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

(57) An expander of the invention includes: n-number of rotary type fluid mechanisms (where n is an integer equal to or greater than 2), a first suction port (41b) for sucking a working fluid into a suction-side space (55a) of a first fluid mechanism (41), a communication port (43a) connecting a discharge-side space (55b) of a k-th fluid mechanism (where k is an integer from 1 to n-

1) and a (k + 1)-th suction-side space (56a) to form a single space, and a discharge port (51a) for discharging the working fluid from the discharge-side space of an n-th fluid mechanism. The expander further includes a second suction port (72f) being capable of changing its connecting position to the suction-side space (55a) of the first fluid mechanism (41), for sucking the working fluid into the suction-side space (55a).

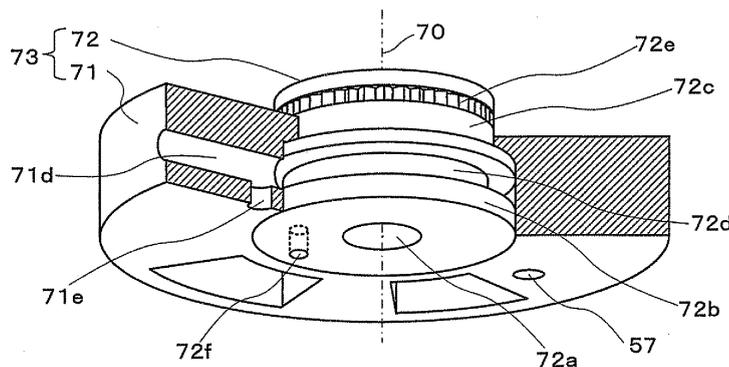


Fig. 3C

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## Description

### TECHNICAL FIELD

**[0001]** The present invention relates to an expander applied to a refrigeration cycle apparatus (heat pump), 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 2a01-116371A).

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

**[0004]** Fig. 20 shows a refrigeration cycle using the conventional expander-compressor unit. In this refrigeration cycle, a main circuit 8 for a working fluid (refrigerant) is constituted by a compressor 1, a gas cooler (radiator) 2, an expander 3, and an evaporator 4. The compressor 1, the expander 3, and a rotation motor 6 are coupled to each other by a shaft 7 to form an expander-compressor unit. The refrigerant circuit is provided with a sub-circuit 9 in addition to the main circuit 8. The sub-circuit 9 branches from the main circuit 8 at the outlet side of the gas cooler 2 and merges with the main circuit 8 at the inlet side of the evaporator 4. The working fluid that passes through the main circuit 8 is expanded at the expander 3, and the working fluid that passes through the sub-circuit 9 is expanded by an expansion valve 5.

**[0005]** The working fluid is compressed in the compressor 1 to convert from a low temperature, low pressure state to a high temperature, high pressure state, and thereafter is cooled in the gas cooler 2 to convert to a low temperature, high pressure state. Then, the working fluid is expanded in the expander 3 or the expansion valve 5 to a low temperature, low pressure state (gas-liquid two phase) and is heated at the evaporator 4 to return to a low temperature, low pressure state (vapor phase). The expander 3 recovers the energy of expansion of the working fluid and converts it into rotation energy for the shaft 7. This rotation energy is utilized as a part of the work for driving the compressor 1. As a result, the power driving the rotation motor 6 can be reduced.

**[0006]** Here, the operation of the refrigeration cycle when the expansion valve 5 is fully closed and the mass flow rate of the working fluid in the sub-circuit 9 is made zero will be described below.

**[0007]** The volume flow rate of the working fluid on the inlet side of the compressor 1 and that of the expander 3 are represented as  $(V_{cs} \times N)$  and  $(V_{es} \times N)$ , respec-

tively, wherein the suction volume of the compressor 1 is denoted as  $V_{cs}$ , the suction volume of the expander 3 is denoted as  $V_{es}$ , and the rotation speed of the shaft 7 is denoted as  $N$ . Since the mass flow rate of the working fluid in the sub-circuit 9 is zero, the mass flow rate in the compressor 1 and the mass flow rate in the expander 3 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 1 and the density of the working fluid on the inlet side of the expander 3 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 3 to the density of the working fluid on the inlet side of the compressor 1 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. 21 shows a Mollier diagram of the refrigeration cycle. In the diagram, the compression process in the compressor 1 corresponds to the line AB, the heat radiation process in the gas cooler 2 corresponds to the line BC, the expansion process in the expander 3 corresponds to the line CD, and the evaporation process in the evaporator 4 corresponds to the line DA. The density ratio of the working fluid at point A on the inlet side of the compressor 1 and that at point C on the inlet side of the expander 3 is constant,  $(V_{es}/V_{cs})$ , so the density  $\rho_c$  at point C can be represented as  $(V_{cs}/V_{es})\rho_0$ , where the density of the working fluid at point A is  $\rho_0$ . 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 1 and the density on the inlet side of the expander 3 is due to the fact that the mass flow rate in the compressor 1 and that in the expander 3 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 9 by opening the expansion valve 5 (see JP 2001-116371 A).

**[0010]** In order to avoid the constraint of the constant density ratio in the power recovery-type heat pump employing the conventional expander-compressor unit, which results from the fact that the compressor and the expander rotate at the same rotation speed, it is neces-

sary 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 an expander including:

n-number of rotary type expansion mechanisms (where "n" is an integer equal to or greater than 2) each having a cylinder, a shaft with an eccentric portion, a piston fitted to the eccentric portion and rotating eccentrically in the cylinder, and a partition member partitioning a space between the cylinder and the piston into a suction-side space and a discharge-side space;

a first suction port for sucking a working fluid into the suction-side space of the first expansion mechanism;

a communication port connecting the discharge-side space of the k-th expansion mechanism (where "k" is an integer from 1 to n-1) and the suction-side space of the (k + 1)-th expansion mechanism to form a single space;

a discharge port for discharging the working fluid from the discharge-side space of the n-th expansion mechanism; and

a second suction port for sucking the working fluid into the suction-side space of the first expansion mechanism, the second suction port being capable of changing its connecting position to the suction-

side space of the first expansion mechanism.

**[0014]** The present invention also provides an expander-compressor unit including: an expander section having an expander according to the present invention; and a compressor section integrally coupled to the expander section by the shaft.

**[0015]** The present invention also provides a heat pump including the expander according to the present invention or the expander-integrated fluid machine.

**[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 changing the connection position at which the suction-side space of the first expansion mechanism and 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 for example, it is possible to avoid the constraint of constant density ratio in a refrigeration cycle employing an expander-compressor unit. Therefore, the energy of expansion of the working fluid can be recovered efficiently by allowing the entire working fluid to flow into the expander without providing a sub-circuit for the working fluid.

**[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 to maintain a high power generation efficiency by 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. 3A is a perspective partial cut-away view of a stationary portion of an upper end plate of the expander section of the expander-compressor unit shown in Fig. 1.

Fig. 3B is a perspective view of a movable portion of the upper end plate of the expander section of the expander-compressor unit shown in Fig. 1.

Fig. 3C is a perspective partial cut-away view illus-

trating the upper end plate in which the stationary portion and the movable portion have been integrated.

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

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

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

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

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

Fig. 6A is a chart illustrating the relationship between the rotation angle of the shaft and the process in 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. 9A is a configuration diagram illustrating a heat pump employing an expander-compressor unit.

Fig. 9B is a configuration diagram illustrating a heat pump employing a separate-type expander.

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 vertical cross-sectional view illustrating an expander according to a second embodiment of the present invention.

Fig. 12A is a cross-sectional view of the expander shown in Fig. 11, taken along line D3-D3 of Fig. 11.

Fig. 12B is a cross-sectional view of the expander shown in Fig. 11, taken along line D4-D4 of Fig. 11.

Fig. 13 is a configuration diagram illustrating a heat pump furnished with a pressure regulator and the expander shown in Fig. 11.

Fig. 14 is a horizontal cross-sectional view illustrating a modified example of an actuator.

Fig. 15 is a configuration diagram illustrating a mod-

ified example of the pressure regulator.

Fig. 16A is a configuration diagram illustrating another modified example of the pressure regulator.

Fig. 16B is a block diagram illustrating an example in which a pressure sensor is provided for the pressure regulator shown in Fig. 16A.

Fig. 17 is a vertical cross-sectional view illustrating an expander-compressor unit according to a third embodiment of the present invention.

Fig. 18 is a plan view illustrating a rotary actuator.

Fig. 19 is a cross-sectional view of the rotary actuator of Fig. 18, taken along line D5-D5 of Fig. 18.

Fig. 20 is a configuration diagram illustrating a heat pump employing a conventional expander-compressor unit.

Fig. 21 is a Mollier diagram illustrating the refrigeration cycle using the heat pump employing the conventional expander-compressor unit.

## 20 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. Fig. 3A is a perspective partial cut-away view of a stationary portion of an upper end plate of the expander section. Fig. 3B is a perspective view of a movable portion of the upper end plate. Fig. 3C is a perspective partial cut-away view illustrating the upper end plate in which the stationary portion and the movable portion have been integrated.

[0021] An expander-compressor unit 100 according to the present embodiment includes a closed casing 11, a scroll type compressor section 20 disposed in an upper portion of the closed casing, a two-stage rotary expander section 40 disposed in a lower portion of the closed casing, a rotation motor 12 disposed between the compressor section 20 and the expander section 40 and having a rotor 12a and a stator 12b, and a shaft 13 for coupling the compressor section 20, the expander section 40, and the rotation motor 12 to one another. The shaft 13 may be formed of a plurality of portions joined uniaxially

[0022] The scroll type compressor section 20 has a stationary scroll 21, an orbiting scroll 22, an Oldham ring 23, a bearing member 24, a muffler (silencer) 25, a suction pipe 26, and a discharge pipe 27. The orbiting scroll 22 is fitted to an eccentric shaft 13a of the shaft 13 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 13. 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 40 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 40 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 13b of the shaft 13 to perform eccentric rotational motion in the first cylinder 41. The first vane 46 is retained reciprocally in a vane groove 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 13c of the shaft 13 to perform eccentric rotational motion in the second cylinder 42. The second vane 47 is retained reciprocally in a vane groove 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]** A first (first stage) expansion mechanism is constituted by the first cylinder 41, the shaft 13, the first piston 44, and the first vane 46. Likewise, a second (second stage) expansion mechanism is constituted by the second cylinder 42, the shaft 13, the second piston 45 and the second vane 47. The pistons 44, 45 and the vanes 46, 47 respectively may be integrated with each other (what are called swing pistons).

**[0025]** The expander section 40 further includes an upper end plate 73 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 73 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 73, 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 73 and the lower end plate 51 serve as closing members for closing the cylinders 41 and 42 and also function as bearing members for retaining the shaft 13 rotatably, together with the bearing member 24 of the compressor section 20. Like the compressor section 20, the expander section 40 is furnished with a muffler 52, a suction pipe 53, and a discharge pipe 54.

**[0026]** 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 13.

**[0027]** As illustrated in Fig. 1, the upper end plate 73 has a stationary portion 71 and a movable portion 72. As illustrated in Fig. 3A, the stationary portion 71 has a through hole 71f for fitting the movable portion 72 therein. The through hole 71f is surrounded by a cylindrical recessed surface 71a, a cylindrical recessed surface 71b having a central axis 70 common to the cylindrical recessed surface 71a and a smaller inner diameter than the cylindrical recessed surface 71a, and a stepped surface 71c connecting these cylindrical recessed surfaces 71a and 71b. It should be noted that when the fluid machine (i.e., the expander-compressor unit 100) is assembled, the central axis 70 aligns the central axis of the shaft 13.

**[0028]** In the stationary portion 71, an inlet passage 71d (first inlet passage) and an inlet passage 71e (second inlet passage), which is a branch passage from the inlet passage 71d, are provided as the inlet passages for guiding the working fluid from the suction pipe 53 to the working chamber 55a. As illustrated in Figs. 1 and 2A, an inlet passage 41a and a first suction port 41b are provided in

the first cylinder 41 as the flow passages in communication with the inlet passage 71e. The first suction port 41b is in communication with the suction-side working chamber 55a in the first cylinder 41.

**[0029]** As illustrated in Fig. 3B, the movable portion 72 of the upper end plate 73 has a through hole 72a for retaining the shaft 13 rotatably, and includes as its outer circumferential surfaces a cylindrical protruding surface 72b abutting on the cylindrical recessed surface 71a of the stationary portion 71, a cylindrical protruding surface 72c abutting on the cylindrical recessed surface 71b of the stationary portion 71, and a stepped surface 72g abutting on the stepped surface 71c of the stationary portion 71 and between the cylindrical protruding surfaces 72b and 72c. A gear 72e formed circumferentially around the cylindrical protruding surface 72c is provided in the cylindrical protruding surface 72c of the movable portion 72 of the upper end plate 73. The movable portion 72 further includes a flow passage groove 72d provided circumferentially in the cylindrical protruding surface 72b and a second suction port 72f connected to the flow passage groove 72d. As illustrated in Figs. 1 and 2A, the second suction port 72f extends from the flow passage groove 72d toward the working chamber 55a of the first cylinder 41 along the axis direction and is in communication with the suction-side working chamber 55a in the first cylinder 41.

**[0030]** As illustrated in Fig. 3C, the stationary portion 71 and the movable portion 72 are integrated with each other by fitting the movable portion 72 into the through hole 71f of the stationary portion 71 rotatably. The stepped surface 71c of the stationary portion 71 and the stepped surface 72g of the movable portion 72 abut onto each other so as to prevent the movable portion 72 from escaping above the stationary portion 71. The lower end face of the stationary portion 71 and the lower end face of the movable portion 72 together constitute the same plane, and this plane constitutes the upper wall of the first cylinder 41.

**[0031]** When the movable portion 72 is rotated, the second suction port 72f rotates with the central axis 70 as the center of rotation while keeping the distance from the central axis 70 of the shaft 13 constant. The rotation of the movable portion 72 brings about a relative change in the position of the second suction port 72 in the suction-side working chamber 55a of the first cylinder 41. That is, while the connection position of the first suction port 41b and the suction-side working chamber 55a of the first cylinder 41 is fixed, the connection position of the second suction port 72f and the working chamber 55a is variable. As will be described later, the change in the connection position of the second suction port 72f makes it possible to avoid the constraint of constant density ratio in the expander-compressor unit.

**[0032]** As described with reference to Figs. 3A, 3B, and 3C, it is recommended that the second suction port 72f be provided in the end plate 73 serving as a closing member for closing an end face of the first cylinder 41,

contained in the first fluid mechanism into which the working fluid flows initially. The reason is that the second suction port 72f that is movable can be constructed with a simple configuration. In addition, since the cylinder 41 side of the upper end plate 73 is a flat surface, it is easy to enhance the processing accuracy even when the end plate 73 is constructed of a plurality of components.

**[0033]** Furthermore, as has been explained above, it is preferable that at least a portion of the end plate 73 be made into the movable portion 72 that can rotate around the shaft 13 being the center of rotation, and that the second suction port 72f be provided in the movable portion 72. The reason is that a large region can be ensured for the movement of the second suction port 72f.

**[0034]** In addition, in the present embodiment, the movable portion 72 includes the cylindrical bearing surface (the inner circumferential surface of the through hole 72a) that supports the shaft 13. Therefore, it is unnecessary to provide a bearing for supporting the shaft 13 separately, and thereby, it is possible to prevent an increase of the parts count.

**[0035]** The stationary portion 71 has an annular configuration and includes therein the inlet passage 71d (first inlet passage) for supplying the working fluid from outside of the expander section 40 to the second suction port 72f provided in the movable portion 72, and the inlet passage 71e (second inlet passage) branched from the inlet passage 71d, for supplying the working fluid to the first suction port 41b. To such a stationary portion 71, the movable portion 72 is united rotatably. By providing the two inlet passages 71d and 71e inside the stationary portion 71, the pipes for guiding the working fluid to the second suction port 72f become unnecessary, which is advantageous in terms of saving the available space in the closed casing 1.1. Moreover, since the inlet passages 71d and 71e are provided inside the stationary portion 71, the problem of working fluid leakage does not arise easily.

**[0036]** Referring back to Fig. 1, the description will proceed further. The stationary portion 71 of the upper end plate 73 further includes a gear 75 meshing with the gear 72e of the rotating portion 72, and a rotation motor 76 (electric actuator) for driving the gear 75. The movable portion 72 is driven by the rotation motor 76 via the gears 72e and 75. In this way, the expander section 40 further may include drive mechanisms 75 and 76 for rotating the movable portion 72. The drive mechanisms 75 and 76 are connected to a controller (not shown) provided outside the closed casing 11, for controlling the rotation angle of the movable portion 72. The drive mechanisms 75 and 76 receive control signal from the controller and rotate the movable portion 72 to control the connection position to the working chamber 55a. When using a stepping motor or a servomotor as the rotation motor 76, the position of the second suction port 72f can be controlled with high accuracy. It is also possible to provide a detector (e.g., an encoder) for detecting the rotation angle of the movable portion 72. It is also possible to employ a means

other than the rotation motor 76, such as an actuator that makes use of the pressure difference of fluid, as the driving means for the movable portion 72.

