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# (54) REFRIGERATION CYCLE DEVICE

(57) A refrigeration cycle apparatus 100 includes a low-pressure compressor 113, a high-pressure compressor 101, a radiator 103, a gas-liquid separator 107, an expansion valve 109, an expander 105 and an evaporator 111. The low-pressure compressor 113 and the expander 105 are coupled by a shaft 116, and the low-pressure compressor 113 is driven using power recovered by the expander 105 from a refrigerant. The low-pressure compressor 113 and the high-pressure compressor 101 are serially connected by an intermediate-pressure flow path 114. The gas-liquid separator 107 and the intermediate-pressure flow path 114 are connected by the reciprocating flow path 115. The reciprocating flow path 115 is configured to allow the refrigerant to circulate bidirectionally. It is possible to regulate the refrigerant flow rate in the reciprocating flow path 115 by controlling the opening degree of the expansion valve 109.

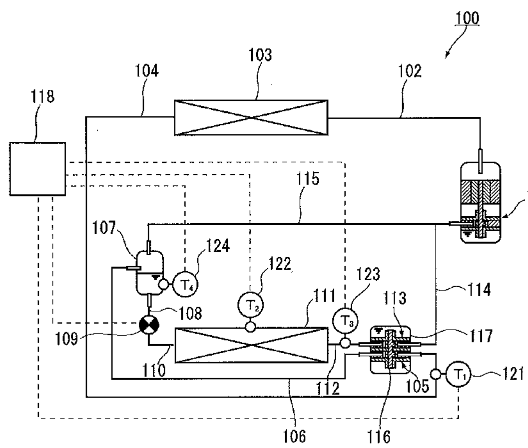


FIG.1

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## Description

TECHNICAL FIELD

**[0001]** The present invention relates to a refrigeration cycle apparatus.

BACKGROUND ART

**[0002]** As indicated in FIG. 13, a refrigeration cycle apparatus having a first compressor 801a, a radiator 802, an expander 803, a heat absorber 804 and a second compressor 801b is known to be used for an air-conditioner, a water heater or the like (Patent literature 1). The second compressor 801b and the expander 803 are coupled to each other by a rotary shaft 806, and the driving power for the second compressor 801b is supplied from the power generated at the time of the expansion of a refrigerant in the expander 803. This makes it possible to reduce the power to be consumed in the first compressor 801a for increasing the refrigerant pressure to a particular pressure.
**[0003]** In the refrigeration cycle apparatus illustrated in FIG. 13, the rotational speed of the expander 803 and the rotational speed of the second compressor 801b are consistent. In addition, the refrigerant that has been expanded in the expander 803 passes through the heat absorber 804 and is compressed in the second compressor 801b, so that the mass flow rate of the refrigerant passing through the expander 803 and the mass flow rate of the refrigerant passing through the second compressor 801b are consistent. Further, the suction volume of the expander 803 and the suction volume of the second compressor 801b each are determined at the time of design, and therefore there is a constraint on the suction refrigerant density of the expander 803 and the suction refrigerant density of the second compressor 801b.
**[0004]** That is, the product of the suction volume $V_{exi}$ of the expander 803 and the suction refrigerant density $\rho_{exi}$ of the expander 803 is always equal to the product of the suction volume $V_{C2i}$ of the second compressor 801b and the suction refrigerant density $\rho_{C2i}$ of the second compressor 801b. The relationship expressed by $(\rho_{exi}/\rho_{C2i}) = (V_{C2i}/V_{exi})$ is always valid between the suction refrigerant density of the expander 803 and the suction refrigerant density of the second compressor 801b. This relationship is referred to as a constraint for constant density ratio. In order to operate the refrigeration cycle apparatus with optimal efficiency, it is indispensable that the density $\rho_{exi}$ and the density $\rho_{C2i}$ are freely adjustable corresponding to external conditions such as season and weather. However, the constraint for constant density ratio makes it impossible to adjust the density $\rho_{exi}$ and the density $\rho_{C2i}$ freely, so that efficient operation is difficult to achieve.
**[0005]** In order to solve such a problem, a refrigeration cycle apparatus indicated in FIG. 14 is proposed (Patent literature 2). The refrigeration cycle apparatus indicated in FIG. 14 includes a two-stage compressor 903, a radiator 904, an expander 905, a gas-liquid separator 906, an evaporator 908, a gas-injection circuit 910 and a bypass circuit 911. The two-stage compressor 903 includes a low-pressure compressor 901 and a high-pressure compressor 902. The low-pressure compressor 901 and the expander 905 are coupled by a rotatable shaft. The bypass circuit 911 is provided with a flow-control valve 913. It is possible to avoid the constraint for constant density ratio by appropriately controlling the opening degree of the flow-control valve 913 and allowing a part of the refrigerant to flow through the bypass circuit 911.

CITATION LIST

Patent Literature

**[0006]** Patent literature 1: JP 2003-307358 A Patent literature 2: JP 2006-71257 A

SUMMARY OF INVENTION

Technical Problem

**[0007]** However, if a part of the refrigerant flows through the bypass circuit 911, the amount of the refrigerant that contributes to power recovery in expander 905 decreases, resulting in a problem of decreased power-recovery efficiency. This problem is more significant, for example, in the case of applying refrigeration cycle apparatuses with the same design respectively to a heat pump hot water floor heater and an air-conditioner. With respect to a $CO_2$ refrigeration cycle apparatus of a particular design, the inventors have calculated the density ratio $(\rho_{exi}/\rho_{C2i})$ at which the optimal efficiency can be achieved, and found it to be 7.13 at rated conditions of floor heating, 3.59 at rated conditions of cooling, and 2.98 at rated conditions of heating. Assuming that the low-pressure compressor 901 and the expander 905 are designed for floor heating, it is unavoidable to allow 49.6% of the refrigerant to flow through the bypass circuit 911 in cooling and 58.2% of the refrigerant to flow through the bypass circuit 911 in heating, and thus the power to be recovered in these cases is reduced to about half of that in floor heating.

**[0008]** It is an object of the present invention to provide a refrigeration cycle apparatus capable of efficient power recovery while avoiding the constraint for constant density ratio.

Solution to Problem

**[0009]** That is, the present invention provides a refrigeration cycle apparatus including: a positive displacement low-pressure compressor for pre-compressing a refrigerant; a high-pressure compressor for further compressing the refrigerant that has been pre-compressed in the low-pressure compressor; an intermediate-pressure flow path serially connecting the low-pressure compressor and the high-pressure compressor so as to allow the refrigerant that has been pre-compressed in the low-pressure compressor to be delivered to the high-pressure compressor; a radiator for cooling the refrigerant that has been compressed in the high-pressure compressor; a positive displacement expander for recovering power by allowing the refrigerant to expand, the expander being coaxially coupled to the low-pressure compressor for power transmission and configured to allow the entire amount of the refrigerant that has been cooled in the radiator to pass through itself; a gas-liquid separator for separating the refrigerant that has been expanded in the expander into gas refrigerant and liquid refrigerant; an evaporator for allowing the liquid refrigerant that has been separated in the gas-liquid separator to evaporate; an expansion valve with variable opening degree, the expansion valve being provided on a flow path between a liquid refrigerant outlet of the gas-liquid separator and an inlet of the evaporator; a reciprocating flow path connecting the intermediate-pressure flow path and the gas-liquid separator so as to allow switching between a first circulation state in which the refrigerant stored in the gas-liquid separator is introduced into an inlet of the high-pressure compressor without passing through the evaporator and the low-pressure compressor and a second circulation state in which a part of the refrigerant that has been pre-compressed in the low-pressure compressor flows back to the gas-liquid separator; and a controller for regulating the refrigerant flow rate in the reciprocating flow path in each state of the first circulation and the second circulation by controlling the opening degree of the expansion valve.

