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- **SAKAMOTO, Katsumi**
Saitama 360-0193 (JP)
- **TAKAGI, Nobukazu**
Saitama 360-0193 (JP)

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(71) Applicant: **Valeo Japan Co., Ltd.**
Saitama 360-0193 (JP)

(72) Inventors:
• **KOWADA, Kazutaka**
Saitama 360-0193 (JP)

(74) Representative: **Delplanque, Arnaud et al**
Valeo Systemes Thermiques
BG THS - Service Propriété Industrielle
8, rue Louis Lormand
B.P. 517 - La Verrière
78321 Le Mesnil Saint-Denis (FR)

(54) **VARIABLE DISPLACEMENT SWASH PLATE-TYPE COMPRESSOR AND AIR CONDITIONING SYSTEM USING SAID COMPRESSOR**

(57) This invention provides a variable-capacity swash plate compressor that does not allow the compressor performance to be compromised by ensuring that the suction passage is not constricted in an operating area where low pressure pulsations do not occur and, at the same time, makes it possible to fully reduce low pressure pulsations by constricting the suction passage in an operating area where low pressure pulsations tend to occur readily and an air-conditioning system equipped with such a variable-capacity swash plate compressor.

A variable-capacity swash plate compressor includes a suction throttle valve 62 that is disposed in a suction passage 61, through which a coolant taken in through a suction port 60 is guided to a suction chamber 54, and adjusts the passage area of the suction passage 61 through which the coolant passes, and a solenoid valve 70 that freely adjusts the degree of opening of the suction throttle valve 62. The solenoid valve 70 may adjust the degree of opening of the suction throttle valve 62 by adjusting the pressure in a back pressure chamber 68 of the suction throttle valve 62 or it may directly control the degree of opening of the suction throttle valve 62 in correspondence to supplied electric energy.

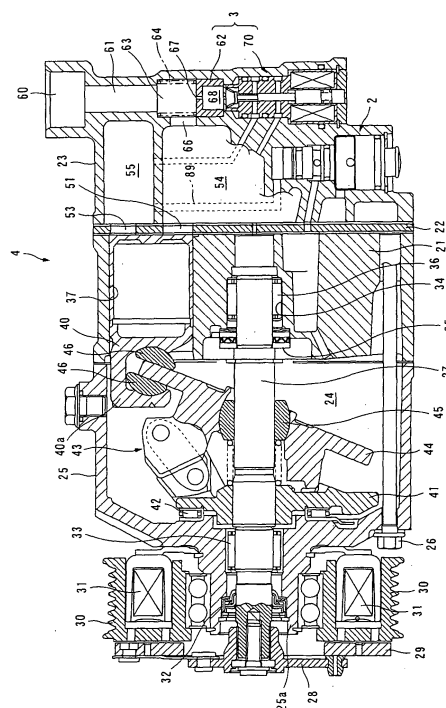


Fig. 2

Description

[0001] The present invention relates to a piston-type compressor that includes a noise-suppression mechanism installed in order to prevent abnormal noise occurring as pressure pulsations, attributable to self-induced vibration of a suction valve, are propagated to the outside of the compressor, and more specifically, it relates to a variable-capacity swash plate compressor that reduces propagation of pressure pulsations to the outside of the compressor by constricting a suction passage in an operating area where the pressure pulsations occur and sustains a satisfactory level of compressor performance in an operating area where pressure pulsations do not occur by ensuring that the suction passage does not become constricted. The present invention further relates to an air-conditioning system equipped with such a variable-capacity swash plate compressor.

[0002] In a typical piston-type compressor, a stopper with a predetermined depth is formed in a cylinder block at a position facing opposite the end of a suction valve so that self-induced vibration of the suction valve is prevented by causing the end of the suction valve to come into contact with the stopper as a coolant gas is drawn into a cylinder bore.

[0003] However, the quantity of gas taken into the cylinder bore under a variable displacement condition is different from the quantity of gas sucked into the cylinder bore under the maximum displacement condition, in the piston-type variable-capacity compressor. This means that if the depth of the stopper is set in correspondence to the maximum displacement condition, the end of the suction valve will not come into contact with the stopper under a small displacement condition, in which the suction valve becomes displaced only by a small extent. Thus, self-induced vibration of the suction valve may occur, which, in turn, cause fluctuation in the pressure of the gas inside a suction chamber and ultimately result in pressure pulsations being propagated to the outside of the compressor and causing abnormal noise.

[0004] The measures taken to address this issue in the related art include those disclosed in the Japanese Patent Applications JP2001-136776-A1 and JP2005-337232-A2 outlined below.

[0005] JP2001-136776-A1 discloses a structure that includes an opening degree control valve disposed in a suction passage of a compressor so as to control the opening area of the suction passage by using the pressure difference attributable to the gas flow through the suction passage in combination with a spring force. Via the opening degree control valve, the opening area of the suction passage is reduced when the intake flow rate is low so as to suppress propagation of suction pulsations to the outside of the compressor under a small displacement condition and the opening area of the suction passage is increased when the intake flow rate is high.

[0006] The structure disclosed in JP2005-337232-A2 includes an opening degree control valve disposed in a

suction passage so as to adjust the opening area in the suction passage based upon the pressure difference between the suction pressure and the crank chamber pressure. In the structure disclosed in this patent, the opening degree control valve is controlled based upon the crank chamber pressure, which changes in correspondence to the capacity, so as to allow the opening degree control valve to readily achieve the maximum opening degree under the maximum displacement condition by lessening the influence of the force applied by the spring and allow the opening degree control valve to achieve a smaller opening degree with ease under a small displacement condition by increasing the role of the force applied by the spring as an influencing factor.

[0007] The structure disclosed in JP2001-136776-A1, in which the opening degree control valve is engaged in operation by using the pressure difference created by the gas flow through the suction passage and the spring force in combination, gives rise to a problem in that if the spring force is set to a high level in order to effectively reduce pulsations, the suction passage will be constricted to result in lowered air cooling performance even under maximum displacement condition, whereas if the spring force is set to a low level so as to maximize the air cooling performance under the maximum displacement condition, the pulsations will not be fully reduced under a small displacement condition, in which the suction passage needs to be constricted effectively.

[0008] In addition, the tilt angle of the swash plate in a standard variable-capacity compressor is altered based upon the difference between the pressure inside cylinder bores applied to the individual pistons and the crank chamber pressure. The pressure inside a cylinder bore is substantially equal to the pressure in the suction chamber when the corresponding piston is positioned at the bottom dead center, and as the piston compresses the coolant gas, the cylinder bore pressure gradually rises. Then, once the cylinder bore pressure exceeds the pressure in the discharge chamber, the pressure difference acting on the front and behind of the discharge valve causes the valve to open and, as a result, the coolant gas is discharged into the discharge chamber. In other words, the pressure inside the cylinder bore changes within a range defined by the suction pressure and the discharge pressure (slightly higher than the actual discharge pressure, to be exact, due to the delay with which the discharge valve opens and the resistance) while the swash plate rotates by a single turn, and the cylinder bore pressure is applied to the piston at all times.

[0009] The cylinder bore pressure applied to the piston act on the swash plate so as to increase the tilt angle of the swash plate, and thus, assuming that the pressure difference between the pressure in the suction chamber and the pressure in the crank chamber remains unchanged, the swash plate will be controlled to assume a greater tilt angle (to achieve a greater discharge capacity) at a higher discharge pressure.

[0010] This means that if the opening degree control

valve disclosed in JP2005-337232-A2, which controls the opening degree based upon the pressure difference between the suction chamber pressure and the crank chamber pressure irrespective of the discharge pressure, is set to constrict the suction passage when the pressure difference between the crank chamber pressure and the suction chamber pressure exceeds, for instance, 0.1 MPa, the opening degree control valve may not be engaged in operation until the swash plate tilt angle becomes 30% or less of the maximum tilt angle when the pressure in the discharge chamber is low (e.g., 0.8 MPa) and the suction passage constriction via the opening degree control valve may start with the swash plate tilt angle at 70% or less of the maximum tilt angle when the pressure in the discharge chamber is high (e.g., 2.5 MPa). In other words, the suction passage may be constricted and the performance of the compressor may be lowered under a high load condition, in which no low pressure pulsations occur and thus, the suction passage does not need to be constricted. Thus, the opening degree control valve disclosed in JP2005-337232-A2 does not assure both full air-cooling performance and pulsation reduction in correspondence to each specific load condition.

