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(54) **EXPANDER AND HEAT PUMP USING THE SAME**

(57) An expander of the present invention includes a plurality of suction ports for guiding a working fluid to a working chamber, and the plurality of suction ports includes a first suction port (71) and a second suction port (73) with a differential pressure valve (72). The ratio R2 of time length of an expansion process of expanding the working fluid in the working chamber to time length of a suction process in which the working fluid is sucked into the working chamber (55a) from the first suction port (71) and the second suction port (73) by opening the differential pressure valve (72) is smaller than the ratio R1 of the time length of the expansion process to a suction process in which the working fluid is sucked into the working chamber (55a) only from the first suction port (71) by closing the differential pressure valve (72).

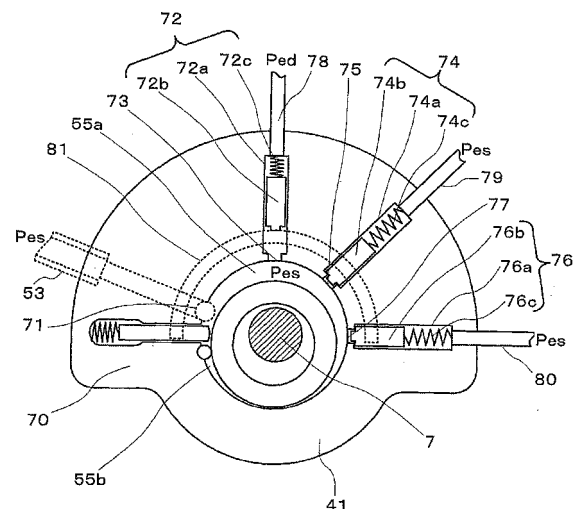


FIG. 4

Description

TECHNICAL FIELD

[0001] The present invention relates to an expander used for a refrigeration cycle apparatus (heat pump) usable as an air-conditioner, a water heater, and the like, and more particularly to a heat pump using the expander.

BACKGROUND ART

[0002] A power recovery type refrigeration cycle in which the energy of expansion of a working fluid (refrigerant) is recovered by an expander and the recovered energy is made use of as a part of the work of a compressor has been proposed. A refrigeration cycle that employs a fluid machine in which an expander and a compressor are coupled to each other by a shaft (hereinafter also referred to as an "expander-compressor unit") has been known as such a refrigeration cycle (see JP 2001-116371 A).

[0003] Hereinbelow, the refrigeration cycle employing the expander-compressor unit is described.

[0004] Fig. 12 shows the refrigeration cycle using the conventional expander-compressor unit. In this refrigeration cycle, a main circuit 108 for a working fluid (refrigerant) is constituted by a compressor 101, a gas cooler (radiator) 102, an expander 103, and an evaporator 104. The compressor 101, the expander 103, and a rotation motor 106 are coupled to each other by a shaft 107 to form an expander-compressor unit. The refrigerant circuit is provided with a sub-circuit 109 in addition to the main circuit 108. The sub-circuit 109 branches from the main circuit 108 at the outlet side of the gas cooler 102 and merges with the main circuit 108 at the inlet side of the evaporator 104. The working fluid that passes through the main circuit 108 is expanded at the expander 103, and the working fluid that passes through the sub-circuit 109 is expanded by an expansion valve 105.

[0005] The working fluid is compressed in the compressor 101 to convert from a low temperature, low pressure state to a high temperature, high pressure state, and thereafter is cooled in the gas cooler 102 to convert to a low temperature, high pressure state. Then, the working fluid is expanded in the expander 103 or the expansion valve 105 to a low temperature, low pressure state (gas-liquid two phase) and is heated at the evaporator 104 to return to a low temperature, low pressure state (vapor phase). The expander 103 recovers the energy of expansion of the working fluid and converts it into rotation energy for the shaft 107. This rotation energy is utilized as a part of the work for driving the compressor 101. As a result, the power driving the rotation motor 106 can be reduced.

[0006] Here, the operation of the refrigeration cycle when the expansion valve 105 is fully closed and the mass flow rate of the working fluid in the sub-circuit 109 is made zero will be described below.

[0007] The volume flow rate of the working fluid on the inlet side of the compressor 101 and that of the expander 103 are represented as $(V_{cs} \times N)$ and $(V_{es} \times N)$, respectively, wherein the suction volume of the compressor 101 is denoted as V_{cs} , the suction volume of the expander 103 is denoted as V_{es} , and the rotation speed of the shaft 107 is denoted as N . Since the mass flow rate of the working fluid in the sub-circuit 109 is zero, the mass flow rate in the compressor 101 and the mass flow rate in the expander 103 are equal to each other. Where the mass flow rate is denoted as G , the density of the working fluid on the inlet side of the compressor 101 and the density of the working fluid on the inlet side of the expander 103 are represented as $\{G/(V_{cs} \times N)\}$ and $\{G/(V_{es} \times N)\}$, respectively, from the ratios of the respective volume flow rates to mass flow rates. From these formulae, the ratio of the density of the working fluid on the inlet side of the expander 103 to the density of the working fluid on the inlet side of the compressor 101 can be represented as $\{G/(V_{cs} \times N)\}/\{G/(V_{es} \times N)\}$, and thus, (V_{es}/V_{cs}) , which means that the ratio is constant.

[0008] Fig. 13 shows a Mollier diagram of the refrigeration cycle. In the diagram, the compression process in the compressor 101 corresponds to the line AB, the heat radiation process in the gas cooler 102 corresponds to the line BC, the expansion process in the expander 103 corresponds to the line CD, and the evaporation process in the evaporator 104 corresponds to the line DA. The density ratio of the working fluid at point A on the inlet side of the compressor 101 and that at point C on the inlet side of the expander 103 is constant, (V_{es}/V_{cs}) , so the density ρ_c at point C can be represented as $(V_{cs}/V_{es})\rho_0$, where the density of the working fluid at point A is ρ_0 . Assuming that the density at point A is constant, increasing the pressure at point C means a shift from point C to point C' on the line $\rho_c = (V_{cs}/V_{es})\rho_0$. That is, it is impossible to shift the process from point C to point C', at which only the pressure is increased along the isothermal line ($T = T_c$). Thus, the refrigeration cycle is hindered from being controlled freely. In a refrigeration cycle, there is an optimal high pressure at which the coefficient of performance (COP) becomes maximum at a certain heat source temperature (for example, see JP 2002-81766 A). Therefore, the refrigeration cycle cannot be operated efficiently if the temperature and the pressure cannot be controlled freely.

[0009] The constraint of the constant ratio between the density on the inlet side of the compressor 101 and the density on the inlet side of the expander 103 is due to the fact that the mass flow rate in the compressor 101 and that in the expander 103 are equal to each other and also the ratio of the volume flow rates is constant. This constraint can be avoided by allowing a portion of the working fluid circulating in the refrigerant circuit to flow through the sub-circuit 109 by opening the expansion valve 105 (see JP 2001-116371 A).

[0010] In order to avoid the constraint of the constant density ratio in the power recovery-type heat pump em-

playing the conventional expander-compressor unit, which results from the fact that the compressor and the expander rotate at the same rotation speed, it is necessary to allow the working fluid to flow in the sub-circuit provided with an expansion valve as well as to the main circuit provided with an expander. In this configuration, however, the energy of expansion of the working fluid that passes through the sub-circuit cannot be recovered.

[0011] The problem of the inefficiency in recovering the energy of expansion of the working fluid is noticeable in the case of using an expander-compressor unit, but the problem also arises in the case of using a separate-type expander, which is not coupled to a compressor by a shaft. In the case of using a separate-type expander, the energy of expansion of the working fluid is recovered by a power generator connected to the expander. Since the power generation efficiency of the power generator becomes poorer when the rotation speed is more distant from the rated rotation speed, it is desirable that the power generator be operated at a speed in the vicinity of the rated rotation speed. In a refrigeration cycle, however, the circulation amount and the density of the working fluid change depending on the operation conditions, so it is difficult to operate the power generator only in the vicinity of the rated rotation speed. Thus, even in the separate-type expander, achieving efficient recovery of the energy of expansion of the working fluid is not easy.

DISCLOSURE OF THE INVENTION

[0012] The present invention has been accomplished in view of the foregoing circumstances, and it is an object of the invention to provide an expander capable of recovering the energy of expansion of the working fluid efficiently. It is another object of the present invention to provide a heat pump using the expander.

[0013] Accordingly, the present invention provides a rotary type expander including:

- a first cylinder;
- a first piston disposed in the first cylinder so as to rotate eccentrically in the first cylinder, and forming a first working chamber between itself and the first cylinder;
- a first partition member partitioning the first working chamber into a first suction-side space and a first discharge-side space;
- a second cylinder disposed so as to be concentric with the first cylinder;
- a second piston disposed in the second cylinder so as to rotate eccentrically in the second cylinder, and forming a second working chamber between itself and the second cylinder, the second working chamber having a greater volume than the first working chamber;
- a second partition member partitioning the second working chamber into a second suction-side space and a second discharge-side space;

a communication passage forming a working chamber, for expanding the working fluid, by permitting the first discharge-side space and the second intake-side space to communicate with each other;

a plurality of suction ports for guiding the working fluid to the first suction-side space, comprising a first suction port and a second suction port, the second suction port provided with a differential pressure valve and disposed at a position advanced from a position of the first suction port by a predetermined angle in a direction of rotation of the first piston; and a control pressure passage connected to the differential pressure valve, for supplying the differential pressure valve with a control pressure for opening and closing the differential pressure valve.

[0014] The present invention also provides an expander-compressor unit including an expander according to the present invention, a compressor, and a shaft coupling the expander and the compressor.

[0015] The present invention also provides a heat pump including the expander or the expander-compressor unit according to the present invention.

[0016] According to the expander of the present invention, it is possible to adjust the timing for shifting from the suction process for the working fluid to the expansion process for the working fluid by opening/closing the differential pressure valve of the second suction port. Specifically, it is possible to control the ratio of the time length for which the expansion process is performed to the time length for which the suction process is performed. As a result, according to the present invention, it becomes possible to change the foregoing ratio (V_{es}/V_{cs}), and it is possible to avoid the constraint of constant density ratio in, for example, a refrigeration cycle employing an expander-compressor unit. Therefore, the energy of expansion of the working fluid can be recovered efficiently by allowing the total amount of the working fluid to flow into the expander.

[0017] When using the expander according to the present invention as a separate-type expander, the rotation speed of the expander can be controlled while at the same time maintaining the amount of the working fluid flowing into the expander. As a result, it becomes easy to set the rotation speed of the power generator connected to the expander in the vicinity of the rated rotation speed and to maintain a high power generation efficiency of the power generator.

BRIEF DESCRIPTION OF THE DRAWINGS

[0018]

Fig. 1 is a vertical cross-sectional view illustrating an expander-compressor unit according to a first embodiment of the present invention.

Fig. 2A is a cross-sectional view of an expander section of the expander-compressor unit shown in Fig.

1, taken along line D1-D1 of Fig. 1.

Fig. 2B is a cross-sectional view of the expander section of the expander-compressor unit shown in Fig. 1, taken along line D2-D2 of Fig. 1.

Fig. 3 is a view illustrating one example of the configuration of a refrigeration cycle apparatus (heat pump) employing the expander-compressor unit shown in Fig. 1.

Fig. 4 is a partially enlarged view illustrating the cross section of the expander section of the expander-compressor unit shown in Fig. 1, taken along line D1-D1 of Fig. 1.

Fig. 5A is a view illustrating the operating principle of the expander section of the expander-compressor unit shown in Fig. 1.

Fig. 5B is a view illustrating the operating principle of the expander section of the expander-compressor unit shown in Fig. 1, along with Fig. 5A.

Fig. 6A is a chart illustrating the relationship between the rotation angle of the shaft and the process of the working chamber, according to the expander section of the expander-compressor unit shown in Fig. 1.

Fig. 6B is a chart illustrating the relationship between the rotation angle of the shaft and the volumetric capacity of the working chamber, according to the expander section of the expander-compressor unit shown in Fig. 1.

Fig. 7 is a Mollier diagram illustrating the refrigeration cycle using the expander-compressor unit shown in Fig. 1.

Fig. 8 is a P-V diagram illustrating the relationship between the pressure and the volumetric capacity of the working chamber, according to the expander section of the expander-compressor unit shown in Fig. 1.

Fig. 9 is a view illustrating the configuration of a heat pump according to a second embodiment of the present invention.

Fig. 10 is a graph illustrating an example of the relationship between the efficiency of a power generator and the rotation speed of the power generator.

Fig. 11 is a view illustrating the configuration of a heat pump according to a third embodiment of the present invention.

Fig. 12 is a view illustrating the configuration of a heat pump employing a conventional expander-compressor unit.

Fig. 13 is a Mollier diagram of the heat pump employing the conventional expander-compressor unit.

BEST MODE FOR CARRYING OUT THE INVENTION

[0019] Hereinbelow, preferred embodiments of the present invention are described with reference to the drawings.

(First Embodiment)

[0020] Fig. 1 is a vertical cross-sectional view illustrating an expander-compressor unit according to a first embodiment of the present invention. Fig. 2A is a horizontal cross-sectional view of an expander section of the expander-compressor unit shown in Fig. 1, taken along line D1-D1 of Fig. 1. Fig. 2B is a horizontal cross-sectional view of the expander section of the expander-compressor unit, taken along line D2-D2.