Advantageous Effects of Invention

**[0010]** According to the present invention, in the case where the suction volume of the low-pressure compressor is insufficient compared to the suction volume of the expander, a gas refrigerant is delivered from the gas-liquid separator to the intermediate-pressure flow path through the reciprocating flow path to be drawn into the high-pressure compressor. This establishes the balance of the flow rate in a refrigeration cycle. On the other hand, in the case where the suction volume of the expander is insufficient compared to the suction volume of the low-pressure compressor, a part of the gas refrigerant that has been pre-compressed in the low-pressure compressor is delivered to the gas-liquid separator through the intermediate-pressure flow path and the reciprocating flow path. This establishes the balance of the flow rate in a refrigeration cycle. Whatever the value (design value) of the ratio between the suction volume of the low-pressure compressor and the suction volume of the expander may be, the flow rate balance is established in a refrigeration cycle due to the function of the reciprocating flow path.

**[0011]** Meanwhile, the pressure inside the gas-liquid separator can be freely adjusted by an expansion valve. By adjusting the pressure inside the gas-liquid separator, it is possible to arbitrarily adjust the refrigerant pressure on the radiator side. For example, if the expansion valve is completely opened under arbitrary operational conditions, the pressure inside the gas-liquid separator is closest to the evaporation pressure of the refrigerant in the evaporator. Then, the pressure difference between the inlet and the outlet of the low-pressure compressor is closest to zero because the gas-liquid separator and the intermediate-pressure flow path are connected by the reciprocating flow path. That is, the compression work of the low-pressure compressor decreases. On the other hand, the pressure difference between the inlet and the outlet of the expander increases, so that the amount of power recovery in the expander increases. The rotational speed of each of the expander and the low-pressure compressor increases based on the relationship expressed by (the amount of power recovery) > (the compression work). This results in a decrease in the refrigerant pressure on the radiator side because the discharge refrigerant flow rate of the expander is excessive with respect to the discharge refrigerant flow rate of the high-pressure compressor. As a result, the amount of power recovery of the expander decreases to be balanced with the compression work of the low-pressure compressor, thereby stabilizing the refrigeration cycle. That is, it is possible to decrease the refrigerant pressure on the radiator side by opening the expansion valve.

**[0012]** Conversely, the pressure inside the gas-liquid separator is steadily increased by gradually closing the expansion valve. The pressure difference between the inlet and the outlet of the low-pressure compressor increases because the gas-liquid separator and the intermediate-pressure flow path are connected by the reciprocating flow path. That is, the compression work of the low-pressure compressor increases. On the other hand, the pressure difference between the inlet and the outlet of the expander decreases, so that the amount of power recovery of the expander decreases. The rotational speed of each of the expander and the low-pressure compressor decreases based on the relationship expressed by (the amount of power recovery) < (the compression work). This results in an increase in the refrigerant pressure on the radiator side because the discharge refrigerant flow rate of the expander falls short with respect to the discharge refrigerant flow rate of the high-pressure compressor. As a result, the amount of power recovery in the expander increases to be balanced with the compression work of the low-pressure compressor, thereby stabilizing the refrigeration cycle. That is, it is possible to increase the refrigerant pressure on the radiator side by closing the expansion valve.

**[0013]** In this way, it is possible always to adjust the refrigerant pressure on the radiator side optimally by controlling the opening degree of the expansion valve appropriately and thereby controlling the rotational speed of each of the expander and the low-pressure compressor. Moreover, the entire amount of the refrigerant passes through the expander, so that efficient power recovery is feasible. Even if the refrigerant flows back in the reciprocating flow path (the second circulation state) and a part of the recovered power is consumed for the circulation of the refrigerant, the present invention can achieve an improved energy budget compared to the conventional example (cf. FIG. 14) in which a refrigerant is allowed to flow into a bypass circuit. Accordingly, a refrigeration cycle apparatus including an expander and a low-pressure compressor at an appropriate volume ratio suitable for application can be operated under desirable pressure and temperature conditions in view of energy efficiency.

**[0014]** Further, the above-described theory is valid whatever the volume ratio between the low-pressure compressor and the expander may be. Therefore, according to the present invention, a low-pressure compressor and an expander can be designed so as to have an arbitrary volume ratio that can minimize their annual power consumption theoretically That is, the refrigeration cycle apparatus of the present invention has an enhanced degree of design freedom.

BRIEF DESCRIPTION OF DRAWINGS

**[0015]**

FIG. 1 is a configuration diagram indicating a refrigeration cycle apparatus according to a first embodiment of the present invention.
FIG. 2 is a configuration diagram indicating a multi-functional heat pump system.
FIG. 3 is a control flowchart of the first embodiment.
FIG. 4 is a Mollier diagram indicating the control of an intermediate pressure.
FIG. 5A is a Mollier diagram indicating a refrigeration cycle in a floor-heating cycle condition.
FIG. 5B is a Mollier diagram indicating the refrigeration cycle in a cooling cycle condition.
FIG. 5C is a Mollier diagram indicating the refrigeration cycle in a heating cycle condition.
FIG. 6A is a graph indicating each variation of cycle properties with respect to the variation of the volume ratio in the floor-heating cycle condition.
FIG. 6B is a graph indicating the variation of the discharge refrigerant temperature of a high-pressure compressor with respect to the variation of the volume ratio in the floor-heating cycle condition.
FIG. 7A is a graph indicating each variation of cycle properties with respect to the variation of the volume ratio in the cooling cycle condition.
FIG. 7B is a graph indicating the variation of the discharge refrigerant temperature of a high-pressure compressor with respect to the variation of the volume ratio in the cooling cycle condition.
FIG. 8A is a graph indicating each variation of cycle properties with respect to the variation of the volume ratio in the heating cycle condition.
FIG. 8B is a graph indicating the variation of the discharge refrigerant temperature of a high-pressure compressor with respect to the variation of the volume ratio in the heating cycle condition.
FIG. 9 is a configuration diagram indicating a refrigeration cycle apparatus according to a second embodiment of the present invention.
FIG. 10 is a control flowchart of the second embodiment.
FIG. 11 is a configuration diagram indicating a refrigeration cycle apparatus according to a third embodiment of the present invention.
FIG. 12A is a configuration diagram indicating a refrigeration cycle apparatus according to a fourth embodiment of the present invention.
FIG. 12B is a partially enlarged view of FIG. 12A indicating a detailed configuration of a two-stage rotary expander.
FIG. 13 is a configuration diagram indicating a conventional refrigeration cycle apparatus.
FIG. 14 is a configuration diagram indicating another conventional refrigeration cycle apparatus.

DESCRIPTION OF EMBODIMENTS

**[0016]** Hereinafter, embodiments of the present invention will be described with reference to the attached drawings.

First embodiment

**[0017]** As indicated in FIG. 1, a refrigeration cycle apparatus 100 includes a high-pressure compressor 101, a radiator 103, an expander 105, a gas-liquid separator 107, an expansion valve 109, an evaporator 111 and a low-pressure compressor 113.

**[0018]** The low-pressure compressor 113 pre-compresses gas refrigerant that has been evaporated in the evaporator 111. The high-pressure compressor 101 further compresses the refrigerant (working fluid) that has been pre-compressed in the low-pressure compressor 113. The expander 105 recovers power by allowing the refrigerant that has been cooled in the radiator 103 to expand. Further, the expander 105 is configured to allow the entire amount of the refrigerant that has been cooled in the radiator 103 to pass therethrough. That is, no bypass circuit is provided for allowing the refrigerant to flow bypassing the expander 105. Since the entire amount of the refrigerant contributes to power recovery, the effect of improving the COP (coefficient of performance) based on the power recovery is enhanced. It should be noted that although the entire amount of the refrigerant passes through the expander 105 in normal operation such as refrigeration and heating, there may be a case where the refrigerant does not pass through the expander 105 in a particular operation such as defrosting.

**[0019]** The low-pressure compressor 113 and the expander 105 each are constituted by a positive displacement fluid machine. The low-pressure compressor 113 and the expander 105 are coupled by a shaft 116 so that the power recovered from the refrigerant in the expander 105 can be transmitted to the low-pressure compressor 113, as well as being accommodated in a common closed casing 117. The low-pressure compressor 313 and the expander 105 each have a constant cylinder volume. Specifically, in this embodiment, the low-pressure compressor 113 and the expander 105 each have a constant suction volume. The suction volume of the low-pressure compressor 113 is larger than the suction volume of the expander 105. Conventionally, it is known that use of a variable displacement fluid machine makes it possible to avoid the constraint for constant density ratio. However, the configuration of such a variable displacement fluid machine is complicated, which causes an increase in cost. Therefore, it is preferable to use a fluid machine having a constant suction volume as a compressor or an expander, as is the case of this embodiment.