[0011] Primary objects of the present invention, having been achieved by addressing the issues described above, are to provide a variable-capacity swash plate compressor that does not allow the compressor performance to be lowered due to constriction of the suction passage in an operating area where low pressure pulsations do not occur and, at the same time, makes it possible to fully reduce low pressure pulsations by constricting the suction passage in an operating area where low pressure pulsations tend to occur readily and to provide an air-conditioning system equipped with such a variable-capacity swash plate compressor.

[0012] The inventors of the present invention, in an attempt to address the issues discussed above, have completed the present invention based upon a finding that the problem of the related art described above can be solved by allowing the passage area of a suction passage, through which coolant taken into the compressor is guided into a suction chamber, to be adjusted freely from the outside.

[0013] Namely, the variable-capacity swash plate compressor according to the present invention, comprising a housing, pistons that each moves reciprocally within a cylinder bore formed at the housing, a crank chamber, a suction chamber and a discharge chamber all formed within the housing, a shaft passing through the crank chamber and rotatably supported at the housing, a swash plate that is housed in the crank chamber and rotates as the shaft rotates, thereby causing the pistons to move reciprocally, a suction port formed at the housing, through which a working fluid is taken in, and a discharge port formed at the housing, through which the working fluid is discharged, wherein the working fluid taken in through the suction port is guided into the suction chamber via a

suction passage formed at the housing and then is compressed by the pistons before being discharged through the discharge port via the discharge chamber. This variable-capacity swash plate compressor is **characterized in that** a suction throttle valve, which adjusts the passage area of said suction passage through which the coolant passes, is disposed in said suction passage, and that said variable-capacity swash plate compressor further includes an external adjustment means for freely adjusting an opening degree of said suction throttle valve in accordance with a request (command) issued from outside.

[0014] In this structure, which includes a suction throttle valve disposed at the suction passage in the compressor and an external adjustment means for freely adjusting the opening degree of the suction throttle valve in accordance with a request issued from the outside, the suction throttle valve can be controlled to sustain a fully open state via the external adjustment means even if the crank chamber pressure rises by a predetermined extent during a middle to high displacement operation executed under a high discharge pressure condition, whereas propagation of suction pulsations can be suppressed in a reliable manner by engaging the suction throttle valve in operation via the external adjustment means under a condition in which low pressure pulsations tend to occur readily.

[0015] The external adjustment means may be a solenoid valve that adjusts the opening degree of the suction throttle valve by adjusting the pressure in a back pressure chamber of the suction throttle valve.

[0016] As the opening degree of the suction throttle valve is adjusted through adjustment of the pressure in the back pressure chamber of the suction throttle valve, the suction throttle valve can be engaged in operation with a sufficient force attributable to the pressure in the back pressure chamber, even when the extent of valve element displacement at the solenoid valve is small.

[0017] As an alternative, the external adjustment means may directly control the opening degree of the suction throttle valve in accordance with a request (command) issued from the outside. In this case, since the opening degree of the suction throttle valve is directly controlled in accordance with a request issued from the outside, the opening degree of the suction throttle valve can be controlled with a high degree of accuracy.

[0018] The air-conditioning system according to the present invention, comprising the variable-capacity swash plate compressor described above and at least a condenser, an expansion device and an evaporator, all connected together via a piping to configure a refrigerating cycle, is **characterized in that** the variable-capacity swash plate compressor further includes a pressure control valve that controls the pressure in the crank chamber based upon the pressure in the suction chamber and that the air conditioning system further comprises a suction throttle control means for driving the external adjustment means in response to a request issued on the vehicle side.

[0019] The external adjustment means may be a solenoid valve that adjusts the opening degree of the suction throttle valve by adjusting the pressure in the back pressure chamber of the suction throttle valve, or the external adjustment means may directly control the opening degree of the suction throttle valve in response to electric energy applied thereto.

[0020] The pressure control valve included in this structure controls the crank chamber pressure based upon the pressure in the suction chamber, i.e., based upon the pressure on a downstream side of the suction throttle valve, and thus, the capacity at the compressor is controlled based upon the pressure on the downstream side of the suction throttle valve. As the external adjustment means is driven in response to a request from the suction throttle control means and the suction passage is constricted via the suction throttle valve, the resulting constricting effect induces a decrease in the pressure in the suction chamber, the crank chamber pressure is raised via the pressure control valve with the lowered suction pressure applied thereto and, as a result, the discharge capacity decreases, which sustains the pressure in the suction chamber in balance at a pressure level prior to the suction passage constriction.

[0021] By taking advantage of the action of the variable-capacity compressor whereby it autonomously controls its capacity so as to sustain the suction chamber pressure at a constant level and creating a pressure difference via the suction throttle valve, the capacity can be freely decreased and, at the same time, propagation of low pressure pulsations under a small displacement condition can be suppressed.

[0022] According to the present invention described above, which includes a suction throttle valve disposed in a suction passage through which a working fluid taken in through a suction port is guided to a suction chamber, so as to adjust a passage area of the suction passage via the suction throttle valve, the opening degree of the suction throttle valve is freely adjusted via an external adjustment means in accordance with a request issued from the outside. As a result, it is ensured that any decrease in the performance level of the compressor caused by suction passage constriction does not occur in an operating area where low pressure pulsations do not occur, and that low pressure pulsations can be suppressed by a sufficient extent by engaging the suction throttle valve in operation via the external adjustment means and constricting the suction passage with the suction throttle valve in an operating area where low pressure pulsations tend to occur readily.

[0023] The invention will now be described with reference to the enclosed drawings. In the drawings :

- FIG. 1 presents an example of a configuration that may be adopted in the air-conditioning system according to the present invention;
- FIG. 2 is a sectional view of an example of a structure that may be adopted in the variable-capacity swash

plate compressor according to the present invention;

- FIG. 3 is a sectional view of an area of the valve plate that faces opposite a piston in the variable-capacity swash plate compressor;
- 5 - FIG. 4 is a sectional view of an example of a structure that may be adopted for the suction throttle valve and the external adjustment means in the variable-capacity swash plate compressor according to the present invention;
- 10 - FIG. 5 presents a flowchart of a control operation that may be executed to control the constriction varying mechanism;
- FIG. 6 is a sectional view of another example of a structure that may be adopted for the suction throttle valve and the external adjustment means in the variable-capacity swash plate compressor according to the present invention;
- 15 - FIG. 7 is a sectional view of yet another example of a structure that may be adopted for the suction throttle valve and the external adjustment means.
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[0024] The following is a description of embodiments of the present invention, given in reference to the attached drawings.

- 25 **[0025]** FIG. 1 presents an example of a configuration that may be adopted in a refrigerating cycle 1 installed in a vehicle. This refrigerating cycle 1 includes a variable-capacity swash plate compressor (hereafter to be referred to as a compressor) 4 equipped with a pressure control valve 2 via which the discharge capacity is adjusted and a constriction varying mechanism 3 that varies the passage area of a suction path through which coolant is guided to a suction chamber, a condenser 5 that cools the coolant, an expansion device 6 that depressurizes the coolant, and an evaporator 8 that is installed in an air-conditioning passage 7 and evaporates a working fluid. In the refrigerating cycle 1, the discharge side of the compressor 4 is connected to the expansion device 6 via the condenser 5 and a high pressure line is constituted with a path extending from the discharge side of the compressor 4 to the inflow side of the expansion device 6. In addition, the outflow side of the expansion device 6 is connected to the evaporator 8, the outflow side of the evaporator 8 is connected to the suction side of the compressor 4, and a low pressure line is constituted with a path extending from the outflow side of the expansion device 6 to the suction side of the compressor 4.
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- [0026]** Thus, the coolant, having been compressed at the compressor 4 and assuming a high temperature and a high pressure, enters the condenser 5 where it is cooled before being delivered to the expansion device 6 in the refrigerating cycle 1. The coolant is depressurized at the expansion device 6 and thus becomes a low temperature, low pressure moist vapor. The low temperature/low pressure moist vapor exchanges heat with air passing through the evaporator 8 and becomes gasified as a result of the heat exchange. The gasified coolant then returns to the compressor 4.
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[0027] Reference numeral 10 indicates a temperature sensor disposed in the air-conditioning passage to detect the temperature of the air on the outlet side of the evaporator 8. A signal output from the temperature sensor, a signal output from another sensor 11 that detects the cabin internal temperature or the like, and a signal output from an operation panel 12, via which the air-conditioning apparatus is turned on/off or a target temperature to be achieved in the cabin is set, are all input to a control device 13.