[0021] An expander-compressor unit according to the present embodiment includes a closed casing 11, a scroll type compressor section 1 disposed in an upper portion of the closed casing, a two-stage rotary expander section 3 disposed in a lower portion of the closed casing, a rotation motor 6 disposed between the compressor section 1 and the expander section 3 and having a rotor 6a and a stator 6b, and a shaft 7 for coupling the compressor section 1, the expander section 3, and the rotation motor 6 to one another.

[0022] The scroll type compressor section 1 has a stationary scroll 21, an orbiting scroll 22, an Oldham ring 23, a bearing member 24, a muffler 25 (silencer), a suction pipe 26, and a discharge pipe 27. The orbiting scroll 22 is fitted to an eccentric shaft 7a of the shaft 7 and its self rotation is restrained by the Oldham ring 23. The orbiting scroll 22, which has a vortex-shaped lap 22a meshing with a lap 21a of the stationary scroll 21, scrolls in association with the rotation of the shaft 7. A crescent-shaped working chamber 28 formed between the laps 21a and 22a moves from outside to inside so as to reduce its volumetric capacity, thereby compressing the working fluid sucked from the suction pipe 26. The compressed working fluid passes through a discharge port 21b provided at the center of the stationary scroll 21, an internal space 25a of the muffler 25, and a flow passage 29 penetrating through the stationary scroll 21 and the bearing member 24, in that order. The working fluid is then discharged to an internal space 11a of the closed casing 11. While the working fluid discharged to the internal space 11a is remaining in the internal space 11a, the lubricating oil mixed in the working fluid is separated from the working fluid by gravitational force or centrifugal force, and thereafter, the working fluid is discharged from the discharge pipe 27 to the refrigeration cycle.

[0023] The two-stage rotary expander section 3 includes a first cylinder 41, a second cylinder 42 having a greater thickness than the first cylinder 41, and an intermediate plate 43 for separating these cylinders 41 and 42. The first cylinder 41 and the second cylinder 42 are disposed concentrically with each other. The expander section 3 further includes a first piston 44, a first vane 46, a first spring 48, a second piston 45, a second vane 47, and a second spring 49. The first piston 44 is fitted to an eccentric portion 7b of the shaft 7 to perform eccentric rotational motion in the first cylinder 41. The first vane 46 is retained freely reciprocally in a vane groove 41a (see Fig. 2A) of the first cylinder 41 and is in contact

with the first piston 44 at one end. The first spring 48 is in contact with the other end of the first vane 46 and pushes the first vane 46 toward the first piston 44. The second piston 45 is fitted to an eccentric portion 7c of the shaft 7 to perform eccentric rotational motion in the second cylinder 42. The second vane 47 is retained freely reciprocally in a vane groove 42a (see Fig. 2B) of the second cylinder 42 and in contact with the second piston 45 at one end. The second spring 49 is in contact with the other end of the second vane 47 and pushes the second vane 47 toward the second piston 45.

[0024] The expander section 3 further includes an upper end plate 50 and a lower end plate 51 that are disposed so as to sandwich the first and second cylinders 41, 42 and the intermediate plate 43. The upper end plate 50 and the intermediate plate 43 sandwiches the first cylinder 41 from the top and bottom, and the intermediate plate 43 and the lower end plate 51 sandwiches the second cylinder 42 from the top and bottom. Sandwiching the first cylinder 41 and the second cylinder 42 by the upper end plate 50, the intermediate plate 43, and the lower end plate 51 forms working chambers, the volumetric capacities of which vary according to the rotations of the pistons 44 and 45, in the first cylinder 41 and the second cylinder 42. The upper end plate 50 and the lower end plate 51 also function as bearing members for retaining the shaft 7 rotatably, together with the bearing member 24 of the compressor section 1. Like the compressor section 1, the expander section 3 is furnished with a muffler 52, a suction pipe 53, and a discharge pipe (not shown).

[0025] As illustrated in Figs. 2A and 2B, a suction-side working chamber 55a (first suction-side space) and a discharge-side working chamber 55b (first discharge-side space), which are demarcated by the first piston 44 and the first vane 46, are formed in the first cylinder 41. Likewise, a suction-side working chamber 56a (second suction-side space) and a discharge-side working chamber 56b (second discharge-side space), which are demarcated by the second piston 45 and the second vane 47, are formed in the second cylinder 42. The total volumetric capacity of the two working chambers 56a and 56b in the second cylinder 42 is greater than the volumetric capacity of the two working chambers 55a and 55b in the first cylinder 41. The discharge-side working chamber 55b of the first cylinder 41 and the suction-side working chamber 56a of the second cylinder 42 are in communication with each other through a communication port 43a provided in the intermediate plate 43, so they can function as a single working chamber (expansion chamber). A high pressure working fluid flows into the working chamber 55a, and thereafter, it expands and reduces its pressure in the working chamber formed by the working chamber 55b and the working chamber 56a while rotating the shaft 7. Then, it is discharged from a discharge port 51a, which is provided in the lower end plate 51 so as to communicate with the working chamber 56b. The working fluid that has been discharged from the discharge port 51a

runs through an internal space 52a of the muffler 52 and a flow passage 57 penetrating the first and second cylinders 41 and 42 in that order, and then is discharged from the discharge pipe to the refrigeration cycle.

[0026] As illustrated in Fig. 2B, a discharge valve 82 is installed at the discharge port 51a provided in the lower end plate 51. The discharge valve 82 is made of, for example, a metal thin plate and is disposed so as to close the discharge port 51a from the internal space 52a side of the muffler 52. The discharge valve 82 is a differential pressure valve that opens when the pressure in the upstream side (i.e., the working chamber 56b side of the discharge-side of the second cylinder 42) becomes higher than the pressure in the downstream side (i.e., the internal space 52a side of the muffler 52). The discharge valve 82 has the function to prevent overexpansion of the working fluid in the expander section 3.

[0027] In the expander section 3, the working fluid is sucked into the working chamber 55a at least through a first suction port 71 being in communication with the suction pipe 53. In addition to the first suction port 71, the expander section 3 further has a second suction port 73, a third suction port 75, and a fourth suction port 77, serving as the suction ports for guiding the working fluid to the suction-side working chamber 55a of the first cylinder 41. The second suction port 73, the third suction port 75, and the fourth suction port 77 are provided at positions advanced from the position of the first suction port 71 by a predetermined angle in a direction of rotation of the pistons 44 and 45. These additional suction ports 73, 75, and 77 are provided with differential pressure valves 72, 74, and 76 so that the opening and closing of them are controlled by these valves 72, 74, and 76. The differential pressure valves 72, 74, and 76 have respective plungers 72b, 74b, and 76b and respective springs 72c, 74c, and 76c.

[0028] Specifically, in the present embodiment, the differential pressure valves 72, 74, and 76 are disposed within the first cylinder 41 and between an outer circumferential surface of the first cylinder and an inner circumferential surface of the first cylinder 41. It is possible to inhibit the size increase of the expander section 3 that may caused by providing the differential pressure valves 72, 74, and 76, and also, it is easy to design the expander section 3. It is also possible that the second suction port 73, the third suction port 75, and the fourth suction port 77 may be provided in the upper end plate 50, in which case the differential pressure valves 72, 74, and 76 may also be disposed within the upper end plate 50.

[0029] The plungers 72b, 74b, and 76b are disposed respectively in grooves 72a, 74a, and 76a, which are in communication with the working chamber 55a, so that they can reciprocate freely along the grooves 72a, 74a, and 76a. The grooves 72a, 74a, and 76a are formed in the first cylinder 41 so as to connect the working chamber 55a to pressure pipes 78, 79, and 80. One end of each of the springs 72c, 74c, and 76c is fitted to each of the end faces of the grooves 72a, 74a, and 76a on the sides

of the pressure pipes 78, 79, and 80, and the other end thereof is in contact with each of the end faces of the plungers 72b, 74b, and 76b. The springs 72c, 74c, and 76c are contracted and squeezed into the grooves 72a, 74a, and 76a so that they can keep the pushing force applied to the plungers 72b, 74b, and 76b even when the plungers 72b, 74b, and 76b occupy the positions closest to the working chamber 55a.

[0030] The pressure pipes 78, 79, and 80, which are connected to the differential pressure valves 72, 74, and 76, serve the role of control pressure passage for supplying a control pressure for opening and closing the differential pressure valves 72, 74, and 76 to the differential pressure valves 72, 74, and 76. The control pressure to be supplied to the differential pressure valves 72, 74, and 76 is equal to the pressure (P1) of the working fluid before expansion or the pressure (P2) of the working fluid after expansion. A control pressure equal to one of the pressure (P1) and the pressure (P2) is supplied selectively to the differential pressure valves 72, 74, and 76 through the pressure pipes 78, 79, and 80. The differential pressure valves 72, 74, and 76 close at least when the control pressure is equal to the pressure (P1), and the differential pressure valves 72, 74, and 76 open at least when the control pressure is equal to the pressure (P2). Thereby, the opening and closing of the differential pressure valves 72, 74, and 76 can be controlled easily.

[0031] The expander section 3 further includes: a the main passage 90 for guiding the working fluid to the first suction port 71; and a sub-passage 81 branching from the main passage 90 and provided along the outer circumference of the working chamber 55a of the first cylinder 41 in an arc-like shape, for guiding the working fluid to the second, third, and fourth suction ports 73, 75, and 77. By providing the main passage 90 and the sub-passage 81 in this way, the working fluid to be expanded can be guided from the suction pipe 53 to each of the second, third, and fourth suction ports 73, 75, and 77 at substantially the shortest distance, and an increase in the pressure loss can be prevented.

[0032] Specifically, the main passage 90 for guiding the working fluid from the suction pipe 53 to the first suction port 71 and the sub-passage 81 for guiding the working fluid from the suction pipe 53 to the differential pressure valves 72, 74, and 76 are formed in the upper end plate 50 as a bearing member. The sub-passage 81 extends in the upper end plate 50 and along the outer circumference of the working chamber 55a of the first cylinder 41 in an arc-like shape, to bring the suction pipe 53 and the grooves 72a, 74a, and 76a in communication with one another.

[0033] Thus, in the present embodiment, the differential pressure valves 72, 74, and 76 include (a) the grooves 72a, 74a, and 76a following the suction ports 73, 75, and 77 and extending outwardly in a radial direction of the first cylinder 41 so that at least a portion thereof overlaps with the sub-passage 81 regarding the axis direction of the shaft 7, and (b) plungers disposed in the grooves

72a, 74a, and 76a and being capable of reciprocating between two positions, one position being an open position for permitting the working fluid to flow through the sub-passage 81 to the suction ports 73, 75, and 77 from the main passage 90 and the other position being a close position for prohibiting the working fluid from flowing through the sub-passage 81 to the second suction port 73, 75, and 77 from the main passage 90. The reciprocating motion of the plungers is controlled by the control pressure. Thus, the working fluid can be guided from each of the second, third, and fourth suction ports 73, 75, and 77 to the suction-side working chamber 55a of the first cylinder 41.

[0034] In the present embodiment, the main passage 90 and the sub-passage 81 are provided in the upper end plate 50 as a bearing member that closes an end face of the first cylinder 41 on an opposite side to the side where the second cylinder 42 is positioned and supports the shaft 7 for rotating the first piston 44 and the second piston 45. The upper end plate 50 has a greater degree of freedom in shape and size than the first cylinder 41, so it is easy to provide the main passage 90 and the sub-passage 81 therein and connect the suction pipe 53 to the main passage 90. Of course, the main passage 90 and the sub-passage 81 as described above may be provided in the first cylinder 41. Alternatively, one of the main passage 90 and the sub-passage 81 may be provided in the first cylinder 41 while the other one is provided in the upper end plate 50.

[0035] When the plungers 72b, 74b, and 76b are pressed from the working chamber 55a side and move backward to the sides of the pressure pipes 78, 79, and 80, the differential pressure valves 72, 74, and 76 are opened and the working fluid is sucked also from the suction ports 73, 75, and 77 into the working chamber 55a. When the plungers 72b, 74b, and 76b are pressed from the sides of the pressure pipes 78, 79, and 80 and move forward to the working chamber 55a side, the differential pressure valves 72, 74, and 76 are closed and the working fluid is sucked only from the first suction port 71 into the working chamber 55a. The diameter of the suction ports 73, 75, and 77 is designed to be smaller than the diameter of the plungers 72b, 74b, and 76b so that the plungers 72b, 74b, and 76b do not jut out into the working chamber 55a.

[0036] The first suction port 71 and the second, third, and fourth suction ports 73, 75, and 77 are formed in an inner side wall inside the first cylinder 41. As has been mentioned previously, the suction ports 71, 73, 75, and 77 be formed in the upper end plate 50 adjoining the first cylinder 41 because it is sufficient that they face the working chamber 55a of the first cylinder 41. The first suction port 71 is formed at the position where $\phi = 20^\circ$ apart from the first vane 46 in the direction of rotation of the shaft 7 (clockwise in Fig. 2A) with respect to the central axis 70 of the shaft 7. According to the same manner of representation, the second suction port 73, the third suction port 75, and the fourth suction port 77 are formed at the

positions where $\varphi = 90^\circ$, $\varphi = 135^\circ$, and $\varphi = 1.50^\circ$, respectively. The angle φ that represents the position of the suction port by the foregoing manner is, more precisely, an angle formed by a first linear line 70a and a second linear line when the first linear line 70a is rotated to the second linear line in the direction of rotation of the shaft 7, taking the central axis 70 being the center of rotation. The first line is a line that connects the central axis 70 of the shaft 7 with the contact point between the first vane 46 and the first piston 44. The second linear line is a line that connects a second suction port with the central axis 70 of the shaft 7 (e.g., the linear line 70b in the case of the second suction port 73). The second, third, and fourth suction ports 73, 75, and 77 are formed at positions where angle φ is greater than that of the first suction port 71, in other words at downstream-side positions.