**[0020]** It should be noted that a "cylinder volume" means the volume of a working chamber (which is an expansion chamber or a compression chamber) at the time of completing the suction stroke, which often is referred to also as a "confined volume". A "suction volume" means the volume of refrigerant to be drawn into a compressor or an expander during one cycle (which means: suction, compression or expansion, and discharge) of the compressor or the expander. In this embodiment, the second embodiment and the third embodiment, the "cylinder volume" is equal to the "suction volume". However, as described in the fourth embodiment, there may be a case where high-pressure refrigerant is injected into the expansion chamber in the course of the expansion of the refrigerant. In this case, the volume (suction volume) of the refrigerant to be drawn into the expander during one cycle of the expander surpasses the cylinder volume.

**[0021]** Further, although a rotary compressor is used for the high-pressure compressor 101 in this embodiment, the type of the high-pressure compressor 101 is not limited in any way, and other positive displacement compressors such as a scroll compressor or other centrifugal compressors such as a turbo compressor may be used. The type of the low-pressure compressor 113 and the type of the expander 105 also are not limited to a rotary type, as long as they can be coupled to each other by the shaft 116 so that power can be transmitted. Other positive displacement fluid machines such as a scroll fluid machine may be used for the low-pressure compressor 113 and the expander 105.

**[0022]** The radiator 103 is provided for cooling the refrigerant that has been compressed in the high-pressure compressor 101, and typically is constituted by a water-refrigerant heat exchanger or an air-refrigerant heat exchanger. The gas-liquid separator 107 is provided for the separation of the refrigerant that has been expanded in the expander 105 into gas refrigerant and liquid refrigerant. The gas-liquid separator 107 is provided with a liquid refrigerant outlet at a bottom, a refrigerant inlet/outlet at an upper part and a refrigerant inlet at a lateral part. The evaporator 111 is provided for the evaporation of the liquid refrigerant that has been separated in the gas-liquid separator 107, and typically is constituted by an air-refrigerant heat exchanger. The expansion valve 109 is a valve that can vary its opening degree, such as an electric expansion valve, and is provided on a flow path between the liquid refrigerant outlet of the gas-liquid separator 107 and the inlet of the evaporator 111.

**[0023]** The above-mentioned devices are connected to one another by refrigerant pipes so as to form a refrigerant circuit. Specifically, the outlet of the high-pressure compressor 101 and the inlet of the radiator 103 are connected by a refrigerant pipe 102. The outlet of the radiator 103 and the inlet of the expander 105 are connected by a refrigerant pipe 104. The outlet of the expander 105 and the inlet of the gas-liquid separator 107 are connected by a refrigerant pipe 106. The liquid refrigerant outlet of the gas-liquid separator 107 and the inlet of the expansion valve 109 are connected by a refrigerant pipe 108. The outlet of the expansion valve 109 and the inlet of the evaporator 111 are connected by a refrigerant pipe 110. The outlet of the evaporator 111 and the inlet of the low-pressure compressor 113 are connected by a refrigerant pipe 112. The outlet of the low-pressure compressor 113 and the inlet of the high-pressure compressor 101 are connected by a refrigerant pipe 114.

**[0024]** Further, the refrigerant inlet/outlet located at an upper part of the gas-liquid separator 107 and the refrigerant pipe 114 are connected by a refrigerant pipe 115. Hereinafter in this description, the flow path formed by the refrigerant pipe 114 is referred to as an intermediate-pressure flow path 114, and the flow path formed by the refrigerant pipe 115 is referred to as a reciprocating flow path 115, respectively.

**[0025]** The reciprocating flow path 115 is not provided with a valve for determining the circulation direction of refrigerant or regulating the flow rate of refrigerant. Accordingly, in the case where the pressure inside the gas-liquid separator 107 is higher than the discharge refrigerant pressure of the low-pressure compressor 113, the refrigerant flows from the gas-liquid separator 107 to the intermediate-pressure flow path 114 through the reciprocating flow path 115 (the first circulation state: injection). In other words, the refrigerant that has been separated in the gas-liquid separator 107 is introduced into the inlet of the high-pressure compressor 101 without passing through the evaporator 111 and the low-pressure compressor 113. On the other hand, in the case where the pressure inside the gas-liquid separator 107 is lower than the discharge refrigerant pressure of the low-pressure compressor 113, the refrigerant flows from the intermediate-pressure flow path 114 to the gas-liquid separator 107 through the reciprocating flow path 115. In other words, a part of the refrigerant that has been pre-compressed in the low-pressure compressor 113 flows back into the gas-liquid separator 107 (the second circulation state: flowback). In this way, the reciprocating flow path 115 is configured to allow the refrigerant to circulate bidirectionally, that is, in the direction from the gas-liquid separator 107 to the intermediate-pressure flow path 114 and the direction from the intermediate-pressure flow path 114 to the gas-liquid separator 107.

**[0026]** The refrigeration cycle apparatus 100 further includes a first temperature sensor 121, a second temperature sensor 122, a third temperature sensor 123, a fourth temperature sensor 124 and a controller 118. The first temperature sensor 121 detects the suction refrigerant temperature of the expander 105. The second temperature sensor 122 detects the refrigerant temperature in the evaporator 111. The third temperature sensor 123 detects the suction refrigerant temperature of the low-pressure compressor 113. The fourth temperature sensor 124 detects the refrigerant temperature in the gas-liquid separator 107. Specific examples of each temperature sensor include a thermocouple and a thermistor. Each temperature sensor is connected to the controller 118. Specific examples of the controller 118 include a DSP (digital signal processor). The controller 118 controls the opening degree of the expansion valve 109 based on signals obtained from each temperature sensor.

**[0027]** In the following description, the suction volume of the low-pressure compressor 113 is denoted by $V_{lc}$, the suction volume of the expander 105 is denoted by $V_{ex}$, the volume ratio between the low-pressure compressor 113 and the expander 105 is denoted by $(V_{lc}/V_{ex})$, the suction refrigerant density of the low-pressure compressor 113 is denoted by $\rho_{lci}$, the suction refrigerant density of the expander 105 is denoted by $\rho_{exi}$, and the degree of dryness of the discharge refrigerant of the expander 105 is denoted by $Q_{exo}$. Further, the pressure inside the gas-liquid separator 107 is referred to as the intermediate pressure.

**[0028]** In this embodiment, the low-pressure compressor 113 and the expander 105 are designed so that the volume ratio $(V_{lc}/V_{ex})$ is equal to or more than a value obtained from $(1- Q_{exo}) \times (\rho_{exi}/\rho_{lci})$ but not more than the density ratio $(\rho_{exi}/\rho_{lci})$.

**[0029]** First, if the volume ratio $(V_{lc}/V_{ex})$ is not more than the density ratio $(\rho_{exi}/\rho_{lci})$, the mass flow rate of the refrigerant in the low-pressure compressor 113 falls below the mass flow rate of the refrigerant in the expander 105, and therefore the refrigerant flows from the gas-liquid separator 107 to the intermediate-pressure flow path 114 through the reciprocating flow path 115 (injection). In this case, the refrigerant that has passed through the reciprocating flow path 115 does not have to be compressed, thereby reducing the load of the low-pressure compressor 113. Further, the flow rate of the refrigerant passing through the evaporator 111 is reduced, so that the pressure loss that occurs when the refrigerant passes through the evaporator 111 is reduced.

**[0030]** Conversely, if the volume ratio $(V_{lc}/V_{ex})$ exceeds the density ratio $(\rho_{exi}/\rho_{lci})$, the mass flow rate of the refrigerant in the low-pressure compressor 113 is larger than the mass flow rate of the refrigerant in the expander 105, and therefore the refrigerant flows from the intermediate-pressure flow path 114 to the gas-liquid separator 107 through the reciprocating flow path 115 (flowback). In this case, the refrigerant that has been pre-compressed in the low-pressure compressor 113 is expanded again in the expansion valve 109. The recovery power of the expander 105 is consumed for the circulation of the refrigerant, thereby decreasing the effect of improving the COP.