**[0037]** The working fluid that has flowed from the suction pipe 53 into the expander section 40 branches into two paths from the inlet passage 71d of the stationary portion 71 of the upper end plate 73 and flows into the working chamber 55a. The first path is a path that follows the inlet passage 71d and the branch inlet passage 71e in the stationary portion 71, and the inlet passage 41a and the first suction port 41b in the first cylinder 41. The second path is a path that follows the inlet passage 71d in the stationary portion 71, the flow passage groove 72d and the second suction port 72f in the movable portion 72. Thus, in the expander section 40, the working fluid is supplied from the suction pipe 53 to the working chamber 55a through the first suction port 41b, whose connection position to the working chamber 55a is fixed, and the second suction port 72f, whose connection position to the working chamber 55a is variable. In these two paths, it is unnecessary to provide a flow rate control mechanism that can be opened and closed such as a solenoid valve or a differential pressure valve.

**[0038]** The working fluid that has been sucked into the first cylinder 41 runs through the second cylinder 42, then passes through the discharge port 51a provided in the lower end plate 51, an internal space 52a of the muffler 52, a flow passage 57 extending through the first and second cylinders 41 and 42 in that order, and then is discharged from the discharge pipe 54 to the refrigeration cycle. It should be noted that the discharge port 51a may be provided in the second cylinder 42.

**[0039]** As illustrated in Fig. 2B, a discharge valve 74 is installed at the discharge port 51a provided in the lower end plate 51. The discharge valve 74 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 74 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 74 has the function of preventing overexpansion of the working fluid in the expander section 40.

**[0040]** Figs. 4A, 4B, and 4C illustrate the positions of the first suction port 41b and the second suction port 72f. The position of the second suction port 72f is regulated at 20° (Fig. 4A), 90° (Fig. 4B), and 180° (Fig. 4C), represented by angle  $\varphi$  with respect to the position of the first vane 46 with the shaft 13 being the center. The angle  $\varphi$  is, more precisely, an angle formed by a first linear line 80 connecting the central axis 70 of the shaft 13 with the contact point between the first vane 46 and the first piston 44, and a second linear line 90 connecting the second suction port 72f and the central axis 70 of the shaft 13, when the first linear line 80 is rotated in the direction of rotation (the clockwise direction in the example shown in

the figure) to the second linear line 90, taking the central axis 70 of the shaft 13 as the center of rotation. According to this manner of representation, the first suction port 41b is fixed at the 20° position in the example shown in the figures. The discharge port 51a is fixed at the 340° position in the second cylinder 42, according to the same manner of representation. In contrast, the position of the second suction port 72f can be set freely from 0° to 360°.

**[0041]** Fig. 5A shows a view illustrating the operating principle of the first cylinder 41 when the angle  $\varphi$  of the second suction port 72f is 90°, and Fig. 5B shows a view illustrating the operating principle of the second cylinder 42, which corresponds to the foregoing. Here, the rotation angle  $\theta$  of the shaft 13 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, that is, what is called at the top dead center, and the direction of rotation of the shaft 13, i.e., the clockwise direction is represented as the forward direction.

**[0042]** After  $\theta = 20^\circ$ , the working fluid flows from the first suction port 41b into the working chamber 55a, which is generated after  $\theta = 0^\circ$ . After  $\theta = 90^\circ$ , the working fluid flows through the first suction port 41b and the second suction port 72f into the working chamber 55a. After  $\theta = 360^\circ$ , the working chamber 55a is turned into the working chamber 55b and is brought into communication with the working chamber 56a of the second cylinder 42 through the communication port 43a. As the shaft 13 rotates further, the contact point between the first cylinder 41 and the first piston 44 passes the first suction port 41b when  $\theta = 380^\circ$  (not shown), breaking the communication between the working chamber 55b and the first suction port 41b. In the conventional two-stage rotary expander section, the suction process for the working fluid finishes at this point.

**[0043]** In contrast, because the second suction port 72f is provided in the expander section 40 of the present embodiment, the inflow of the working fluid from the second suction port 72f continues even after the angle  $\theta$  reaches  $\theta = 380^\circ$ . In this expander section 40, the suction process for the working fluid finishes when the angle  $\theta$  reaches  $\theta = 450^\circ$ , at the point where the contact point between the first cylinder 41 and the first piston 44 passes the second suction port 72f and where the communication between the working chamber 55b and the second suction port 72f is broken.

**[0044]** When the suction process is completed, an expansion process for the working fluid is started. As the shaft 13 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, as the shaft 13 rotates, 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  $\theta$  reaches  $\theta = 700^\circ$  (not shown), the contact point between the second cylinder

42 and the second piston 45 passes the discharge port 51a, bringing the working chamber 56a into communication with the discharge port 51a. The expansion process finishes at this point.

**[0045]** When the expansion process is completed, a discharge process for the working fluid is started. When  $\theta = 720^\circ$ , the working chamber 55b disappears, and the working chamber 56a changes into the working chamber 56b. As the shaft 13 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.

**[0046]** Fig. 6A illustrates the relationship between the rotation angles  $\theta$  of the shaft 13 and the shifting points of the processes from the suction to the discharge in the cases that the angle  $\varphi$  of the second suction port 72f is  $20^\circ$ ,  $90^\circ$ , and  $180^\circ$ . As is clear from the foregoing description, the rotation angle  $\theta$  of the shaft 13 at which the suction process finishes is an angle at which the contact point between the first cylinder 41 and the first piston 44 passes the second suction port 72f for the second time. This angle can be represented as  $\theta = (360 + \varphi)$ . Accordingly, as the angle  $\varphi$  of the second suction port 72f increases, the timing for shifting from the suction process to the expansion process delays, 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 becomes smaller.

**[0047]** Fig. 6B illustrates the relationship between the rotation angle  $\theta$  of the shaft 13 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. Shown on the vertical axis of the graph are the suction volume  $V_{es\varphi}$ , which represents the volumetric capacity of the working chamber at the end of the suction process, for each of the cases that the angle  $\varphi$  of the second suction port 72f is  $20^\circ$ ,  $90^\circ$ , and  $180^\circ$ , and the discharge volume  $V_{ed}$ , which represents the volumetric capacity of the working chamber at the start of the discharge process. As the angle  $\varphi$  increases, the suction volume  $V_{es\varphi}$  accordingly increases, but the discharge volume  $V_{ed}$  is constant irrespective of the angle  $\varphi$ .

**[0048]** As described above, in the present embodiment, the suction volume  $V_{es\varphi}$ , the volumetric capacity of the working chamber 55a, 55b, 56a and 56b at the end of the suction process, is made variable by providing the second suction port 72f that is movable, in addition to the first suction port 41b provided in the conventional two-stage rotary expander section 40. This makes it possible to control the density ratio ( $V_{cs}/V_{es\varphi}$ ) of the working fluid on the inlet side of the compressor section 20 and that of the expander section 40.

**[0049]** 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 varied, point C, which corresponds to the state on the inlet side of the two-stage rotary expander section 40, can be shifted to point C' or point C'' 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 40 can be controlled freely. As a result, it becomes possible to operate the refrigeration cycle with a high efficiency that has been impossible with the refrigeration cycle employing the conventional expander-compressor unit.

**[0050]** Particularly when the second suction port 72f is formed in the movable portion 72 that is rotatable with the axis of the shaft 13 being the center of rotation and the angle  $\varphi$  is made adjustable from  $0^\circ$  to  $360^\circ$ , as in the present embodiment, the range of the controlling becomes wide and therefore a highly efficient refrigeration cycle can be achieved easily.

**[0051]** Next, the effect obtained by providing the discharge valve 74 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 subscripts  $\varphi$  for the symbols in the diagram denote angles  $\varphi$  of the second suction port 72f. Point  $P_\varphi$  denotes the start of an expansion process, point  $S_\varphi$  denotes the end of an expansion process, and point T denotes the start of a discharge process. It should be noted that an inflection point  $Q_\varphi$  originating from a phase change is shown in the middle of each expansion process since the refrigeration cycle using carbon dioxide as the working fluid is assumed.

**[0052]** As the suction volume  $V_{es\varphi}$  becomes greater in association with movement of the second suction port 72f, the volumetric capacity ratio ( $= V_{ed}/V_{es\varphi}$ ) before and after the expansion process becomes smaller and the pressure  $S_\varphi$  at the end of the expansion process becomes higher, because the discharge volume  $V_{ed}$  is constant. For this reason, when, for example, the angle  $\varphi$  of the second suction port 72f is controlled in the range of from  $20^\circ$  to  $180^\circ$ , it is desirable that the expander section 40 be designed in such a manner that the pressure  $S_{180}$ , which is the pressure at the end of the expansion process in the case of selecting the maximum angle  $180^\circ$ , is lower than the low pressure  $P_{ed}$  of the refrigeration cycle to prevent underexpansion. The reason is that if underexpansion occurs, part of the energy of the working fluid originating from the pressure difference cannot be recovered.

**[0053]** In such a design, overexpansion occurs at least in the case that the angle  $\varphi$  is set at  $180^\circ$  or less. The overexpansion refers to a phenomenon in which the pressure  $P_{ed\varphi}$  becomes lower than the low pressure  $P_{ed}$  of 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  $R\phi S\phi T$  in Fig. 8.

**[0054]** When the discharge valve 74 is provided for the discharge port 51a, however, recompression is carried out in the discharge process when overexpansion  $R\phi S\phi$  occurs in the working chamber 56b. In the discharge process, the volumetric capacity of the working chamber 56b reduces as the shaft 13 rotates. When the discharge valve 74 is provided for the discharge port 51a, the discharge valve 74 does not open until the pressure of the working chamber 56b that has been reduced by overexpansion becomes equal to the low pressure  $P_{ed}$  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 74.

**[0055]** Hereinbelow, other features of the expander-compressor unit according to the present embodiment will be described.

**[0056]** In the present embodiment, it is recommended that the movable portion 72 of the upper end plate 73 provided with the second suction port 72f should be rotatable in the same direction as the direction of rotation of the shaft 13, taking the shaft 13 as its center of rotation. The reason is that the friction force between the shaft 13 and the movable portion 72 enables the movable portion 72 to be rotated with a smaller mechanical power. Thereby, the size of the rotation motor 76 may be reduced so that it can be accommodated easily in the closed casing 11.

**[0057]** In the present embodiment, the movable portion 72 returns to the original position when it rotates  $360^\circ$ . The movable portion 72 needs to be driven to rotate only in the same direction, so the controlling of the rotation motor 76 is easy. Moreover, the friction force between the shaft 13 and the movable portion 72 does not hinder the rotation driving.

**[0058]** In the present embodiment, the suction volume  $V_{es\phi}$  of the expander section 40 is made variable, and thereby the compressor section 20 is allowed to have a common structure used for the refrigeration cycle that does not employ an expander. The common structure may be used without alteration for the compressor section 20 and therefore the development costs can be reduced.

**[0059]** When using the expander-compressor unit of the present embodiment, the suction volume  $V_{es\phi}$  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 20 and while the expander section 40 is being rotated at the same rotation speed as that of the compressor section 20. Therefore, it is possible that the compressor section 20 and the expander section 40 serve different roles in controlling the refrigeration cycle, and also, the control algorithm for the refrigeration

cycle using the expander-compressor unit becomes simple.

**[0060]** Although there is no particular limitation to 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.

**[0061]** In the present embodiment, the second suction port 72f that is movable is provided along with the first suction port 41b. However, two or more movable suction ports may be provided, in which case the suction volume  $V_{es\phi}$  is determined by the suction port disposed at the most downstream position. Although the expander section 40 has two stages in the present embodiment, the same advantageous effects as described above may be obtained by providing a second suction port that is movable for the first stage cylinder even in the case where the expander section has three or more stages.

**[0062]** Next, Fig. 9A illustrates the configuration of a power recovery type heat pump employing the expander-compressor unit according to the present embodiment.

The heat pump shown in Fig. 9A includes the expander-compressor unit 100, a gas cooler (radiator) 2, an evaporator 4, and piping 88 (refrigerant pipes) connecting these components to one another. In the conventional example shown in Fig. 20, the sub-circuit 9 connected parallel to the expander 3 is indispensable, but the heat pump employing the expander-compressor unit according to the present embodiment does not require such a sub-circuit as an essential item. That said, a sub-circuit may be provided for different purposes, for example, for the purpose of carrying out the start-up and stopping of the heat pump stably.

**[0063]** Furthermore, the expander section 40 according to the present embodiment may be used alone; in other words, it may be used as an expander separate from a compressor. Fig. 9B illustrates the configuration of a power recovery type heat pump employing a separate-type expander. This system includes a compressor 81, a gas cooler (radiator) 82, an expander 83, and an evaporator 84. It further includes piping 88 (refrigerant pipes) that connects the compressor 81, the gas cooler 82, the expander 83, and the evaporator 84 in that order and in which the working fluid circulates. The expander 83 contains the expander section 40, described with reference to Fig. 1 and so forth. In this heat pump, the energy of expansion of the working fluid, which is obtained by the expander 83, is converted by a power generator 86 into electric energy, which is used as part of the input to a rotation motor 85 for rotating the compressor 81.

**[0064]** Fig. 10 shows an efficiency curve for a common power generator 86. Since the power generator 86 is designed so that the power generation efficiency becomes highest at a predetermined rated rotation speed  $N_r$ , 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 86 be as close as possible to the rated rotation speed  $N_r$ . 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  $V_{es}$  is difficult to operate only at a speed in the vicinity of the rated rotation speed  $N_r$ . When the expander section 40 according to the first embodiment is used as the expander 83, it becomes possible to control the rotation speed to a speed in the vicinity of the rated rotation speed  $N_r$  by adjusting the suction volume  $V_{es\phi}$ .

(Second Embodiment)

**[0065]** As has been mentioned in the foregoing embodiment, the position of the second suction port for varying the suction volume of the expander can be varied by an actuator that makes use of pressure difference of a fluid. The actuator making use of pressure difference of a fluid enhances reliability under severe conditions, such as under a high temperature, high pressure environment. Another advantage is that the working fluid that should be expanded by an expander can be utilized as it is for the power source of the actuator. The present embodiment describes a variable suction volume-type expander including such an actuator. In the present embodiment, the same components as described in the first embodiment are designated by the same reference numerals.

**[0066]** Fig. 11 is a vertical cross-sectional view illustrating an expander according to the second embodiment. As illustrated in Fig. 11, an expander 303 is a rotary type expander. The expander 303 includes a closed casing 11, a power generator 86 disposed in the closed casing 11, and an expander section 400 connected to the power generator 86. The expander section 400 includes a port member 412b (movable member), a housing 413 for accommodating the port member 412b, and an actuator 406.

**[0067]** The port member 412b closes the cylinder 41 (first cylinder) of the first expansion mechanism and is capable of rotating independently of the shaft 13, taking the shaft 13 as the center of rotation. The port member 412b is provided with an additional second suction port 412c. The actuator 406 is a fluid pressure actuator that makes use of the pressure difference of a fluid as its power source, and imparts a rotational force with a magnitude corresponding to the pressure difference between a high pressure fluid and a low pressure fluid to the port member 412b. When switching the rotation angle of the port member 412b, the position of the second suction port 412c around the central axis line O changes. Thereby, in the expander section 400, the timing for shifting from the suction process to the expansion process for the working fluid changes, and the ratio of the time length for which the expansion process is performed to the time

length for which the suction process is performed accordingly changes.

**[0068]** It is possible to use the working fluid, which should be expanded by the expander 303, as the high pressure fluid and the low pressure fluid, which are the power source of the actuator 406. In this way, it becomes unnecessary to prepare the fluid for operating the actuator 406 separately. Moreover, a stringent sealing structure for preventing different types of fluids from mixing together is also unnecessary. The mechanism for using the working fluid as the power source of the actuator 406 will be made clear in the following description.

**[0069]** In the present embodiment, the actuator 406, the port member 412b, and the cylinder 41 (first cylinder) of the first expansion mechanism are disposed in that order and lined up concentrically, along a direction parallel to the central axis line O of the shaft 13. This arrangement makes it possible to minimize the size increase arising from the provision of the actuator 406 and the port member 412b, and is therefore suitable for a small-sized expander 303.

**[0070]** Hereinbelow, the components of the expander 303 will be described individually. The power generator 86 includes a stator 86b fixed to a side wall of the closed casing 11, and a rotor 86a disposed inside the stator 86b. A shaft 13 is fixed to a center portion of the rotor 86a. The shaft 13 extends downwardly from the rotor 13a. The shaft 13 is shared with the expander section 400.

**[0071]** An oil reservoir 405 for holding lubricating oil is formed in a bottom portion of the closed casing 11. The lower end of the shaft 13 is disposed in the oil reservoir 405. An oil pump, which is not shown in the drawings, is formed at a lower end portion of the shaft 13, and an oil supply passage, which is not shown in the drawings, is formed in the interior and/or in the outer circumference portion of the shaft 13. When the shaft 13 rotates, the lubricating oil in the oil reservoir 405 is pumped up by the oil pump and is supplied to various sliding parts in the expander section 400 through the oil supply passage.

**[0072]** The basic structure of the expander section 400 and its workings for expanding working fluid are the same as explained in the first embodiment, and are not further elaborated on here. It should be noted that the present embodiment differs from the first embodiment in that the actuator 406 and the port member 412b for varying the position of the second suction port 412c are disposed between the first the cylinder 41 and an upper end plate 424, serving as a bearing member, and in that an end face of the first the cylinder 41 is closed by the port member 412b.

