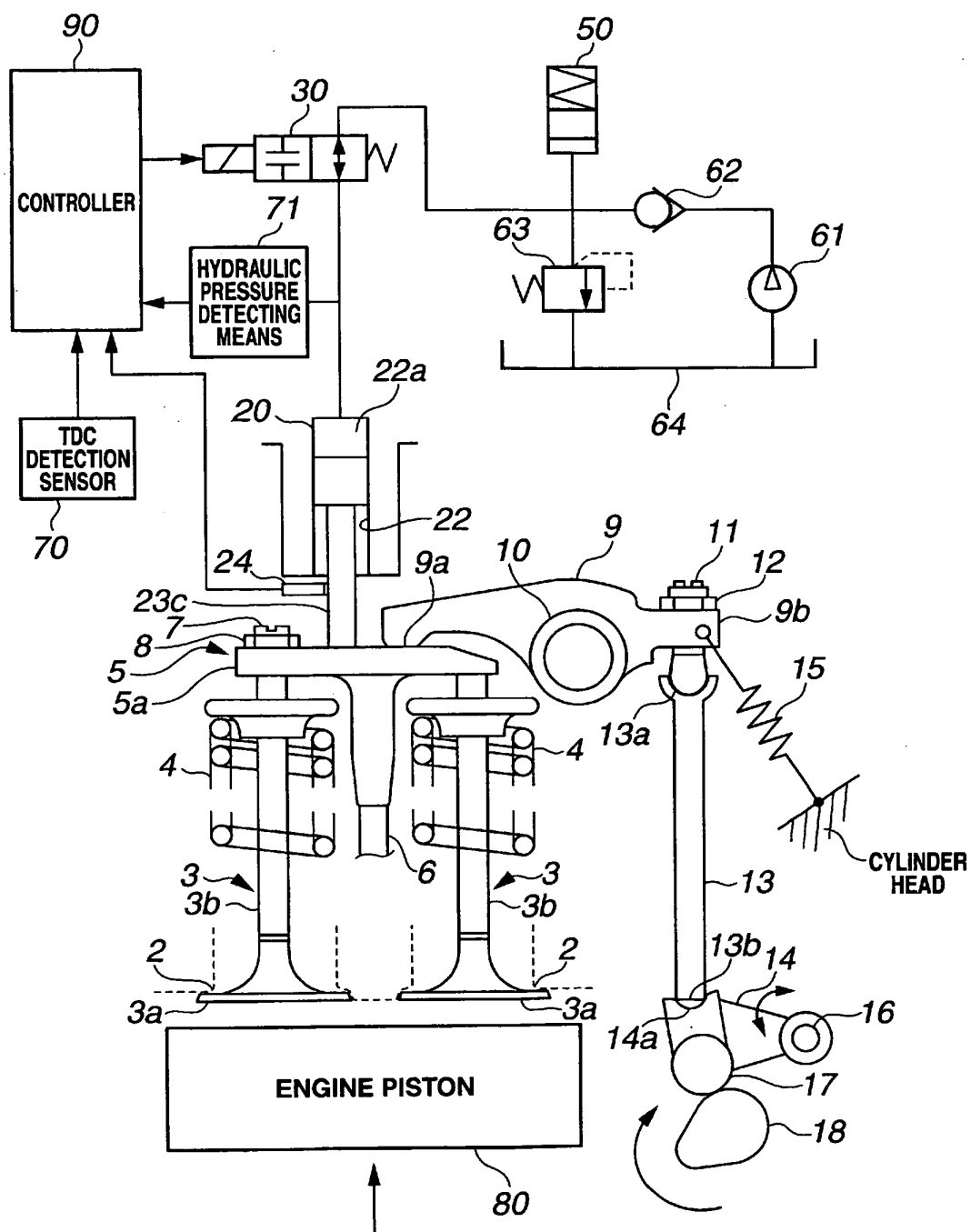


FIG.3

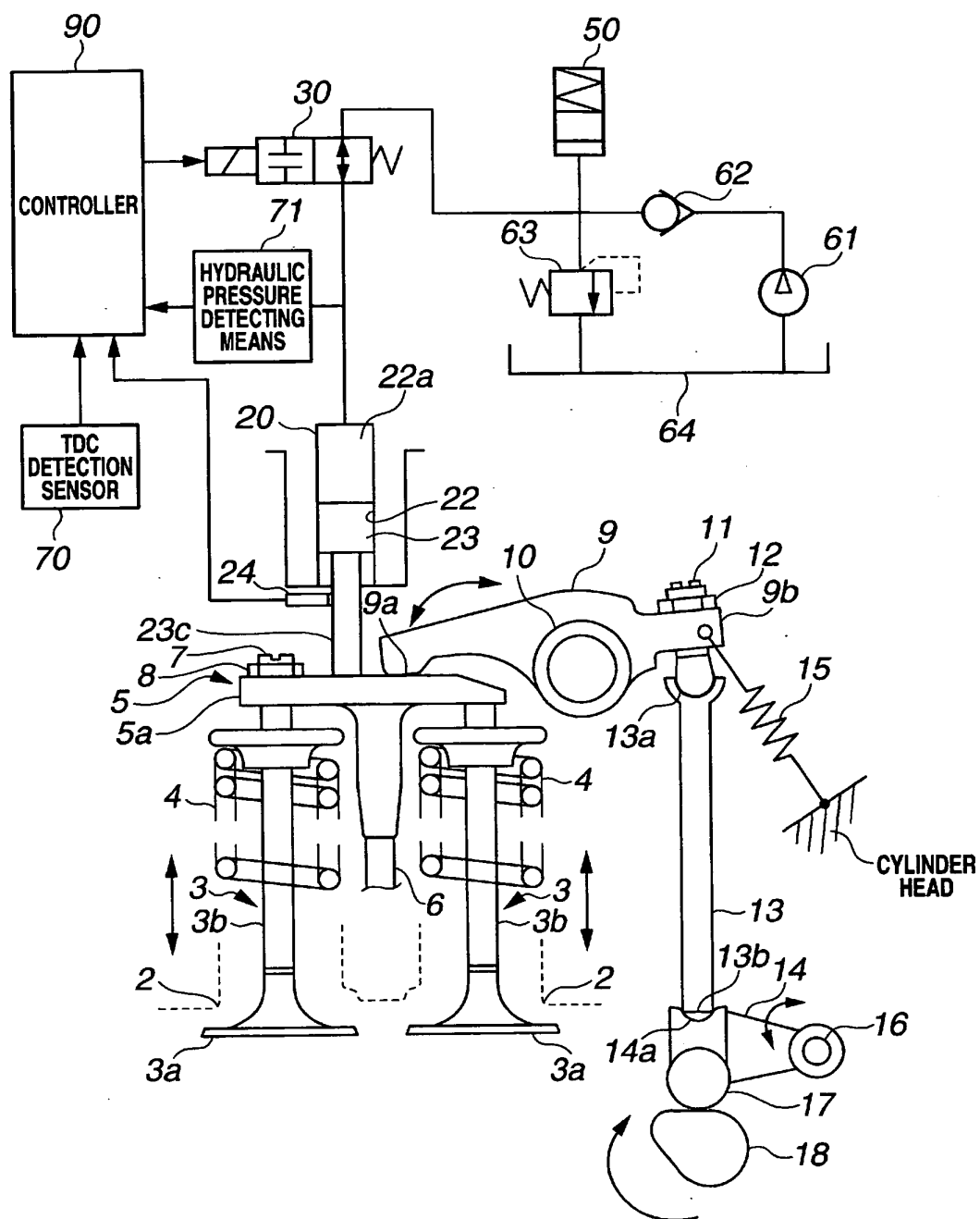


FIG.4