**[0031]** For these reasons, it is desirable to satisfy the relationship expressed as follows in designing, so as to prevent flowback from occurring. In other words, when operation is carried out so as to cause injection, the following relationship is satisfied: $(V_{lc}/V_{ex}) \leq (\rho_{exi}/\rho_{lci})$.

**[0032]** On the other hand, if the volume ratio $(V_{lc}/V_{ex})$ is equal to or more than a value obtained from $(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci})$, the ratio of the refrigerant flowing from the gas-liquid separator 107 to the intermediate-pressure flow path 114 through the reciprocating flow path 115 is equal to or less than the degree of dryness $Q_{exo}$ of the discharge refrigerant of the expander 105. That is, only gas refrigerant is injected into the high-pressure compressor 101. Assuming that the mass flow rate of the refrigerant in the expander 105 is $(V_{ex} \times \rho_{exi})$ and the mass flow rate of the refrigerant in the low-pressure compressor 113 is $(V_{lc} \times \rho_{lci})$, the ratio $R_i$ of the refrigerant flowing from the gas-liquid separator 107 to the intermediate-pressure flow path 114 through the reciprocating flow path 115 can be expressed as: $(V_{ex} \times \rho_{exi} - V_{lc} \times \rho_{lci})/(V_{ex} \times \rho_{exi})$. The volume ratio $(V_{lc}/V_{ex})$ in the case of the ratio $R_i$ being less than the degree of dryness $Q_{exo}$ is larger than a value obtained from $(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci})$. Therefore, if the volume ratio $(V_{lc}/V_{ex})$ is equal to or more than a value obtained from $(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci})$, the ratio $R_i$ of the refrigerant flowing from the gas-liquid separator 107 to the intermediate-pressure flow path 114 through the reciprocating flow path 115 does not exceed the degree of dryness $Q_{exo}$ of the discharge refrigerant of the expander 105, and thus the injection of liquid refrigerant can be avoided. In other words, when operation is carried out so as to prevent liquid injection, the following relationship is satisfied: $(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci}) \leq (V_{lc}/V_{ex})$.

**[0033]** As described above, according to this embodiment, when operation is carried out so as to cause gas injection, the following relationship is satisfied: $(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci}) \leq (V_{lc}/V_{ex}) \leq (\rho_{exi}/\rho_{lci})$.

**[0034]** It should be noted that, as described below, in the case where the refrigeration cycle apparatuses 100 having the same design are used for two or more different applications, or the single refrigeration cycle apparatus 100 is used for two or more applications, there may be a case where flowback is allowed intentionally. If the injection of liquid refrigerant occurs, the actual COP possibly falls below the COP in the case without power recovery. Therefore, the injection of liquid refrigerant should be avoided, On the other hand, even in the case of flowback, the COP of the refrigeration cycle apparatus 100 theoretically never falls below the COP in the case without power recovery.

**[0035]** Specific examples of the applications of the refrigeration cycle apparatus 100 includes heat pump water heaters and air-conditioners. Some heat pump water heaters have a water-heating function for supplying heated water to a tap and/or a floor-heating function for heating indoor space by circulating heated water in a pipe running throughout the floor of a house. Air-conditioners are configured to adjust the indoor temperature by heat exchange between indoor air and refrigerant, and typically have a cooling function and a heating function.

**[0036]** The inventors calculated the volume ratio that enables the annual power consumption to be reduced sufficiently, in the case where the refrigeration cycle apparatus 100 is applied as a heat pump water heater or an air-conditioner. Specifically, it was assumed that the outdoor air temperature was 7°C, the return temperature of the hot water for floor heating was 25°C, the suction refrigerant temperature of the low-pressure compressor 113 was 7°C, and the refrigerant was carbon dioxide in the floor-heating condition of the heat pump water heater. A desirable volume ratio $(V_{lc}/V_{ex})$ resulting from the relationship between the degree of dryness $Q_{exo}$ and the density ratio $(\rho_{exi}/\rho_{lci})$ at an intermediate pressure determined with respect to an arbitrary volume ratio was 4.7 to 7.1.

**[0037]** It was assumed that the outdoor air temperature was 35°C, the suction air temperature of the indoor equipment (evaporator 111) was 27°C, the suction refrigerant temperature of the low-pressure compressor 113 was 27°C, and the refrigerant was carbon dioxide in the cooling condition of the air-conditioner. A desirable volume ratio $(V_{lc}/V_{ex})$ resulting from the relationship between the degree of dryness $Q_{exo}$ and the density ratio $(\rho_{exi}/\rho_{lci})$ at an intermediate pressure determined with respect to an arbitrary volume ratio was 2.4 to 3.6.

**[0038]** It was assumed that the outdoor air temperature was 7°C, the suction air temperature of the indoor equipment (radiator 103) was 20°C, the suction refrigerant temperature of the low-pressure compressor 113 was 7°C, and the refrigerant was carbon dioxide in the heating condition of the air-conditioner. A desirable volume ratio $(V_{lc}/V_{ex})$ resulting from the relationship between the degree of dryness $Q_{exo}$ and the density ratio $(\rho_{exi}/\rho_{lci})$ at an intermediate pressure determined with respect to an arbitrary volume ratio was 2.1. to 2.9.

**[0039]** Here, if the volume ratio $(V_{lc}/V_{ex})$ falls below a value obtained from $(1- Q_{exo}) \times (\rho_{exi}/\rho_{lci})$, the injection of the liquid refrigerant occurs, so that the enthalpy of the suction refrigerant of the high-pressure compressor 101 decreases considerably. As a result, the discharge refrigerant temperature of the high-pressure compressor 101 decreases, so that the heating performance necessary for the floor-heating function of the heat pump water heater or the heating function of the air-conditioner becomes insufficient. In addition, the liquid refrigerant that is to pass originally through the evaporator 111 passes through the reciprocating flow path 115, thereby decreasing the cooling performance necessary for the cooling function of the air-conditioner. Accordingly, in the case where the refrigeration cycle apparatus 100 is applied to a plurality of applications, it is preferable to set the volume ratio $(V_{lc}/V_{ex})$ to at least a value to be obtained from $(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci})$ in each condition and a value that can prevent flowback as much as possible. The value is 4.7 in the above-mentioned example.

**[0040]** In the case where the refrigeration cycle apparatuses 100 each are designed to be dedicated to one of the applications of a heat pump water heater, a cooling air-conditioner and heating air-conditioner, an appropriate volume ratio for the application can be set. However, in the case where the refrigeration cycle apparatus 100 is used for a multi-functional heat pump system as indicated in FIG. 2, there is a problem in selecting the volume ratio. In the above-mentioned example, by selecting the desirable volume ratio 4.7 in the floor-heating condition, it is possible to decrease the flowback rate as much as possible while surely avoiding the injection of the liquid refrigerant. Of course, since an optimal volume ratio varies depending on external conditions such as season and weather, even when applying the present invention to a refrigeration cycle apparatus for a single application, the benefits thereof can be enjoyed sufficiently.

**[0041]** It should be noted that the multi-function heat pump system indicated in FIG. 2 includes a heat pump water heater 12 with a floor-heating function and an air-conditioner 14, and the refrigeration cycle apparatus 100 is used commonly for these water heater 12 and air-conditioner 14. However, the radiator 103 (FIG. 1) is provided exclusively for each of the water heater 12 and the air-conditioner 14.

**[0042]** Next, the operation of the refrigeration cycle apparatus 100 will be described.

**[0043]** First, at the time of the start, the expansion valve 109 is closed completely. Next, power supply to the motor of the high-pressure compressor 101 is started, thereby driving the high-pressure compressor 101. The high-pressure compressor 101 draws the refrigerant of the intermediate-pressure flow path 114 to compress it. The compressed refrigerant passes through the refrigerant pipe 102, the radiator 103 and refrigerant pipe 104 to be delivered to the expander 105. The inside of the refrigerant pipe 104 on the inlet side of the expander 105 is filled with the refrigerant that has been discharged from the high-pressure compressor 101. Therefore, the pressure inside the refrigerant pipe 104 increases. Further, the high-pressure compressor 101 draws the refrigerant from the gas-liquid separator 107 through the reciprocating flow path 115, thereby causing the liquid refrigerant inside the gas-liquid separator 107 to evaporate. Therefore, the temperature and the pressure inside the refrigerant pipe 106 on the outlet side of the expander 105 decrease. That is, if the high-pressure compressor 101 is operated with the expansion valve 109 being closed, the pressure difference between the inlet and the outlet of the expander 105 increases. The pressure difference thus generated drives the expander 105.