**[0073]** Hereinbelow, the port member 412b and the actuator 406 will be described in detail. The port member 412b is in a substantially disk shape and has a hole for accepting the shaft 13 in the center portion. It is disposed inside the housing 413, the outer shape of which substantially corresponds with the first the cylinder 41. The inner diameter of the housing 413 and the outer diameter of the port member 412b are approximately equal to each

other, so that displacement of the port member 412b in radial directions is restrained by the housing 413. The port member 412b can, however, rotate smoothly in the housing 413. The second suction port 412c is formed in the port member 412b so as to pass through the port member 412b vertically (axially) at a location that does not overlap with a piston 430 of the actuator 406. As the port member 412b rotates, the second suction port 412c shifts in the direction of rotation of the shaft 13.

**[0074]** Fig. 12A is a cross-sectional view of the expander shown in Fig. 11, taken along line D3-D3. As illustrated in Fig. 12A, the actuator 406 includes an eccentric portion 412a for driving the port member, a piston 430 for driving the port member, a cylinder 432 for driving the port member, a vane 433 for driving the port member, a spring 434 for driving the port member, a suction pipe 53, and a pressure controlled pipe 435. The shaft 13 is located at a center portion of the cylinder 432 for driving the port member.

**[0075]** In the following description, the phrase "for driving the port member" suffixed to the parts names of the actuator 406 will be omitted for brevity.

**[0076]** As illustrated in Fig. 12A, the eccentric portion 412a is off-centered with respect to the shaft 13 and is disposed in the cylinder 432. The top side of the cylinder 432 is closed by the upper end plate 424 (see Fig. 11). The piston 430 is fitted into the eccentric portion 412a so as to form pressure chambers 431 (431a, 431b) between the piston 430 and the cylinder 432. The eccentric portion 412a and the piston 430 rotate (more specifically, swings eccentrically) in the cylinder 432 while keeping an eccentric state with respect to the central axis line O of the shaft 13. A through hole through which the shaft 13 extends is formed in the eccentric portion 412a. The eccentric portion 412a and the shaft 13 are not joined so that they can rotate independently of each other.

**[0077]** The vane 433 is retained reciprocally in a vane groove provided in the cylinder 432 so that its leading end makes contact with the piston 430. The spring 434 pushes the vane 433 toward the piston 430.

**[0078]** The pressure chambers 431a and 431b formed in the cylinder 432 are divided by the vane 433 into two spaces, a first pressure chamber 431a and a second pressure chamber 431b. A high-pressure side inlet port 450 and a low-pressure side inlet port 451 also are provided in the cylinder 432. The high-pressure side inlet port 450 and the low-pressure side inlet port 451 are spaced circumferentially at a predetermined angle, and both penetrate the cylinder 432. The suction pipe 53 is connected to the first pressure chamber 431a via the high-pressure side inlet port 450. The suction pipe 53 is for supplying a high pressure working fluid before expansion to the first pressure chamber 431a. The pressure controlled pipe 435 is connected to the second pressure chamber 431b via the low-pressure side inlet port 451. The pressure controlled pipe 435 is for supplying the second pressure chamber 431b with a working fluid having a lower pressure than that of the working fluid supplied

to the first pressure chamber 431a. The pressure difference between the first pressure chamber 431a and the second pressure chamber 431b imparts a rotational force to the piston 430. The piston 430 that has received the rotational force originating from the pressure difference of the working fluid rotates the eccentric portion 412a and the port member 412b.

**[0079]** Also formed in the cylinder 432 is a suction passage 437 for sucking the working fluid into the working chamber 55a of the first the cylinder 41, which passes from the suction pipe 53 via the upper end plate 424, the cylinder 432, the housing 413, and the first cylinder 41.

**[0080]** In other words, the expander section 400 in the expander 303 of the present embodiment includes the suction passage 437 that is connected to the first suction port 41b formed in the first cylinder 41 and is for sending the working fluid (refrigerant) to the first cylinder 41, and the high-pressure side inlet port 450 as a branch passage, branched from the suction passage 437. The high pressure chamber 431a of the actuator 406 and the high-pressure side inlet port 450 are connected to each other, and the high pressure working fluid supplied through the high-pressure side inlet port 450 to the actuator 406 is utilized as the high pressure fluid for driving the actuator 406. Moreover, the actuator 406 and the port member 412b are disposed vertically adjacent to each other so that one end of the second suction port 412c provided in the port member 412b can be connected to the high pressure chamber 431a of the actuator 406. The working fluid supplied to the actuator 406 as the high pressure fluid is supplied through the second suction port 412c provided in the port member 412b to the working chamber 55a in the first cylinder 41 (see Fig. 2A).

**[0081]** In this way, it becomes unnecessary to prepare the fluid for operating the actuator 406 separately. It is unnecessary to provide a stringent sealing structure for preventing different kinds of fluids from mixing with each other, and in addition, the problem of the changes in the characteristics of the refrigeration cycle caused by mixing of different kinds of fluids does not arise. Moreover, because the working fluid used in the expander 303 is used as the power source of the actuator 406, no energy, such as electric power, needs to be supplied from outside. This is advantageous for improving the efficiency in recovering the energy of expansion of the working fluid.

**[0082]** On the inner circumferential surface of the cylinder 432, a first stopper 436a and a second stopper 436b protruding toward the central axis line O of the shaft 13 are provided circumferentially spaced apart at a predetermined angle. These stoppers 436a and 436b restrict the movable range (the rotation angle around the central axis line O) of the piston 430 when it is rotated by the pressure difference (when the working fluid (refrigerant) is carbon dioxide, the high pressure is higher than about 10 MPa and the low pressure is about 3 MPa to 5 MPa during a rated operation) of the working fluid. Thereby, the port member 412b is permitted to rotate only within the range of a predetermined angle (for example, about

180°).

**[0083]** It should be noted that the center of rotation of the piston 430 of the actuator 406 may correspond to the center of rotation of the shaft 13. However, when the structure in which the piston 430 eccentrically rotates is employed as in the present embodiment, the space for forming the second suction port 412c that penetrates the port member 412b vertically can be ensured easily, which is also advantageous for size reduction of the expander.

**[0084]** Fig. 12B is a cross-sectional view of the expander 303 shown in Fig. 11, taken along line D4-D4. As illustrated in Fig. 12B, a rotation spring 439 (pushing means) is attached to the port member 412b. It is preferable that the rotation spring 439 be incorporated in the port member 412b. The rotation spring 439 is interposed between the port member 412b and the housing 413 (or the cylinder 432). It applies elastic force to the port member 412b, the eccentric portion 412a, and the piston 430 in a predetermined direction of rotation at all times. As illustrated in Fig. 12A, the first pressure chamber 431a serves as the high-pressure side while the second pressure chamber 431b serves as the low-pressure side in the present embodiment. Therefore, the direction in which the rotation spring 439 applies elastic force is set to be the direction in which the volumetric capacity of the first pressure chamber 431a reduces, in other words, in the direction in which the position of the second suction port 412c moves closer to the first suction port 41b (see Fig. 2A). Because of the workings of the rotation spring 439, the position of the port member 412b can be varied continuously within the movable range determined by the stoppers 436a and 436b. In addition, it becomes possible to rotate the port member 412b in both forward and reverse directions under the condition in which the working fluid supplied to the first pressure chamber 431a is at a high pressure and the working fluid supplied to the second pressure chamber 431b is at a low pressure.

**[0085]** Of course, in the case that the rotation spring 439 is not provided as well, the port member 412b can be rotated in both forward and reverse directions by reversing the magnitude relationship between the pressure of the working fluid supplied to the first pressure chamber 431a and the pressure of the working fluid supplied to the second pressure chamber 431b. It is also possible to restrict the range of rotation of the port member 412b by providing the stoppers 436a and 436b. However, in such a configuration, it is difficult to utilize the working fluid used in the expander 303 for the power source of the actuator 406, and moreover, the structure becomes complicated. Therefore, it is preferable that the configuration as in the present embodiment be employed.

**[0086]** Moreover, with the rotation spring 439 such as described above, the magnitude of the rotational force imparted to the piston 430 is changed according to the position taken by the piston 430 in the cylinder 432. When the rotational force in the forward direction (or the reverse direction) that is imparted to the eccentric portion 412a and the piston 430 by the pressure difference between

the high pressure working fluid supplied to the first pressure chamber 431a and the low pressure working fluid supplied to the second pressure chamber 431b is counterbalanced with the repulsive force by the rotation spring 439, that is, the rotational force imparted to the port member 412b in the reverse direction (or the forward direction), the position of the port member 412b is determined at a predetermined rotation angle. In this way, it becomes possible to control the position of the port member 412b freely by adjusting the pressure difference between the working fluid supplied to the first pressure chamber 431a and the working fluid supplied to the second pressure chamber 431b of the actuator 406. In other words, it becomes possible to adjust the second suction port 412c to be at the optimal position according to the operation condition of the expander 303.

**[0087]** As described above, a high pressure working fluid passes through the suction pipe 53 to the suction passage 437 and flows into the working chamber 55a from the first suction port 41b provided in the first cylinder 41 (see Fig. 2A). Apart from that passage, a high pressure working fluid passes through the high-pressure side inlet port 450 branched from the suction pipe 53 and flows into the first pressure chamber 431a in the cylinder 432, and it passes through the second suction port 41.2c provided in the port member 412b and flows into the working chamber 55a. Since the position of the second suction port 412c changes as the port member 412b rotates, the suction volume of the working fluid to the first cylinder 41 changes.

**[0088]** In the present embodiment, the eccentric portion 412a and the port member 412b are coupled or integrated with each other vertically, parallel to the central axis line O. As illustrated in Figs. 11 and 12B, the port member 412b is in a substantially disk shape. One of main surfaces of the port member 412b closes the first cylinder 41 while the other one of main surfaces closes the cylinder 432. On the other one of main surfaces side, the port member 412b is coupled (or united) with the eccentric portion 412a. The portion positioned distant from the first cylinder 41 is the eccentric portion 412a. The portion positioned near the first cylinder 41 is the port member 412b. In this way, the power transfer mechanism from the actuator 406 to the port member 412b can be eliminated, contributing to reducing the parts count and simplifying the structure, and a highly reliable expander 303 can be provided. It should be noted that the eccentric portion 412a can also serve the role of the piston 430, and in that case, the port member 412b can be constructed as a component integrated with the piston 430.

**[0089]** The positional relationship between the first suction port 41b and the second suction port 412c is as described above with reference to Figs. 4A, 4B, and 4C.

**[0090]** Also, the operation principles of the first cylinder 41 and the second cylinder 42 are as described above with reference to Figs. 5A and 5B.

**[0091]** The relationship between the rotation angle  $\theta$  of the shaft 13 and the time point for shifting the processes

from suction to discharge is as described above with reference to Fig. 6A.

**[0092]** The relationship between the rotation angle  $\theta$  of the shaft 13 and the volumetric capacity of the working chamber is as described above with reference to Fig. 6B.

**[0093]** Next, a pressure regulator for controlling the pressure of the working fluid to be supplied to the pressure controlled pipe 435 of the actuator 406 will be described below. A heat pump 300 shown in Fig. 13 includes the compressor 81, the gas cooler 82, the expander 303 described with reference to Fig. 11, the evaporator 84, and a pressure regulator 500A. The pressure regulator 500A regulates the pressure difference between a high pressure fluid and a low pressure fluid to be supplied to the actuator 406 of the expander 303. By providing such a pressure regulator 500A, it becomes possible to control the workings of the actuator 406 from the outside of the expander 303. In the example shown in Fig. 13, the pressure regulator 500A is installed outside the expander 303, but it may be installed inside the expander 303.

**[0094]** The pressure regulator 500A includes a first pressure pipe 501, one end of which is connected to the suction pipe 53 of the expander 303, a second pressure pipe 502, one end of which is connected to the pressure controlled pipe 435 of the expander 303, a third pressure pipe 503, one end of which is connected to the discharge pipe 54 of the expander 303, and a hollow housing 513, to which the other ends of the pressure pipes 501, 502, and 503 are connected. In other words, the outlet pipe of the gas cooler 82 is branched into the first pressure pipe 501 and the suction pipe 53 of the expander 303. In addition, the discharge pipe 54 of the expander 303 and the third pressure pipe 503 merge with each other, forming an inlet pipe of the evaporator 84. The interior of the housing 513 is formed into three pressure regulating chambers, a first pressure regulating chamber 504, a second pressure regulating chamber 505, and a third pressure regulating chamber 506. The first pressure pipe 501 is connected to the first pressure regulating chamber 504. The second pressure pipe 502 is connected to the second pressure regulating chamber 505. The third pressure pipe 503 is connected to the third pressure regulating chamber 506.

**[0095]** An elastic body 507 (spring) is disposed in the third pressure regulating chamber 506. A piston 508, one end of which is connected to the elastic body 507, is disposed between the second pressure regulating chamber 505 and the third pressure regulating chamber 506, for separating the two pressure regulating chambers. The piston 508 can move back and forth between the second pressure regulating chamber 505 and the third pressure regulating chamber 506. A micro flow passage 514 for bring the second pressure regulating chamber 505 and the third pressure regulating chamber 506 in communication with each other is formed in the piston 508. A valve 509 for adjusting the amount of the working fluid flowing between the first pressure regulating chamber 504 and the second pressure regulating chamber 505 is provided

between the two pressure regulating chambers. One end of a coupling shaft 512 is connected to the valve 509. The other end of the coupling shaft 512 is connected to an iron core 511. A coil 510 is disposed around the iron core 511. The iron core 511 and the coil 510 together form a plunger solenoid.

**[0096]** In the pressure regulator 500A, the pressure of the first pressure regulating chamber 504 is equal to the high pressure of the refrigerant circuit, and the pressure of the third pressure regulating chamber 506 is equal to the low pressure of the refrigerant circuit. The pressure of the second pressure regulating chamber 505 controlled by the pressure regulator 500A is supplied to the pressure controlled pipe 435 of the expander 303 and is used for changing the suction volume of the expander section 400.

**[0097]** In the configuration as shown in Fig. 13, the resilient force of the elastic body 507, the pressure resulting from the pressure difference between the second pressure regulating chamber 505 and the third pressure regulating chamber 506, and the driving force imparted by the current passed through the coil 510 are applied to the coupling shaft 512. The coupling shaft 512 stops at the position where these forces are balanced. The pressure of the second pressure regulating chamber 505 can be controlled by varying the current passed through the coil 510.

**[0098]** Specifically, the pressure regulator 500A acquires a portion of the high pressure working fluid to be sent to the first suction port 41b of the expander section 400, and decompresses the acquired working fluid to thereby produce a low pressure working fluid to be supplied to the second pressure chamber 431b of the actuator 406. Then, by adjusting the degree of decompression of the working fluid, the pressure of the second pressure chamber 431b formed in the cylinder 432 for driving the port member is regulated, and the positions of the port member 412b and the second suction port 412c provided in the port member 412b are controlled around the central axis line O. In this way, the control process of the position of the second suction port 412c can be carried out easily and accurately.

**[0099]** As described above, the heat pump 300 includes: the first pressure pipe 501, having one end connected to the main pipe (the suction pipe 53) for sending the working fluid to the first suction port 41b of the expander section 400 and the other end connected to the pressure regulator 500A, and supplying a portion of the high pressure working fluid that is to be expanded to the first pressure regulating chamber 504 of the pressure regulator 500A; and the second pressure pipe 502, having one end connected to the second pressure regulating chamber 505 of the pressure regulator 500A and the other end connected to the actuator 406 (more precisely the pressure controlled pipe 435), and supplying the low pressure chamber 431b of the actuator 406 (see Fig. 14) with the working fluid that has been decompressed to a low pressure by the pressure regulator 500A.

**[0100]** The workings of the pressure regulator 500 will be described. For example, when it is desired to increase the suction volume of the expander 303 (the expander section 400), the current flowing through the coil 510 should be increased. Then, the force applied to the iron core 511 toward the elastic body 507 increases and compresses the elastic body 507, and the valve 509 narrows the passage between the first pressure regulating chamber 504 and the second pressure regulating chamber 505. Thereby, the pressure of the second pressure regulating chamber 505 reduces and becomes close to the pressure of the third pressure regulating chamber 506. Accordingly, the pressure difference between the pressure in the pressure controlled pipe 435 and the pressure in the suction pipe 53 increases. The piston 430 for driving the port member and the eccentric portion 412a for driving the port member rotate in a direction in which the volumetric capacity of the second pressure chamber 431b decreases. The second suction port 412c arrives at, for example, the position shown in Fig. 4C. As a result, the suction time of the expander 303 becomes longer, and the suction volume increases, according to the principle that has been explained with reference to Figs. 5A and 5B.