[0037] The pressure in the working chamber 55a is applied to each of one end faces (the end faces nearer the working chamber 55a) of the plungers 72b, 74b, and 76b, whereas the pressures in the pressure pipes 78, 79, and 80 and the pressure resulting from the pushing forces of the springs 72c, 74c, and 76c are applied to the other end faces (the end faces distant from the working chamber 55a). The positions of the plungers 72b, 74b, and 76b in the grooves 72a, 74a, and 76a, in other words, the opening and closing of the differential pressure valves 72, 74, and 76, are determined by the balance between these pressures and the pushing forces. For example, when the pressure in the working chamber 55a and the pressures in the pressure pipes 78, 79, and 80 are equal to each other, the plungers 72b, 74b, and 76b move forward by the pushing forces of the springs 72c, 74c, and 76c, closing the differential pressure valves 72, 74, and 76. On the other hand, when the pressure in the working chamber 55a exceeds the total of the pressures in the pressure pipes 78, 79, and 80 and the pushing forces of the springs 72c, 74c, and 76c, the plungers 72b, 74b, and 76b move backward by the pressure in the working chamber 55a, opening the differential pressure valves 72, 74, and 76. By making use of this, the opening and closing of the differential pressure valves 72, 74, and 76 can be controlled individually when the pressures in the pressure pipes 78, 79, and 80 are controlled individually.

[0038] It is preferable to use the pressure of a working fluid that circulates in the heat pump while shifting from a high pressure state to a low pressure state, or vice versa, as the control pressure for controlling the opening and closing of the differential pressure valves 72, 74, and 76. Hereinbelow, an embodiment in which the opening and closing of the differential pressure valves 72, 74, and 76 are controlled by the pressure of the working fluid will be described.

[0039] Fig. 3 illustrates the configuration of a power recovery type refrigeration cycle apparatus (heat pump) employing an expander-compressor unit according to the present embodiment. In addition to the expander-compressor unit, this apparatus includes a gas cooler (radiator) 2 and an evaporator 4, and further includes pipes

8 that connect the compressor section 1, the gas cooler 2, the expander section 3, and the evaporator 4 so that the working fluid flows therethrough in that order.

[0040] A high pressure pipe 83 is connected to the pipes 8 at a portion thereof through which a high pressure working fluid (that is, the working fluid that has been discharged from the compressor section 1 but has not yet flowed into the expander section 3) passes, specifically at a portion between the gas cooler 2 and the expander section 3. In addition, a low pressure pipe 84 is connected to the pipes 8 at a portion thereof through which a low pressure working fluid (that is, the working fluid that has been discharged from the expander 3 but has not yet flowed into the compressor section 1) passes, specifically at a portion between the expander section 3 and the evaporator 4. The high pressure pipe 83 and the low pressure pipe 84 are branched and connected to switching valves (three-way valves) 85, 86, and 87 respectively. The switching valves (three-way valves) 85, 86, and 87 also are connected respectively to the pressure pipes 78, 79, and 80. The pressure pipes 78, 79, 80, the high pressure pipe 83, the low pressure pipe 84, and the switching valves 85, 86, and 87 together constitute a control pressure passage that supplies a control pressure for opening and closing the differential pressure valves 72, 74, and 76 to the differential pressure valves 72, 74, and 76.

[0041] By switching the switching valves 85, 86, and 87, the high pressure pipe 83 or the low pressure pipe 84 can be connected to the pressure pipes 78, 79, and 80. Thus, a high pressure or a low pressure of the refrigeration cycle is supplied to the pressure pipes 78, 79, and 80. In the example shown in Fig. 3, the switching valve 85 connects the pressure pipe 78 to the low pressure pipe 84, and the switching valves 86 and 87 respectively connect the pressure pipes 79 and 80 to the high pressure pipe 83. In this state, a low pressure equal to a discharge pressure P_{ed} of the expander section 3 is applied to the plunger 72b of the differential pressure valve 72 through the pressure pipe 78, and a high pressure equal to a suction pressure P_{es} of the expander section 3 is applied to the plungers 74b and 76b of the differential pressure valves 74 and 76 through the pressure pipes 79 and 80.

[0042] The cross-sectional view of Fig. 4 also shows the pressures applied to the plungers 72b, 74b, and 76b of the differential pressure valves 72, 74, and 76 when the switching valves 85, 86, and 87 are set in the just-described manner. The working fluid with a suction pressure P_{es} is sucked into the working chamber 55a. As a result, the plunger 72b moves backward, reflecting the difference between the suction pressure P_{es} and the discharge pressure P_{ed} ($P_{es} > P_{ed}$), and correspondingly the differential pressure valve 72 opens, whereby the working fluid is sucked from the second suction port 73 into the working chamber 55a. On the other hand, the plungers 74b and 76b are pressed forward by the pushing forces of the springs 74c and 76c since the same pres-

sure P_{es} is applied to the opposite end faces of the plungers 74b and 76b, whereby the differential pressure valves 74 and 76 are in a closed state. Therefore, the working fluid is not sucked into the working chamber 55a from the third suction port 75 and the fourth suction port 77.

[0043] As is clear from the foregoing description, the pressure P_{sp} resulting from pushing forces of the springs 72c, 74c, and 76c is set to be smaller than the difference between the pressure P_{es} and the pressure P_{ed} (i.e., $P_{sp} < (P_{es} - P_{ed})$). In addition, the pressure P_{sp} resulting from pushing forces of the springs 72c, 74c, and 76c is set at a level that is sufficient for the differential pressure valves 72, 74, and 76 to close the plungers 72b, 74b, and 76b when the pressure difference that may be caused by other pressures becomes zero.

[0044] As described above, the pressures in the back side spaces of the plungers 72b, 74b, and 76b of the differential pressure valves 72, 74, and 76 can be switched by merely adding the three-way valves 85, 86, and 87 and the pipes 83 and 84 as shown in Fig. 3 to the basic configuration of the refrigeration cycle apparatus.

[0045] It should be noted that in the configuration shown in Fig. 3, the working fluid does not flow directly from the high pressure pipe 83 to the low pressure pipe 84 because the three-way valves 85, 86, and 87 are disposed between the high pressure pipe 83 and the low pressure pipe 84. Therefore, all the working fluid circulating in the refrigeration cycle passes through the expander section 3.

[0046] When the differential pressure valves are driven in the above-described manner, the need for actuators for driving the differential pressure valves is eliminated. It may be possible to drive the differential pressure valves using actuators such as represented by electric actuators, the use of actuators necessitates enlargement of the volume of the closed casing 11, resulting in an increase in the size of the overall apparatus. Moreover, in order to use electric actuators in a high temperature, high pressure working fluid, it is necessary to prevent deterioration of the insulating resin caused by permeation of the working fluid into the resin, which leads to deterioration in reliability. Therefore, the manufacturing costs increase significantly since off-the-shelf products cannot be used therefor. Particularly when carbon dioxide is used as the working fluid, the working fluid permeates into the resin considerably, promoting the deterioration, because the interior of the closed casing 11 reaches a pressure of 100 atm or higher and a temperature of 100°C or higher and also the carbon dioxide assumes a supercritical state.

[0047] For that reason, as the differential pressure valve, it is recommended to use such a differential valve as described above that is opened and closed by a control pressure applied thereto in such a manner that the differential pressure valve closes at least when the control pressure is equal to the pressure of the working fluid to be sucked into the working chamber and the differential pressure valve opens at least when the control pressure

is equal to the pressure of the working fluid discharged from the working chamber. More specifically, it is suitable to use a differential pressure valve including a plunger having one end face to which the pressure in the working chamber is applied and another end face to which the control pressure is applied, and a spring for pushing the plunger toward the working chamber side, as described above. This differential pressure valve is compact in size, simple in structure, and has high reliability. In addition, when at least a portion of the interior wall of the working chamber is formed by a cylinder and a piston that rotates eccentrically in the cylinder, it is advantageous to dispose the differential pressure valve within the cylinder in terms of size reduction of the apparatus.

[0048] Figs. 5A and 5B show views illustrating operation principles of the first cylinder 41 and the second cylinder 42. Figs. 5A and 5B show the states of the cylinders 41 and 42 at each 45° of rotation angle θ of the shaft 7. Here, the rotation angle θ of the shaft 7 is represented as 0° when the contact point between the first cylinder 41 and the first piston 44 is positioned at the first vane 46, i.e., when the contact point is at what is called the top dead center. The forward direction refers to the direction of rotation of the shaft 7, which is the clockwise direction.

[0049] The expander section 3 carries out 1 cycle that is from a suction process to a discharge process while the shaft 7 rotates 3 times. Therefore, in Figs. 5A and 5B, the rotation angle θ is denoted using an integer n ($n = 0, 1, 2$). Figs. 5A and 5B show the operation principle in the state in which the second suction port 73 is open while the third suction port 75 and the fourth suction port 77 are closed, as has been described with reference to Fig. 3.

[0050] The cycle starts from $\theta = 0^\circ$ at the first turn ($n = 0$) of the pistons 44 and 45. When the contact point between the first cylinder 41 and the first piston 44 passes the first suction port 71 at $\theta = 20^\circ$ (not shown), the working chamber 55a and the first suction port 71 are brought into communication with each other, and a suction process starts. When the contact point between the first cylinder 41 and the first piston 44 passes the second suction port 73 at $\theta = 90^\circ$, the working chamber 55a and the second suction port 73 are brought into communication with each other, and the working fluid flows into the working chamber 55a from the first suction port 71 and the second suction port 73 thereafter. The working chamber 55a is brought into communication with the third suction port 75 at $\theta = 135^\circ$ and the fourth suction port 77 at $\theta = 180^\circ$, but these suction ports 75 and 77 are kept closed by the differential pressure valves 74 and 76.

[0051] As the angle θ increases, the volumetric capacity of the working chamber 55a increases. After $\theta = 360^\circ$, at which the second turn ($n = 1$) starts, the working chamber 55a changes into the working chamber 55b, and the working chamber 55b is brought into communication with the working chamber 56a of the second cylinder 42 via the communication port 43a, forming a single working

chamber. As the shaft 7 rotates further, the contact point between the first cylinder 41 and the first piston 44 passes the first suction port 71 at $\theta = 380^\circ$ (not shown), and the communication between the working chamber 55b and the first suction port 71 is broken. In the conventional two-stage rotary expander, the suction process finishes at this point because the second suction port 73 is not provided.

[0052] In contrast, in the present embodiment, the working fluid continues to flow in from the second suction port 73 even after the first suction port 71 is closed. Then, at the stage where the angle reaches $\theta = 450^\circ$, the contact point between the first cylinder 41 and the first piston 44 passes the second suction port 73, so the communication between the working chamber 55b and the second suction port 73 is broken. The suction process finishes at this point.

[0053] When the suction process is completed, an expansion process for the working fluid is started. As the shaft 7 rotates further, the volumetric capacity of the working chamber 55b reduces, but the volumetric capacity of the working chamber 56a increases at a greater rate because the second cylinder 42 is axially higher and therefore has a greater volumetric capacity than the first cylinder 41. As a result, the total of the volumetric capacities of the working chamber 55b and the working chamber 56a increases, and the working fluid expands. When the angle θ reaches $\theta = 700^\circ$ (not shown), the contact point between the second cylinder 42 and the second piston 45 passes the discharge port 51a and the working chamber 56a is brought into communication with the discharge port 51a. The expansion process finishes at this point.

[0054] When the expansion process is completed, a discharge process for the working fluid is started. When $\theta = 720^\circ$, at which the third turn ($n = 2$) starts, the working chamber 55b of the first cylinder 41 disappears, and the working chamber 56a of the second cylinder 42 changes into the working chamber 56b. As the shaft 7 rotates further, the volumetric capacity of the working chamber 56b reduces, and the working fluid is discharged from the discharge port 51a. When $\theta = 1080^\circ$, the working chamber 56b disappears, and the discharge process finishes.

[0055] As is clear from the foregoing description, a suction process is completed and an expansion process is started at the point when the contact point between the first cylinder 41 and the first piston 44 passes the most downstream one of the suction ports that are open, among the plurality of suction ports 71, 73, 75, and 77, for the second time.

[0056] Fig. 6A illustrates the relationship between the shaft rotation angles θ and the shift time points for the processes from the suction process to the discharge process, in the cases that the most downstream suction port that is open is the first suction port 71, the second suction port 73, the third suction port 75, or the fourth suction port 77. As shown in Fig. 6A, when a more downstream suction port, i.e., a suction port with a greater

angle ϕ , is open, the timing for shifting from the suction process to the expansion process is later, so the suction process becomes longer while the expansion process becomes shorter. In other words, the ratio of the time length for which the expansion process is performed to the time length for which the suction process is performed is smaller.