**[0044]** Meanwhile, the pressure inside the refrigerant pipe 114 on the outlet side of the low-pressure compressor 113 decreases because the high-pressure compressor 101, the reciprocating flow path 115 and the gas-liquid separator 107 are interconnected via the refrigerant pipe 114 (intermediate-pressure flow path). Further, the inside of the refrigerant pipe 112 on the inlet side of the low-pressure compressor 113 is filled with the refrigerant having an evaporation pressure corresponding to the atmosphere temperature (heat source temperature) of the place where the evaporator 111 is provided (for example, outdoor air). For this reason, at the time of the start, the pressure inside the refrigerant pipe 112 temporarily surpasses the pressure inside the intermediate-pressure flow path 114. Then, the low-pressure compressor 113 acts as an expander, which is driven by the pressure difference between the refrigerant pipe 112 and the intermediate-pressure flow path 114.

**[0045]** As described referring to FIG. 13, a refrigeration cycle apparatus having a low-pressure compressor and an expander that are coupled by a shaft is conventionally known. However, in the case where a rotary expander is used for the expander 803 of the conventional refrigeration cycle apparatus indicated in FIG. 13, its piston possibly stops at an eccentric position on the vane side, that is, the piston stops in the state where the suction port and the discharge port communicate with each other. In such a case, a sufficient initial pressure difference necessary for driving the low-pressure compressor 801b and the expander 803 cannot be obtained, thereby causing a problem of the occurrence of start error. In contrast, according to the starting method of this embodiment, the low-pressure compressor 113 temporarily acts as an expander. Therefore, a design where the piston of the expander 105 and the piston of the low-pressure compressor 113 each are not decentered to the vane side at the same time is possible. That is, by differing the eccentric directions of the pistons from each other, it is possible to generate surely the pressure difference necessary for driving between the suction side and the discharge side of at least one of the expander 105 and the low-pressure compressor 113. This makes it possible to ensure the start of the refrigeration cycle apparatus.

**[0046]** After the low-pressure compressor 113 and the expander 105 start operation, the low-pressure compressor 113 is driven by the recovery power of the expander 105. The low-pressure compressor 113 draws the refrigerant from the refrigerant pipe 112, the evaporator 111 and refrigerant pipe 110. This allows the liquid refrigerant to start evaporating in the evaporator 111, so that the temperature and the pressure inside the evaporator 111 decrease. If the pressure inside the refrigerant pipe 110 falls below the pressure inside the refrigerant pipe 108, the opening degree of the expansion valve 109 is increased gradually until the initial value. In this embodiment, the expansion valve 109 is opened at the time when the detected temperature of the second temperature sensor 122 falls below the detected temperature of the fourth temperature sensor 124.

**[0047]** Thereafter, the controller 118 controls the opening degree of the expansion valve 109. Specifically, it adjusts the pressure inside the gas-liquid separator 107 so that the theoretical recovery power of the expander 105 and the theoretical compression work of the low-pressure compressor 113 are equal in the target cycle condition that has been determined based on the refrigerant evaporation pressure in the evaporator 111, the suction refrigerant temperature of the expander 105, the discharge refrigerant pressure of the high-pressure compressor 101 and the suction refrigerant temperature of the low-pressure compressor 113. This control is carried out in order to match the actual high pressure in the refrigerant circuit with the optimal high pressure in the target cycle condition. The pressure (intermediate pressure) inside the gas-liquid separator 107 can be adjusted by the expansion valve 109. It should be noted that the theoretical recovery power and the theoretical compression work each are a value to be obtained by calculation, and they do not mean the actual recovery power and the actual compression work.

**[0048]** Further detailed description will be given with reference to the flowchart in FIG. 3.

**[0049]** First, in step 101, the suction refrigerant temperature $T_1$ of the expander 105 is obtained from the first temperature sensor 121, the refrigerant evaporation temperature $T_2$ in the evaporator 111 is obtained from the second temperature sensor 122, and the suction refrigerant temperature $T_3$ of the low-pressure compressor 113 is obtained from the third temperature sensor 123. The refrigerant evaporation pressure in the evaporator 111 can be obtained from the refrigerant evaporation temperature $T_2$ in the evaporator 111.

**[0050]** Next, in step 102, the optimal high pressure at which the COP of the refrigeration cycle apparatus 100 is maximized is calculated based on the temperature and the pressure obtained in step 101.

**[0051]** Next, in step 103 and step 104, the target intermediate pressure at which the theoretical recovery power and the theoretical compression work are equal is calculated. First, a certain target intermediate pressure is set in step 103. The recovery power (theoretical recovery power) in the case of expanding the refrigerant in the expander 105 until the set target intermediate pressure is calculated based on the calculated optimal high pressure and the suction refrigerant temperature $T_1$ of the expander 105. As indicated in FIG. 4, the state of the refrigerant at the inlet of the expander 105 is represented by point D. Point D can be specified by the optimal high pressure $P_H$ and the suction refrigerant temperature $T_1$. The target intermediate pressure $P_M$ is a pressure at point E. In the expander 105, the refrigerant is expanded along the isentropic curve (from point D to point E). The theoretical recovery power can be obtained by multiplying the efficiency of the expander 105 by the enthalpy ($h_2 \cdot h_1$) that the refrigerant has lost in the transition process from point D to point E.

**[0052]** Further, in step 103, the compression work (theoretical compression work) in the case of compressing the refrigerant in the low-pressure compressor 113 until the set target intermediate pressure is calculated based on the evaporation pressure $P_L$ of the evaporator 111 and the suction refrigerant temperature $T_3$ of the low-pressure compressor 113. As indicated in FIG. 4, the state of the refrigerant at the inlet of the low-pressure compressor 113 is represented by point A. Point A is specified by the evaporation pressure $P_L$ and the suction refrigerant temperature $T_3$. In the low-pressure compressor 113, the refrigerant is pre-compressed along the isentropic curve (from point A to point B). The theoretical compression work can be obtained by dividing the enthalpy ($h_4$-$h_3$) that the refrigerant has gained in the transition process from point A to point B by the efficiency of the low-pressure compressor 113, and further multiplying the ratio of the mass flow rate of the refrigerant in the low-pressure compressor 113 with respect to the mass flow rate of the refrigerant in the expander 105 by it.

**[0053]** It should be noted that the mass flow rate of the refrigerant in the expander 105 can be calculated from the refrigerant density at the inlet of the expander 105 and the suction volume of the expander 105. The refrigerant density at the inlet of the expander 105 can be calculated, for example, from the optimal high pressure and the suction refrigerant temperature $T_1$. Similarly, the mass flow rate of the refrigerant in the low-pressure compressor 113 can be calculated from the refrigerant density at the inlet of the low-pressure compressor 113 and the suction volume of the low-pressure compressor 113. The refrigerant density at the inlet of the low-pressure compressor 113 can be calculated, for example, from the evaporation temperature $T_2$ and the suction refrigerant temperature $T_3$. Further, the efficiencies of the expander 105 and the low-pressure compressor 113 each are a design value.

**[0054]** Next, in step 104, whether or not the theoretical recovery power and the theoretical compression work match is determined. If they match, the process proceeds to step 105. If they do not match, the process returns to step 103. While another target intermediate pressure is set, the processes of step 103 and step 104 are repeated until the theoretical recovery power and the theoretical compression work match. In this way, the controller 118 calculates an arbitrary optimal high pressure $P_H$ and an arbitrary target intermediate pressure $P_M$ based on the detection results of each temperature sensor.