**[0101]** Conversely, when it is desired to decrease the suction volume of the expander 303 (the expander section 400), the current flowing through the coil 510 should be decreased. Then, the force applied to the iron core 511 toward the elastic body 507 decreases, so that the elastic body 507 elongates, and at the same time, the valve 509 widens the passage between the first pressure regulating chamber 504 and the second pressure regulating chamber 505. Thereby, the pressure of the second pressure regulating chamber 505 increases and becomes close to the pressure of the first pressure regulating chamber 504. Accordingly, the pressure difference between the pressure in the pressure controlled pipe 435 and the pressure in the suction pipe 53 reduces. The piston 430 for driving the port member and the eccentric portion 412a for driving the port member rotate in a direction in which the volumetric capacity of the second pressure chamber 431b increases. The second suction port 412c arrives at, for example, the position shown in Fig. 4A. As a result, the suction time of the expander 303 becomes shorter, and the suction volume decreases, according to the principle that has been explained with reference to Figs. 5A and 5B.

**[0102]** It is also possible to employ a pressure regulator having the configuration as shown in Fig. 15. First, as illustrated in Fig. 14, a micro passage 440 that bypasses the pressure controlled pipe 435 and the suction pipe 53 is provided in an actuator 406'. As illustrated in Fig. 15, a pressure regulator 500B includes a housing 515, the coil 510, the iron core 511, the coupling shaft 512, a piston 516, and the elastic body 507 (spring). The interior of the housing 515 is partitioned into two pressure regulating chambers 520 and 521. A valve 509 for adjusting the amount of the working fluid flowing between the two pres-

sure regulating chambers 520 and 521 is provided between the two pressure regulating chambers. The coil 510 and the iron core 511 together form a plunger solenoid. The elastic body 507 pushes the valve 509 in the direction in which it opens, via the piston 516. On the other hand, when passing electric current through the coil 510, the iron core 511 pushes the valve 509 in the direction in which it closes, via the coupling shaft 512. That is, the opening of the valve 509 can be controlled by controlling the electric current passing through the coil 510. The pressure of a pressure regulating chamber 521, to which the pressure controlled pipe 435 is connected, can be changed according to the opening of the valve 509.

**[0103]** It should be noted that in the examples shown in Figs. 13 and 15, it is necessary to keep the control pressure by causing the working fluid to flow in a small amount at all times, and therefore, the efficiency in recovering the energy of expansion lowers inevitably. In view of this, when the control pressure is produced with the configuration as shown in Fig. 16A, it is possible to enhance the efficiency in recovering the energy of expansion further.

**[0104]** A pressure regulator 500C shown in Fig. 16A includes a housing 560 in which the interior thereof is partitioned into three pressure regulating chambers, a first pressure regulating chamber 561, a second pressure regulating chamber 562, and a third pressure regulating chamber 563. The first pressure pipe 501 in which the working fluid before expansion circulates is connected to the first pressure regulating chamber 561. The second pressure pipe 502, which brings the second pressure regulating chamber 562 and the second pressure chamber 431b of the actuator 406 in the expander 303 (see Fig. 12A) into communication with each other, is connected to the second pressure regulating chamber 562. The third pressure pipe 503 in which the working fluid after expansion circulates is connected to the third pressure regulating chamber 563. A first valve 580 is disposed between the first pressure regulating chamber 561 and the second pressure regulating chamber 562. The open/close operation of the first valve 580 can be controlled by actuating a plunger solenoid including a coil 570, an iron core 573, an elastic body 584 (spring), and a coupling shaft 576. A high pressure working fluid can be sent into the second pressure regulating chamber 562 by opening the first valve 580. On the other hand, a second valve 581 is disposed between the second pressure regulating chamber 562 and the third pressure regulating chamber 563. As in the case of the first valve 580, the open/close operation of the second valve 581 can be controlled by a plunger solenoid including a coil 571, an iron core 574, an elastic body 585 (spring), and a coupling shaft 577. A working fluid can be sent from the second pressure regulating chamber 562 to the third pressure regulating chamber 563 by opening the second valve 581. Thus, it becomes possible to produce a control pressure between the pressure of the working fluid before expansion and

the pressure of the working fluid after expansion by controlling the open/close operation of the two (a plurality of) valves 580 and 581 so that the interior of the second pressure regulating chamber 562, and accordingly the interior of the second pressure chamber 431b of the actuator 406, can be kept at the produced control pressure. When the pressure of the second pressure regulating chamber 562 is higher than a desired pressure, the first valve 580 should be closed while the second valve 581 should be opened. When the pressure is lower than a desired pressure, the first valve 580 should be opened while the second valve 581 should be closed.

**[0105]** In addition, as illustrated in the block diagram of Fig. 16B, the pressure regulator 500C may include a pressure sensor 590 for detecting the pressure in the second pressure regulating chamber 562 and a controller 591 for controlling the open/close operation of the valves 580 and 581 based on the result detected by the pressure sensor 590. The pressure sensor 590 may be disposed in the second pressure chamber 431b of the actuator 406 in the expander 303. The controller 591 acquires sensor signal from the pressure sensor 590 and calculates the difference between the target pressure value and the current pressure value. If the calculated difference is out of a predetermined permissible range, it controls the open/close operation of the first valve 580 and the second valve 581 so that the difference becomes smaller. Specifically, if the current pressure value is smaller than the target pressure value, the solenoid on the first valve 580 side is driven for a certain time so that a certain amount of a high pressure working fluid flows from the first pressure regulating chamber 561 into the second pressure regulating chamber 562. Conversely, if the current pressure value is greater than the target pressure value, the solenoid on the second valve 581 side is driven for a certain time to move the working fluid from the second pressure regulating chamber 562 to the third pressure regulating chamber 563.

**[0106]** By repeating such a process, the pressure of the second pressure regulating chamber 562 can be adjusted quickly and accurately to the desired pressure. Since the solenoids (the coils 570 and 571) for opening/closing the first valve 580 and the second valve 581 do not require electric current at all times, it is possible to reduce power consumption of the pressure regulator 500C, which is advantageous for improving the efficiency in recovering the energy of expansion of the working fluid. Moreover, when a program that monitors the input from the pressure sensor 590 periodically is installed in the controller 591, the pressure in the second pressure regulating chamber 562 can be recovered to a desired pressure automatically even if a pressure variation occurs due to unavoidable working fluid leakage or the like.

(Third Embodiment)

**[0107]** The features of the expander illustrated in the second embodiment may be employed suitably for an

expander-compressor unit in which an expander section and a compressor section are integrated with each other by a shaft, as explained in the first embodiment. Fig. 17 is a vertical cross-sectional view illustrating such an expander-compressor unit.

**[0108]** An expander-compressor unit 700 includes a closed casing 11, a scroll type compressor section 20 disposed in an upper portion of the closed casing 11, a two-stage rotary expander section 400 disposed in a lower portion of the closed casing 11, a rotation motor 12 disposed between the compressor section 20 and the expander section 400, and a shaft 13 commonly used for the compressor section 20, the expander section 400, and the rotation motor 12. As the rotation motor 12 drives rotation of the shaft 13, the compressor section 20 operates. This expander-compressor unit 700 makes use of the rotational force imparted to the shaft 13 by the working fluid (refrigerant) when expanding at the expander section 400 as auxiliary power for the compressor section 20. High energy recovery efficiency is expected because the energy of expansion of the working fluid is transferred directly to the compressor section 20 without being converted to electric energy.

**[0109]** As described in the second embodiment, the expander section 400 includes the port member 412b provided with the second suction port 412c for changing a suction volume, and the actuator 406 for rotating the port member 412b to change the position. The structures and functions of the port member 412b and the actuator 406 are the same as described in the second embodiment.

**[0110]** The basic structures and operation principle of the compressor section 20 and the expander section 400 are also the same as described in the first embodiment.

**[0111]** With the expander-compressor unit 700 shown in Fig. 17, the actuator 406 is driven by the pressure difference between the working fluid fed from the pressure controlled pipe 435 and the working fluid fed from the suction pipe 53, and thereby the position of the port member 412b (rotation angle around the central axis line O) can be changed. By controlling the position of the port member 412b, the suction volume of the expander section 400 can be controlled freely. A heat pump employing such an expander-compressor unit 700 makes it possible to control the flow rate of the working fluid flowing through the expander section 400 freely without providing a bypass circuit, and accordingly realizes a highly efficient heat pump system.

(Fourth Embodiment)

**[0112]** The actuator incorporated in the expander according to the second embodiment may be employed for an expander or an expander-compressor unit suitably, but it also may be configured as a rotary actuator for other applications.

**[0113]** Fig. 18 is a vertical cross-sectional view illustrating a rotary actuator according to a fourth embodi-

ment. Fig. 19 is a cross-sectional view taken along line D5-D5 in Fig. 18. As illustrated in Figs. 18 and 19, a rotary actuator 800 includes a cylinder 806, a shaft 801 penetrating the cylinder 806, a piston 807 that swings eccentrically in the cylinder 806 to rotate the shaft 801, and a vane 812 partitioning a pressure chamber 808 formed between the cylinder 806 and the piston 807 into a first pressure chamber 808a and a second pressure chamber 808b.

**[0114]** The shaft 801 has an eccentric portion 802 protruding radially outwardly at its intermediate portion. One end of the shaft 801 penetrates an upper end plate 803 while its other end penetrates a lower end plate 804. A closing member 805 is disposed below the lower end plate 804. The upper end plate 803 and/or the closing member 805 may contain a bearing for the shaft 801. The eccentric portion 802 of the shaft 801 is disposed in the cylinder 806. A ring-shaped piston 807 is fitted onto the eccentric portion 802.

**[0115]** The rotary actuator 800 includes the vane 812, a spring 809, a suction pipe 810, and a pressure controlled pipe 811. The vane 812 is retained reciprocally in a vane groove provided in the cylinder 806 so that its leading end makes contact with the piston 807. The spring 809 pushes the vane 812 toward the piston 807. In the upper end plate 803 that closes the top of the cylinder 806, a first inlet port 820 in communication with the first pressure chamber 808a and a second inlet port 821 in communication with the second pressure chamber 808b are formed. The suction pipe 810 is connected to the first pressure chamber 808a via the first inlet port 820. The pressure controlled pipe 811 is connected to the second pressure chamber 808b via the second inlet port 821. The pressure difference between the first fluid flowing into the first pressure chamber 808a and the second fluid flowing into the second pressure chamber 808b produces a force applied to the piston 807, rotating the eccentric portion 802 and consequently the overall shaft 801. A first stopper 813a and a second stopper 813b are formed circumferentially spaced at a predetermined angle on the inner circumferential surface of the cylinder 806. These stoppers 813a and 813b restrain the range of rotation of the piston 807 when it is rotated by the pressure difference of the working fluid.

**[0116]** It should be noted that the elastic body for imparting a repulsive force to the rotation of the shaft 801 is not provided in the present embodiment, but it is possible to provide an elastic body (a rotation spring 439; see Fig. 12B) as described in the second embodiment. In this way, it becomes possible to control the rotation angle of the shaft 801 by adjusting the pressure difference between the first fluid flowing into the first pressure chamber 808a and the second fluid flowing into the second pressure chamber 808b. The first fluid and the second fluid may be either the same or different kinds of fluids. Possible examples of such fluids include oil in the hydraulic pressure circuit, refrigerant in the refrigerant circuit, and air in the air pressure circuit.

## INDUSTRIAL APPLICABILITY

**[0117]** 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. An expander comprising:

n-number of rotary type expansion mechanisms (where "n" is an integer equal to or greater than 2) each having a cylinder, a shaft with an eccentric portion, a piston fitted to the eccentric portion and rotating eccentrically in the cylinder, and a partition member partitioning a space between the cylinder and the piston into a suction-side space and a discharge-side space;  
a first suction port for sucking a working fluid into the suction-side space of the first expansion mechanism;  
a communication port connecting the discharge-side space of the k-th expansion mechanism (where "k" is an integer from 1 to n-1) and the suction-side space of the (k + 1)-th expansion mechanism to form a single space;  
a discharge port for discharging the working fluid from the discharge-side space of the n-th expansion mechanism; and  
a second suction port for sucking the working fluid into the suction-side space of the first expansion mechanism, the second suction port being capable of changing its connecting position to the suction-side space of the first expansion mechanism.

2. The expander according to claim 1, further comprising:

an intermediate plate disposed between the cylinder of the k-th expansion mechanism and the cylinder of the (k + 1)-th expansion mechanism, for separating the cylinders, and wherein the first suction port is provided in the cylinder of the first expansion mechanism, the communication port is provided in the intermediate plate, and the discharge port is provided in the cylinder of the n-th expansion mechanism and/or in a closing member for closing the cylinder of the n-th expansion mechanism.

3. The expander according to claim 1, further comprising:

- a closing member for closing the cylinder of the first expansion mechanism at an end face thereof, and wherein the second suction port is provided in the closing member.
4. The expander according to claim 3, wherein:
- the closing member includes a movable portion capable of rotating, taking the shaft as the center of rotation, and the second suction port is provided in the movable portion.
5. The expander according to claim 4, wherein the movable portion includes a cylindrical bearing surface for supporting the shaft.
6. The expander according to claim 4, further comprising a drive mechanism for rotating the movable portion.
7. The expander according to claim 6, wherein the drive mechanism includes an electric actuator.
8. The expander according to claim 4, wherein the closing member further includes an annular stationary portion to which the movable portion is united rotatably, the annular stationary portion having therein a first inlet passage for supplying the working fluid from outside of the expander to the second suction port provided in the movable portion, and a second inlet passage branched from the first inlet passage, for supplying the working fluid to the first suction port.
9. An expander-compressor unit comprising: an expander section comprising an expander according to claim 1; and a compressor section coupled integrally with the expander section by the shaft.
10. A heat pump comprising an expander according to claim 1.
11. A heat pump comprising an expander-compressor unit according to claim 9.
12. The expander according to claim 1, further comprising:
- a movable member that closes the cylinder of the first expansion mechanism and that is capable of rotating independently of the shaft, taking the shaft as the center of rotation; and an actuator imparting a rotational force having a magnitude corresponding to a pressure difference between a high pressure fluid and a low pressure fluid to the movable member, wherein the second suction port is provided in the movable member.
13. The expander according to claim 12, wherein the working fluid is used as the high pressure fluid and the low pressure fluid.
14. The expander according to claim 12, wherein the actuator, the movable member, and the first expansion mechanism are disposed in that order along a direction parallel to the central axis line of the shaft.
15. The expander according to claim 12, further comprising:
- a suction passage connected to the first suction port, for sending the working fluid to the cylinder of the first expansion mechanism; and a branch passage branched from the suction passage, and wherein:
- a high pressure chamber of the actuator and the branch passage are connected to each other, and a high pressure working fluid supplied to the actuator through the branch passage is utilized as the high pressure fluid; the actuator and the movable member are disposed adjacent to each other so that one end of the second suction port is connected to the high pressure chamber of the actuator; and the working fluid supplied to the actuator as the high pressure fluid is supplied through the second suction port provided in the movable member to the suction-side space of the first fluid mechanism.
16. The expander according to claim 12, wherein the actuator comprises:
- a cylinder for driving the movable member; a piston for driving the movable member, the piston forming a pressure chamber between the piston and the cylinder for driving the movable member and rotating the movable member by swinging eccentrically in the cylinder for driving the movable member; and a partition member for driving the movable member, that partitions the pressure chamber into a high pressure chamber in which the high pressure fluid flows, and a low pressure chamber in which the low pressure fluid flows.
17. The expander according to claim 16, wherein:
- the movable member is in a substantially disk-shaped configuration; one of main surfaces of the movable member closes the cylinder of the first expansion mech-

anism while the other one of the main surfaces closes the cylinder for driving the movable member; and  
 on the other one of the main surfaces side, the movable member is coupled or united with the piston for driving the movable member or an eccentric portion disposed in the piston for driving the movable member.

**18.** The expander according to claim 16, further comprising a stopper provided on an inner circumferential surface of the cylinder for driving the movable member and having a protruding shape toward the central axis line of the shaft, for restricting a movable range of the piston for driving the movable member in the cylinder for driving the movable member.

**19.** The expander according to claim 16, further comprising pushing means for pushing the movable member in a predetermined direction of rotation.

**20.** The expander according to claim 19, wherein:

the pushing means is configured to vary the magnitude of the rotational force imparted to the movable member according to the position taken by the movable member around the central axis line of the shaft; and  
 the position of the movable member around the central axis line of the shaft is controlled by counterbalancing a rotational force in a forward direction, imparted to the piston for driving the movable member by the pressure difference between the high pressure fluid and the low pressure fluid, and a rotational force in a reverse direction, imparted to the movable member by the pushing means.

**21.** An expander-compressor unit comprising: an expander section comprising an expander according to claim 12; and a compressor section integrally coupled to the expander section by the shaft.

**22.** A heat pump comprising an expander according to claim 12.

**23.** The heat pump according to claim 22, further comprising a pressure regulator for regulating a pressure difference between the high pressure fluid and the low pressure fluid to be supplied to the actuator.

**24.** The heat pump according to claim 23, wherein the pressure regulator acquires a portion of the working fluid to be sent to the first suction port and decompresses the acquired working fluid to produce a low pressure fluid, and by adjusting the degree of the decompression, the pressure regulator controls the position of the second suction port around the central

axis line of the shaft.

**25.** The heat pump according to claim 23, further comprising:

a first pressure pipe having one end connected to a main pipe for sending the working fluid to the first suction port and another end connected to the pressure regulator and supplying a portion of a high pressure working fluid to be expanded to a first chamber of the pressure regulator; and a second pressure pipe having one end connected to a second chamber of the pressure regulator and another end connected to the actuator and supplying, to the low pressure chamber of the actuator, the working fluid that has been decompressed to a low pressure by the pressure regulator.