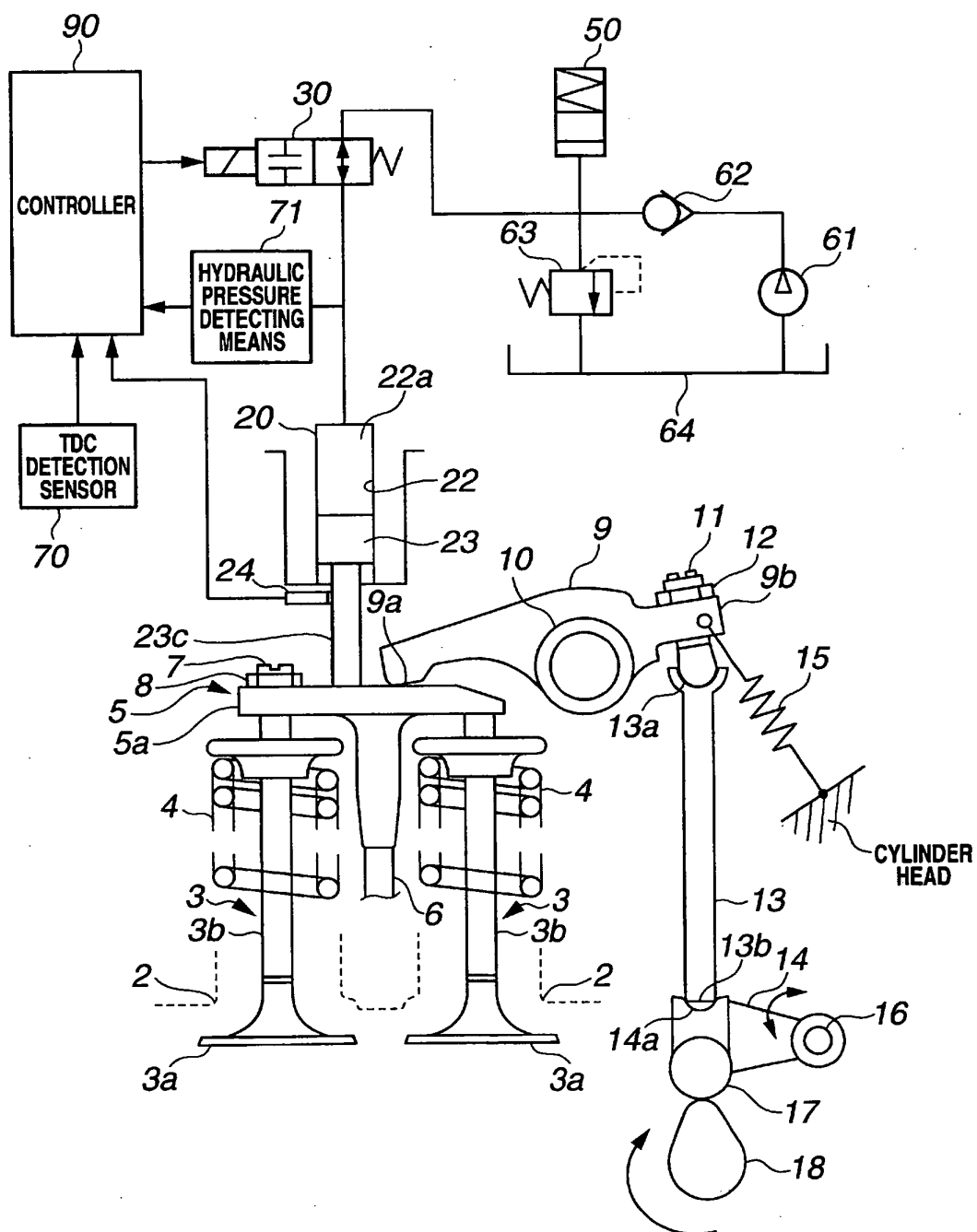


FIG.5

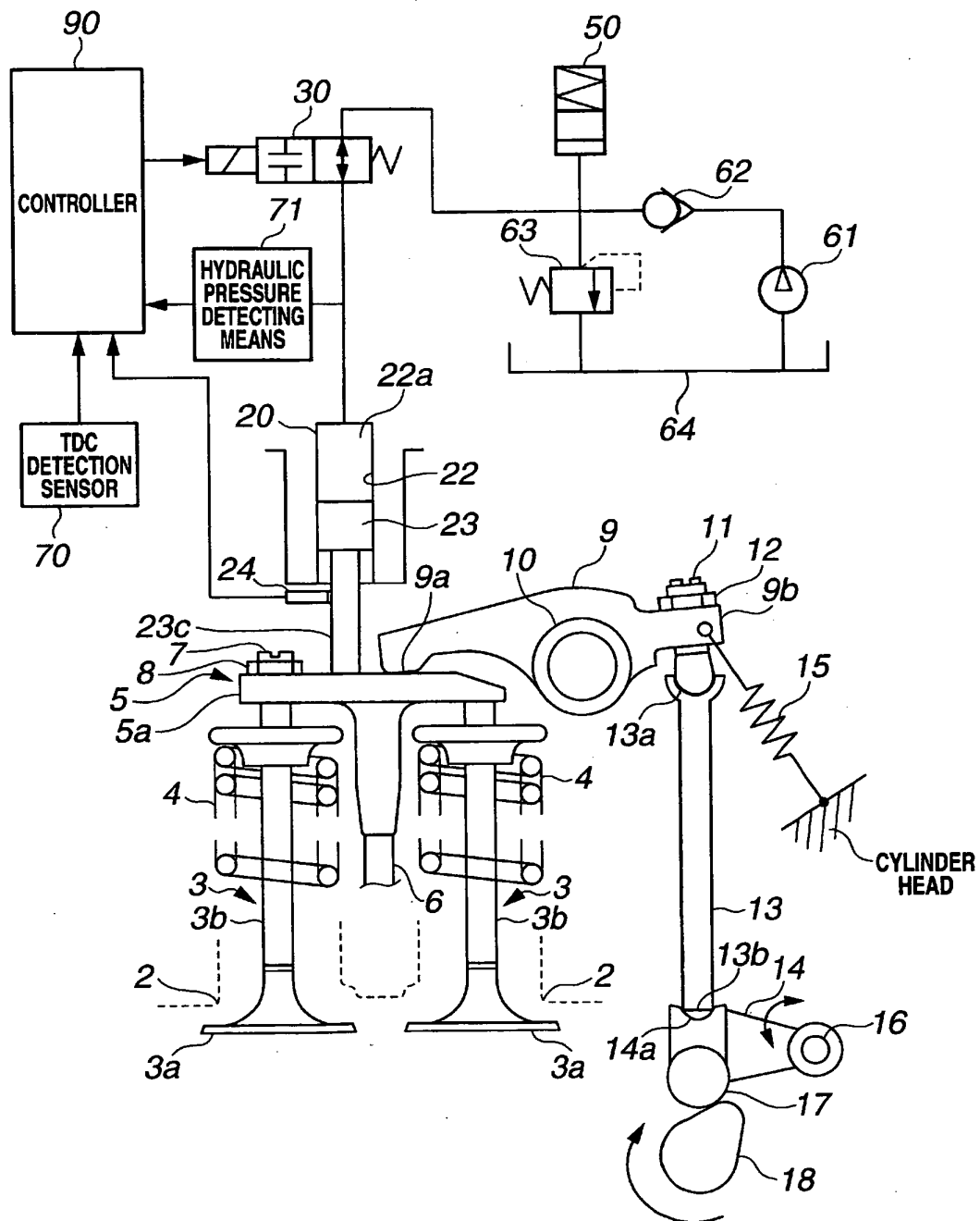


FIG.6

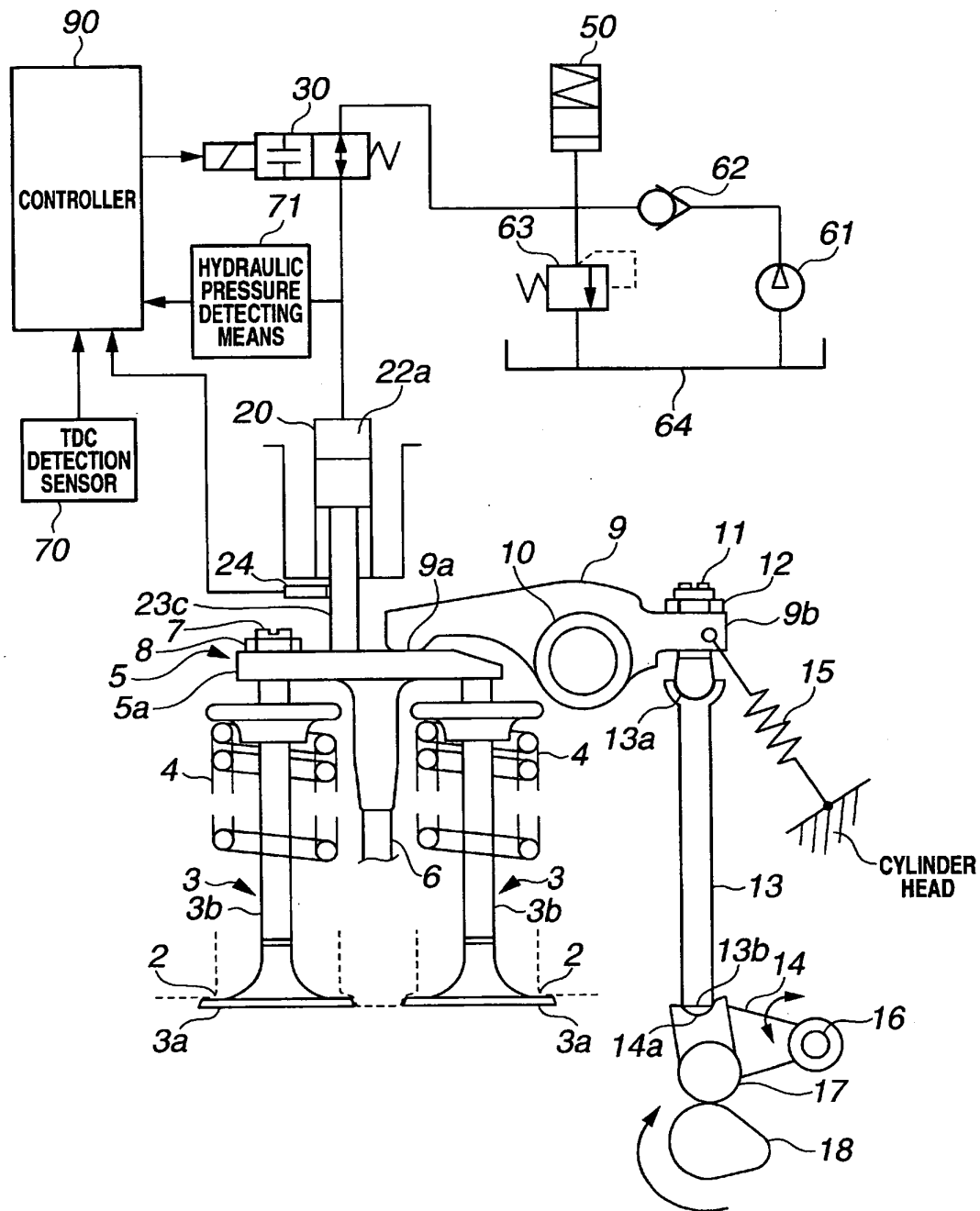