**[0055]** Next, in step 105, the pressure inside the gas-liquid separator 107 (actual intermediate pressure) is calculated. Specifically, first, the refrigerant evaporation temperature $T_4$ in the gas-liquid separator 107 is obtained from the fourth temperature sensor 124. The refrigerant pressure can be calculated from the refrigerant evaporation temperature $T_4$. That is, the controller 108 that serves as a means for controlling the opening degree of the expansion valve 109 calculates the actual pressure inside the gas-liquid separator 107 based on the detection results of the fourth temperature sensor 124.

**[0056]** Next, in step 106, the actual intermediate pressure and the target intermediate pressure $P_M$ are compared. If the actual intermediate pressure exceeds the target intermediate pressure $P_M$, the process proceeds to step 107. If it falls therebelow, the process proceeds to step 107'. In step 107, the set opening degree of the expansion valve 109 is increased. In step 107', the set opening degree of the expansion valve 109 is decreased.

**[0057]** Next, in step 108, the set opening degree is output to the expansion valve 109, thereby causing the opening degree of the expansion valve 109 to vary. The variation of the opening degree of the expansion valve 109 causes the variation of the pressure inside the gas-liquid separator 107 as well. By periodically carrying out the process shown in this flowchart, it is possible to adjust the pressure inside the gas-liquid separator 107 as well as maintaining the optimal high pressure so that the recovery power of the expander 105 and the compression work of the low-pressure compressor 113 are theoretically equal.

**[0058]** As described above, the controller 118 includes a means for calculating the target intermediate pressure $P_M$ at which the theoretical recovery power of the expander 105 at an arbitrary optimal high pressure of the refrigeration cycle and the theoretical compression work of the low-pressure compressor 113 at the arbitrary optimal high pressure are equal, and a means for controlling the opening degree of the expansion valve 109 so that the actual pressure inside the gas-liquid separator 107 approaches the calculated target intermediate pressure $P_M$. Specifically, the controller 118 controls the opening degree of the expansion valve 109 based on the detection results of the fourth temperature sensor 124 so that the pressure inside the gas-liquid separator 107 and the target intermediate pressure $P_M$ are consistent.

**[0059]** Further, in the case where the volume ratio ($V_{lc}/V_{ex}$) is excessive with respect to the cycle conditions of the refrigeration cycle apparatus 100, each mass flow rate of the refrigerant in the high-pressure compressor 101 and the expander 105 is insufficient with respect to the mass flow rate of the refrigerant in the low-pressure compressor 113. In other words, the suction amount into the high-pressure compressor 101 falls short with respect to the discharge amount from the low-pressure compressor 113. For this reason, the pressure of the intermediate-pressure flow path 114 increases, so that the refrigerant that cannot be drawn into the high-pressure compressor 101 flows back from the intermediate-pressure flow path 114 to the gas-liquid separator 107 through the reciprocating flow path 115.

**[0060]** On the other hand, in the case where the volume ratio ($V_{lc}/V_{ex}$) falls short with respect to the cycle conditions of the refrigeration cycle apparatus 100, each of the mass flow rate of the refrigerant in the high-pressure compressor 101 and the expander 105 is excessive with respect to the mass flow rate of the refrigerant in the low-pressure compressor 113. In other words, the suction amount into the high-pressure compressor 101 is excessive with respect to the discharge amount from the low-pressure compressor 113. For this reason, the pressure of the intermediate-pressure flow path 114 decreases, so that the shortage of the refrigerant is injected from the gas-liquid separator 107 to the intermediate-pressure flow path 114 through the reciprocating flow path 115.

**[0061]** The operation of the refrigeration cycle apparatus 100 in each application will be described with reference to the Mollier diagrams indicated in FIGs. 5A to 5C. In this regard, the volume ratio ($V_{lc}/V_{ex}$) is assumed to be set to 4.7.

**[0062]** As indicated in FIG. 5A, the volume ratio 4.7 matches the value expressed by $(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci})$ in the aforedescribed floor-heating condition. Accordingly, the entire gas refrigerant in the gas-liquid separator 107 is injected to the intermediate-pressure flow path 114 through the reciprocating flow path 115, and the liquid refrigerant is delivered to the evaporator 111 through the expansion valve 109.

**[0063]** As indicated in FIG. 5B, the volume ratio 4.7 exceeds the value expressed by $(\rho_{exi}/\rho_{lci})$ in the aforedescribed cooling condition. That is, the mass flow rate of the refrigerant in the low-pressure compressor 113 is excessive with respect to the mass flow rate of the refrigerant in the expander 105. For this reason, a part of the refrigerant that has been compressed in the low-pressure compressor 113 flows back to the gas-liquid separator 107 through the reciprocating flow path 115, and is expanded again in the expansion valve 109.

**[0064]** As indicated in FIG. 5C, the volume ratio 4.7 exceeds the value expressed by $(\rho_{exi}/\rho_{lci})$ in the aforedescribed heating condition in the same manner as in the cooling condition. For this reason, a part of the refrigerant that has been compressed in the low·pressure compressor 113 flows back to the gas-liquid separator 107 through the reciprocating flow path 115, and is expanded again in the expansion valve 109.

**[0065]** As described above, according to this embodiment, the refrigerant can circulate bidirectionally in the reciprocating flow path 115, thereby balancing each flow rate in the refrigeration cycle. This allows the low-pressure compressor 113 and the expander 105 to be designed so that the minimum annual power consumption can be achieved regardless of the constraint for constant density ratio. Further, it is possible to adjust the intermediate pressure easily by controlling the opening degree of the expansion valve 109 and to operate the refrigeration cycle apparatus 100 so that the actual high pressure in the refrigerant circuit matches the optimal high pressure whatever the volume ratio ($V_{lc}/V_{ex}$) may be.

**[0066]** Further, it is possible to avoid the injection of the liquid refrigerant by setting the volume ratio ($V_{lc}/V_{ex}$) to the value expressed by $(1\ Q_{exo}) \times (\rho_{exi}/\rho_{lci})$ or more.

**[0067]** FIG. 6A and FIG. 6B each are a graph indicating each variation of cycle properties (calculated value) with respect to the variation of the volume ratio in the floor-heating cycle condition (the outdoor air temperature: 7°C, the return temperature of hot water for floor heating: 25°C, the suction refrigerant temperature of the low-pressure compressor: 7°C, and the refrigerant: $CO_2$) The vertical axis in FIG. 6A indicates the intermediate pressure, the COP and the refrigerant flow rate in the reciprocating flow path as cycle properties. The vertical axis in FIG. 6B indicates the discharge refrigerant temperature of the high-pressure compressor 101 as a cycle property

**[0068]** In the above-mentioned the floor-heating cycle condition, the sign of the refrigerant flow rate in the reciprocating flow path 115 switches at the border of the volume ratio ($V_{lc}/V_{ex}$) = 7.1, as indicated in the graph of FIG. 6A. During a positive refrigerant flow rate, the refrigerant flows from the gas-liquid separator 107 to the intermediate-pressure flow path 114 (injection). During a negative refrigerant flow rate, the refrigerant flows from the intermediate-pressure flow path 114 to the gas-liquid separator 107 (flowback). If gas injection occurs, both the COP and the intermediate pressure increase. This is because the compression load of the low-pressure compressor 113 is reduced due to a decrease of the mass flow rate of the refrigerant in the low-pressure compressor 113, resulting in a reduction of the compression load of the high-pressure compressor 101.

**[0069]** Further, in the above-mentioned the floor-heating cycle condition, the discharge refrigerant temperature of the high-pressure compressor 101 sharply decreases from the border of the volume ratio ($V_{lc}/V_{ex}$) = 4.7 as indicated in FIG. 6B. That is, switching between the injection of the liquid refrigerant (including the gas refrigerant) and the injection of the gas refrigerant occurs at the volume ratio ($V_{lc}/V_{ex}$) = 4.7. As aforedescribed, in the case where the value expressed by (1 - Qexo) $\times$ $(\rho_{exi}/\rho_{lci})$ is less than the volume ratio ($V_{lc}/V_{ex}$), the injection of the liquid refrigerant does not occur. Conversely, in the case where the value expressed by (1

Qexo) $\times$ $(\rho_{exi}/\rho_{lci})$ is greater than the volume ratio ($V_{lc}/V_{ex}$), the injection of the liquid refrigerant occurs.