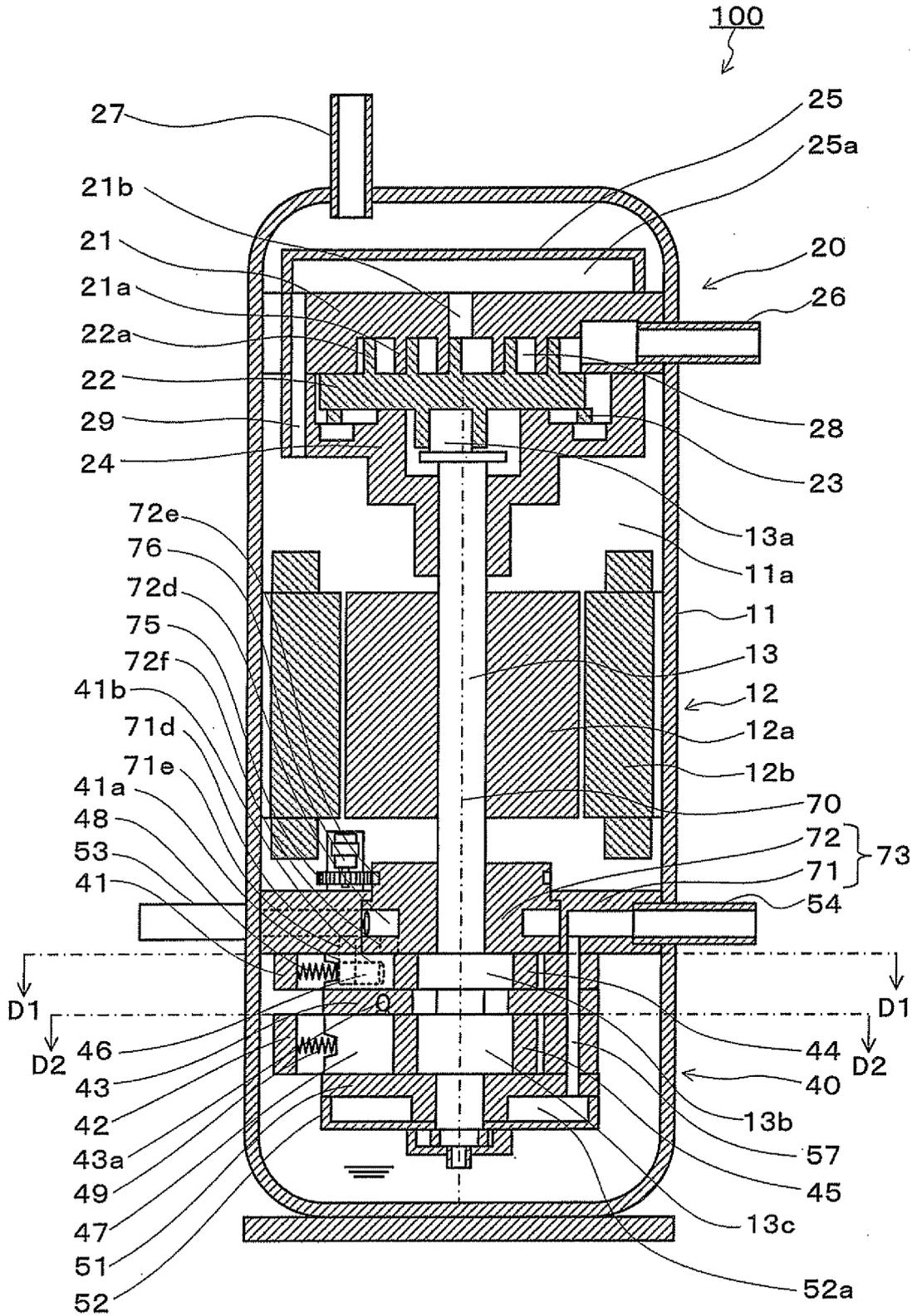


Fig. 1

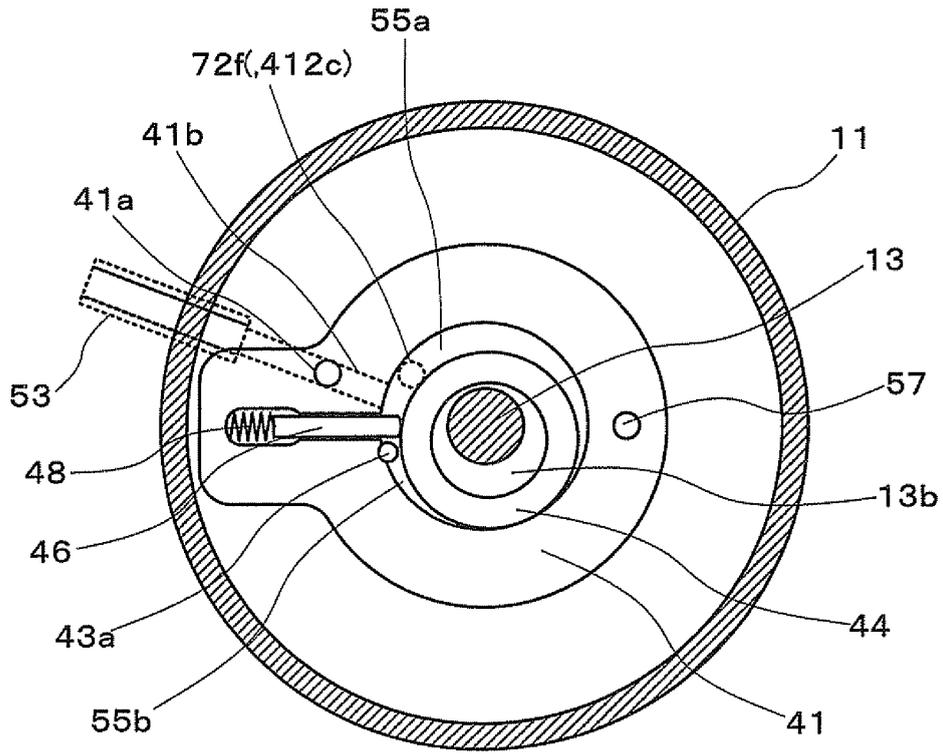


Fig. 2A

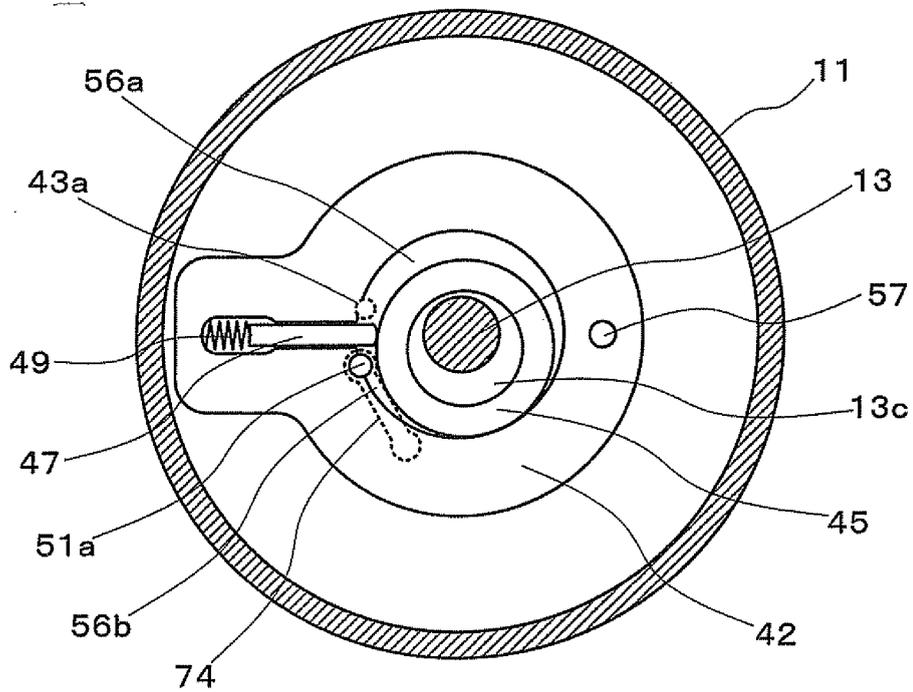


Fig. 2B

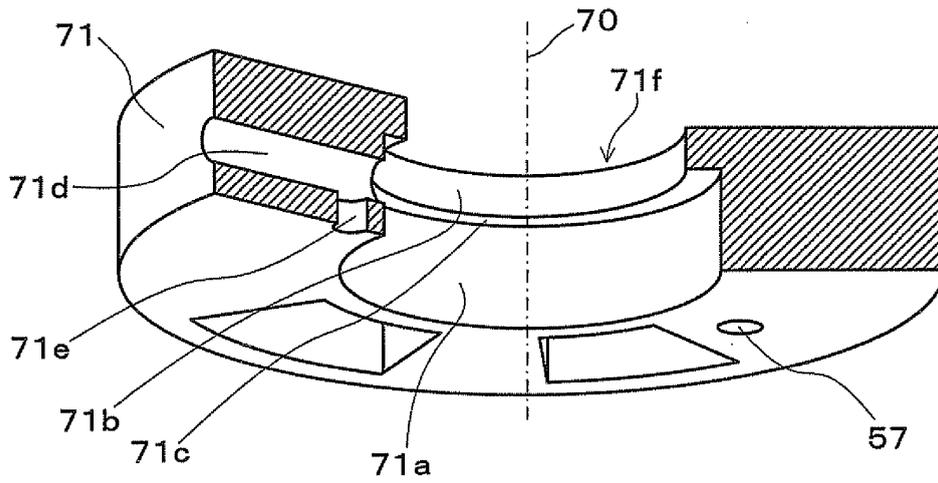


Fig. 3A

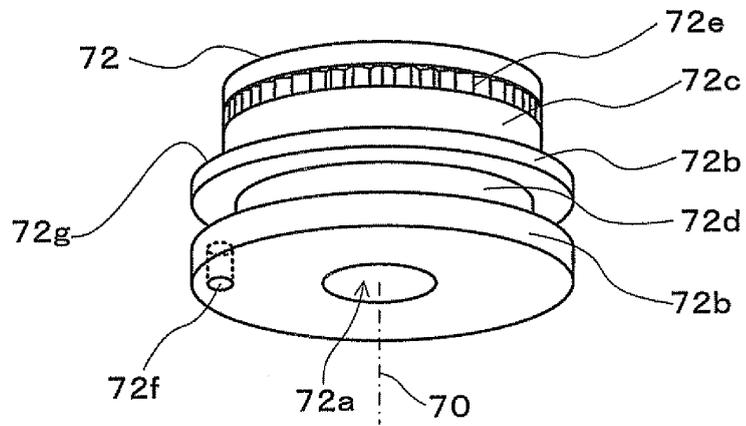


Fig. 3B

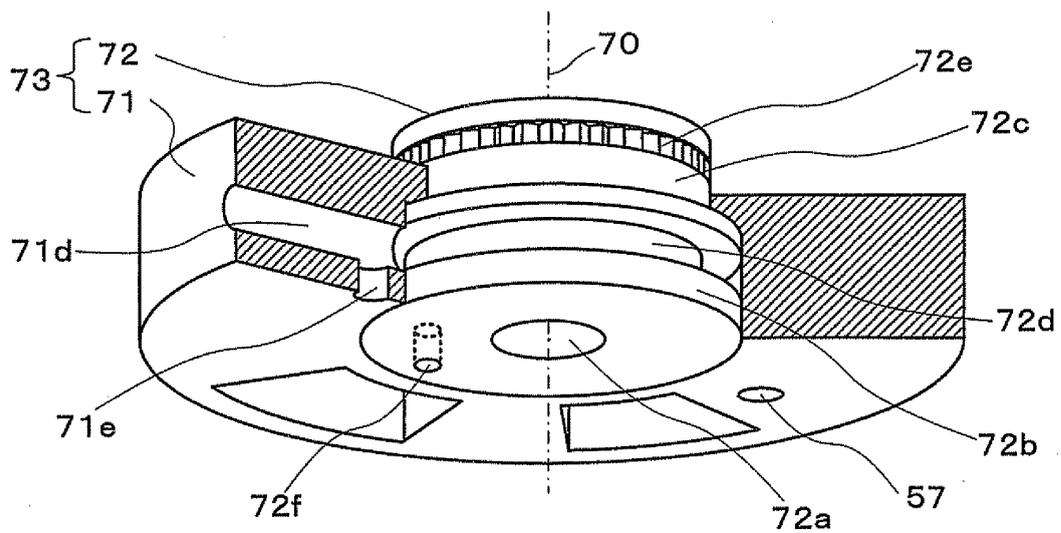


Fig. 3C

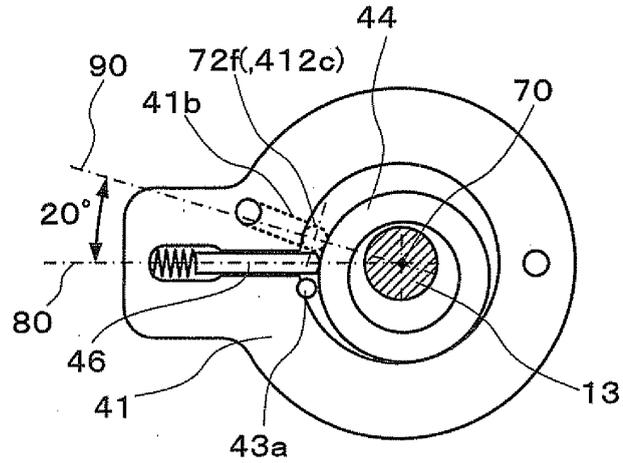


Fig. 4A

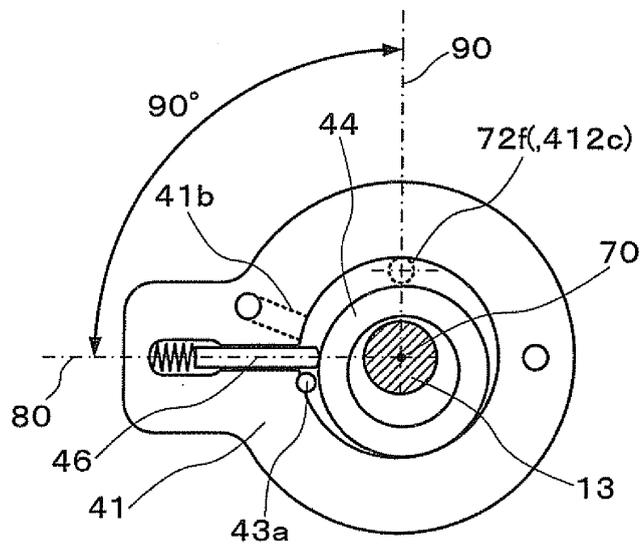


Fig. 4B

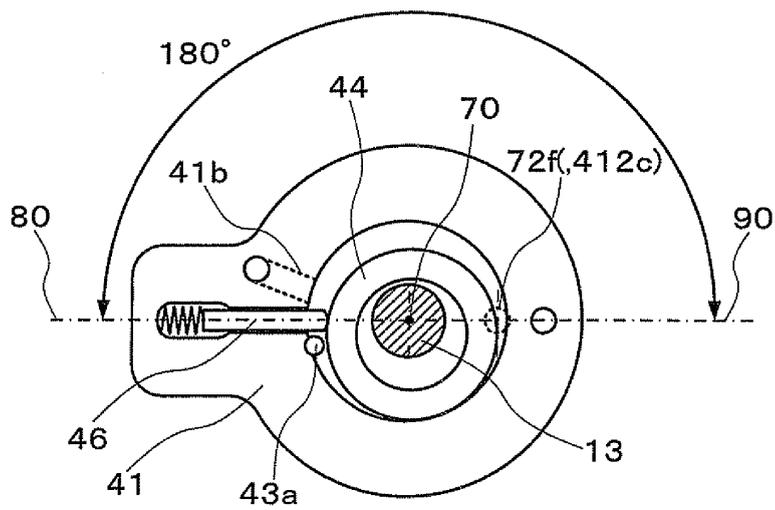


Fig. 4C

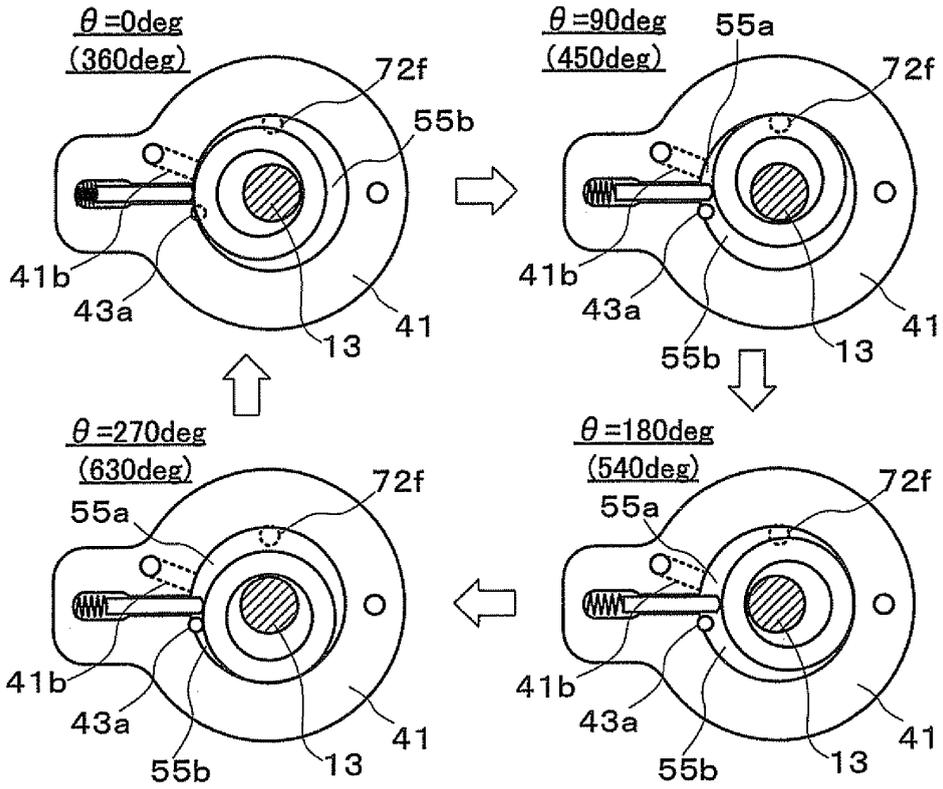


Fig. 5A

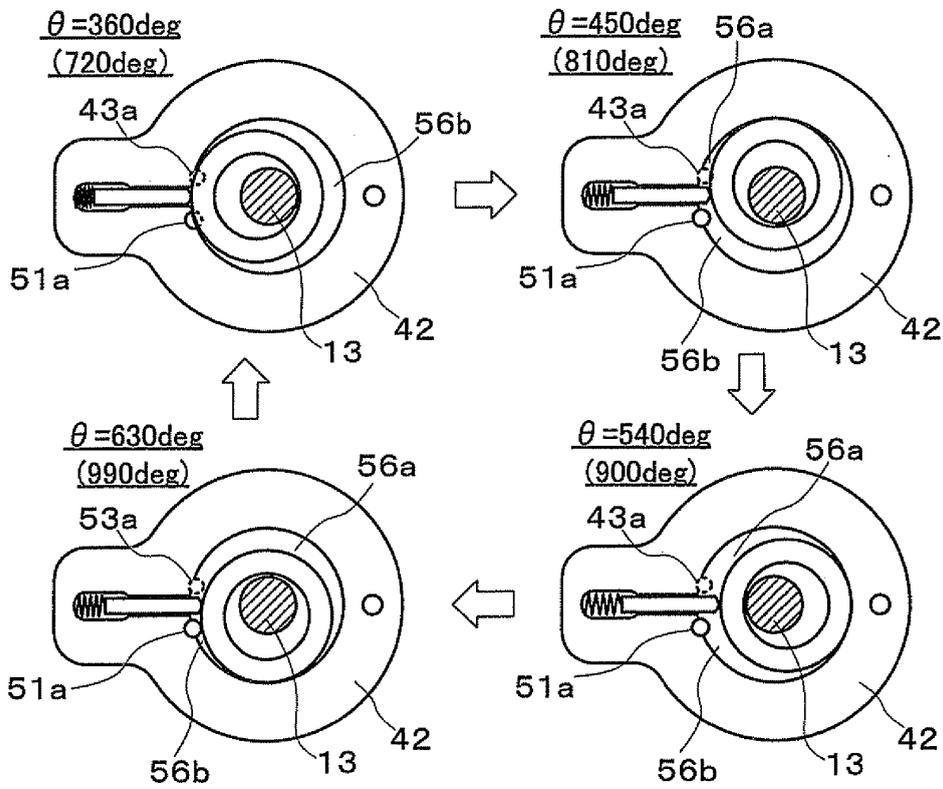


Fig. 5B

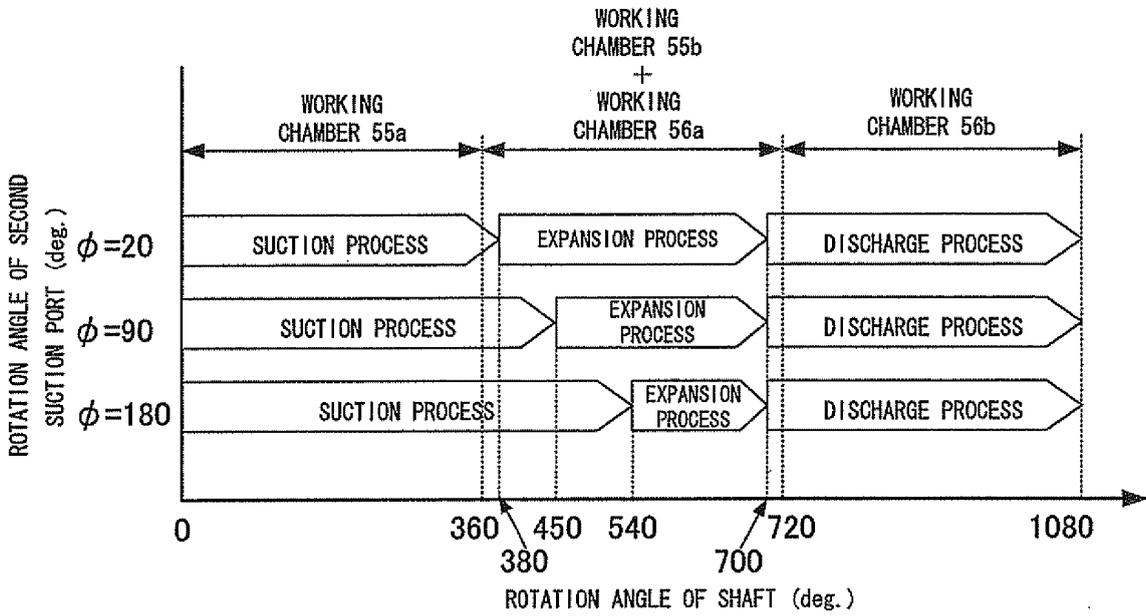


Fig. 6A

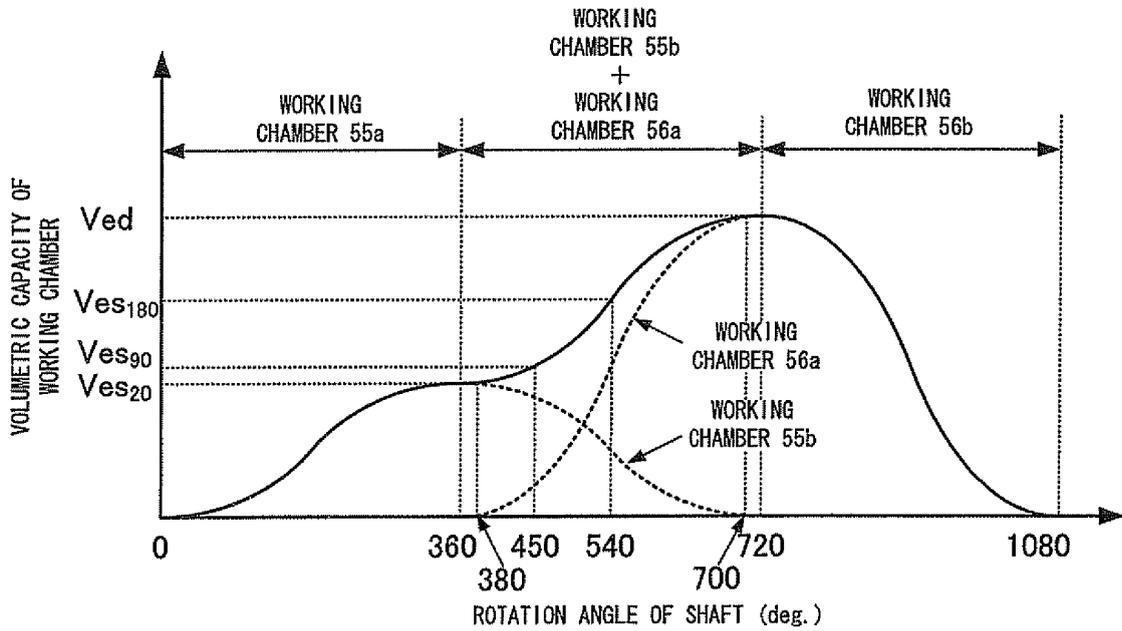


Fig. 6B

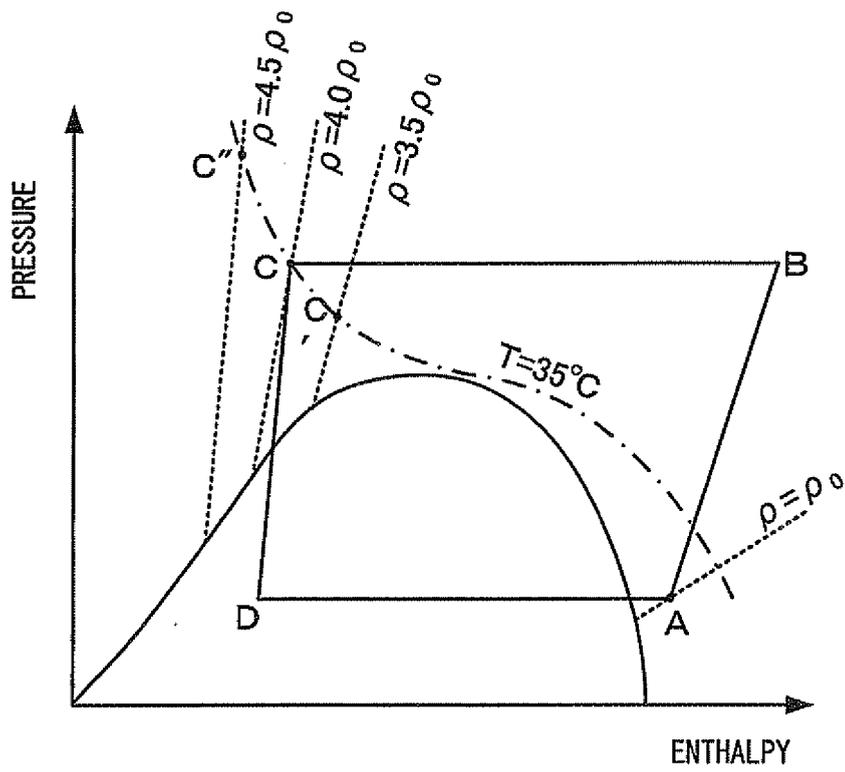


FIG. 7

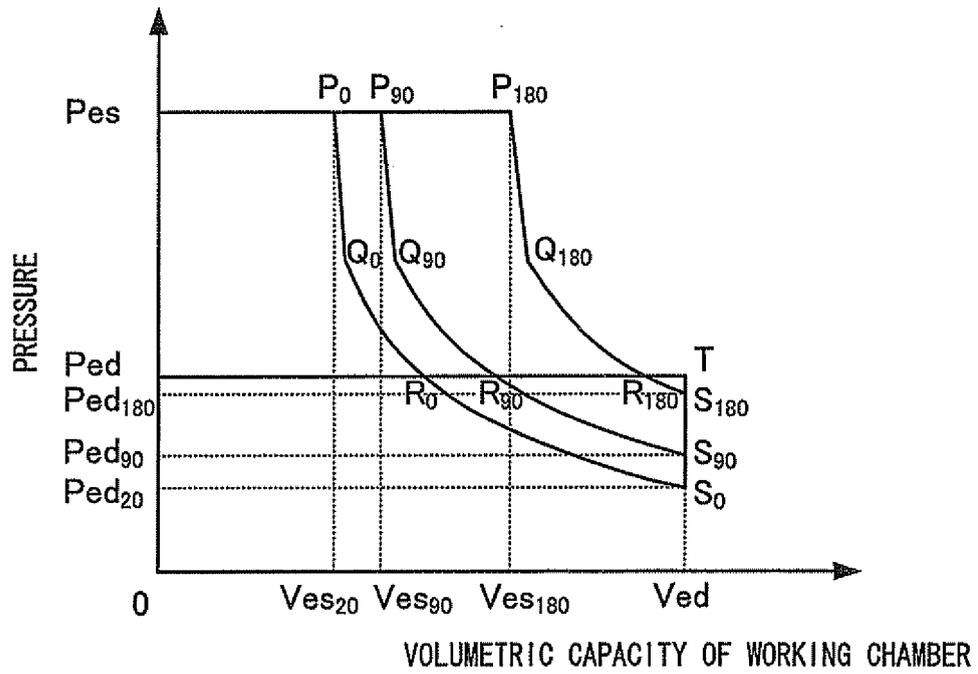


FIG. 8

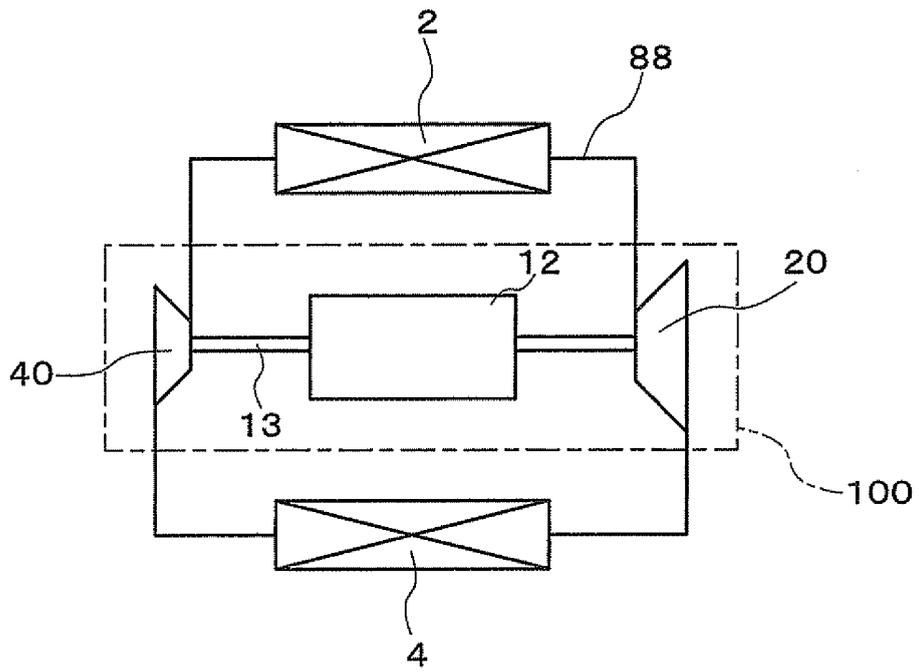


FIG. 9A

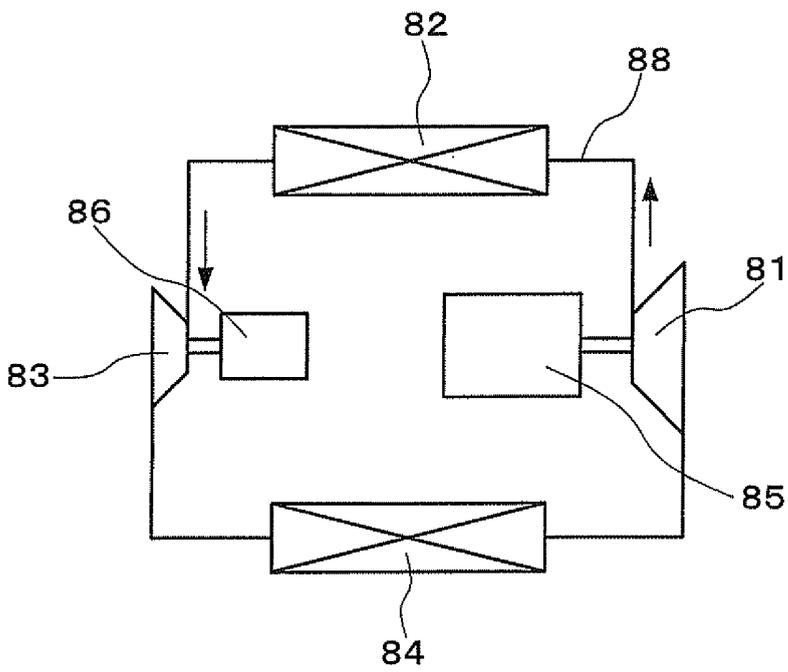


FIG. 9B

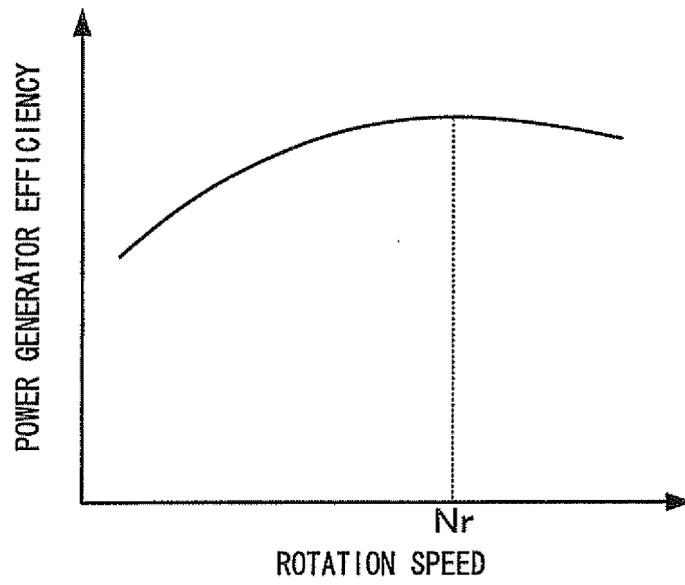


FIG. 10