[0028] The control device 13, comprising an input circuit that inputs the various signals described above as data, a memory unit constituted with a read only memory (ROM) and a random access memory (RAM), a central processing unit (CPU) that processes the data and generates a control signal through arithmetic operation based upon a program called up from the memory unit where it is stored, a control signal output circuit that outputs a control signal to the constriction varying mechanism 3 and the like, controls the constriction varying mechanism 3 in accordance with the signals output from the various sensors 10 and 11 and the operation panel 12.

[0029] FIG. 2 presents a specific example of a structure that may be adopted in the compressor 4. This compressor includes a cylinder block 21, a rear housing 23 mounted on the rear side of the cylinder block 21 via a valve plate 22, and a front housing 25 mounted on the front side of the cylinder block, which defines a crank chamber 24 together with the cylinder block 21. The front housing 25, the cylinder block 21, the valve plate 22 and the rear housing 23, fastened together along the axial direction with fastening bolts 26, together constitute the housing for the compressor 4.

[0030] In the crank chamber 24 defined by the front housing 25 and the cylinder block 21, a shaft 27, one end of which projects out from the front housing 25, is housed. A clutch plate 29 is fixed over the area of the shaft 27 projecting out from the housing 25 via a connecting member 28 mounted along the axial direction. A drive pulley 30 is rotatably fitted around a boss portion 25a of the front housing 25 so as to face opposite the clutch plate 29, and as power is supplied to an excitable electromagnetic coil 31 embedded in the drive pulley 30, the clutch plate 29 is pulled to the drive pulley 30 and the rotating motive force imparted to the drive pulley 30 is transmitted to the shaft 27.

[0031] In addition, a side of the shaft 27 where the one end is present is sealed via a seal member 32 disposed between the one end of the shaft 27 and the front housing 25 so as to assure a high level of airtightness between the one end and the front housing 25.

[0032] The side of the shaft 27 where the one end is present is also rotatably supported by a radial bearing 33, whereas the side of the shaft 27 where the other end is present is rotatably supported by a thrust bearing 35 housed inside a housing hole 34 formed at a substantial center of the cylinder block 21 and a radial bearing 36

that is disposed next to the thrust bearing 35 toward the rear side.

[0033] The housing hole 34 and a plurality of cylinder bores 37 that set over equal intervals on the circumference of a circle centered on the housing hole 34, are formed at the cylinder block 21. A single-ended piston 40 inserted at each cylinder bore 37 is allowed to slide reciprocally within the cylinder bore.

[0034] A thrust flange 41, which rotates as one with the shaft 27, is fixed to the shaft 27 within the crank chamber 24. The thrust flange 41 is rotatably supported via a thrust bearing 42 at the inner wall surface of the front housing 25 formed substantially perpendicular to the shaft 27. A swash plate 44 is linked via a link member 43 to the thrust flange 41.

[0035] The swash plate 44, which is tiltably held via a hinge ball 45 disposed on the shaft 27, rotates as one with the thrust flange 41 by synchronizing with the rotation of the thrust flange 41. The thrust flange 41 and the swash plate 44 together constitute a motive power transmission mechanism that rotates synchronously with the rotation of the shaft 27. The circumferential edge portion of the swash plate 44 is held by an engaging portion 40a of the single-ended piston 40 via a pair of shoes 46 disposed at the front and rear sides of the swash plate 44.

[0036] Thus, as the shaft 27 rotates, the swash plate 44 also rotates, the rotating motion of the swash plate 32 is converted to linear reciprocal motion of the single-ended piston 40 via the shoes 46 and, as a result, the volumetric capacity of a compression space 47 (see FIG. 3) formed between the piston 40 and the valve plate 22 inside each cylinder bore 37 changes.

[0037] As shown in FIG. 3, a suction hole 51, which is opened/closed via a suction valve 50 disposed at an end surface of the valve plate 22 on the side further toward the cylinder block and a discharge hole 53 which is opened/closed via a discharge valve 52 disposed at an end surface of the valve plate 22 on the side further toward the rear housing, are formed in correspondence to each cylinder bore 37. In addition, a suction chamber 54 where the coolant to be supplied into the compression space 47 is stored and a discharge chamber 55 where the coolant discharged from the compression space 47 is stored are formed at the rear housing 23. In the current example, the suction chamber 54 is formed substantially at the center of the rear housing 23, whereas the discharge chamber 55 is formed around the suction chamber 54.

[0038] The suction valve 50 is held, together with a suction valve-side gasket 56, between the cylinder block 21 and the valve plate 22, whereas the discharge valve 52 is held, together with a discharge valve-side gasket 57, between the valve plate 22 and the rear housing 23.

[0039] The suction valve side gasket 56, inserted between the cylinder block 21 and the suction valve 50, is disposed so as to enclose the corresponding cylinder bore 37. In addition, a suction valve stopper 58, which regulates valve-opening operation of the suction valve

50, is formed at an end of the cylinder bore 37 located on a side further toward the valve plate. The suction valve stopper 58 is formed at a position facing opposite the tip of the suction valve 50 by grinding the end of the cylinder bore to a predetermined depth. In this example, the stopper depth is set in correspondence to the quantity of gas taken in through the suction valve under a middle displacement condition.

[0040] In addition, the discharge valve side gasket 57, inserted between the discharge valve 52 and the rear housing 23, is disposed so as to enclose the discharge hole 53, and a discharge valve stopper 59 used to regulate the valve opening operation of the discharge valve 52 is formed as an integrated part of the discharge valve side gasket 57.

[0041] At the rear housing 23, a suction port 60, through which the coolant is taken in from an outside cycle, a suction passage 61, through which the coolant having been taken in through the suction port 60 is guided to the suction chamber 54, a discharge port (not shown), through which the coolant is discharged to the outside cycle, and a discharge passage (not shown), through which the coolant having been discharged into the discharge chamber is guided to the discharge port, are formed.

[0042] As shown in FIG. 4, a suction throttle valve 62 is disposed on the suction passage 61 connecting the suction port 60 with the suction chamber 54. This suction throttle valve 62 is housed inside a valve housing chamber 62 formed within the suction passage 61 so as to allow displacement thereof along the coolant flow direction, and a force imparted from a spring 64 mounted at the valve housing chamber 63 is applied to the suction throttle valve 62 along the valve opening direction.

[0043] In more specific terms, the valve housing chamber 63, assuming a diameter greater than the diameter of the suction passage 61, is formed by aligning its axial center with the axial center of the suction passage 61 extending from the suction port 60, a communicating portion 66 opening in communication with the suction chamber 54 is formed at a side surface of the valve housing chamber 63, and the suction throttle valve 62 is formed in a tubular shape having one open end and another end that is closed off. The suction throttle valve 62 is slidably housed in the valve housing chamber 63 with a closed-off portion 62a of the suction throttle valve being toward the suction port and the force imparted from the spring 64 mounted at the side of the suction port in the valve housing chamber 63 is applied to the suction throttle valve so as to move it away from the suction port 60. When the suction throttle valve 62 is at the farthest position from the suction port 60 by the force imparted from the spring 64, the communication portion 66 is fully opened without being blocked by the side surface of the suction throttle valve 62. Then as the suction throttle valve 62, moving against the force imparted from the spring 64, is displaced toward the suction port, the communicating portion 66 becomes constricted with the side

surface of the suction throttle valve 62 and thus the opening degree of the communicating portion is reduced.

[0044] An orifice 67 is formed at the closed-off portion 62a of the suction throttle valve 62, a back pressure chamber 68 is formed inside the suction throttle valve 62, and the back pressure chamber 68 and the suction passage 61 are in communication with each other via the orifice 67. The suction throttle valve 62 is displaced to a position at which the pressure on the upstream side of the suction chamber (the pressure at the suction passage 61), the pressure in the back pressure chamber 68 and the spring force imparted from the spring 64 are in balance. Thus, unless the pressure in the back pressure chamber 68 is raised, the pressure in the back pressure chamber 68 and the pressure in the suction passage 61 are equalized via the orifice and the communicating portion 66 sustains a fully open state (the maximum opening degree is sustained at the suction throttle valve 62) with the spring force imparted from the spring 64.

[0045] The adjustment of the opening degree at the suction throttle valve 62, eventually, the adjustment of the pressure in the back pressure chamber 68, is achieved via a solenoid valve 60 constituting an external adjustment means, as described below. It is to be noted that the suction throttle valve 62 and the solenoid valve 70 together constitute the constriction varying mechanism 3 mentioned earlier.