**[0070]** Further, the discharge refrigerant temperature of the high-pressure compressor 101 stops varying at the border of the volume ratio ($V_{lc}/V_{ex}$) = 7.1. That is, switching between the injection of the gas refrigerant and the flowback occurs at the volume ratio ($V_{lc}/V_{ex}$) = 7.1. As aforedescribed, if the density ratio ($\rho_{exi}/\rho_{lci}$) is greater than the volume ratio ($V_{lc}/V_{ex}$), the flowback does not occur. Conversely, in the case where the density ratio ($\rho_{exi}/\rho_{lci}$) is less than the volume ratio ($V_{lc}/V_{ex}$), the flowback occurs.

**[0071]** As seen from above, assuming that operation is carried out in the floor-heating cycle condition, the volume ratio ($V_{lc}/V_{ex}$) may be set in the range of 4.7 to 7.1 for avoiding the injection of the liquid refrigerant and the flowback.

**[0072]** As described referring to FIG. 2, in the case of a multi-function heat pump system, it also should be considered that the single refrigeration cycle apparatus 100 supplies water heating, floor heating and air-conditioning, from the viewpoint of cost or the like. In such a case, by taking only the application for floor heating into consideration in setting the volume ratio, there is a possibility that efficient operation cannot be achieved in other applications. Specifically, as indicated in FIG. 7A and FIG. 7B, it is possible to avoid the injection of the liquid refrigerant and the flowback by setting the volume ratio ($V_{lc}/V_{ex}$) in the range of 2.4 to 3.6 in the cooling cycle condition. Similarly, as indicated in FIG. 8A and FIG. 8B, it is possible to avoid the injection of the liquid refrigerant and the flowback by setting the volume ratio ($V_{lc}/V_{ex}$) in the range of 2.1 to 2.9 in the heating cycle condition.

In this way, each suitable range of the volume ratio differs depending on the cycle condition. Although the flowback may be acceptable, the injection of the liquid refrigerant should be avoided. Therefore, the volume ratio ($V_{lc}/V_{ex}$) = 4.7 at which the liquid injection can be avoided in the floor-heating cycle condition is adequate in the present example.

**[0073]** If the volume ratio ($V_{lc}/V_{ex}$)is set to 4.7, the volume ratio ($V_{lc}/V_{ex}$) is equal to or less than the density ratio ($\rho_{exi}/\rho_{lci}$) in the floor-heating condition, so that all the refrigerant that has been compressed by the low-pressure compressor 113 is drawn into the high-pressure compressor 101. As a result, the refrigeration cycle apparatus 100 can be operated efficiently. The value expressed by (1- $Q_{exo}$) $\times$ ($\rho_{exi}/\rho_{lci}$ is less than the volume ratio ($V_{lc}/V_{ex}$) not only in the floor-heating condition but also in each application for cooling and heating, thus allowing the injection of the liquid refrigerant to be avoided.

**[0074]** Further, in the case of the volume ratio ($V_{lc}/V_{ex}$) = 4.7, in the cooling condition, 23.6% of the refrigerant that has been compressed in the low-pressure compressor 113 returns to the gas-liquid separator 107 through the reciprocating flow path 115 to expand again in the expansion valve 109. If calculated in the same cooling condition for the conventional example (FIG. 14) in which the constraint for constant density ratio is avoided using a bypass circuit, 49.6% of the refrigerant bypasses the expander. Similarly, in the heating condition, 36.6% of the refrigerant that has been compressed in the low-pressure compressor 113 returns to the gas-liquid separator 107 through the reciprocating flow path 115 to expand again in the expansion valve 109. If calculated in the same heating condition for the conventional example (FIG. 14) in which the constraint for constant density ratio is avoided using a bypass circuit, 58.2% of the refrigerant bypasses the expander. Thus, according to the refrigeration cycle apparatus 100 of this embodiment, it is possible to achieve more efficient operation than in the conventional refrigeration cycle apparatus including a bypass circuit.

**[0075]** It should be noted that the injecion of the liquid refrigerant is not intended to be completely inhibited in the present invention.

Second embodiment

**[0076]** FIG. 9 is a configuration diagram indicating a refrigeration cycle apparatus according to the second embodiment of the present invention. A refrigeration cycle apparatus 500 of this embodiment has a configuration similar to that of the refrigeration cycle apparatus 100 according to the first embodiment (see FIG. 1). This embodiment differs from the first embodiment in that a temperature sensor 520 is provided and in how the controller 118 carries out control. Hereinafter, the same functional components each are denoted by the same referential numeral, and the description thereof is omitted.

**[0077]** As indicated in FIG. 9, the refrigeration cycle apparatus 500 includes the temperature sensor 520 for detecting the discharge refrigerant temperature of the high-pressure compressor 101. In the same manner as in the first embodiment, the temperature sensor 122 also is provided for detecting the refrigerant evaporation temperature in the evaporator 111. The controller 118 controls the opening degree of the expansion valve 109 based on the detection results of the temperature sensor 520 and the temperature sensor 122.

**[0078]** Referring to the flowchart in FIG. 10, the operation of the refrigeration cycle apparatus 500 will be described. First, in step 501, the outdoor air temperature is estimated based on the refrigerant evaporation temperature in the evaporator 111. Next, in step 502, the target discharge refrigerant temperature of the high-pressure compressor 101 is calculated. The target discharge refrigerant temperature is determined corresponding to, for example, the outdoor air temperature and the set temperature of a floor heater (or the set temperature of a heater). Next, in step 503, the actual discharge refrigerant temperature of the high-pressure compressor 101 is obtained from the temperature sensor 520. Next, in step 504, the actual discharge refrigerant temperature and the target discharge refrigerant temperature are compared. If the actual discharge refrigerant temperature is higher than the target discharge refrigerant temperature, the process proceeds to step 505. If the actual discharge refrigerant temperature is lower than the target discharge refrigerant temperature, the process proceeds to step 505'.

**[0079]** In step 505 and step 506, the set opening degree of the expansion valve 109 is increased, so that the intermediate pressure is decreased. The decrease of the intermediate pressure causes a reduction in the compression work of the low-pressure compressor 113. This causes an imbalance between the compression work of the low-pressure compressor 113 and the recovery power of the expander 105. In order to eliminate the imbalance, the rotational speed of the shaft 116 increases, and the high pressure of the refrigeration cycle decreases. As a result, the recovery power of the expander 105 is reduced, so that the imbalance is eliminated. Further, the decrease of the high pressure of the refrigeration cycle causes a decrease in the discharge refrigerant temperature of the high-pressure compressor 101.

**[0080]** In step 505' and step 506, the set opening degree of the expansion valve 109 is reduced, so that the intermediate pressure is increased. If the intermediate pressure increases, the compression work of the low-pressure compressor 113 increases. This causes an imbalance between the compression work of the low-pressure compressor 113 and the recovery power of the expander 105. In order to eliminate the imbalance, the rotational speed of the shaft 116 decreases, and the high pressure of the refrigeration cycle increases. As a result, the recovery power of the expander 105 increases, so that the imbalance is eliminated. Further, the increase of the high pressure of the refrigeration cycle causes an increase in the discharge refrigerant temperature of the high-pressure compressor 101.

**[0081]** According to this embodiment, the opening degree of the expansion valve 109 is controlled based on the detection results of the temperature sensors 520 and 123. It is possible to adjust the intermediate pressure by controlling the opening degree of the expansion valve 109. The rotational speed of the shaft 116 varies corresponding to the intermediate pressure. As the rotational speed of the shaft 116 varies, the high pressure of the refrigeration cycle also varies. That is, it is possible to adjust the high pressure of the refrigeration cycle by the expansion valve 109. Therefore, the refrigeration cycle apparatus 500 of this embodiment is suitable for applications where heating performance is required, such as floor heating, air heating and water heating.

## Third embodiment

**[0082]** FIG. 11 is a configuration diagram indicating a refrigeration cycle apparatus according to the third embodiment of the present invention. A refrigeration cycle apparatus 700 has a configuration similar to that of the refrigeration cycle apparatuses described in the first embodiment and the second embodiment. This embodiment differs from the first embodiment in that a high-pressure compressor 701, a low-pressure compressor 713 and an expander 705 are accommodated in a common closed casing 717.