FIG.7

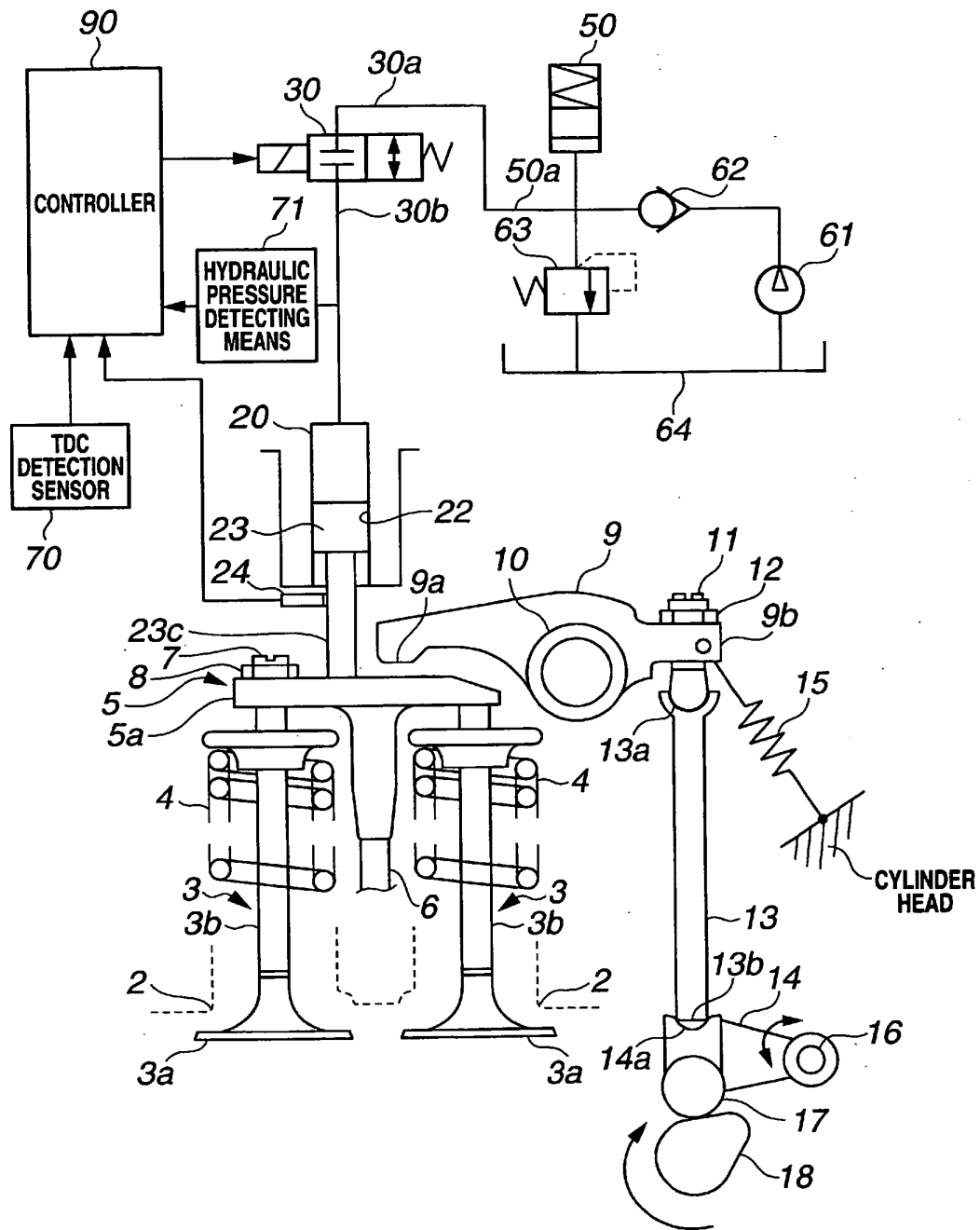


FIG.8

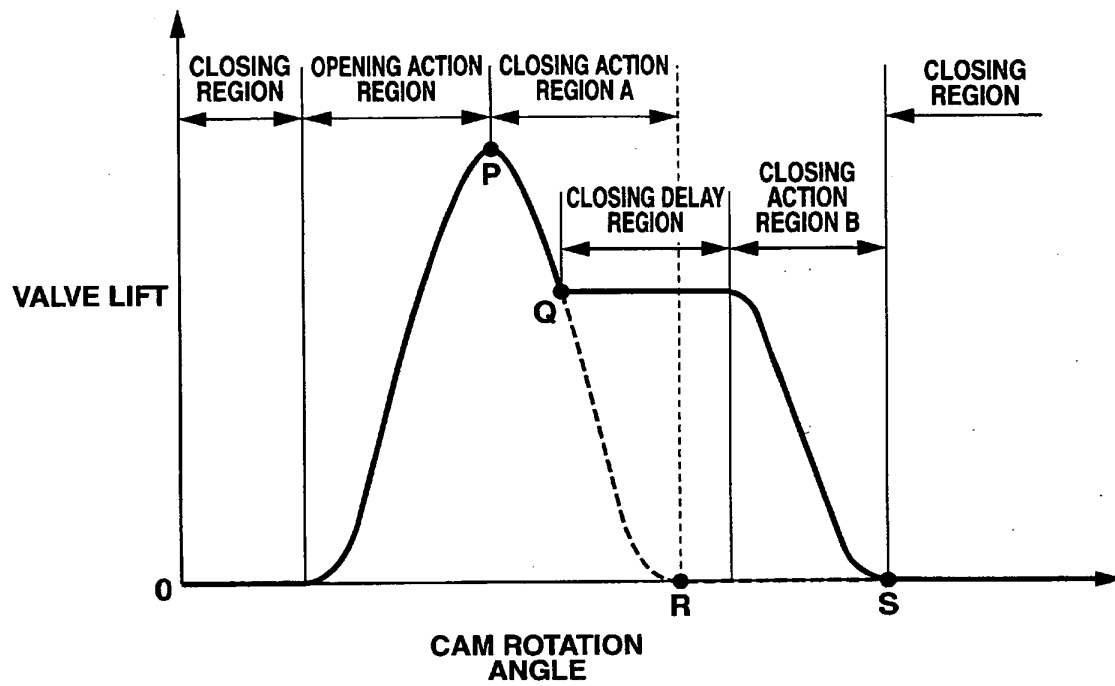


FIG.9