[0057] Fig. 6B illustrates the relationship between the rotation angle θ of the shaft 7 and the volumetric capacity of the working chamber. As the working fluid moves through the working chamber 55a, the working chamber 55b, the working chamber 56a, and the working chamber 56b in that order, the volumetric capacity of the working chamber changes in a sine wave-like curve during that process. Suction volumes V_{esk} (where the suffix "k" is a number from (1) to (4)), which are the volumetric capacity of the working chamber at the end of the suction process for (1) to (4) in Fig. 6A, and a discharge volume V_{ed} , which is the volumetric capacity of the working chamber at the start of the discharge process, are indicated on the vertical axis of the graph. The suction volume V_{esk} increases when a more downstream suction port is opened, but the discharge volume V_{ed} is constant irrespective of the angle ϕ .

[0058] Thus, the present embodiment makes it possible to select the suction volume V_{esk} from four levels by providing the second suction port 73, the third suction port 75, the fourth suction port 77 having the differential pressure valves 72, 74, and 76 respectively in addition to the first suction port 71 as provided in the conventional two-stage rotary expander section 3. This makes it possible to control the density ratio (V_{cs}/V_{esk}) of the working fluid on the inlet side of the compressor section 1 and that of the expander section 3.

[0059] Fig. 7 illustrates a Mollier diagram of the refrigeration cycle employing the expander-compressor unit of the present embodiment. Since the density ratio can be selected, point C, which corresponds to the state on the inlet side of the two-stage rotary expander section 3 can be shifted to a point selected from points C_1 , C_2 , C_3 , and C_4 , by changing only the pressure along the isothermal line ($T = 35^\circ\text{C}$ in the example shown in the figure). Thus, the temperature and pressure on the inlet side of the two-stage rotary expander section 3 can be controlled in this way. As a result, it becomes possible to operate the refrigeration cycle with such a high efficiency as has been impossible with the refrigeration cycle employing the conventional expander-compressor unit.

[0060] In the present embodiment, the number of the differential pressure valves has been described as 4. However, the number of k may be selected from any number equal to or greater than 2 as appropriate.

[0061] When the number of k is 2, in other words, when the first suction port 71 and the second suction port 73 having a differential pressure valve are provided, the foregoing density ratio (V_{cs}/V_{esk}) can be varied between two levels, by controlling the differential pressure valve to select the ratio of the time length for the expansion

process to the time length for the suction process from a ratio R1 (the ratio in the case that the working fluid is sucked only from the first suction port 71) and a ratio R2 (the ratio in the case that the working fluid is sucked from both the first suction port 71 and the second suction port 73).

[0062] In other words, it is possible that a time length t2 of a suction process in which the working fluid is sucked into the working chamber 55a of the first cylinder 41 from the first suction port 71 and the second suction port 73 by opening the differential pressure valve 72 may be made greater than a time length t1 of a suction process in which the working fluid is sucked into the working chamber 55a only from first suction port 41 by closing the differential pressure valve 72.

[0063] If it is necessary to select the density ratio to a larger number of levels, the number of k may be increased by providing a greater number of suction ports, each having a differential pressure valve. For example, along with the first suction port 71 and the second suction port 73 with a differential pressure valve, the third suction port 75 with a differential pressure valve may further be provided at a more downstream position from the second suction port 73. In this case, a ratio R3 of the length of an expansion process for the working fluid to the length of a suction process for the working fluid into the working chamber in which the working fluid is sucked into the working chamber 55a from the first suction port 71, the second suction port 73, and the third suction port 75 by opening the differential pressure valves of the second suction port 73 and third suction port 75, is smaller than the foregoing ratios R1 and R2 (i.e., $R3 < R2 < R1$). Thus, the density ratio (V_{cs}/V_{esk}) can be varied in three levels.

[0064] In other words, it is possible that a length t3 of a suction process in which the working fluid is sucked from the first suction port 71, the second suction port 73, and the third suction port 75 to the working chamber 55a by opening the differential pressure valves 72 and 74 of the second suction port 73 and the third suction port 75 may be made greater than the foregoing length t2.

[0065] Next, the effect obtained by providing the discharge valve 82 for the discharge port 51a will be described below. Fig. 8 illustrates the relationship (P-V diagram) between the pressure and the volumetric capacity of the working chamber. The number suffixed to each symbol in the figure indicates the number of the most downstream suction port that is open, as in the foregoing. Point P ϕ denotes the start of an expansion process, point S ϕ denotes the end of an expansion process, and point T denotes the start of a discharge process. It should be noted that inflection point Q ϕ originating from a phase change is shown in the middle of each expansion process since it is assumed here that the refrigeration cycle uses carbon dioxide as the working fluid.

[0066] As the downstream suction port opens and the suction volume Vesk becomes greater, the volumetric capacity ratio ($= V_{ed}/V_{esk}$) before and after the expansion process becomes smaller and the pressure Pedk at

the end of the expansion process becomes higher, because the discharge volume Ved is constant. For this reason, when, for example, the suction ports 73, 75, and 77 provided with the differential pressure valves are disposed in the range up to 180° as represented by angle ϕ , it is desirable that the expander section 3 be designed in such a manner that the pressure Ped ϕ , which is the pressure at the end of the expansion process when the angle ϕ is the maximum angle 180°, is lower than the low pressure Ped of the refrigeration cycle to prevent under-expansion. The reason is that if underexpansion occurs, part of the energy originating from the pressure difference of the working fluid cannot be recovered.

[0067] In such a design, overexpansion occurs at least in the case that the angle ϕ is set at 180° or less. The overexpansion refers to a phenomenon in which the pressure Pedk becomes lower than the low pressure Ped in the refrigeration cycle. If the overexpansion takes place, overexpansion loss occurs in the discharge process because the working fluid needs to be pushed out from the discharge port 51a to the internal space 52a of the muffler 52, in which the pressure is higher than that in the working chamber 56b. The degree of the overexpansion loss can be represented by the area of the triangle R ϕ S ϕ T in Fig. 8.

[0068] When the discharge valve 82 is provided for the discharge port 51a, however, recompression is carried out in the discharge process when overexpansion R ϕ S ϕ occurs in the working chamber 56b. In the discharge process, the volumetric capacity of the working chamber 56b reduces as the shaft 7 rotates. When the discharge valve 82 is provided for the discharge port 51a, the discharge valve 82 does not open until the pressure of the working chamber 56b that has been reduced by overexpansion becomes equal to the low pressure Ped of the refrigeration cycle, and therefore the working fluid is recompressed in the working chamber 56b. Thus, the overexpansion loss can be prevented by providing the discharge valve 82.

[0069] Hereinbelow, other features of the present embodiment will be described.

[0070] In the present embodiment, the volumetric capacity ratio (V_{ed}/V_{esk}) in the expansion process is made variable in four levels by actuating the three differential pressure valves 72, 74, and 76. However, only two levels of control pressures Pes and Ped are required for the controlling of the four levels. In order to create multi-level control pressures, a complicated mechanism for producing pressures is necessary. Moreover, in a transition period such as at the start-up of the refrigeration cycle or at a change in operation conditions, the pressure of the working chamber 55a of the expander section 3 changes abruptly, and the control pressure needs to be adjusted accordingly. If the control pressure is set to be constant in the case that the controlling is carried out relying on the level of the control pressure, the polarity of the pressure difference may reverse in the transition period as mentioned above, and the differential pressure valves 72, 74, and 76 may fail to function properly. In contrast,

the present embodiment enables stable controlling because the control pressure also changes in a self-aligned manner in the transition periods.

[0071] Thus, it is recommended that the heat pump should be configured to include an expander described in the present embodiment and further include: a compressor; a high pressure pipe in which a high pressure working fluid compressed by the compressor flows; a low pressure pipe in which a low pressure working fluid expanded in the expander flows; a pressure pipe for supplying the control pressure to the differential pressure valve; and a switching valve connected to the pressure pipe, the high pressure pipe, and the low pressure pipe; and wherein, by switching the switching valve, a pressure of the working fluid in the high pressure pipe or a pressure of the working fluid in the low pressure pipe is applied as the control pressure to the differential pressure valve. The control pressure works on the differential pressure valve and operates the differential pressure valve in such a manner that the differential pressure valve closes at least when the control pressure is equal to the pressure P_{es} of the working fluid to be sucked into the working chamber, and the differential pressure valve opens at least when the control pressure is equal to the pressure P_{ed} of the working fluid discharged from the working chamber.

[0072] In the present embodiment, the additional suction ports 73, 75, and 77 are formed in a side wall in the interior of the first cylinder 40, and the grooves 72a, 74a, and 76a are formed so as to be connected to the suction ports 73, 75, and 77. In addition, the pressure pipes 78, 79, and 80 are disposed so as to be in communication with the grooves 72a, 74a, and 76a. As illustrated in Fig. 1, the pressure pipes 78, 79, and 80 can be disposed by inserting them through the holes provided in the side wall of the closed casing 11. Thus, assembling is easy, which is advantageous in reducing manufacturing costs.

[0073] In the present embodiment, a compressor that has a common structure, used for the refrigeration cycle that does not employ an expander, is used as the compressor section 1 since the suction volume V_{es} of the expander section 3 is made variable. For the compressor section 1, a common structure may be used without alteration and therefore the development costs can be reduced.

[0074] When using the expander-compressor unit of the present embodiment, the suction volume V_{es} can be adjusted according to the operation conditions while the circulation amount of the working fluid in the refrigeration cycle is being controlled by the rotation speed of the compressor section 1 and while the expander section 3 is being rotated at the same rotation speed as that of the compressor section 1. Therefore, it is possible for the compressor section 1 and the expander section 3 to serve different roles in controlling of the refrigeration cycle, and also, the control algorithm for the refrigeration cycle using the expander-compressor unit becomes simple.

[0075] Although there is no particular limitation on the

type of the working fluid used in expander-compressor unit of the present embodiment, carbon dioxide is suitable. This makes the effect of power recovery by the expander more significant. Accordingly, when using carbon dioxide as the working fluid, the effect of improving efficiency by avoiding a constant density ratio also becomes more significant.

[0076] In the present invention, a multi-stage rotary type expander is employed for the expander section 3. However, the same advantageous effects may be obtained even when a plurality of suction ports having differential pressure valves as in the present embodiment are provided for a scroll type expander or a sliding vane type expander. The number of cylinders in the rotary type expander is not particularly limited either. However, the rotary type expanders with two or more stages are more advantageous in providing a large number of suction ports having differential pressure valves because in such expanders, the suction process is performed in a working chamber 55a that comes in contact with the first cylinder 41 with a large area. Moreover, the present invention may be employed suitably also for a rotary type expander in which the vane and the piston are formed integrally.

(Second Embodiment)

[0077] The first embodiment has described a two-stage rotary expander (expander section 3) that, in addition to the first expansion mechanism, further includes a second expansion mechanism having a working chamber in communication with a working chamber in the first expansion mechanism through a communication port.

[0078] The expander section 3 may be used alone. In other words, it may be used as an expander separate from a compressor. Fig. 9 illustrates the configuration of a power recovery type refrigeration cycle apparatus employing the separate-type expander. This apparatus has a similar configuration to the refrigeration cycle apparatus shown in Fig. 3 (the same members are denoted by the same reference numerals, and the descriptions thereof are omitted). In place of the expander-compressor unit (reference numerals 1, 3, 6, and 7 in Fig. 3), this apparatus includes a compressor 61 and an expander 63, separated from each other, a rotation motor 66 connected to the compressor 61 via a shaft 7d, and a power generator 67 connected to the expander 63 via a shaft 7e. The refrigeration cycle of this apparatus is constituted by the compressor 61, a gas cooler (radiator) 2, the expander 63, and an evaporator 4. The compressor 61 is driven by the rotation motor 66. In the expander 63, the energy of expansion of the working fluid is converted into electric energy by the power generator 67, and the converted energy is used as a part of the input to the rotation motor 66.

[0079] Fig. 10 shows an efficiency curve for a common power generator 67. Since the power generator 67 is designed so that the power generation efficiency becomes highest at a predetermined rated rotation speed

Nr, its power generation efficiency becomes poorer when the rotation speed is more distant from the rated rotation speed. For this reason, it is desirable that the rotation speed of the power generator 67 be as close as possible to the rated rotation speed Nr. In a refrigeration cycle, however, the circulation amount and the density of the working fluid vary, and therefore, an expander with a constant suction volume Ves is difficult to operate only at a speed in the vicinity of the rated rotation speed Nr. When the expander section 3 according to the first embodiment is used as the expander 63, it becomes possible to control the rotation speed to a speed in the vicinity of the rated rotation speed Nr by adjusting the suction volume Vesk.

(Third Embodiment)

[0080] The present embodiment provides a refrigeration cycle apparatus that has the configuration shown in Fig. 11, in place of the configuration shown in Fig. 3. This apparatus has a similar configuration to that of the refrigeration cycle apparatus shown in Fig. 3 (the same members are denoted by the same reference numerals, and the descriptions thereof will be omitted), but employs a rotary valve 92 in place of the three-way valves 85, 86, and 87.

[0081] The rotary valve 92 has a cylinder 92a and a piston 92c that is disposed therein and is rotatable with a rotating shaft 92b as the center. The contact surface between the inner wall of the cylinder 92a and the piston 92c is sealed, and the piston 92c can be rotated from outside. The interior space of the cylinder 92a is divided by the piston 92c into a low pressure space 93a in communication with the low pressure pipe 84 and a high pressure space 93b in communication with the high pressure pipe 83.