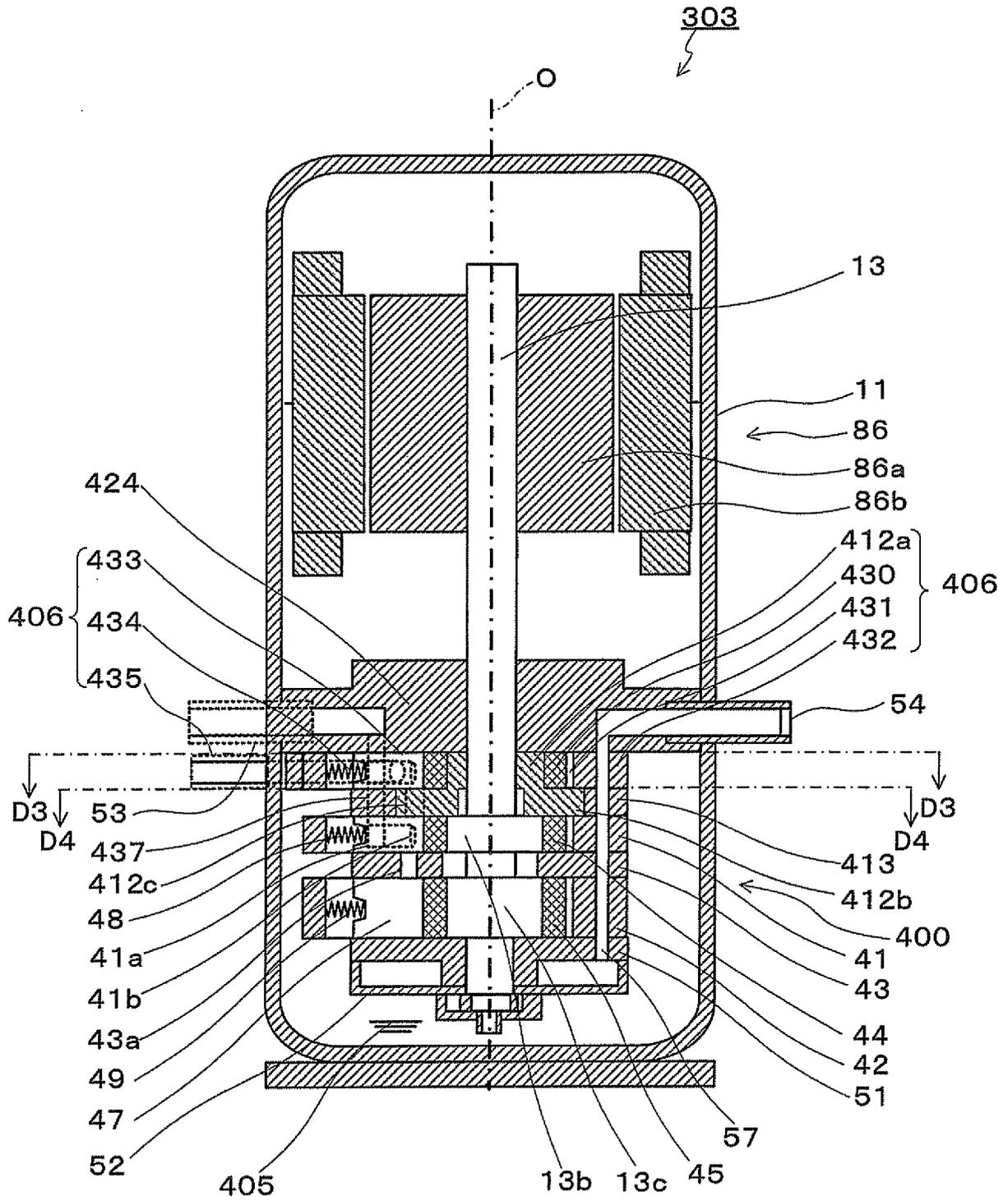


FIG. 11

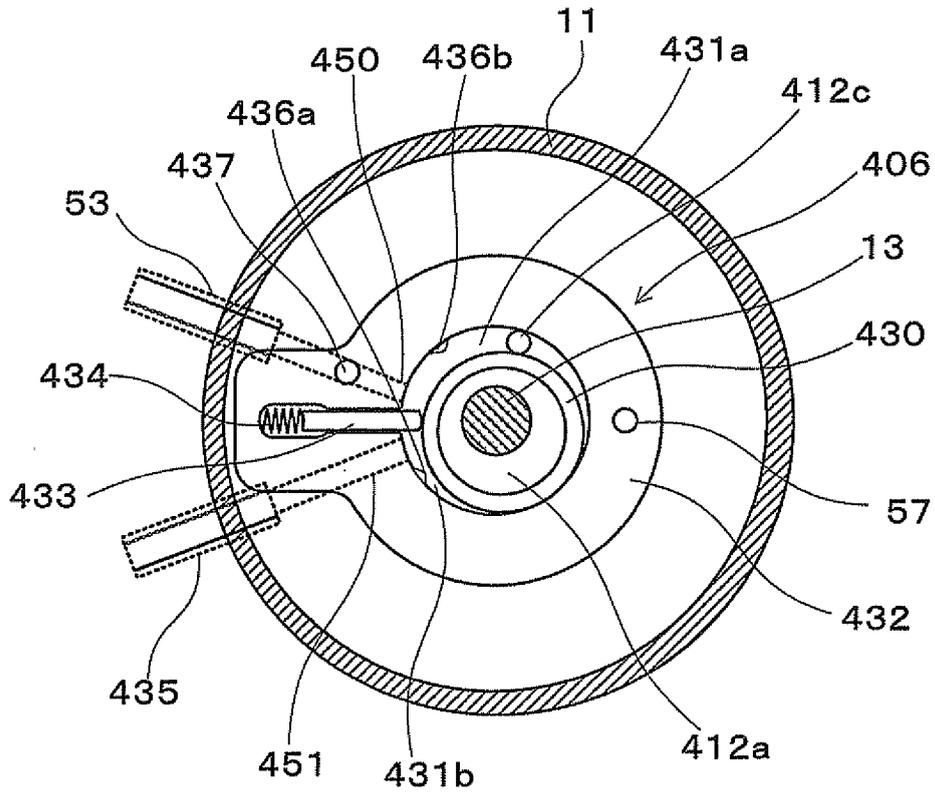


FIG. 12A

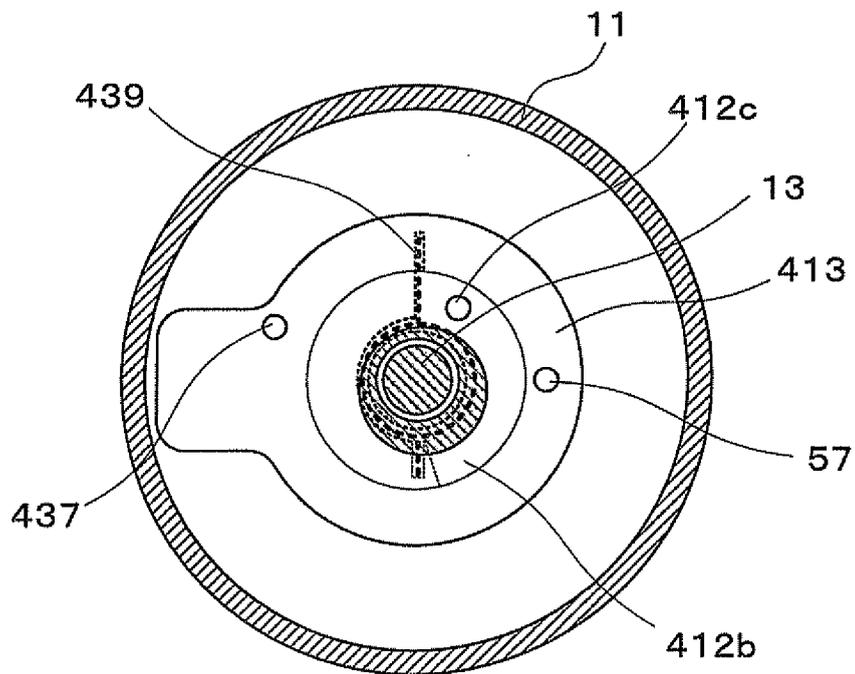


FIG. 12B

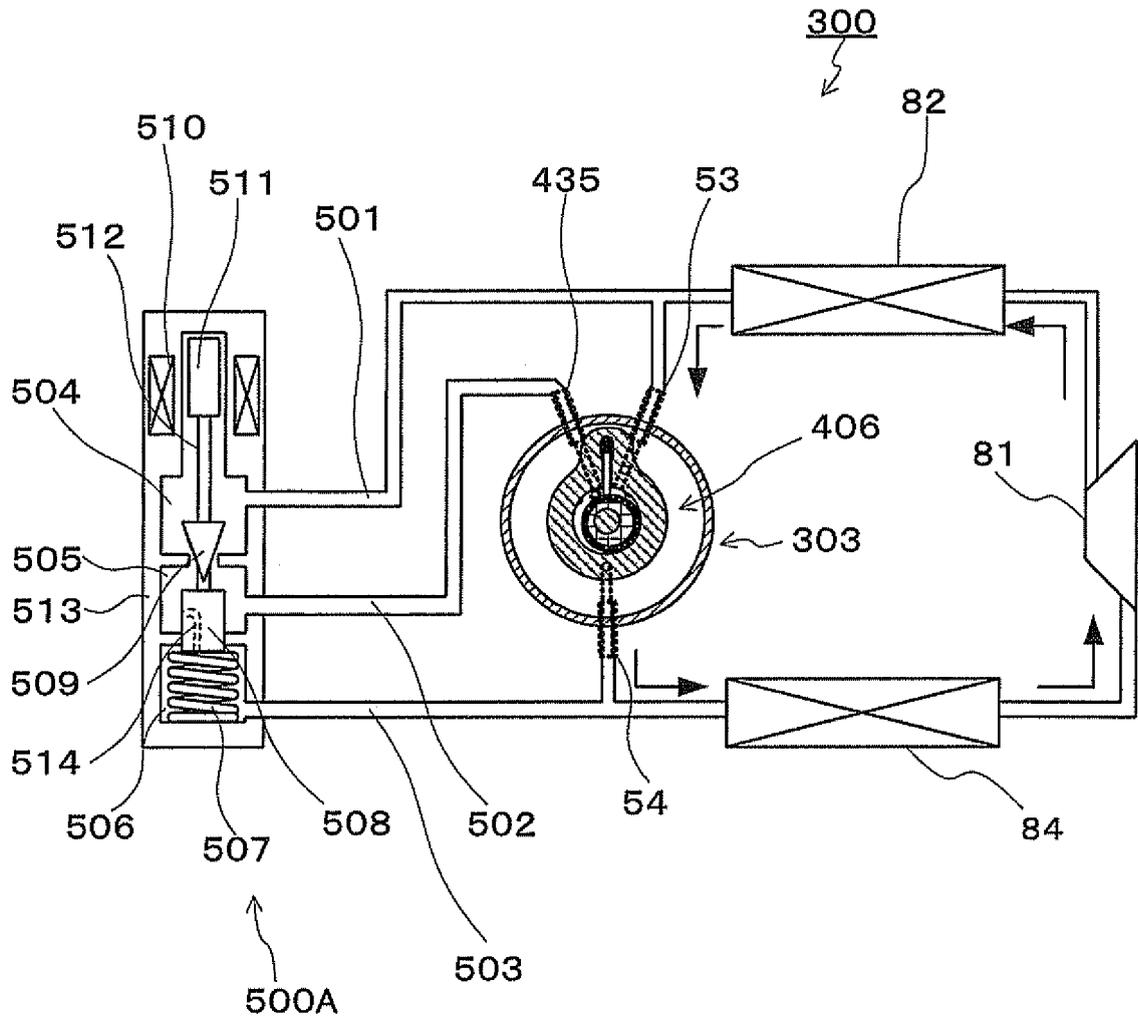


FIG. 13



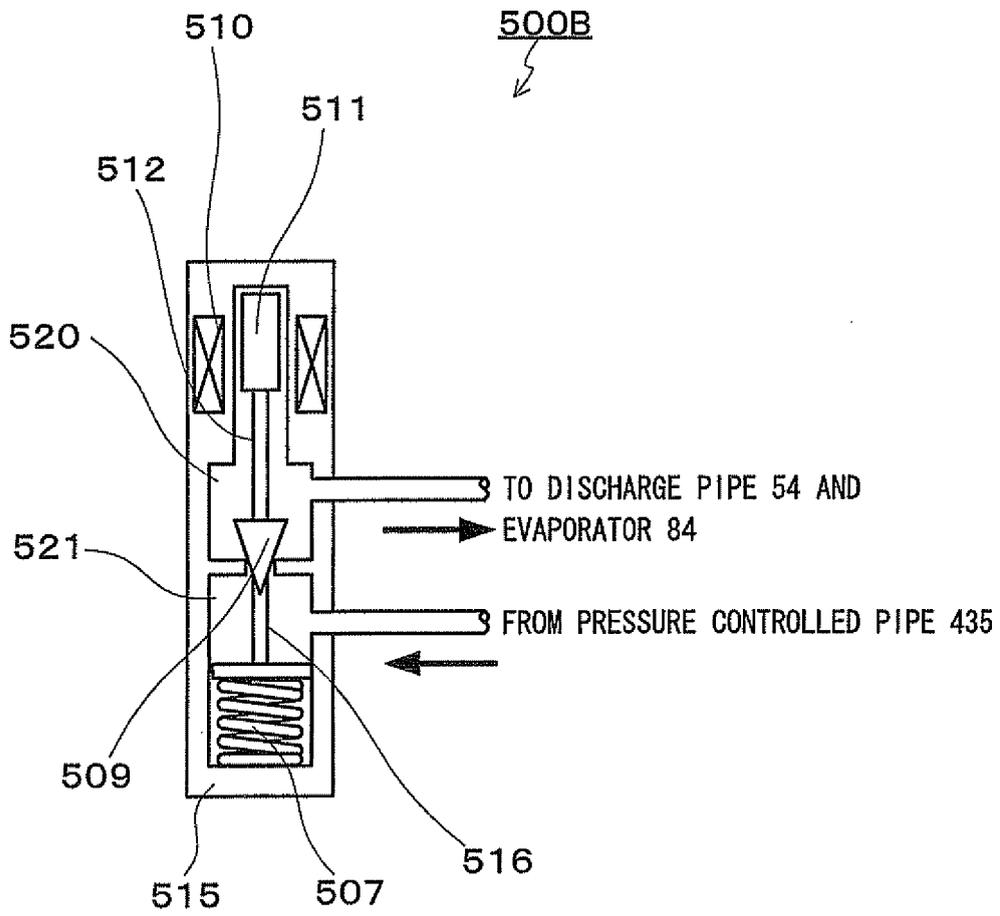


FIG. 15

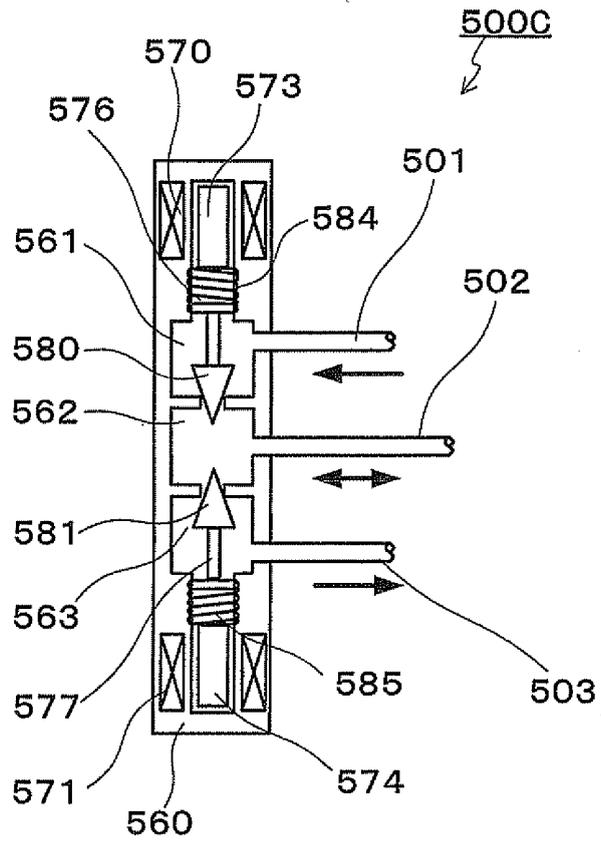


FIG. 16A

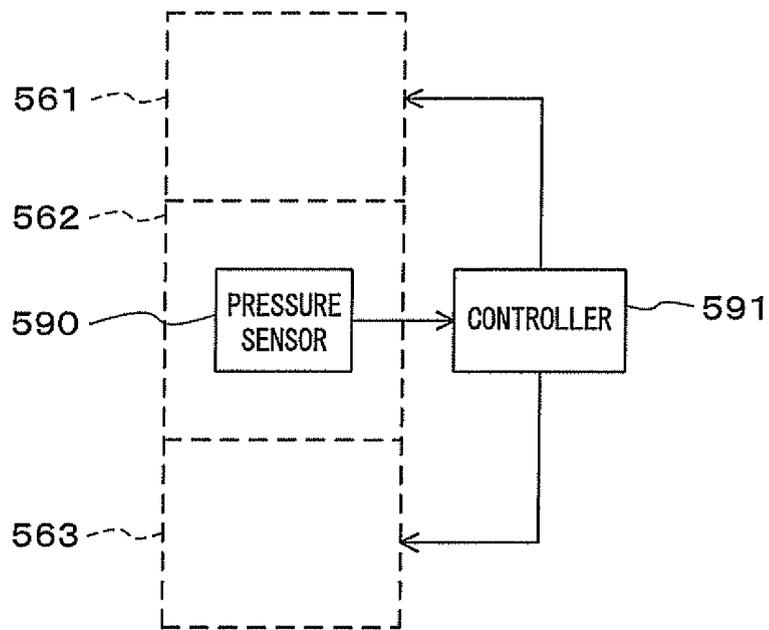


FIG. 16B

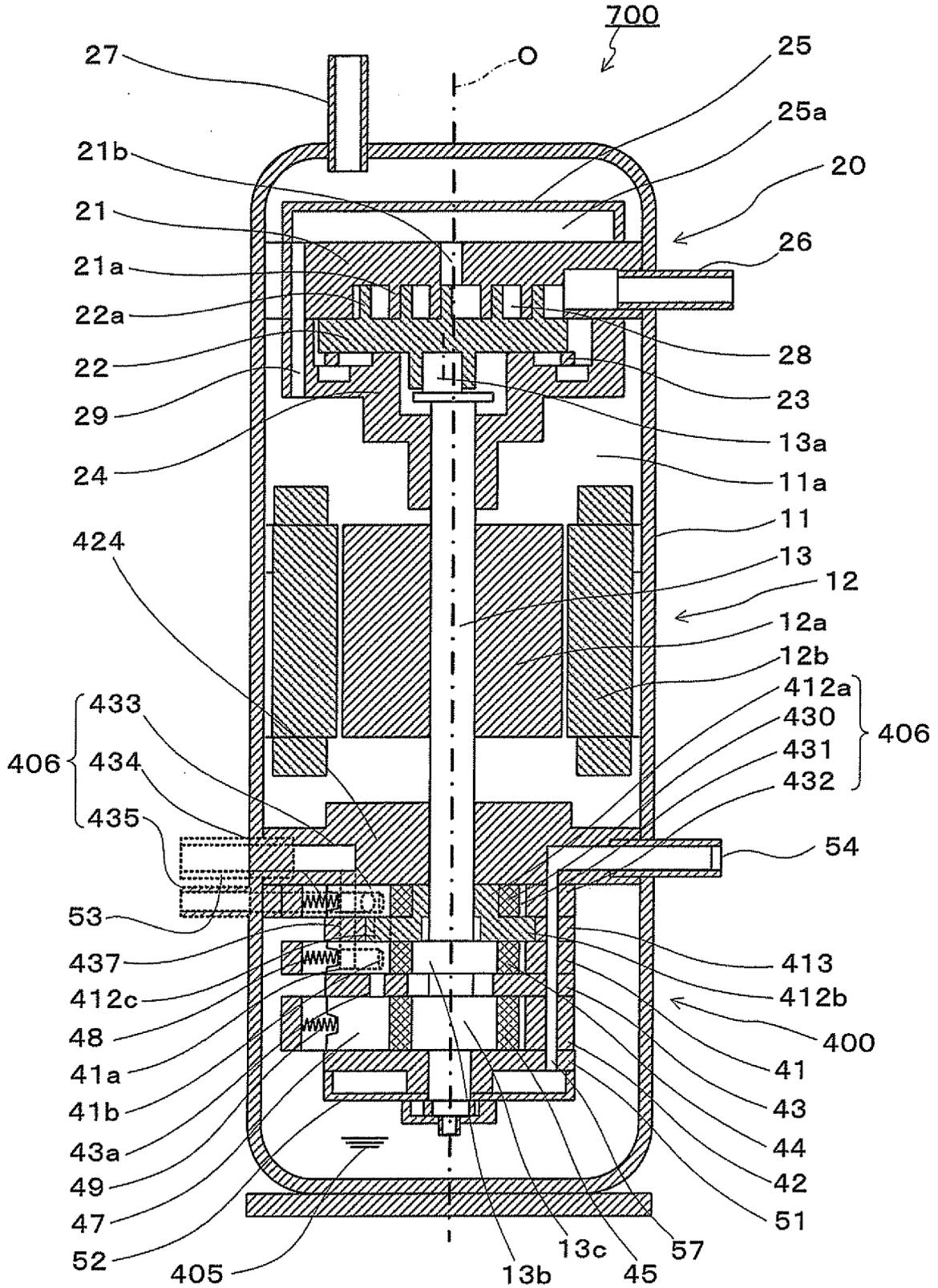


FIG. 17

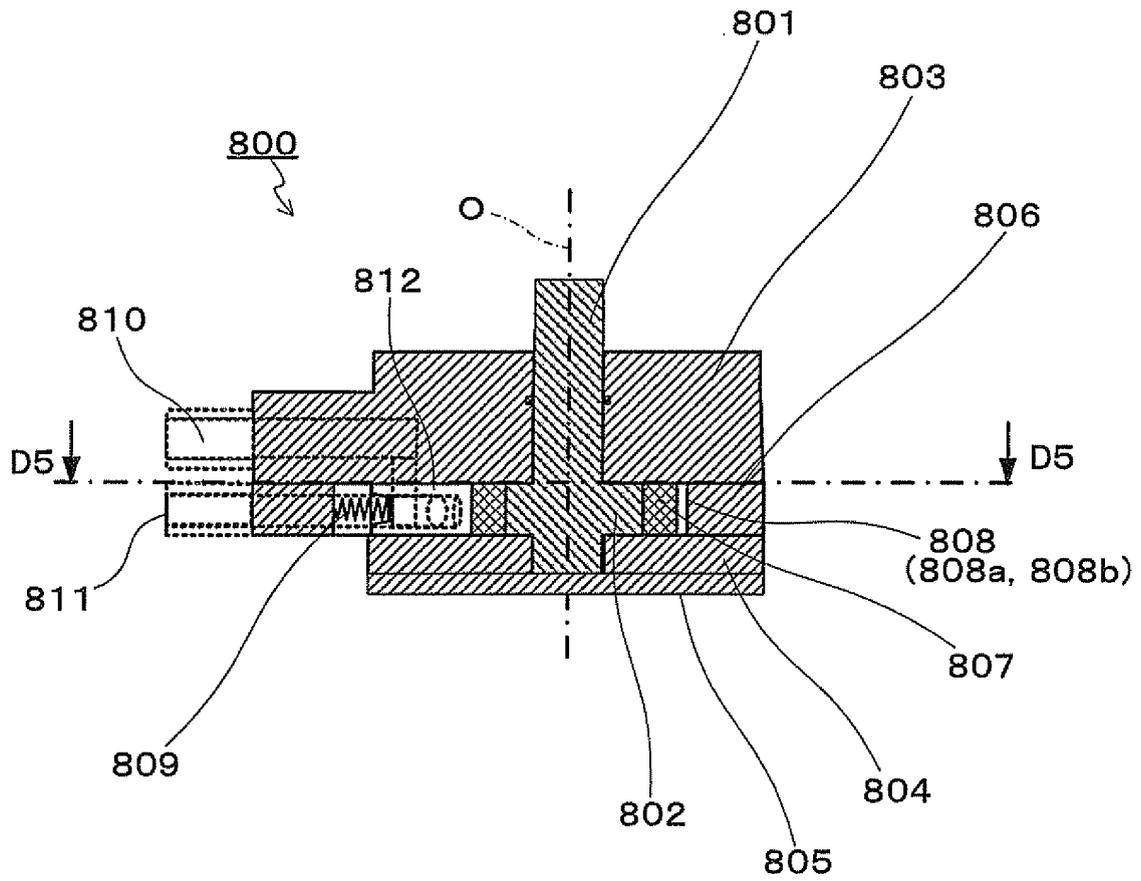


FIG. 18

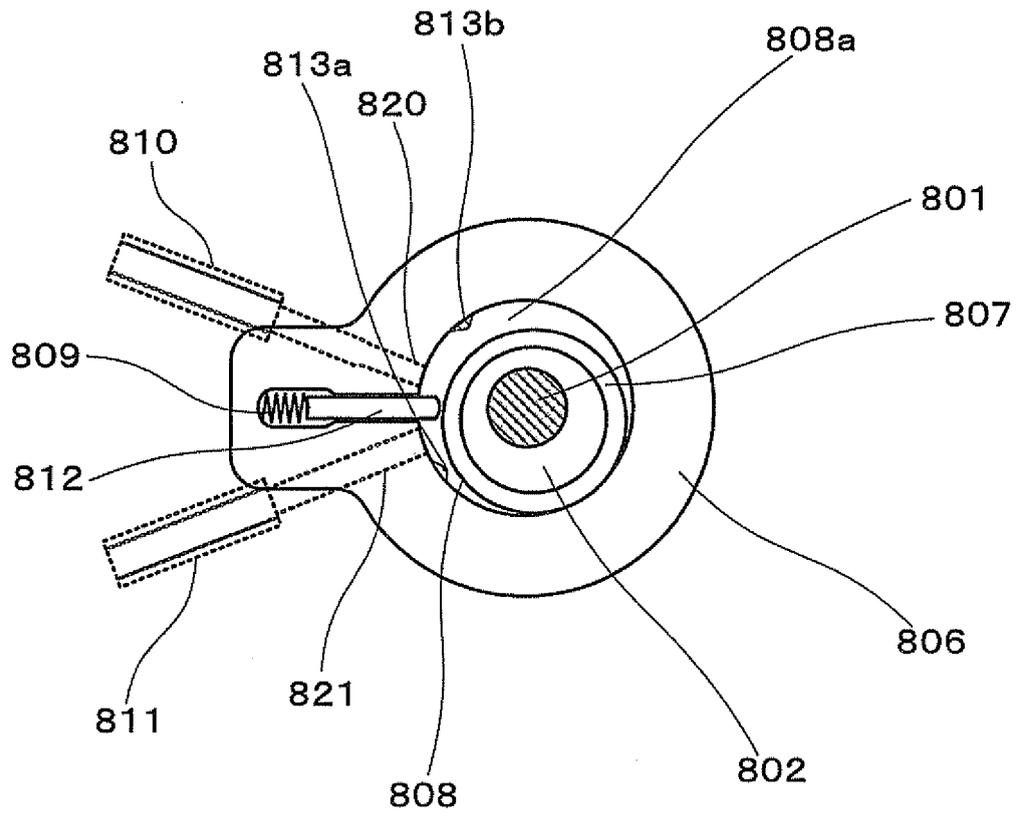


FIG. 19

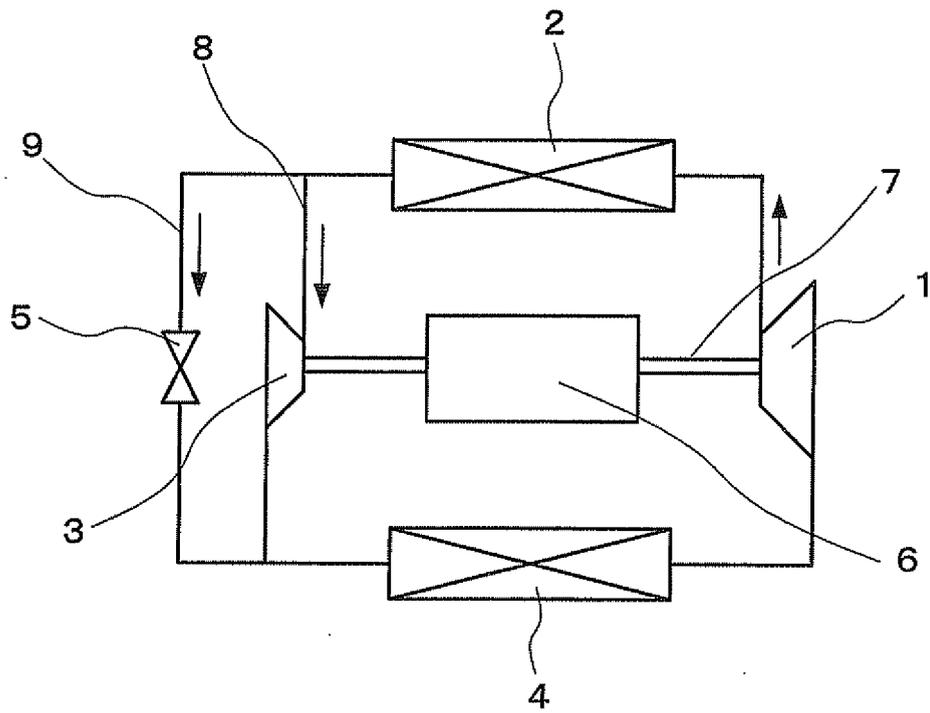


FIG. 20

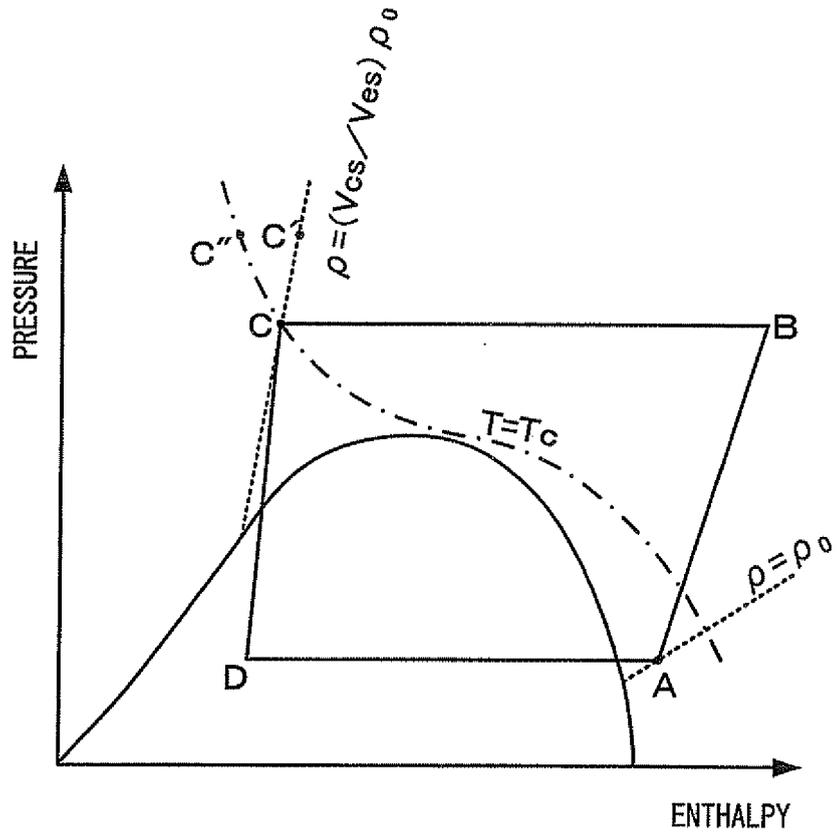


FIG. 21

## INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2006/321561

A. CLASSIFICATION OF SUBJECT MATTER F01C1/356(2006.01)i, F01C13/04(2006.01)i, F01C21/18(2006.01)i, F04C23/02(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, F01C20/10, F01C20/18, F01C20/24, F01C21/18, F04C23/02, F04C29/00, F03C2/00, 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.