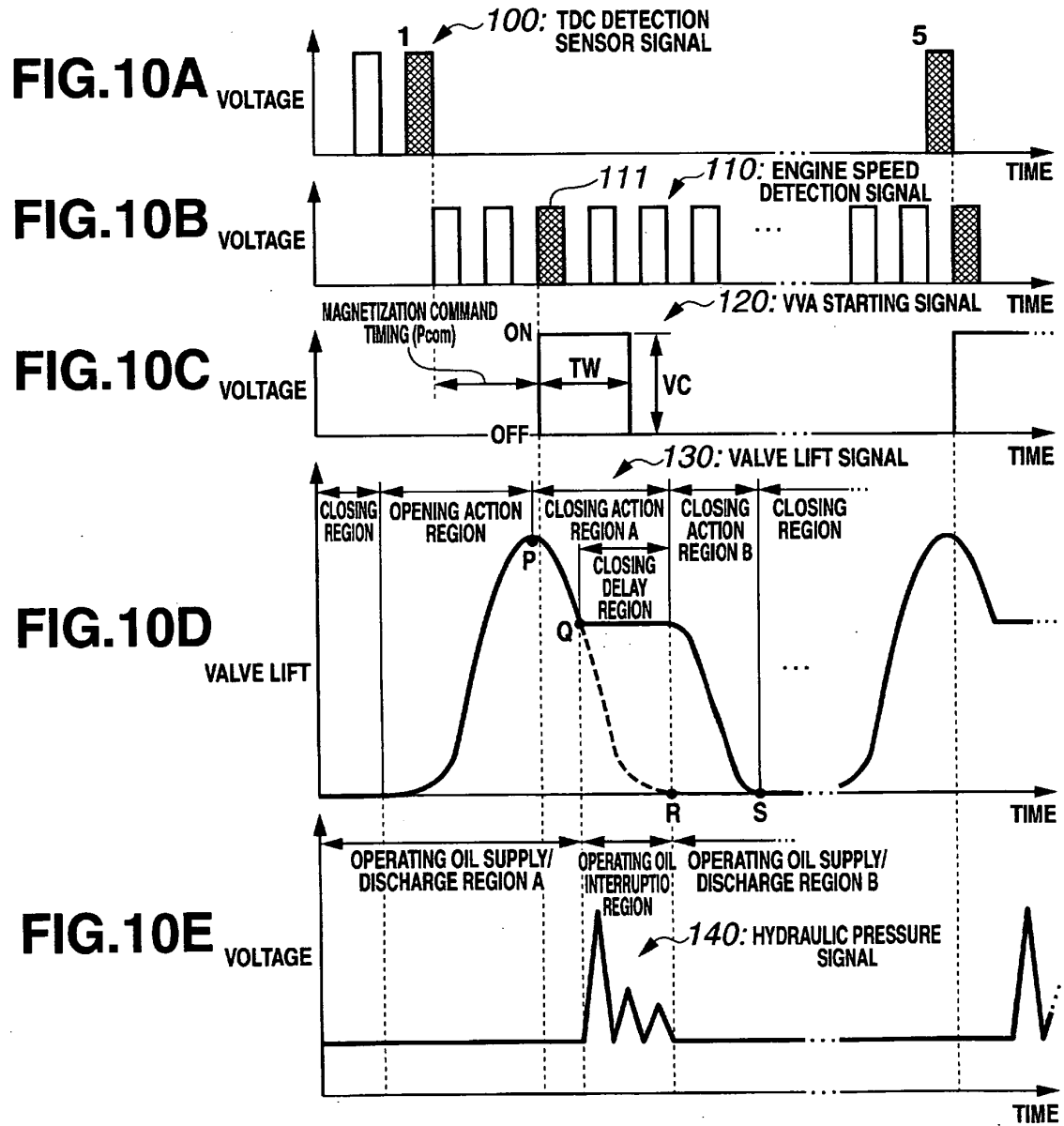


FIG.11A

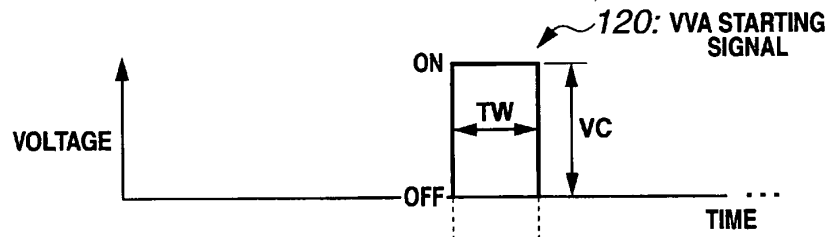


FIG.11B