[0046] At the solenoid valve 70, fitted inside the rear housing 23, a rod portion 75 of a valve element 73 thereof is slidably disposed in an axial hole 72 formed at the center of a body 71 assuming the shape of a circular column. The valve element 73 includes a head portion 74 formed to assume a diameter greater than the diameter of the axial hole 73, which sits at the circumferential edge of the opening of the axial hole 72 and the rod portion 75 extending from the head portion 74. At the rod portion 75, a constricted portion 75a is formed so as to assume a relatively-small diameter over a predetermined area starting from the head portion 74.

[0047] At the upper end of the body 71, a communicating chamber 76, which communicates with the back pressure chamber 68 of the suction throttle valve 62 and also communicates with the axial hole 72, is defined. The head portion 74 of the valve element 73 and a valve spring 77, which imparts a force applied to the valve element 73 along the closing direction (along the downward direction in the figure) are housed in the communicating chamber 76.

[0048] In addition, at the body 71, a cylinder 80 assuming a cylindrical shape, which communicates with the axial hole 72 and is securely locked by aligning its axial center with the axial center of the axial hole 72, an electromagnetic coil 81 wound around the cylinder 80, a plunger 82, which is slidably inserted through the cylinder 80 and comes into contact with the rod portion 75 of the valve element 73 inserted at the axial hole 72, an adjuster 83 mounted at an open end of the cylinder 80 located on the base end side, and a spring 84 mounted at a position

between the rear surface of the plunger 82 and the adjuster 83, which imparts a force to push the plunger 82 toward the valve element (toward the top side in the figure) are provided.

[0049] A discharge pressure drawing port 85, through which the pressure in the discharge chamber 55 is drawn in is disposed at a central area of the side surface of the solenoid valve 70, and a suction pressure drawing port 86, through which the pressure in the suction chamber 54 is drawn in, is disposed under the discharge pressure drawing port 85 (on the side further away from the back pressure chamber).

[0050] The discharge pressure drawing port 85, which is formed along the radius of the body 71, opens at the side surface of the axial hole 72 facing opposite the constricted portion 75a of the valve element 73. In addition, the suction pressure drawing port 86, extending along the radius of the body 71, communicates with a groove 87 formed at the inner circumferential surface of the axial hole 72 so as to guide the suction pressure, having been drawn in through the suction pressure drawing port 86, toward the bottom surface of the valve element 73.

[0051] Thus, the level of force pulling the plunger 82 downward can be adjusted by controlling the level of power supplied to the electromagnetic coil 81, and while no power is supplied to the electromagnetic coil 81, the force imparted from the spring 84 disposed under the plunger 82 pushes the plunger 82 upward, causing the valve element 73 to open against the force imparted from the valve spring 77. As a result, the high pressure gas drawn in via the discharge pressure drawing port 85 passes through the area around the constricted portion 75a formed at the valve element 73 to reach the communicating chamber 76 and then is guided to the back pressure chamber 68 via the communicating chamber 76.

[0052] In contrast, when power is supplied to the electromagnetic coil 81 so as to generate a pull strong enough to overcome the spring force, the plunger 82 is caused to move downward and the valve element 73 is displaced downward by the spring force imparted from the valve spring 77, thereby closing the axial hole 72. Since the axial hole 72 is formed so as to maintain a uniform diameter, the size of the area where the high pressure, taken in toward the constricted portion 75a of the valve element 73, is applied to the top surface of the constricted portion 75a and the size of the area where the high pressure is applied to the bottom surface of the constricted portion 75a remain the same. Thus, the discharge pressure, even when very high, does not cause the valve element 73 to open.

[0053] In addition, since the suction pressure, having been drawn in through the suction pressure drawing port 86, is directed to the bottom surface of the valve element 73, the influence of the suction pressure is also canceled out as long as the pressure in the back pressure chamber 68 is substantially equal to the pressure in the suction chamber 54. Namely, the opening degree of the solenoid valve can be controlled simply in correspondence to the

level of power supplied to the electromagnetic coil 81 without allowing any pressure to be an influential factor.

[0054] Furthermore, since the back pressure chamber 68 is made to communicate with the suction pressure area via the orifice 67 formed at the center of the suction throttle valve 62, as described earlier, the pressure in the back pressure chamber 68 can be adjusted based upon the balance between the quantity of gas drawn into the back pressure chamber 68 through the solenoid valve 70 and the quantity of gas released through the orifice 67.

[0055] The compressor described above further includes a pressure control valve 2 being well known, which is disposed in the rear housing 23 and is used to adjust the pressure in the crank chamber 24 based upon the pressure in the suction chamber 54. The pressure control valve 2 used in this type of application controls the crank chamber pressure by engaging a pressure-sensitive member such as a bellows or a diaphragm in operation in correspondence to the suction pressure and moving the valve element via the pressure-sensitive member whenever the suction pressure becomes lower than a predetermined value. While the crank chamber pressure may be adjusted through any of various methods, including a method whereby the discharge pressure is drawn into the crank chamber 24 by opening the valve seat whenever the suction pressure becomes lower than the predetermined value, a method whereby the quantity of gas released from the crank chamber 24 into the suction chamber 54 is regulated and a method combining both of the two methods, the present embodiment adopts a method whereby the state of communication at an air supply passage 89 that communicates the discharge chamber 55 with the crank chamber 24 is adjusted via the pressure control valve 2 so as to draw the high pressure gas in the discharge chamber 55 into the crank chamber 24. Via this pressure control valve 2, the pressure in the crank chamber 24 is controlled and thus the piston stroke, i.e., the discharge capacity, is adjusted.

[0056] FIG. 5 presents a flowchart of an example of control operation that may be executed by the control device 13 to control the constriction varying mechanism 3. The suction throttle control means for driving the solenoid valve in response to a request issued from the vehicle side is achieved through this control operation.

[0057] The following is a description of the example of the operation that may be executed to control the constriction varying mechanism 3, given in reference to the flowchart. The control device 13 first makes a decision as to whether or not the compressor 4 has been set in an OFF state in response to an air-conditioning apparatus OFF command (step S01). If it is decided in this decision-making step that the air-conditioning apparatus is in the OFF state, a control signal (Dt) for the solenoid valve 70 at the constriction varying mechanism 3 is set to 0 (the current supply to the solenoid is cleared) (step S02). As a result, the spring force imparted from the spring 84 pushes the plunger 82 upward in the figure and the valve element 73 also moves upward. In this situation,

the high pressure drawn into the back pressure chamber 68 pushes up the suction throttle valve against the force imparted from the spring 64, and the suction throttle valve blocks the communicating portion 66, thereby cutting off the suction passage 61.

[0058] If, on the other hand, it is decided in step S01 that the air-conditioning apparatus is in an operating state, a decision is made in step S03 as to whether or not the air temperature (T_e) at the evaporator outlet is higher than an air temperature setting ($T_e(\text{set})$) for the evaporator outlet and a further decision is made in step S04 as to whether or not the air temperature (T_e) at the evaporator outlet is lower than the air temperature setting ($T_e(\text{set})$) for the evaporator outlet. If the air temperature (T_e) at the evaporator outlet is determined to be higher than the air temperature setting ($T_e(\text{set})$) for the evaporator outlet, i.e., if the temperature of the air output from the evaporator 8 is higher than the temperature setting, a larger value is set for the control signal (duty ratio: D_t) (the current supplied to the solenoid is increased) so as to drive the suction throttle valve 62 toward the full open state (step S05). As a result, the plunger 82 is pulled downward, the valve element 73 is displaced along the closing direction (along the downward direction in the figure) with the spring force imparted from the valve spring 77 and thus the pressure supply into the back pressure chamber 68 becomes restricted, which, in turn, results in a reduction in the pressure difference between the pressure in the back pressure chamber 68 and the pressure in the suction passage 61. In this situation, the spring force imparted from the spring 64 induces a downward displacement of the suction throttle valve 62 and the extent to which the suction passage 61 is constricted is lessened (the constriction resistance is reduced by opening up the communicating portion 66).

[0059] If, on the other hand, the air temperature (T_e) at the evaporator outlet is determined to be lower than the evaporator outlet air temperature setting ($T_e(\text{set})$), i.e., if the temperature of the air output from the evaporator is lower than the temperature setting, a smaller value is set in the control signal (D_t) (the current supplied to the solenoid is reduced) so as to drive the suction throttle valve 62 toward a fully closed state (step S06). As a result, the plunger 82 is displaced upward with the spring force imparted from the spring 84, the valve element 73 is displaced along the opening direction against the spring force imparted from the valve spring 77 and the high pressure coolant supply into the back pressure chamber 68 is increased. Consequently, the pressure difference between the pressure in the back pressure chamber 68 and the pressure in the suction passage 61 increases and the suction throttle valve 62 is displaced upward against the spring force imparted from the spring 64, thereby increasing the extent to which the suction pressure 61 is constricted.