[0082] The pressure pipes 78, 79, and 80 are connected to the cylinder 92a of the rotary valve 92 in that order from a low pressure space 93a side to a high pressure space 93b side. With this connection, the pressures of the pressure pipes 78, 79, and 80 can be switched sequentially between a suction pressure Pes and a discharge pressure Ped by rotating the piston 92c.

[0083] In the state shown in Fig. 11, a low pressure is supplied to the pressure pipe 78 while a high pressure is supplied to the pressure pipes 79 and 80. Therefore, the differential pressure valve 72 opens, but the differential pressure valves 74 and 76 are in a closed state. In this state, the second suction port 73 is the most downstream suction port that is open (state (2) in Fig. 6A). As the piston 92c is rotated anticlockwise in Fig. 11 from the state shown in Fig. 11, a low pressure is also supplied to the pressure pipe 79, so that the differential pressure valve 74 as well as the differential pressure valve 72 opens. In this state, the third suction port 75 is the most downstream suction port that is open (state (3) in Fig. 6A). As the piston 92c is further rotated anticlockwise, all the differential pressure valves 72, 74, and 76 open, and then the fourth suction port 77 is the most downstream

suction port that is open (state (4) in Fig. 6A). Conversely, when the piston 92c is rotated clockwise from the state shown in Fig. 11, a high pressure also is supplied to the pressure pipe 78, so that all the differential pressure valves 72, 74, and 76 are closed and the working fluid is supplied only from the first suction port 71, for which no differential pressure valve is provided (state (1) in Fig. 6A).

[0084] When the rotary valve 92 is used as in the present embodiment, a plurality of differential pressure valves can be controlled with only one valve. The controlling becomes simple because only one actuator needs to be controlled, and therefore, the structure of the piping also becomes simple.

INDUSTRIAL APPLICABILITY

[0085] As has been described above, the expander according to the present invention has great utility value since it provides an efficient means for recovering the energy of expansion of the working fluid in a refrigeration cycle and, in particular, achieves high efficiency in a heat pump employing an expander-compressor unit.

Claims

1. A rotary type expander comprising:

- a first cylinder;
- a first piston disposed in the first cylinder so as to rotate eccentrically in the first cylinder, and forming a first working chamber between itself and the first cylinder;
- a first partition member partitioning the first working chamber into a first suction-side space and a first discharge-side space;
- a second cylinder disposed so as to be concentric with the first cylinder;
- a second piston disposed in the second cylinder so as to rotate eccentrically in the second cylinder, and forming a second working chamber between itself and the second cylinder, the second working chamber having a greater volume than the first working chamber;
- a second partition member partitioning the second working chamber into a second suction-side space and a second discharge-side space;
- a communication passage forming a working chamber, for expanding the working fluid, by permitting the first discharge-side space and the second intake-side space to communicate with each other;
- a plurality of suction ports for guiding the working fluid to the first suction-side space, comprising a first suction port and a second suction port, the second suction port provided with a differential pressure valve and disposed at a position

- advanced from a position of the first suction port by a predetermined angle in a direction of rotation of the first piston; and
a control pressure passage connected to the differential pressure valve, for supplying the differential pressure valve with a control pressure for opening and closing the differential pressure valve. 5
2. The rotary type expander according to claim 1, wherein: 10
- the control pressure is equal to a pressure (P1) of the working fluid before expansion or a pressure (P2) of the working fluid after expansion; and 15
- the control pressure that is equal to one selected from the pressure (P1) and the pressure (P2) is supplied selectably to the differential pressure valve through the control pressure passage. 20
3. The rotary type expander according to claim 2, wherein: 25
- the differential pressure valve closes at least when the control pressure is equal to the pressure (P1); and
- the differential pressure valve opens at least when the control pressure is equal to the pressure (P2). 30
4. The rotary type expander according to claim 1, wherein a time length t2 of a suction process in which the working fluid is sucked into the first suction-side space from the first suction port and the second suction port by opening the differential pressure valve is greater than a time length t1 of a suction process in which the working fluid is sucked into the first suction-side space only from the first suction port by closing the differential pressure valve. 35 40
5. The rotary type expander according to claim 1, wherein the differential pressure valve comprises a plunger having one end face to which the pressure in the first working chamber is applied and another end face to which the control pressure is applied, and a spring for pushing the plunger toward the first working chamber. 45
6. The rotary type expander according to claim 1, wherein the differential pressure valve is disposed within the first cylinder and between an outer circumferential surface of the first cylinder and an inner circumferential surface of the first cylinder. 50 55
7. The rotary type expander according to claim 1, further comprising:
- a main passage for guiding the working fluid to the first suction port; and
a sub-passage branching from the main passage and provided along an outer circumference of the first working chamber, for guiding the working fluid to the second suction port.
8. The rotary type expander according to claim 7, wherein:
- the second suction port is provided in the first cylinder or in a closing member closing the first cylinder;
the differential pressure valve comprises (a) a groove following the second suction port and extending outwardly in a radial direction of the first cylinder so that at least a portion thereof overlaps with the sub-passage, and (b) a plunger disposed in the groove and being capable of reciprocating between two positions, one position being an open position for permitting the working fluid to flow through the sub-passage to the second suction port from the main passage and the other position being a close position for prohibiting the working fluid from flowing through the sub-passage to the second suction port from the main passage; and
the reciprocating motion of the plunger is controlled by the control pressure.
9. The rotary type expander according to claim 7, further comprising:
- a shaft for rotating the first piston and the second piston; and
a bearing member closing an end face of the first cylinder on an opposite side to the side where the second cylinder is positioned and supporting the shaft, wherein
the main passage and the sub-passage are provided in the bearing member.
10. The rotary type expander according to claim 1, wherein:
- the plurality of suction ports further comprises a third suction port provided with a differential pressure valve; and
a length t3 of a suction process in which the working-fluid is sucked from the first suction port, the second suction port, and the third suction port to the first suction-side space by opening the differential pressure valves of the second suction port and the third suction port is greater than the length t2.
11. An expander-compressor unit comprising: an expander according to claim 1; a compressor; and a

shaft coupling the expander and the compressor.

12. A heat pump comprising an expander according to claim 1.

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13. A heat pump comprising an expander-compressor unit according to claim 11.

14. A heat pump comprising an expander according to claim 1, further comprising:

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a compressor;

a high pressure pipe in which a high pressure working fluid compressed by the compressor flows;

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a low pressure pipe in which a low pressure working fluid expanded in the expander flows;

a pressure pipe for supplying the control pressure to the differential pressure valve; and

a switching valve connected to the pressure pipe, the high pressure pipe and the low pressure pipe, and wherein:

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the control pressure passage comprises the pressure pipe, the high pressure pipe, the low pressure pipe, and the switching valve; and

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by switching the switching valve, a pressure of the working fluid in the high pressure pipe or a pressure of the working fluid in the low pressure pipe is applied as the control pressure to the differential pressure valve.

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15. The heat pump according to claim 14, wherein the working fluid is carbon dioxide.

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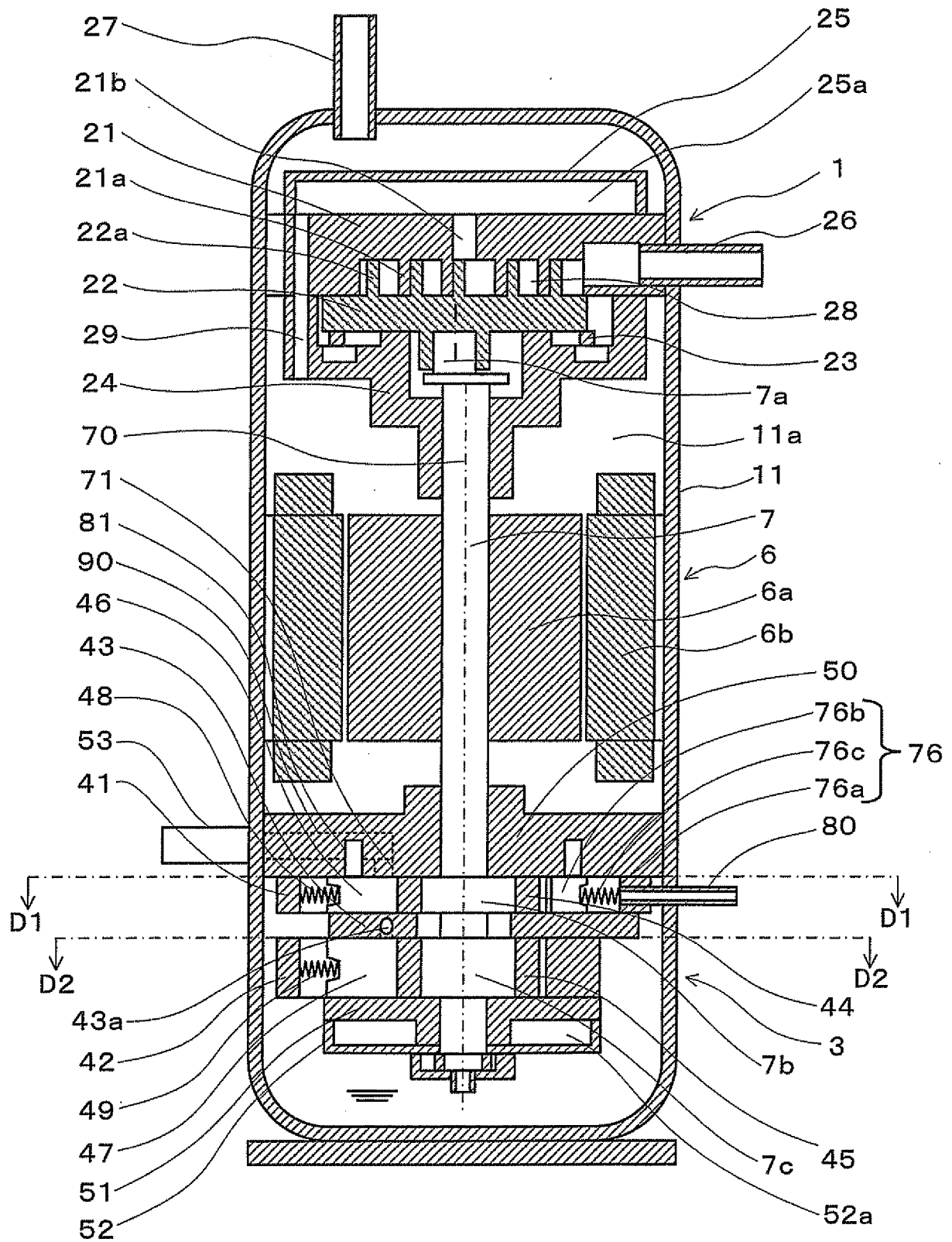