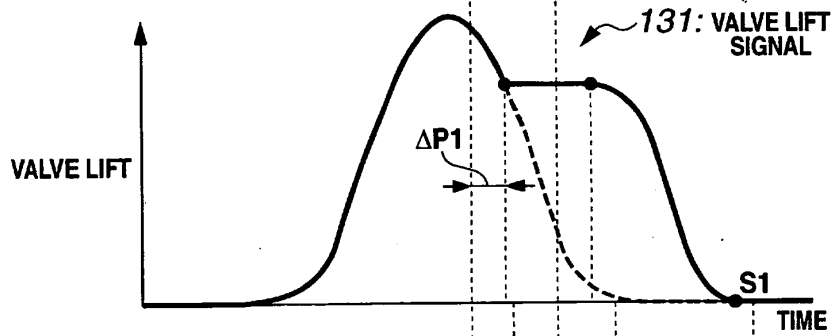
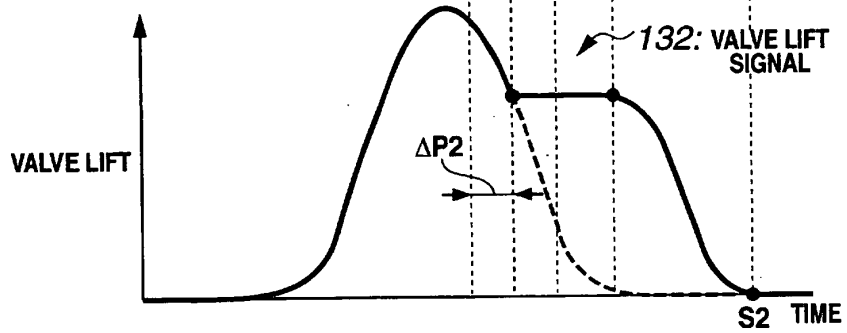
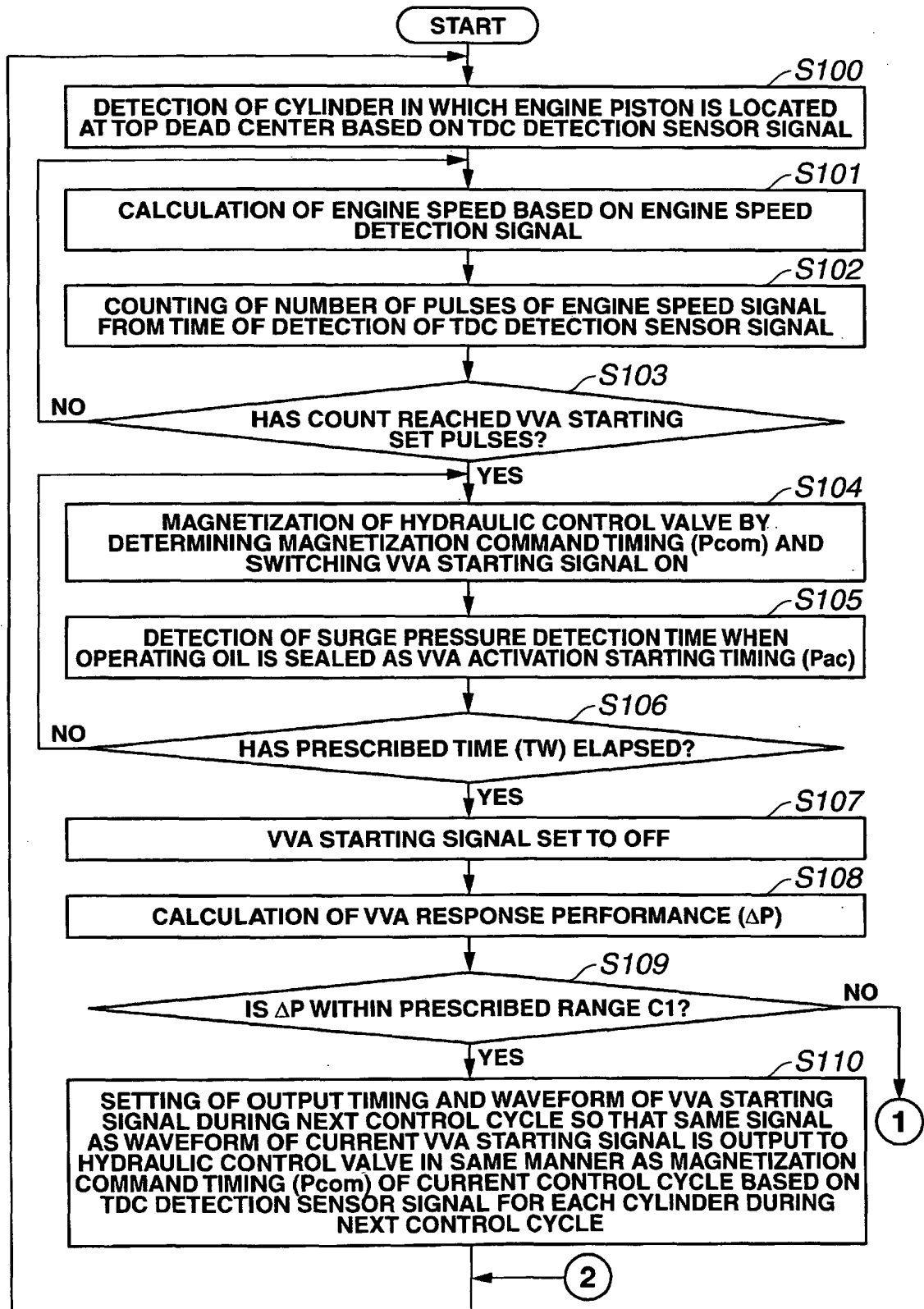