[0060] It is to be noted that if it is decided that the air temperature (T_e) at the evaporator outlet is equal to the evaporator outlet air temperature setting ($T_e(\text{set})$), the

current control condition is sustained (step S07).

[0061] The following is a description of various states of compressor operation, each corresponding to a specific heat load in the cabin, executed under the suction throttle valve control explained above.

Full-capacity operation:

[0062] When the air within the cabin has not been cooled to the full extent, the pressure (suction pressure) on the low pressure side of the refrigerating cycle 1 is relatively high. In this situation, pulsations are not likely to occur and a sufficient level of air cooling performance must be assured. Accordingly, the suction throttle control means applies an electrical current equal to or greater than a predetermined value to the electromagnetic coil 81 so as to sustain the fully closed state at the solenoid valve 70. As a result, the pressure in the back pressure chamber 68 is equalized with the pressure in the suction passage 61, the spring force imparted from the spring 64 sustains the suction throttle valve 62 in the fully open state, the pressure control valve 2 is not engaged in operation due to a sufficiently high pressure in the suction chamber 54, the swash plate 44 is operated at the maximum tilt angle, and the compressor 4 sustains the maximum-capacity operating state.

Medium-capacity operation:

[0063] As the air inside the cabin becomes cooled, the heat load on the evaporator 8, too, is lowered and thus, the suction pressure decreases. At this time, unless the air temperature at the outlet of the evaporator 8 is lower than the target air temperature, it indicates that the cooling capacity is not still sufficient. Accordingly, the suction throttle control means applies an electrical current equal to or greater than the predetermined value to the electromagnetic coil 81 so as to sustain the fully closed state at the solenoid valve 70. As a result, the suction throttle valve 62 sustains the fully open state. As the pressure in the suction chamber 54 at the compressor 4 becomes less than a predetermined value, the bellows in the pressure control valve 2 expands, thereby opening the valve seat. As a result, the high pressure gas in the discharge chamber is drawn into the crank chamber 24 and the tilt angle of the swash plate 44 decreases, leading to a reduction in the capacity of the compressor 4, while preventing any decrease in the suction pressure (excessive cooling).

Low-capacity operation + suction throttle:

[0064] If the air temperature at the outlet of the evaporator 8 becomes lower than the target outlet air temperature due to a further reduction in the heat load, change of the cabin temperature setting or the like, it indicates that the cooling capacity is excessive. Accordingly, the suction throttle control means reduces the level of power

supplied to the solenoid valve 70 and draws the high pressure into the back pressure chamber 68. By constricting the suction passage 61 via the suction throttle valve 62 in correspondence to the pressure difference between the increased pressure in the back pressure chamber 68 and the pressure in the suction passage 61, the pressure in the suction chamber 54 is lowered. Since the pressure control valve 2 raises the pressure in the crank chamber 24 according to the decrease in the pressure in the suction chamber 54 and thus the capacity of the compressor 4 is reduced, the cooling capacity is lowered and, as a result, the pressure in the suction chamber 54 is sustained at a constant level. At this time, the pressure on the upstream side of the suction throttle valve 62, i.e., the pressure on the side of the evaporator 8 in the refrigerating cycle, is higher than the pressure in the suction chamber 54 due to the pressure difference acting on the front and behind of the suction throttle valve 62. Consequently, capacity control equivalent to control under which the cooling capacity is reduced by raising the suction pressure setting of the pressure control valve 2 is achieved.

[0065] While suction pulsations tend to occur readily during low-load, low-capacity operation, propagation of the pulsations to the vehicle side is minimized with the suction throttle valve 62 constricting the suction passage 61.

[0066] It is to be noted that while the suction throttle valve 62 in the embodiment described above opens/closes along a direction matching the direction of the intake gas flow, a spool-type suction throttle valve 62 may be used instead and, in such a case, the suction throttle valve 62 will open/close along a direction perpendicular to the direction of the intake gas flow, as shown in FIG. 6.

[0067] In this example, the suction throttle valve 62 is disposed in a valve element sliding passage 90, which extends perpendicular to the suction passage 61, at the rear housing 23, allowing the spool-type suction throttle valve 62 to slide through the valve element sliding passage 90 and control of the movement of the suction throttle valve 62 is enabled by mounting a solenoid valve 70 similar to that described earlier so as to face opposite the suction throttle valve 62 along the axis of the suction throttle valve 62.

[0068] The suction throttle valve 62 includes a small diameter portion 62a present at a middle area thereof, a first sliding portion 62b formed at one end thereof, which is pushed toward the solenoid valve 70 by a force imparted from a spring 91, and a second sliding portion 62c formed at another end thereof, which alters the section of the suction passage 61 through which coolant flows. The area where the spring 91 is housed is in communication with the suction chamber 54, the back pressure chamber 68 is formed between the second sliding portion 62c and the front end of the solenoid valve 70, and the back pressure chamber 68 communicates with the suction passage 61 via an orifice 67 formed at the second sliding portion 62c. Thus, as long as the pressure in the

back pressure chamber 68 and the pressure in the suction passage 61 are in balance, the second sliding portion 62c stays out of the suction passage 61 and the small diameter portion 62a faces toward the suction passage 61 by the force imparted from the spring 91 to the suction throttle valve 62. In addition, as the pressure in the back pressure chamber 68 rises to a level higher than the pressure in the suction passage 61, the suction throttle valve 62 becomes displaced to the left in the figure against the spring force imparted from the spring 91, causing the second sliding portions 62c to gradually constrict the suction passage 61. When the suction throttle valve 62 is displaced to the leftmost position, the suction passage 61 becomes blocked.

[0069] It is to be noted that since other structural features are similar to those of the embodiment described earlier, the same reference numerals are assigned to those structural features to preclude the necessity for a repeated explanation thereof. Advantages similar to those of the embodiment described earlier are also achieved by adopting this alternative structure.

[0070] In addition, while the back pressure chamber 68 in the embodiment described above is located at the rear of the suction throttle valve 62 and displacement of the suction throttle valve 62 is caused by the pressure in the back pressure chamber, the suction throttle valve 62 may be directly driven with a solenoid, as shown in FIG. 7.

[0071] In this example, the suction throttle valve 62 disposed in the suction passage 61 is formed as an integrated part of the solenoid valve 70 so as to allow the integrated unit to function as a plunger. The suction throttle valve 62 assuming a tubular shape with one end thereof closed off is displaceably housed inside an axial hole 72 formed at a body 71. The suction throttle valve 62 is also slidably inserted in a cylindrical cylinder 80 which is locked by aligning its axial center with the axial center of the axial hole 72 at the body 71. At the body 71, an inflow port 92, which is in communication with the axial hole 72, is formed at the front end thereof, an outflow port 93, which is in communication with the axial hole 72, is formed at the side surface thereof and thus, as the suction throttle valve 62 comes into contact with the front end of the body 71, the inflow port 92 becomes closed off.

[0072] In addition, the constriction varying mechanism 3, used to control the drive of the suction throttle valve 62, includes an electromagnetic coil 81 wound around the cylinder 80, an adjust cap 83 mounted at an opening formed on the base end side of the cylinder 80 and a spring 84 mounted between the adjust cap 83 and the suction throttle valve 62, which imparts a force to push the suction throttle valve 62 toward the inflow port 92. It is to be noted that the pressure in the suction chamber 54 is guided into the suction throttle valve 62 via passing holes 94, 95 and 96 formed at the body 71, the cylinder 80 and the suction throttle valve 62.

[0073] In this structure, the force applied to the suction throttle valve 62 to pull it downward can be adjusted by controlling the level of power supplied to the electromag-

netic coil 81. When no power is supplied to the electromagnetic coil 81, the suction throttle valve 62 is pushed upward by the force imparted from the spring 84 so that it closes off the inflow port 92 formed at the body 71 and ultimately blocks the suction passage 61.