FIG. 1

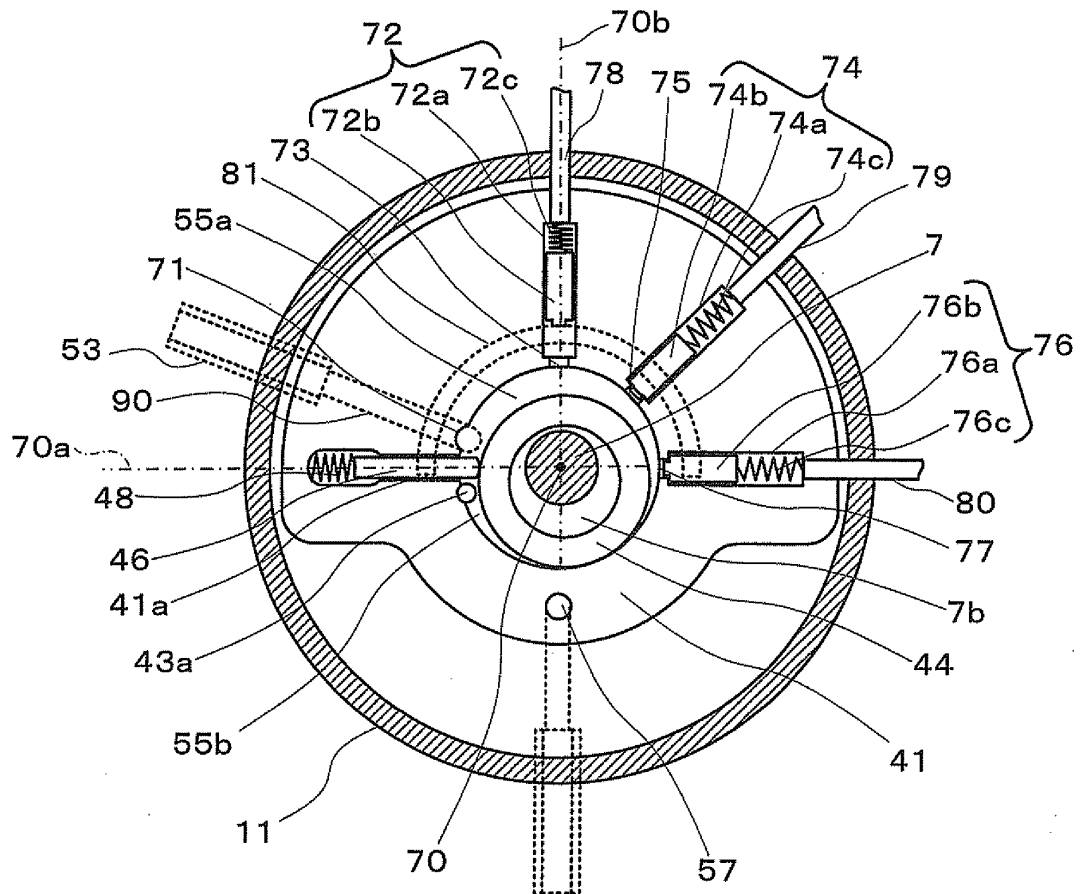


FIG. 2A

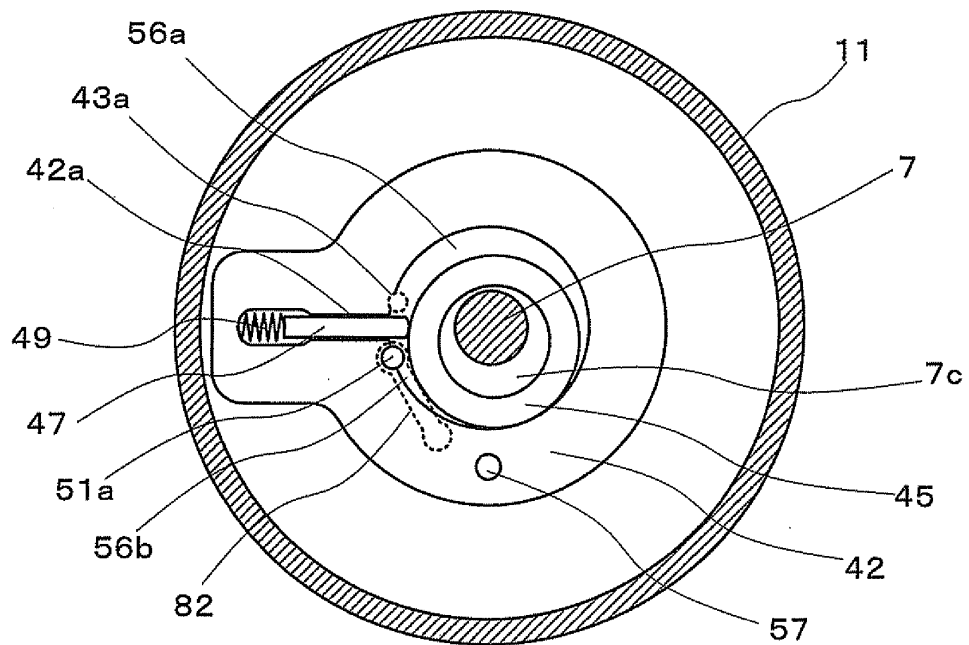


FIG. 2B

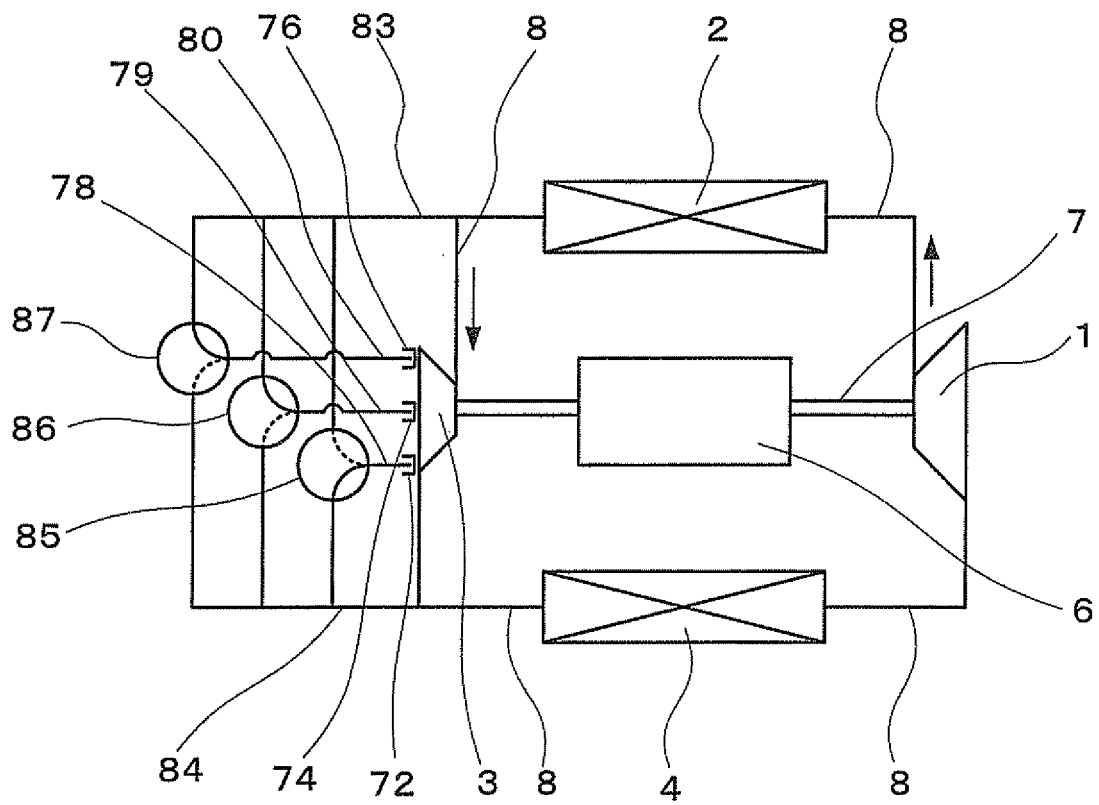


FIG. 3

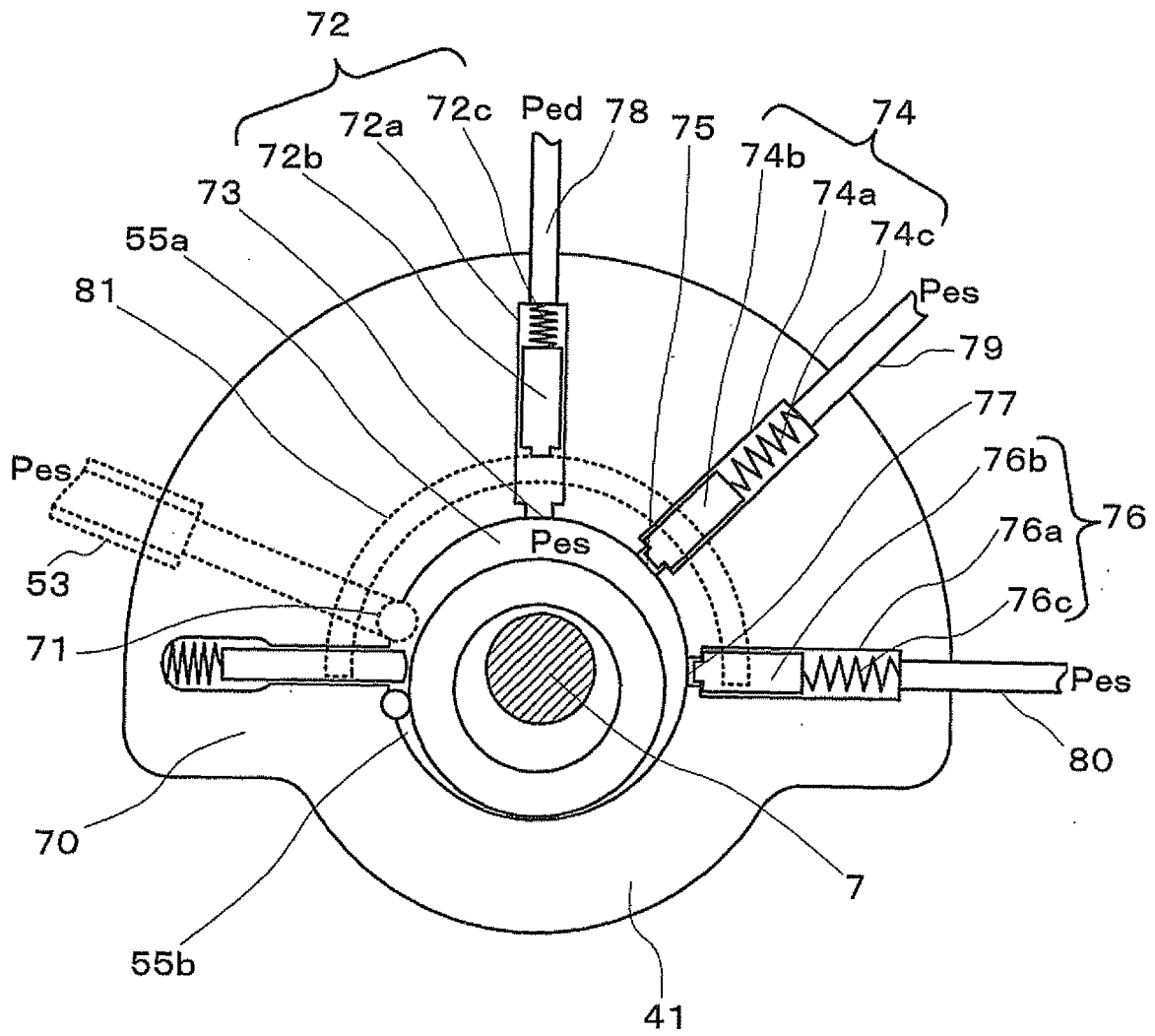


FIG. 4

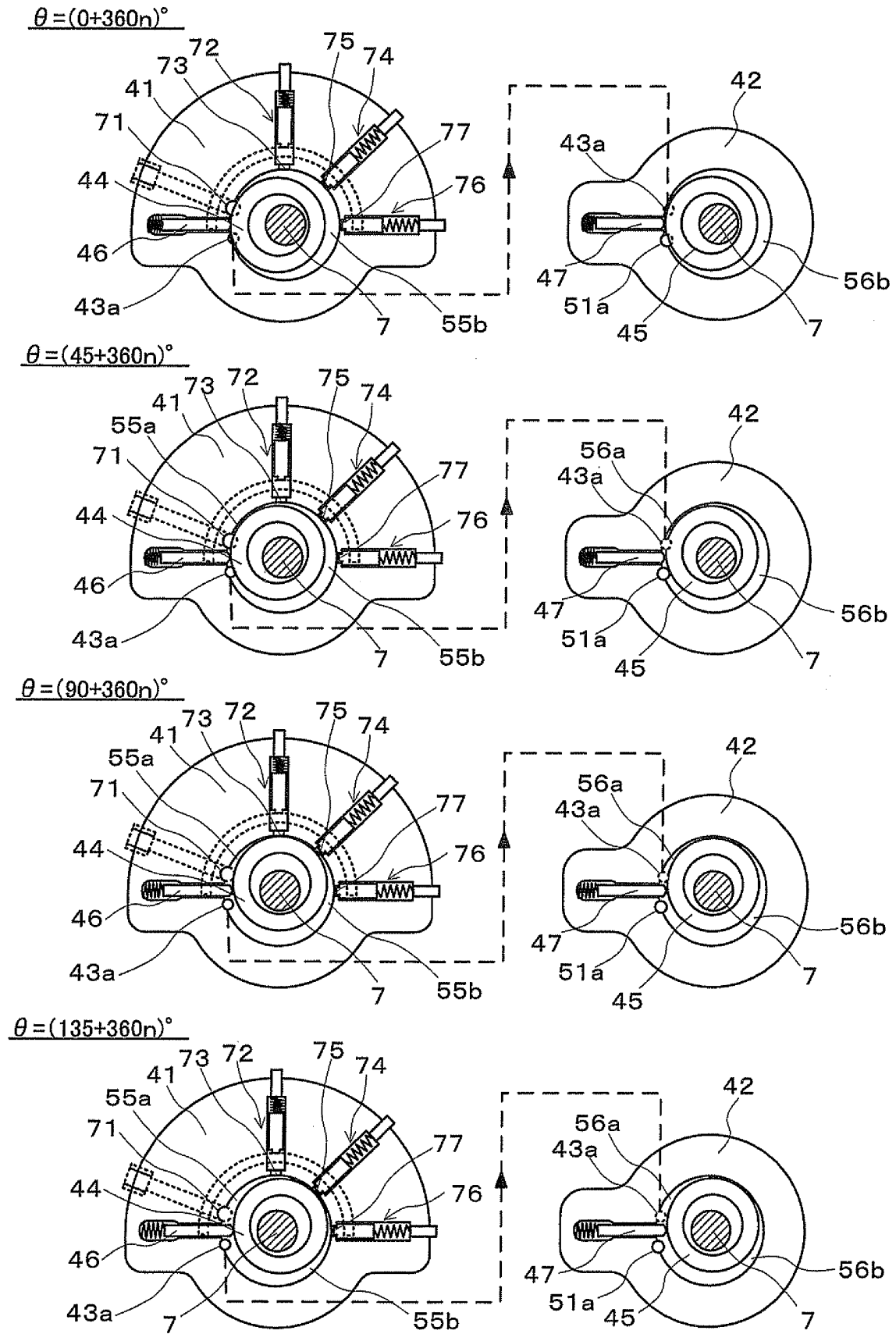
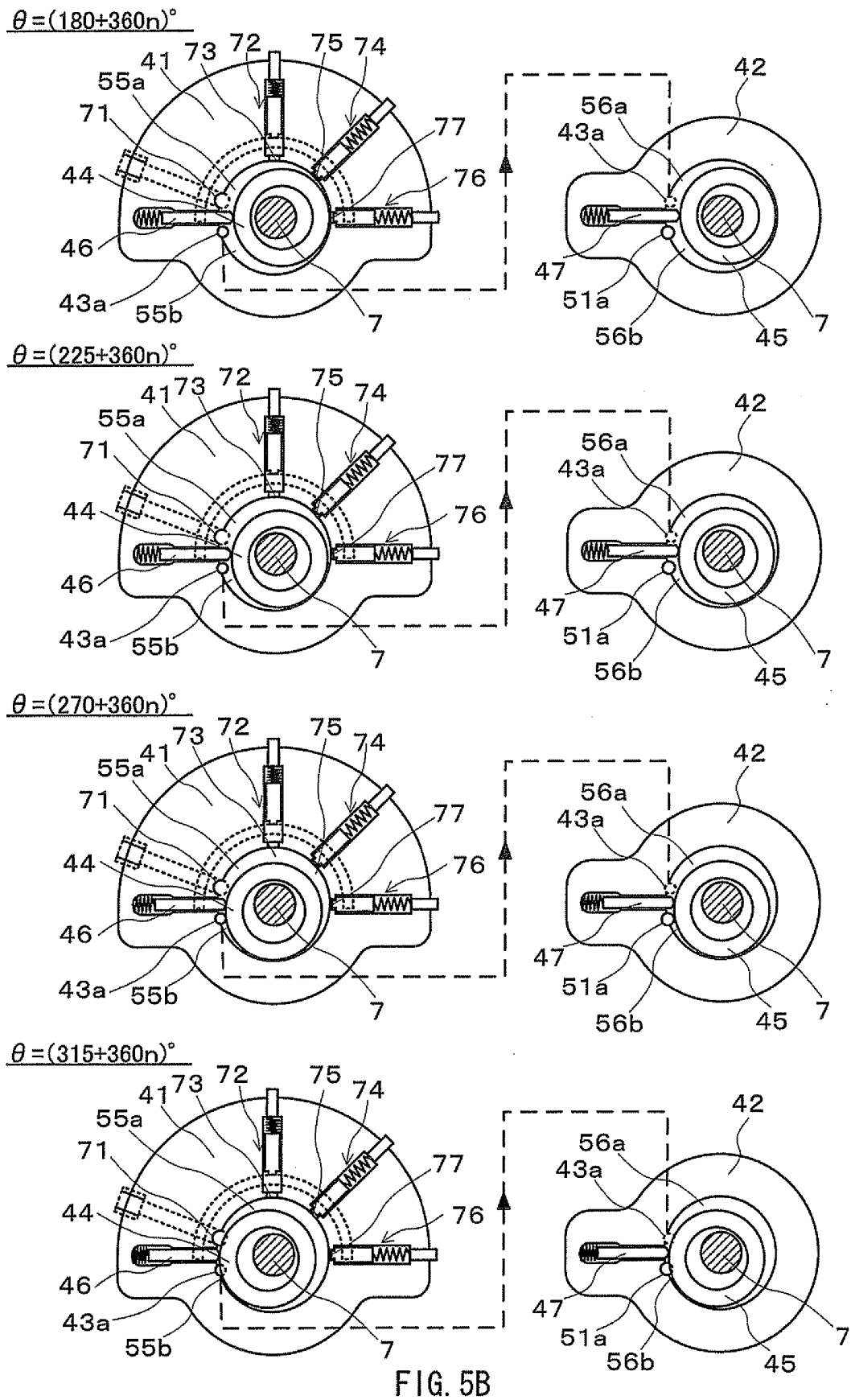