FIG.11C





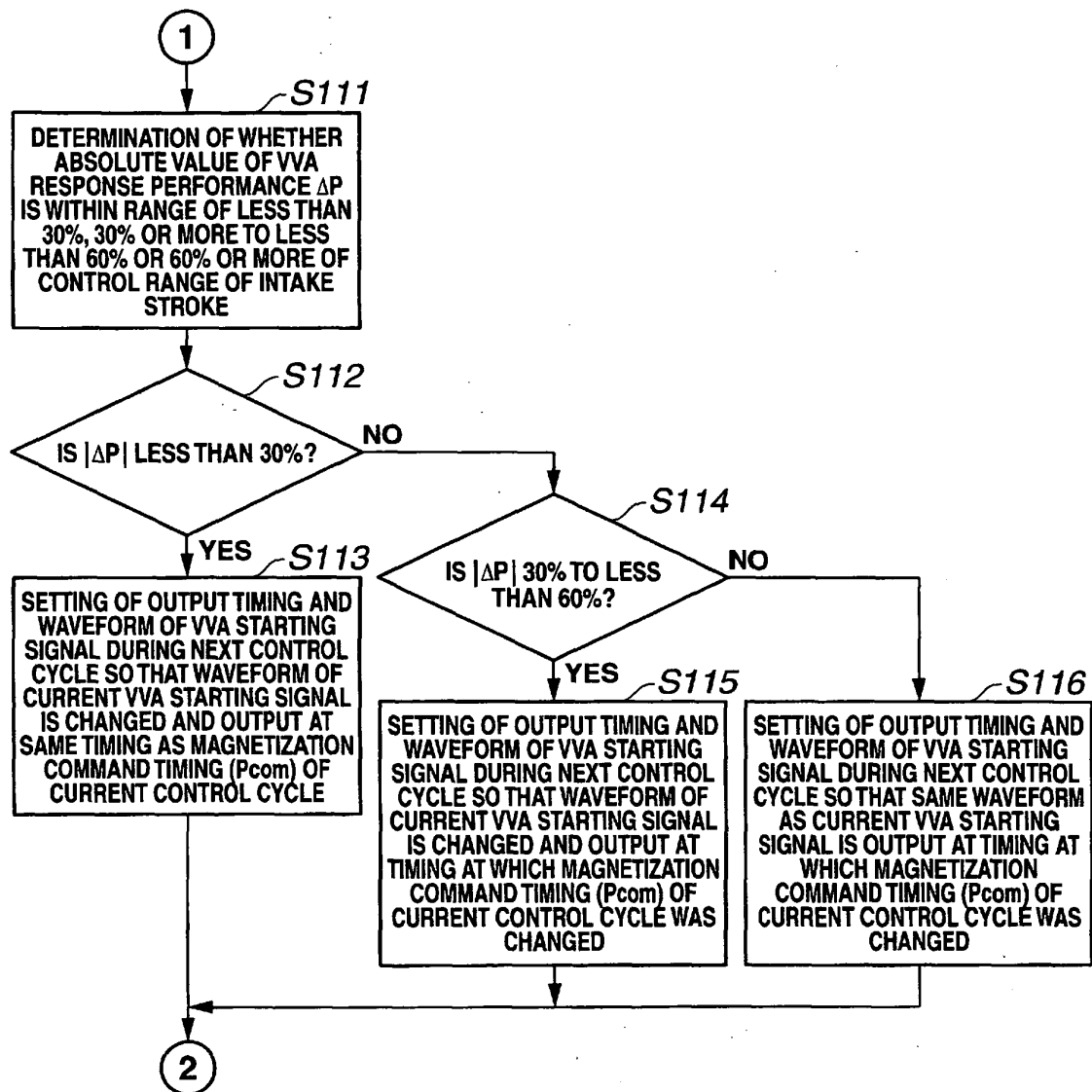


FIG.13

FIG.14A

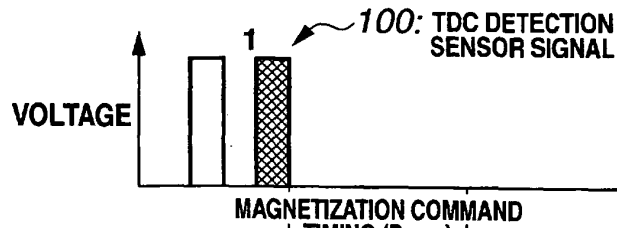


FIG.14B

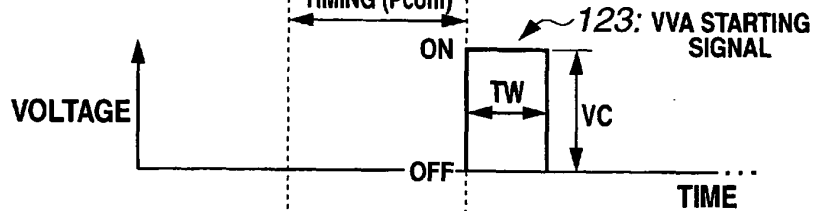


FIG.14C

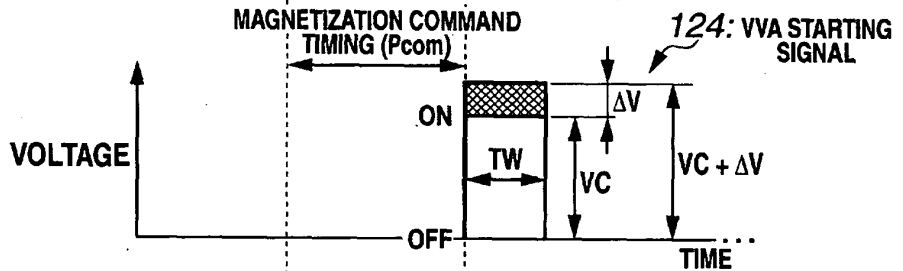


FIG.14D

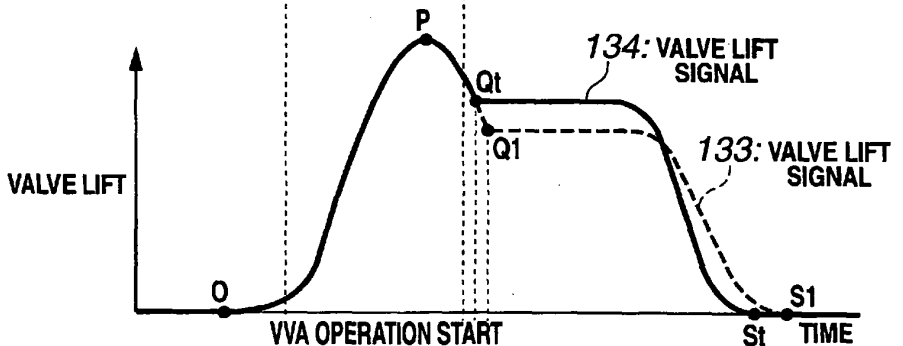


FIG.14E

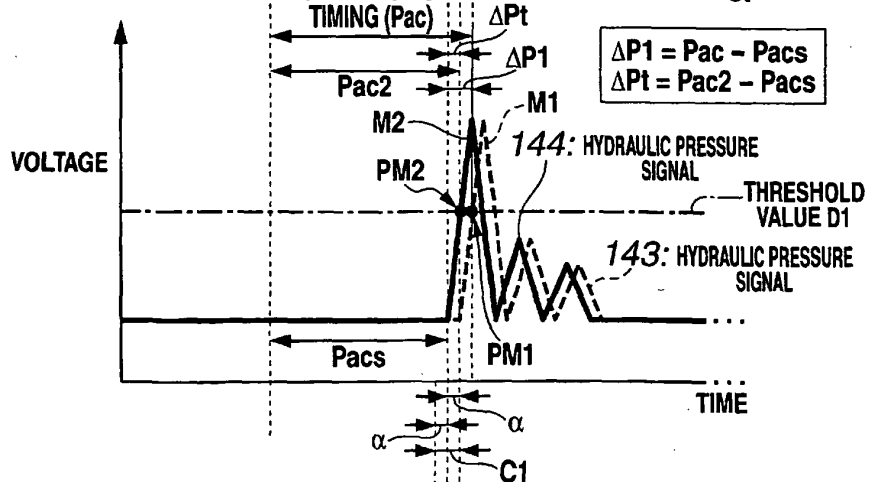


FIG.15A

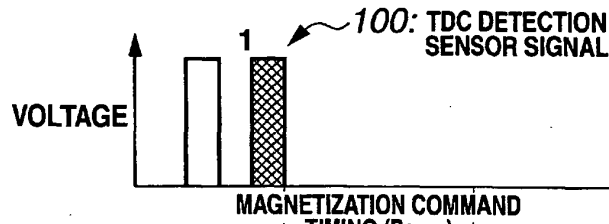


FIG.15B

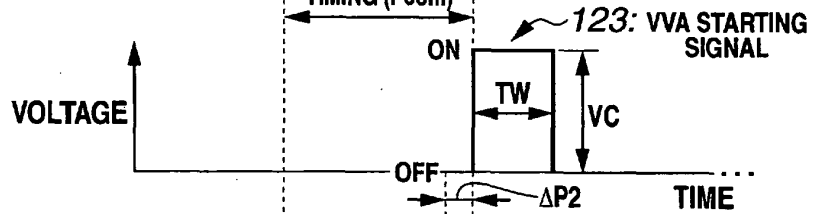


FIG.15C

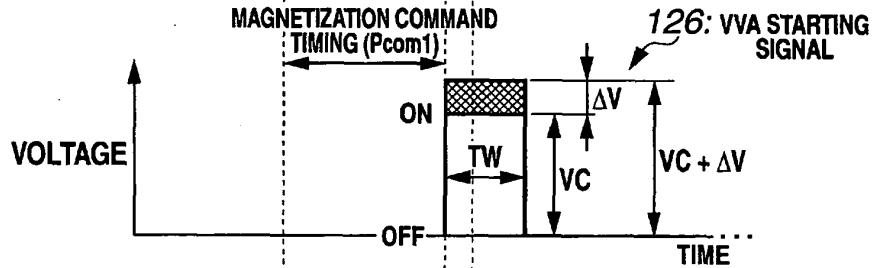


FIG.15D

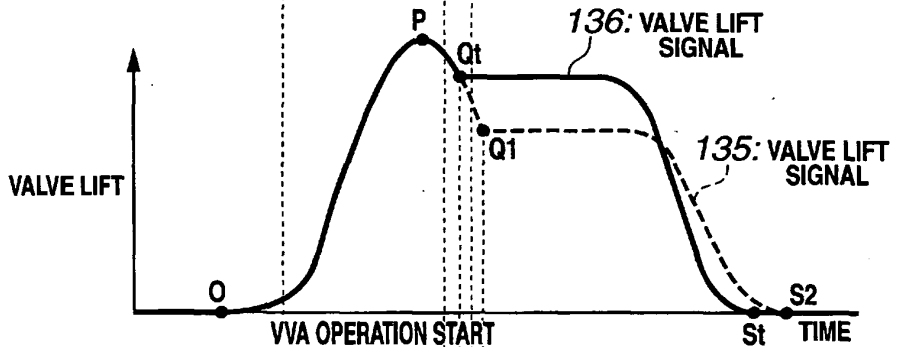


FIG.15E

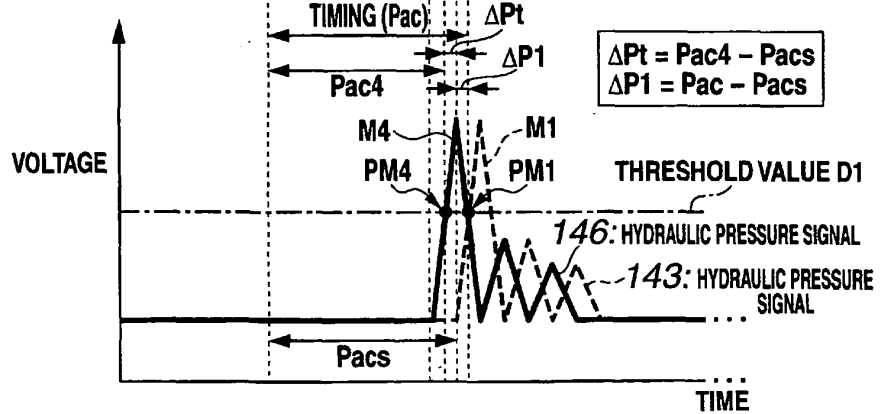


FIG.16A

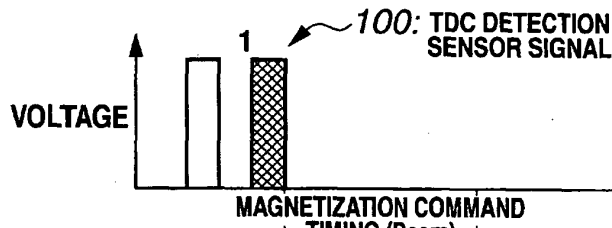


FIG.16B

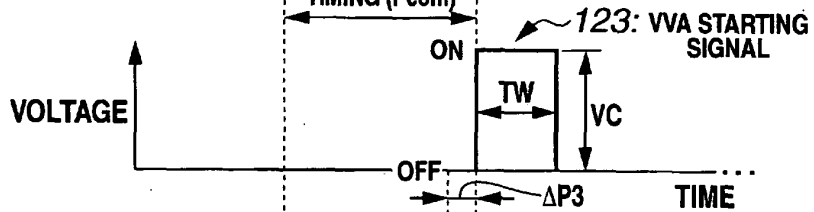


FIG.16C

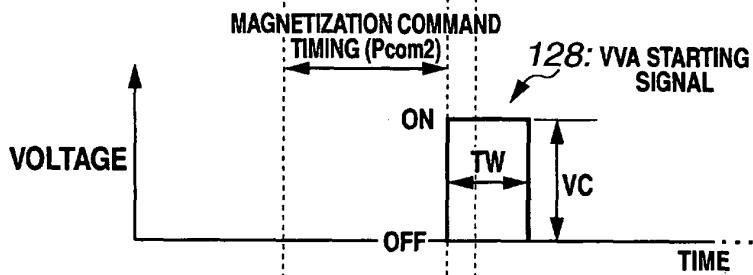


FIG.16D

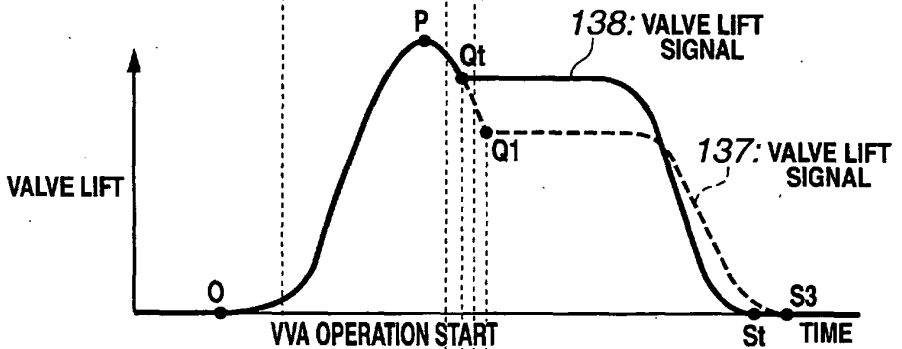


FIG.16E

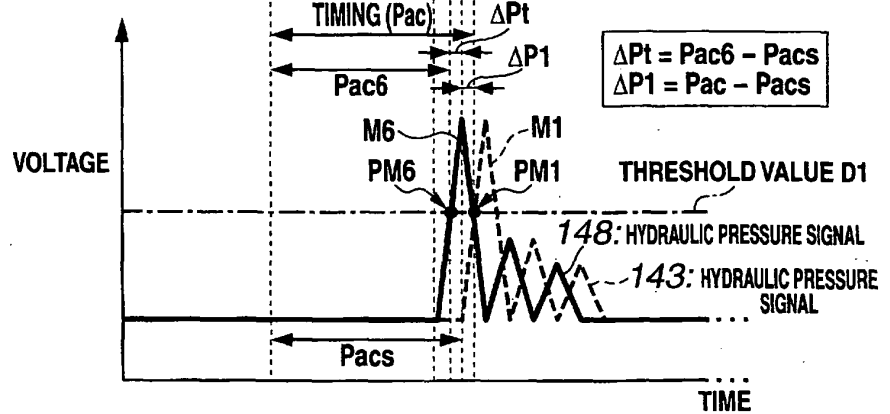


FIG.17A

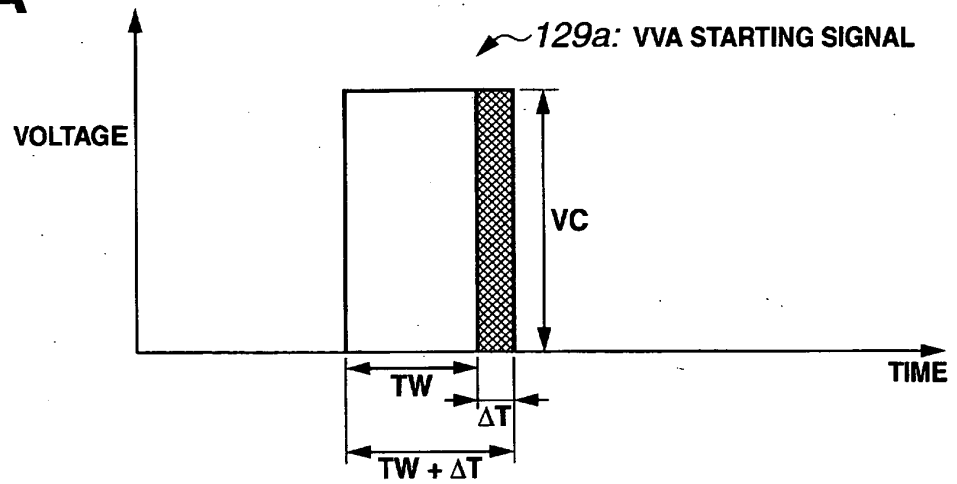
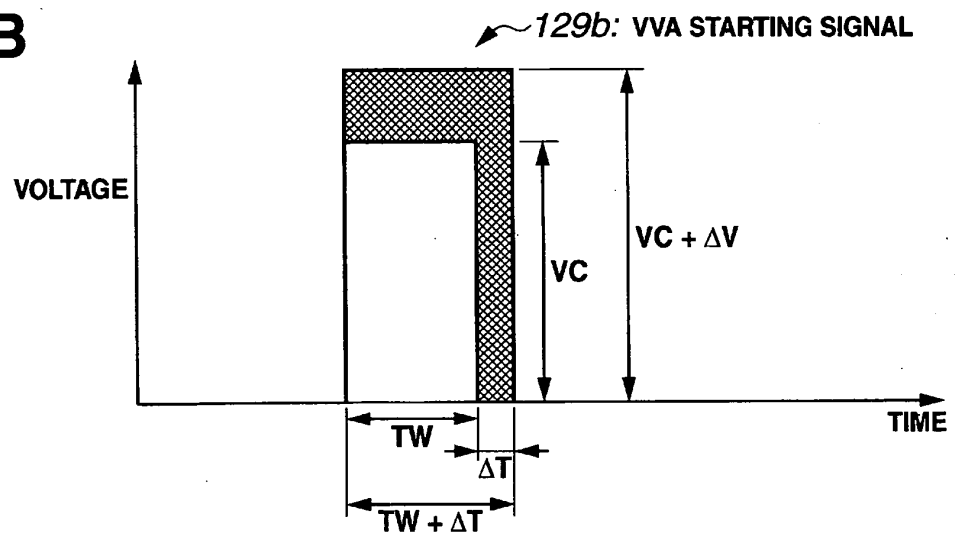