[0074] In contrast, as power is supplied to the electromagnetic coil 81 at a level at which a pulling force great enough to overcome the spring force imparted by the spring 84 is induced, the suction throttle valve 62 moves downward and stops at a position at which the magnetic force, the pressure in the suction passage 61, the pressure in the suction chamber 64 and the spring force from the spring 84 are in balance. Then, if the suction throttle valve 62 moves to the lowermost position, the suction throttle valve 62 becomes disengaged from the outflow port 93 and, in this situation, the extent of communication between the inflow port 92 and the outflow port 93 is maximized. It is to be noted that the inflow port 92 and the outflow port 93 are formed to assure sufficient passage area so as to ensure that no pressure loss occurs even during operation executed at a high coolant flow rate with the inflow port 92 and the outflow port 93 communicating with each other to the maximum extent.

[0075] This structure, which includes the suction throttle valve 62 and the solenoid valve 70 configured as an integrated unit, too, allow the suction throttle valve to sustain the fully open state during middle- to high-capacity operation executed under a high discharge pressure condition and, at the same time, effectively suppresses propagation of the suction pulsations during operation executed under a condition in which low pressure pulsations tend to occur readily.

[0076] It is to be noted that while the back pressure chamber 68 and the solenoid valve 70 are disposed next to each other in the first and second embodiments described above, the back pressure chamber 68 and the solenoid 70 may be disposed at positions set apart from each other and in such a case, the pressure in the communicating chamber 76, linked with the back pressure chamber 68 through a gas passage, may be guided to the back pressure chamber 68 via the gas passage.

[0077] In addition, while the constriction varying mechanism 3 is controlled by designating the heat load in the refrigerating cycle as the control target in the control operation described in reference to FIG. 5, the constriction varying mechanism 3 may be controlled by detecting or estimating whether or not pulsations have occurred.

[0078] Furthermore, while the embodiments have been described by assuming that the pressure control valve is an internal control-type valve, the pressure control valve may instead be configured as an external control valve that allows the pressure setting point to be altered with an external force applied from, for instance, a solenoid and, in such a case, the cooling capacity control corresponding to the heat load in the refrigerating cycle may be achieved via this external control valve instead of the suction throttle valve. Under such circumstances, more accurate capacity control and more reliable sup-

pression of pulsations may be achieved by issuing a command for engaging the suction throttle valve in operation based upon whether or not pulsations have occurred.

[0079] Moreover, while the first and second embodiments have been described by assuming that the suction throttle valve is opened/closed along the direction matching the direction of the intake gas flow or that the suction throttle valve is a spool valve that opens/closes along the direction perpendicular to the intake gas flow, the suction passage may instead be constricted by rotating a butterfly valve by a predetermined angle via an external adjustment means.

EXPLANATION OF REFERENCE NUMERALS

[0080]

1	refrigerating cycle
2	control valve
3	constriction varying mechanism
4	compressor
5	condenser
6	expansion device
8	evaporator
21	cylinder block
22	valve plate
23	rear housing
25	front housing
37	cylinder bore
40	piston
44	swash plate
54	suction chamber
55	discharge chamber
60	suction port
61	suction passage
62	suction throttle valve
68	back pressure chamber
70	solenoid valve

Claims

1. A variable-capacity swash plate compressor, comprising:
 - a housing;
 - pistons that each moves reciprocally within a cylinder bore formed at said housing;
 - a crank chamber, a suction chamber and a discharge chamber all formed within said housing;
 - a shaft passing through said crank chamber and rotatably supported at said housing;
 - a swash plate that is housed in said crank chamber and rotates as said shaft rotates thereby causing said pistons to move reciprocally;
 - a suction port formed at said housing, through which a working fluid is taken in; and

- a discharge port formed at said housing, through which the working fluid is discharged,

wherein the working fluid taken in through said suction port is guided into the suction chamber via a suction passage formed at said housing and then is compressed by said pistons before being discharged through the discharge port via said discharge chamber,

characterized in that a suction throttle valve, which adjusts the passage area of said suction passage through which the coolant passes, is disposed in said suction passage, and that said variable-capacity swash plate compressor further includes an external adjustment means for freely adjusting an opening degree of said suction throttle valve in accordance with a request issued from outside.

2. A variable-capacity swash plate compressor according to claim 1, **characterized in that** said external adjustment means is a solenoid valve that adjusts the opening degree of said suction throttle valve by adjusting the pressure in a back pressure chamber of said suction throttle valve.
3. A variable-capacity swash plate compressor according to claim 1, **characterized in that** said external adjustment means directly controls the opening degree of said suction throttle valve in accordance with a request issued from outside.
4. An air-conditioning system achieved by connecting, via a piping, a variable-capacity swash plate compressor according to any one of claims 1 through 3, with at least a condenser, an expansion device and an evaporator so as to configure a refrigerating cycle, **characterized in that** the variable-capacity swash plate compressor further includes a pressure control valve that controls the pressure in said crank chamber based upon the pressure in the suction chamber; and that said air-conditioning system is further equipped with a suction throttle control means for driving said external adjustment means in response to a request issued on the vehicle side.