A	JP 2005-256667 A (Daikin Industries, Ltd.), 22 September, 2005 (22.09.05), Claims 1, 2; Fig. 2 & WO 2005/088077 A1	1
A	JP 58-133401 A (Matsushita Electric Industrial Co., Ltd.), 09 August, 1983 (09.08.83), Fig. 3 (Family: none)	1
<input checked="" type="checkbox"/> Further documents are listed in the continuation of Box C. <input type="checkbox"/> See patent family annex.		
* Special categories of cited documents: "A" document defining the general state of the art which is not considered to be of particular relevance "E" earlier application or patent but published on or after the international filing date "L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified) "O" document referring to an oral disclosure, use, exhibition or other means "P" document published prior to the international filing date but later than the priority date claimed "T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention "X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone "Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art "&" document member of the same patent family		
Date of the actual completion of the international search 22 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.

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

INTERNATIONAL SEARCH REPORT

International application No.  
PCT/JP2006/321561

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT		
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 2571/1985 (Laid-open No. 118973/1986) (Shimadzu Corp.), 26 July, 1986 (26.07.86), Fig. 3 (Family: none)	1
A	JP 55-66601 A (Yoshiro HOSOYAMA), 20 May, 1980 (20.05.80), Full text; all drawings (Family: none)	1

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**REFERENCES CITED IN THE DESCRIPTION**

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