FIG. 5A



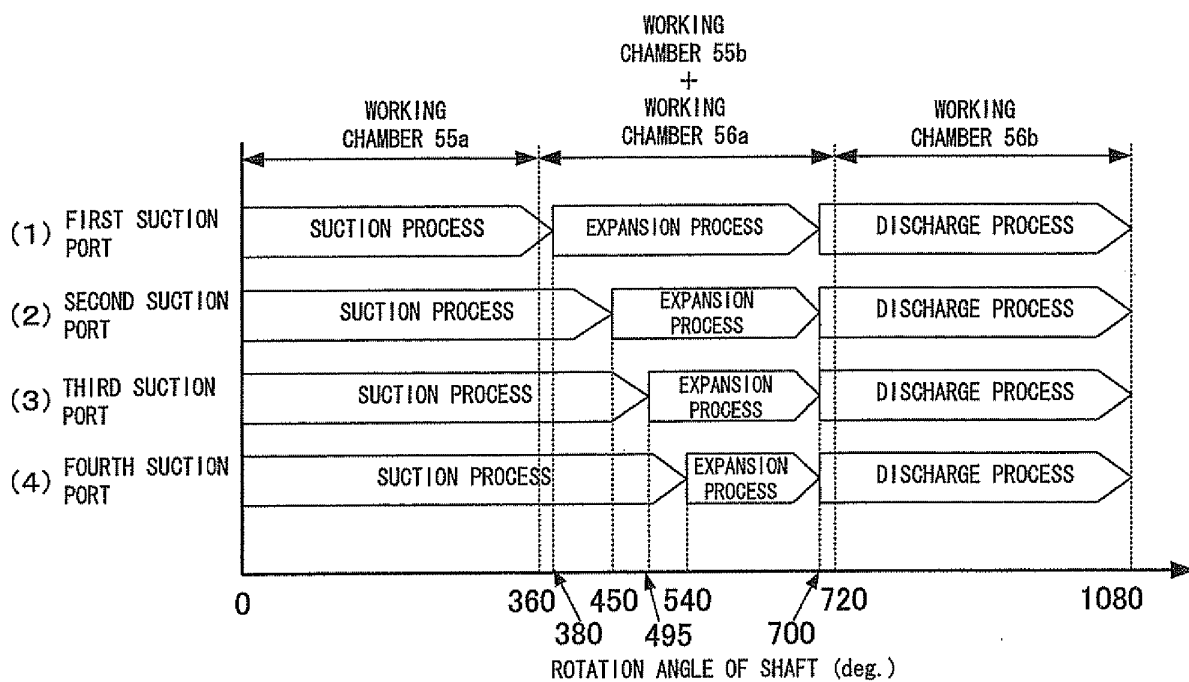


FIG. 6A

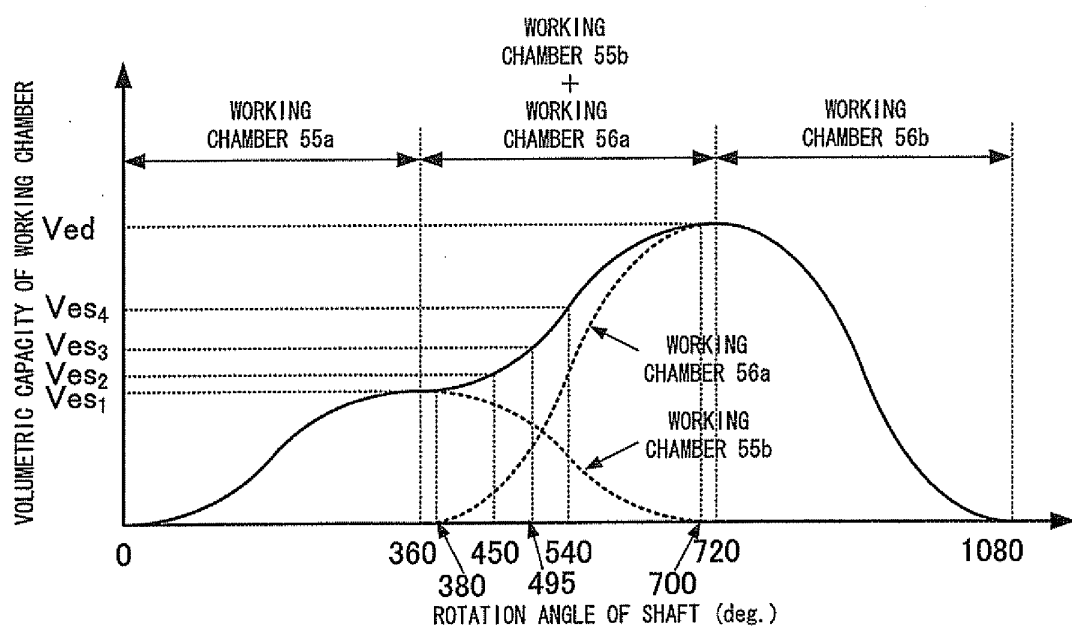


FIG. 6B

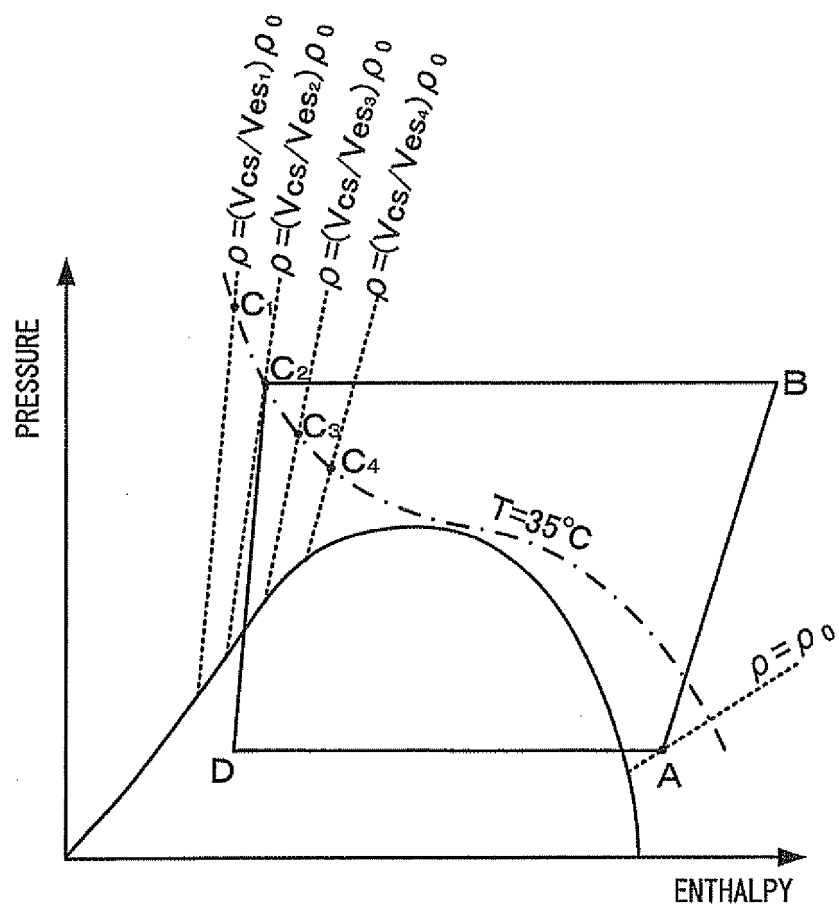


FIG. 7

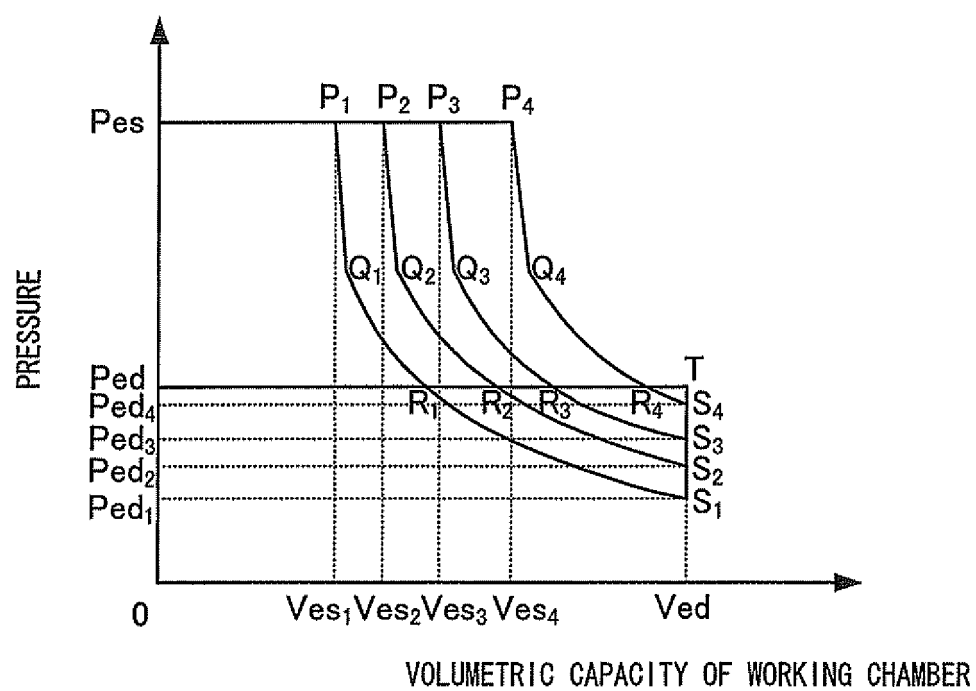


FIG. 8

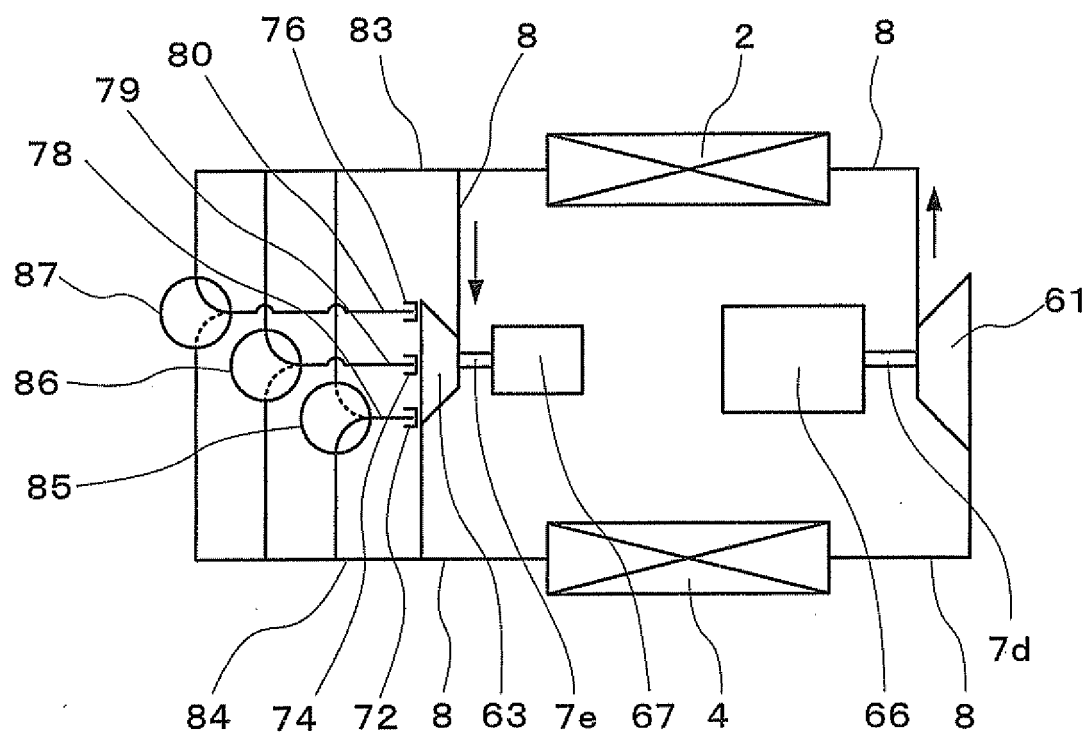


FIG. 9

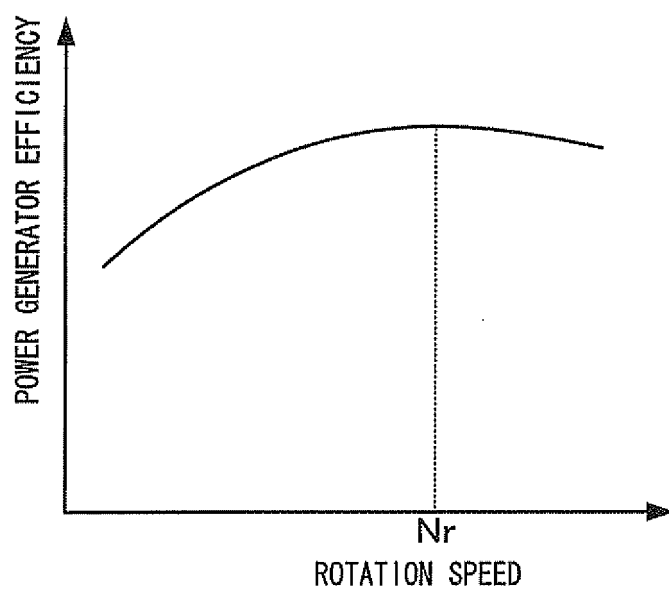


FIG. 10

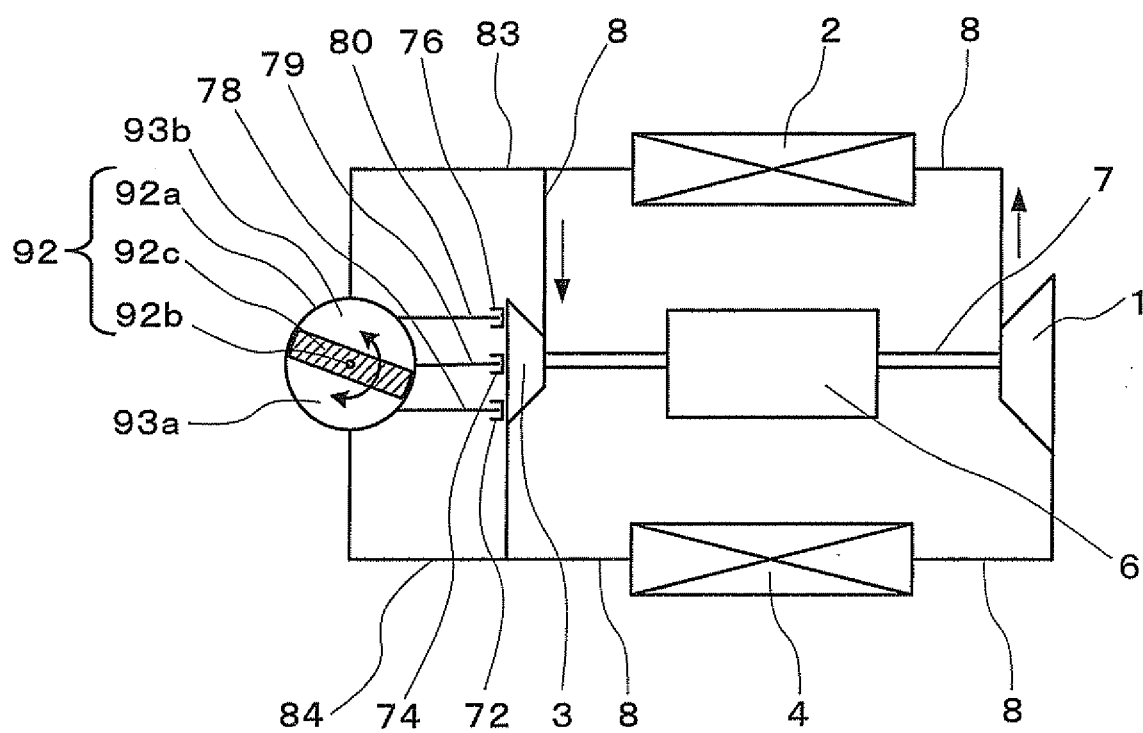


FIG. 11

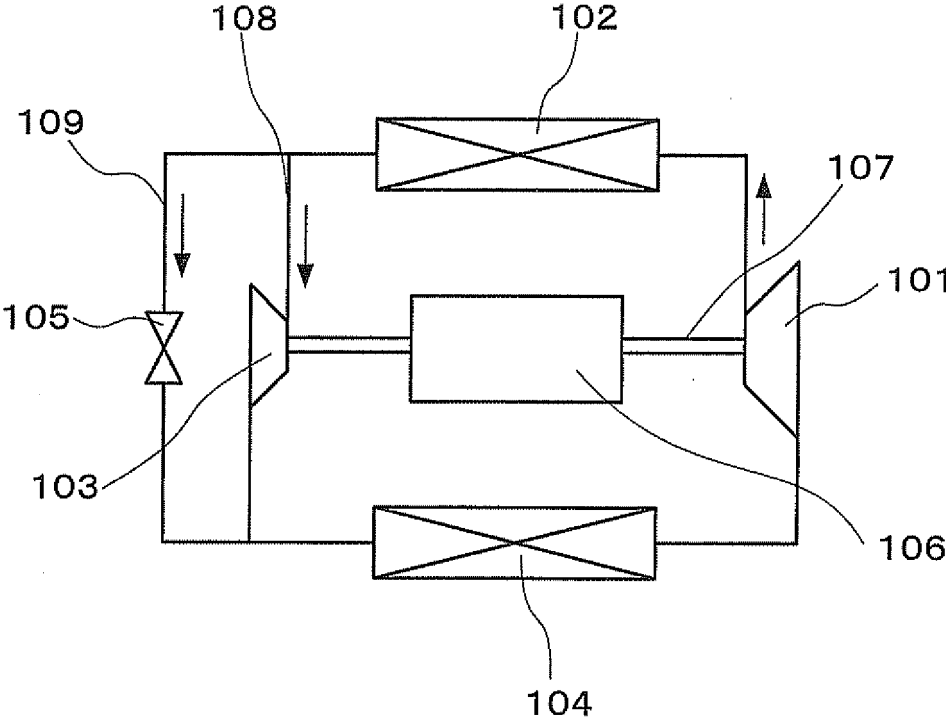


FIG. 12

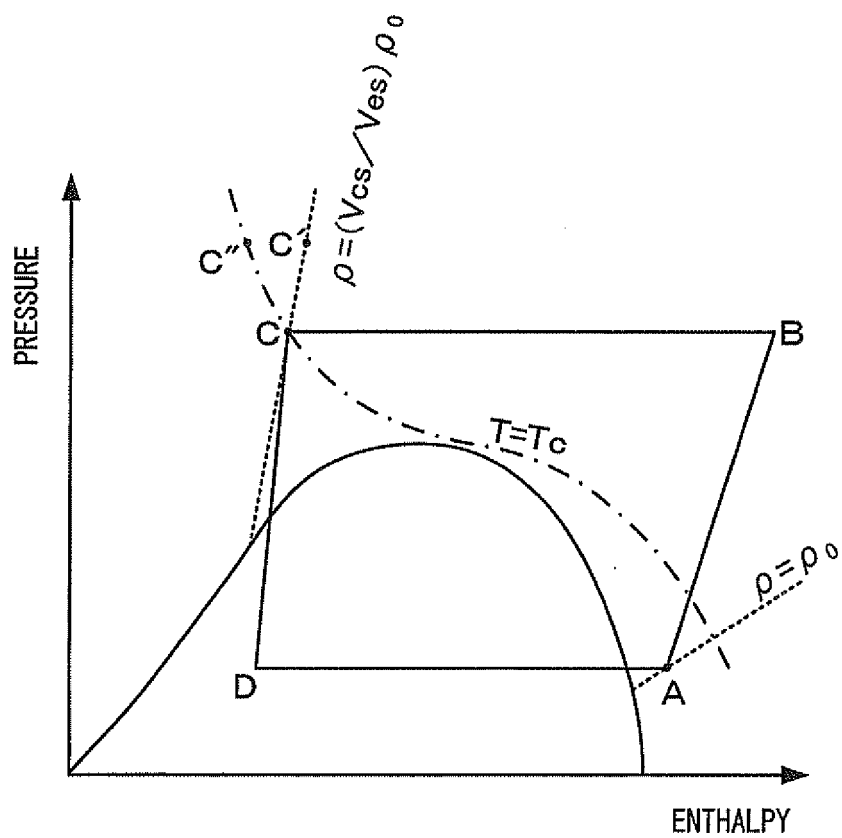


FIG. 13

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2006/321245

A. CLASSIFICATION OF SUBJECT MATTER

F01C13/04(2006.01)i, F01C21/18(2006.01)i, F25B1/00(2006.01)i, F25B11/02(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)

F01C13/04, F01C21/18, F25B1/00, F25B11/02

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	1996-2006
Kokai Jitsuyo Shinan Koho	1971-2006	Toroku Jitsuyo Shinan Koho	1994-2006

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y A	JP 2004-197640 A (Daikin Industries, Ltd.), 15 July, 2004 (15.07.04), Claim 4; Par. Nos. [0025] to [0027], [0096] to [0100]; Figs. 17, 18 & WO 2004/055331 A1	1, 4, 6, 7, 10-13, 15 2, 3, 5, 8, 9, 14
Y A	JP 2005-256667 A (Daikin Industries, Ltd.), 22 September, 2005 (22.09.05), Claim 1; Figs. 2, 12, 13 & WO 2005/088077 A1	1, 4, 6, 7, 10-13, 15 2, 3, 5, 8, 9, 14 1

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

☐ See patent family annex.

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"O" document referring to an oral disclosure, use, exhibition or other means

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

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

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

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"&" document member of the same patent family

Date of the actual completion of the international search
22 November, 2006 (22.11.06)

Date of mailing of the international search report
28 November, 2006 (28.11.06)

Name and mailing address of the ISA/
Japanese Patent Office

Authorized officer

Facsimile No.

Telephone No.

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2006/321245

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 5126/1985 (Laid-open No. 122301/1986) (Nissan Motor Co., Ltd.), 01 August, 1986 (01.08.86), Full text; all drawings (Family: none)	10
A	JP 2004-190559 A (Daikin Industries, Ltd.), 08 July, 2004 (08.07.04), Full text; all drawings & WO 2004/053298 A1 & US 2006/165542 A1	1
A	JP 58-133401 A (Matsushita Electric Industrial Co., Ltd.), 09 August, 1983 (09.08.83), Full text; all drawings (Family: none)	1
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 92218/1986 (Laid-open No. 203901/1987) (Kobe Steel, Ltd.), 26 December, 1987 (26.12.87), Full text; all drawings (Family: none)	1

Form PCT/ISA/210 (continuation of second sheet) (April 2005)

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2001116371 A [0002] [0009]
- JP 2002081766 A [0008]