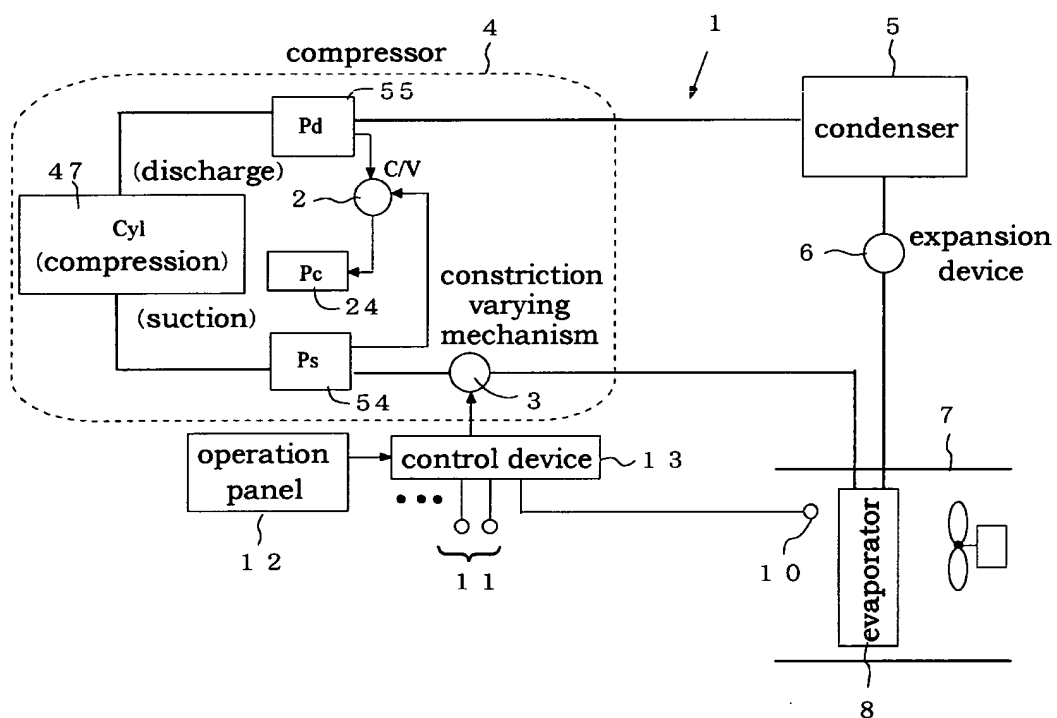


Fig. 1

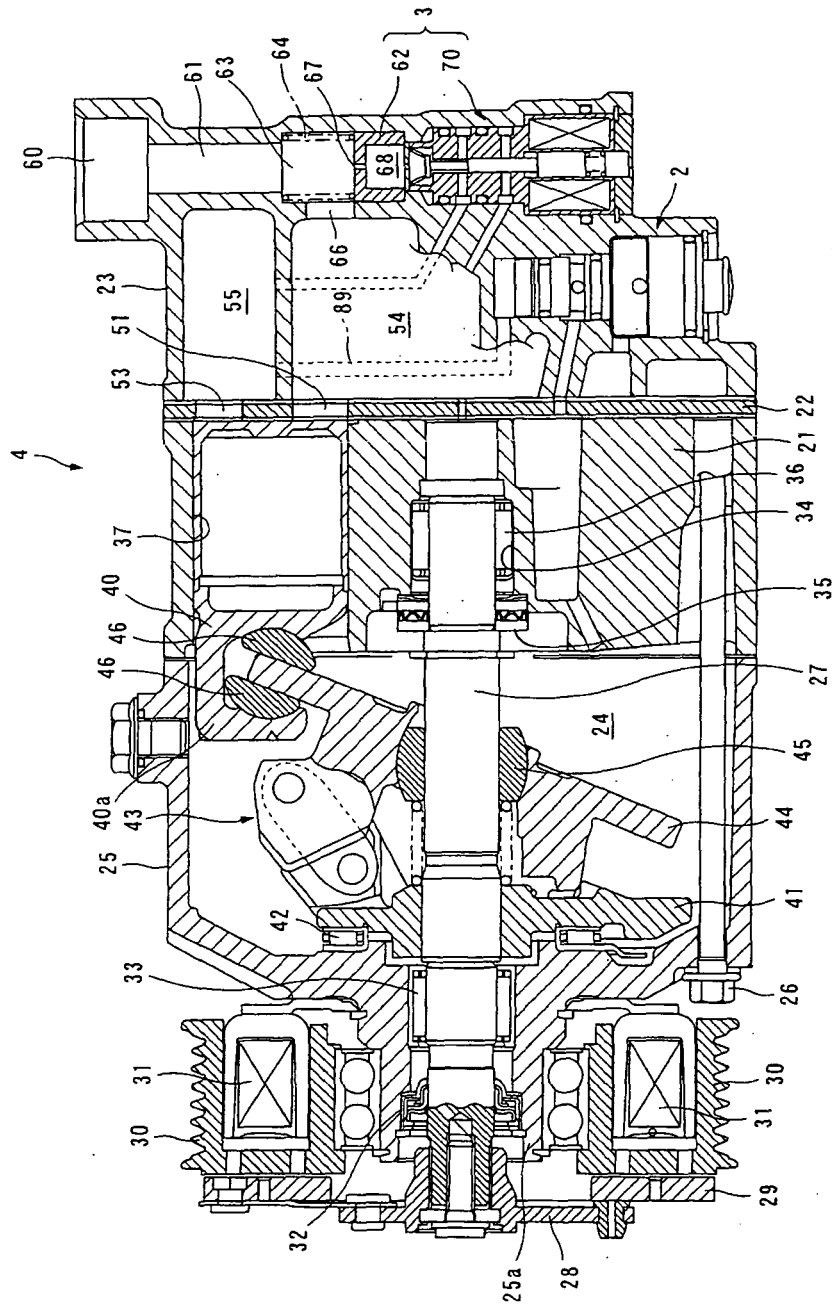


Fig. 2

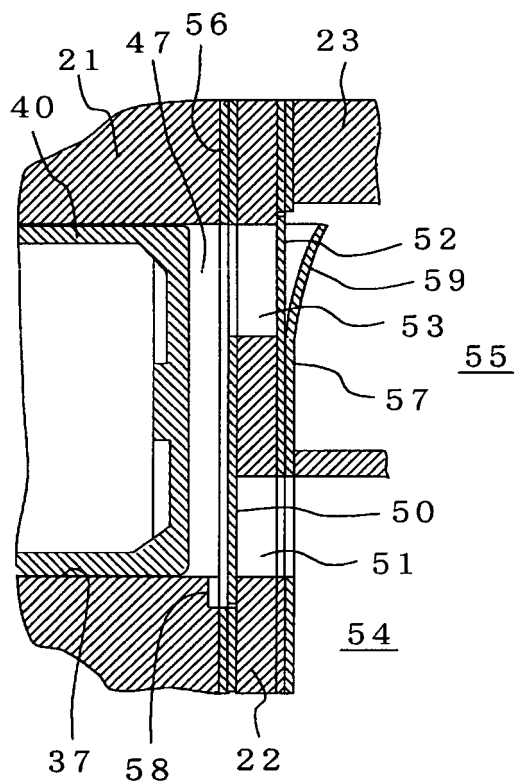


Fig. 3

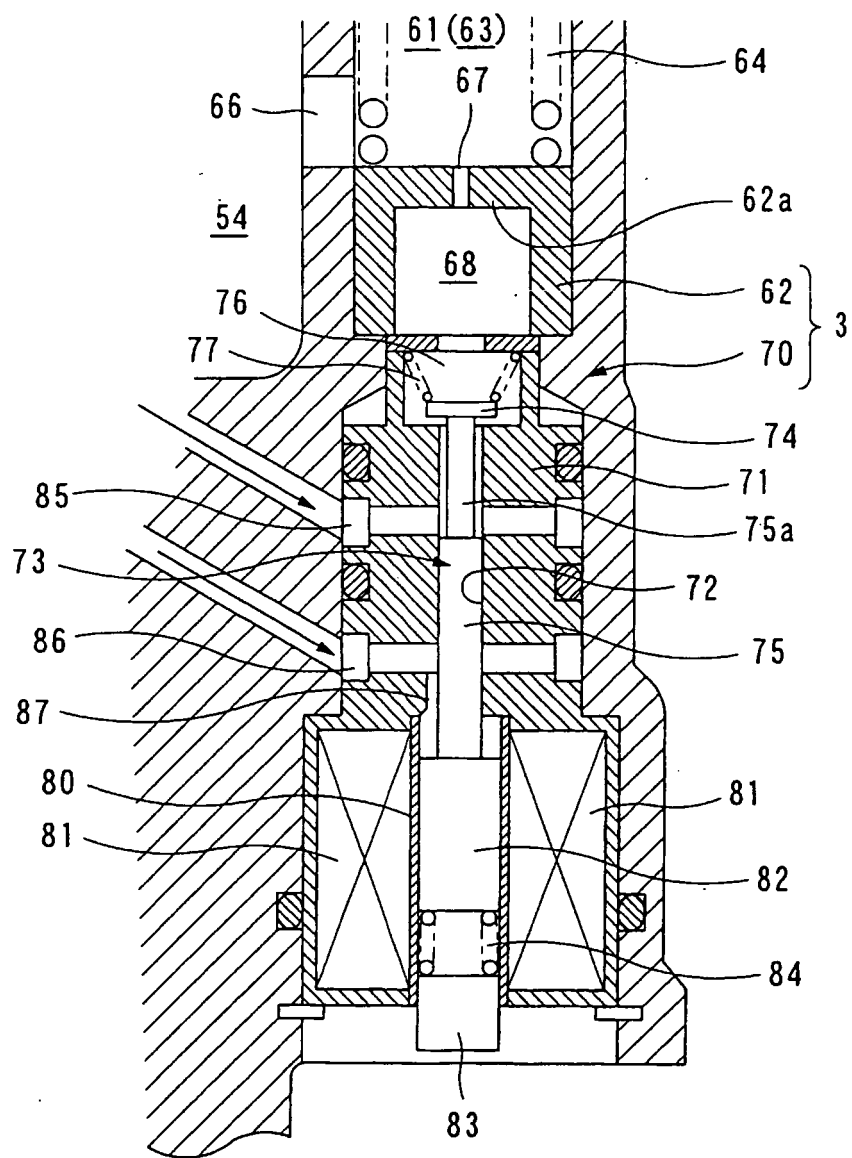


Fig. 4

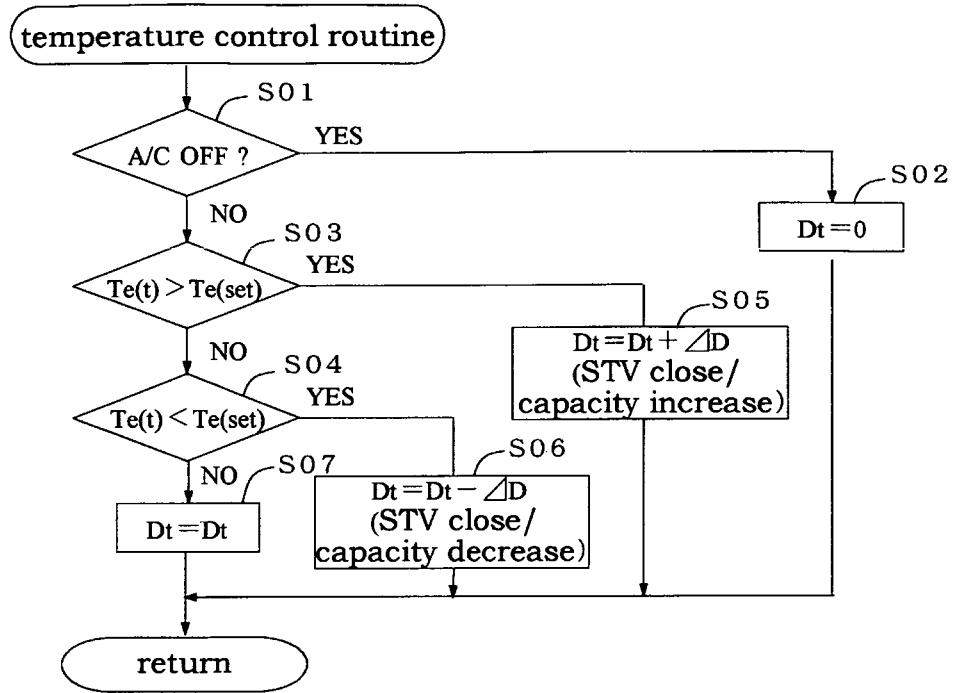


Fig. 5

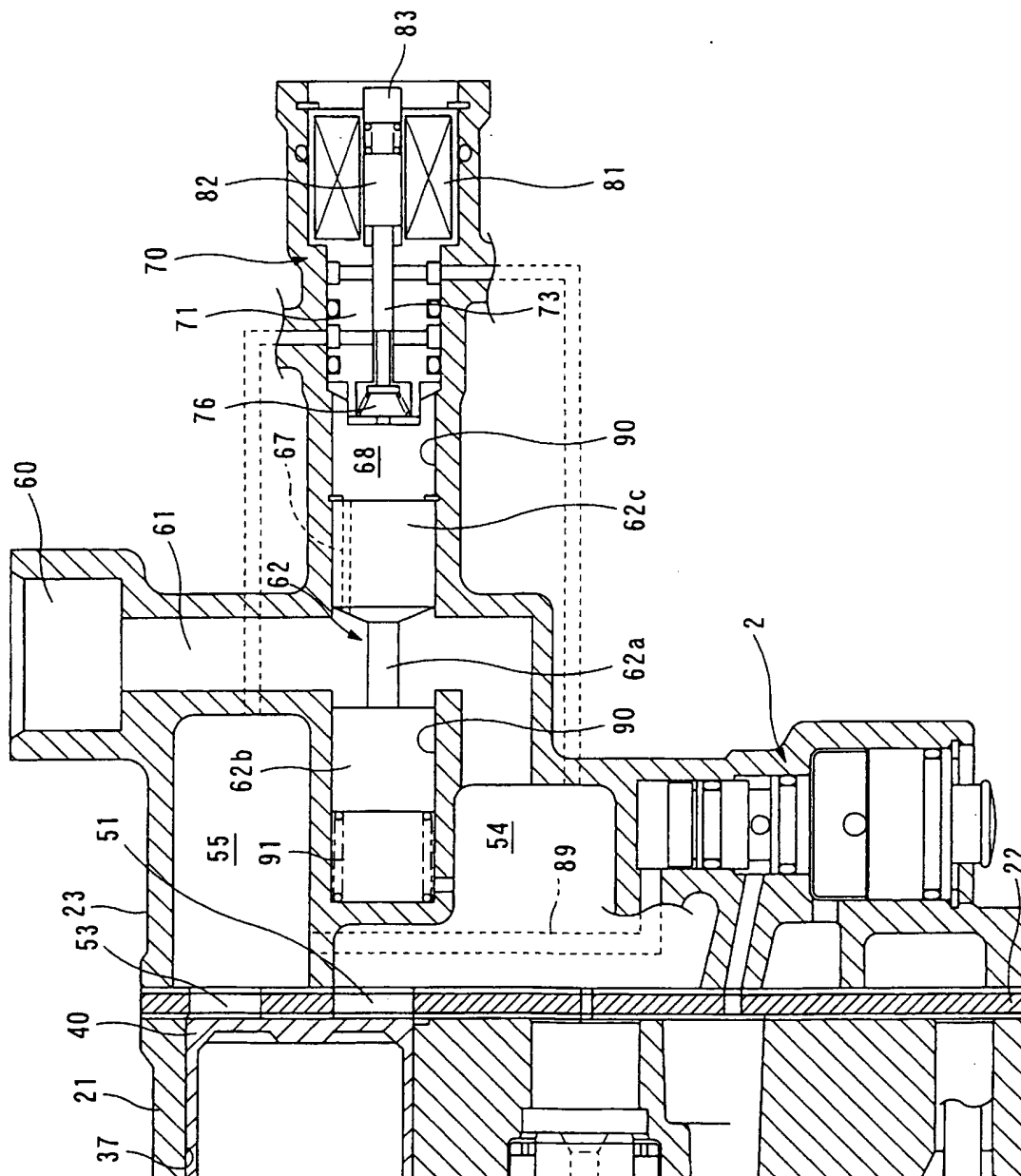


Fig. 6

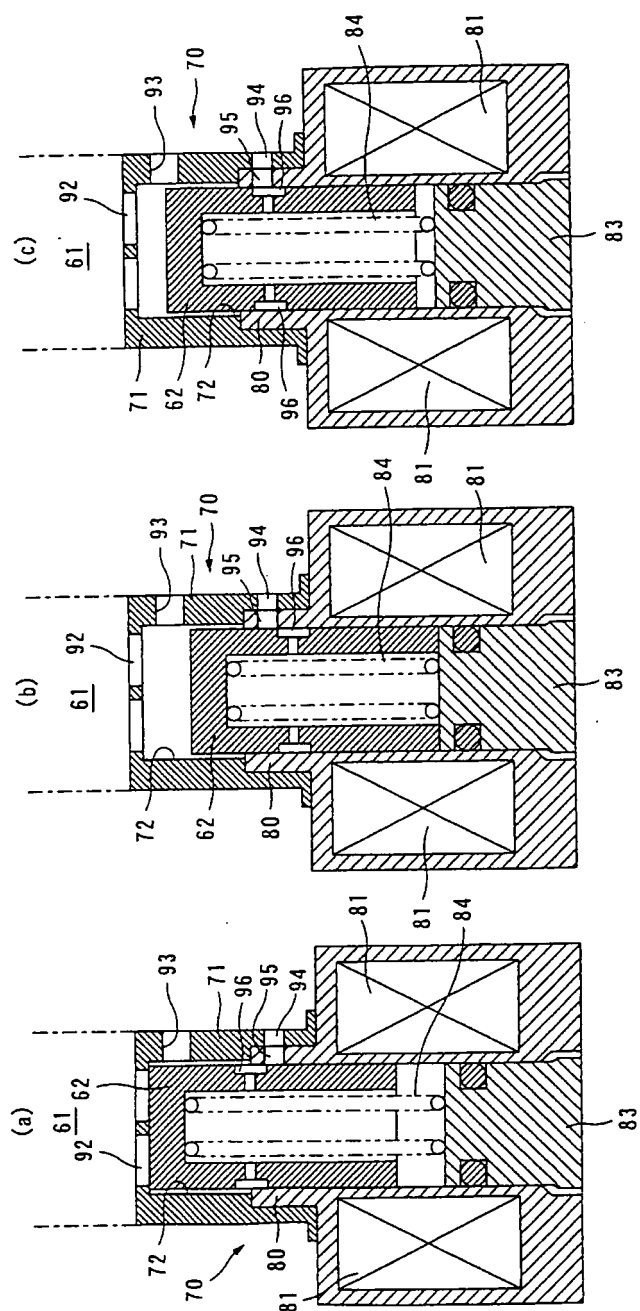


Fig. 7

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2010/004026

A. CLASSIFICATION OF SUBJECT MATTER

F04B27/08 (2006.01) i, B60H1/32 (2006.01) i, F04B27/14 (2006.01) i, F25B1/02 (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)

F04B27/08, F04B27/14

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.
X Y	JP 63-25455 A (Hitachi, Ltd.), 02 February 1988 (02.02.1988), page 4, upper right column, line 15 to page 5, lower right column, line 20; fig. 1 to 2, 5 to 6 (Family: none)	1-3 2
X Y	WO 2002/101237 A1 (TGK Co., Ltd.), 19 December 2002 (19.12.2002), page 6, lines 18 to 45; fig. 6 to 8 & US 2004/0091369 A1 & EP 1394412 A1 & DE 60218659 T2	1, 3-4 2

☐ Further documents are listed in the continuation of Box C.

☐ See patent family annex.

* Special categories of cited documents:

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"E" earlier application or patent but published on or after the international filing date

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"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search
07 October, 2010 (07.10.10)

Date of mailing of the international search report
19 October, 2010 (19.10.10)

Name and mailing address of the ISA/
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REFERENCES CITED IN THE DESCRIPTION

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- JP 2001136776 A [0004] [0005] [0007]
- JP 2005337232 A [0004] [0006] [0010]