FIG.17B



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2009/069170

A. CLASSIFICATION OF SUBJECT MATTER

F01L9/02(2006.01) i, F01L13/00(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)

F01L9/02, F01L13/00

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	1996-2010
Kokai Jitsuyo Shinan Koho	1971-2010	Toroku Jitsuyo Shinan Koho	1994-2010

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	WO 2007/074072 A1 (ROBERT BOSCH GMBH), 05 July 2007 (05.07.2007), claims 5, 10, 15, 19; page 14, lines 18 to 23; page 20, lines 20 to 24; fig. 1, 4 & JP 2009-521643 A & WO 2007/074072 A1 & DE 102006016145 A & DE 102006016145 A1	1-5
Y	WO 2008/001699 A1 (Komatsu Ltd.), 03 January 2008 (03.01.2008), paragraph [0054]; fig. 6 & US 2009/0199796 A & EP 2039892 A1 & WO 2008/001699 A1 & KR 10-2009-0027649 A & CN 101473111 A	1-5

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

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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
03 February, 2010 (03.02.10)Date of mailing of the international search report
16 February, 2010 (16.02.10)Name and mailing address of the ISA/
Japanese Patent Office

Authorized officer

Facsimile No.

Telephone No.

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2009/069170

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT		
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 3785464 B2 (Yokohama TLO Co., Ltd.), 31 March 2006 (31.03.2006), fig. 1 (Family: none)	1-5
A	JP 2004-521275 A (ROBERT BOSCH GMBH), 15 July 2004 (15.07.2004), fig. 1 & US 2004/0035388 A1 & EP 1412620 A & WO 2003/008770 A & WO 2003/008770 A1 & DE 10134644 A	1-5

Form PCT/ISA/210 (continuation of second sheet) (April 2007)

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- US 7178491 B [0007]
- US 7059282 B [0007]