**[0083]** As indicated in FIG. 11, in the refrigeration cycle apparatus 700, the high-pressure compressor 701, the low-pressure compressor 713 and the expander 705 are disposed in the single closed casing 717 from above in this order. The low-pressure compressor 718 and the expander 705 are connected by a shaft 716 so as to be capable of transmitting power. In the bottom of the closed casing 717, oil is stored. The space above the oil level is filled with a discharge refrigerant from the high-pressure compressor 701. The space around each of the low-pressure compressor 713 and the expander 705 is filled with oil.

**[0084]** Upon the high-pressure compressor 701 being driven, the space above the oil level is filled with the discharge refrigerant at high pressure. In the space around the high-pressure compressor 701, the oil that has lubricated the high-pressure compressor 701 at high temperature is held. Meanwhile, the low-pressure compressor 713 and the expander 705 operate at lower temperature than the high-pressure compressor 701. For this reason, oil at lower temperature compared to oil present in the space around the high-pressure compressor 713 is held in the space around the low-pressure compressor 713 or the expander 705.

**[0085]** That is, oil at high temperature is held in the space around the high-pressure compressor 701, and oil at low temperature is held in the space around the low-pressure compressor 713 and the expander 705. The oil forms a thermal stratification along the vertical direction. The formation of the thermal stratification makes it difficult for the oil in the upper layer and the oil in the lower layer to mix with each other. Thus, it is possible to prevent heat transfer from the high-pressure compressor 701 to the expander 705 via the oil. The occurrence of heat transfer causes a decrease in the discharge refrigerant temperature of the high-pressure compressor 701 and an increase in the discharge refrigerant temperature of the expander 705, which is not preferable in view of the efficiency of the refrigeration cycle apparatus. According to this embodiment, heat transfer can be prevented effectively, and therefore the efficiency of the refrigeration cycle apparatus 700 can be improved further.

## Fourth embodiment

**[0086]** FIG. 12A is a configuration diagram indicating a refrigeration cycle apparatus according to the fourth embodiment of the present invention. A refrigeration cycle apparatus 600 of this embodiment employs a multi-stage rotary type expander 605 capable of changing its suction volume. Further, the refrigeration cycle apparatus 600 includes an expander injection path 630 and an expander injection valve 631. As indicated in FIG. 12B, the expander injection path 630 connects the suction path (pipe 104) of the expander 605 and the expander injection port 632 opening into an expansion chamber 611 of the expander 605. The expander injection valve 631 is provided on the expander injection path 630. It is possible to change the suction volume of the expander 605 by controlling the expander injection valve 631. Other configurations are as described in the first embodiment and the second embodiment.

**[0087]** As indicated in FIG. 12B, the expander 605 includes a two-stage rotary expander having a first-stage cylinder 605a and a second-stage cylinder 605b. The expander injection port 632 is provided in the first-stage cylinder 605a and opens into the expansion chamber 611 of the first-stage cylinder 605a. Although the first-stage cylinder 605a and the second-stage cylinder 605b are separated by an intermediate plate 605c, the expansion chamber 611 of the first-stage cylinder 605a communicates into an expansion chamber 612 of the second-stage cylinder 605b through a communication hole 605d formed in the intermediate plate 605c. Thus, the expansion chambers 611 and 612 form a single expansion chamber. The expander injection path 630 branching from the pipe 104 is connected to the first-stage cylinder 605a so as to be capable of injecting the refrigerant from the expander injection port 632 into the expansion chamber 611. The expander injection port 632 is provided in the vicinity of the communication hole 605d along the circumferential direction of the shaft 116.

**[0088]** When the expander injection valve 631 is closed, the refrigerant is not allowed to inflow from the expander injection port 632 to the expansion chamber 611. Thus, the cylinder volume of the first-stage cylinder 605a serves as the suction volume $V_{ex}$. On the other hand, when the expander injection valve 631 is opened, the refrigerant is allowed to inflow from the expander injection port 632 to the expansion chamber 611. Thus, the cylinder volume of the second-stage cylinder 605b serves as the suction volume $V_{ex}$'. For example, assuming that the cylinder volume of the first-stage cylinder 605a is twice the cylinder volume of the second-stage cylinder 605b, the suction volume $V_{ex}$ can be changed into the double suction volume $V_{ex}$', as needed.

**[0089]** According to the first embodiment, the desirable volume ratio ($V_{lc}/V_{ex}$) in the cooling condition is in the range of 2.4 to 3.6. In this embodiment, the desirable volume ratio ($V_{lc}/V_{ex}$') in the same cooling condition is in the range of 2.4 to 3.6, using the suction volume $V_{ex}$' with the expander injection valve 631 being opened. Using the suction volume $V_{ex}$, the desirable volume ratio ($V_{lc}/V_{ex}$) in the same cooling condition is in the range of 4.8 to 7.2. Further, according to the first embodiment, the desirable volume ratio ($V_{lc}/V_{ex}$) in the heating condition is in the range of 2,1 to 2.9. In this embodiment, the desirable volume ratio ($V_{lc}/V_{ex}$') in the same heating condition is in the range of 2.1 to 2.9, using the suction volume $V_{ex}$' with the expander injection valve 631 being opened. Using the suction volume $V_{ex}$, the desirable volume ratio ($V_{lc}/V_{ex}$) in the same heating condition is in the range of 4.2 to 5.8. Further, according to the first embodiment, the desirable volume ratio ($V_{lc}/V_{ex}$)in the floor-heating condition is in the range of 4.7 to 7.1. In this embodiment, the desirable volume ratio ($V_{lc}/V_{ex}$) in the same floor-heating condition also is in the range of 4.7 to 7.1, using the suction volume $V_{ex}$ with the expander injection valve 631 being closed.

**[0090]** That is, in the cooling condition and the heating condition, the expander injection valve 631 is opened so that the suction volume is increased. On the other hand, in the floor-heating condition, the expander injection valve 631 is closed. Such control enables the desirable volume ratio ($V_{lc}/V_{ex}$) in each condition to approximate one another.

**[0091]** Specifically, the low-pressure compressor 113 and the expander 605 can be designed so that the volume ratio ($V_{lc}/V_{ex}$) is 4.8. In such a design, the volume ratio ($V_{lc}/V_{ex}$) of 4.8 is lower than the upper limit of the volume ratio ($V_{lc}/V_{ex}$) in each condition of the cooling, heating and floor-heating. Therefore, no flowback occurs in each condition of the cooling, heating and floor-heating, so that the compression work in the low-pressure compressor 113 can be used effectively. Further, the volume ratio ($V_{lc}/V_{ex}$) of 4.8 in the design is higher than the lower limit of the volume ratio ($V_{lc}/V_{ex}$) in each condition of the cooling, heating and floor-heating. Therefore, the injection of the liquid refrigerant also can be prevented.

**[0092]** Accordingly, the gas refrigerant flows from the gas-liquid separator 107 to the intermediate-pressure flow path 114 through the reciprocating flow path 115. It is possible to operate the refrigeration cycle apparatus 600 in a steady gas-injection state. In the gas-injection state operation, the low-pressure compressor 113 does not have to compress the refrigerant that has flowed through the reciprocating flow path 115, resulting in a reduction in the load of the low-pressure compressor 113. Further, the entire amount of the refrigerant that has been cooled in the radiator 103 passes through the expander 605, so that efficient power recovery can be achieved. As seen from above, according to the refrigeration cycle apparatus 600 of this embodiment, the operation can be carried out more efficiently than in the conventional refrigeration cycle apparatus having a bypass circuit.

**[0093]** It should be noted that other types of expanders, such as a scroll expander and a reciprocating expander, can be used as the expander 605. The expander injection valve 631. may be a valve capable of changing its opening degree in multiple stages, or may be a simple on-off valve.

## INDUSTRIAL APPLICABILITY

**[0094]** The present invention is useful for refrigeration cycle apparatuses to be used for air-conditioners, refrigerator-freezers, heat pump water heaters, heat pump heaters, vending machines, vehicle air-conditioners, and the like. Above all, when using a single refrigeration cycle apparatus commonly for two or more applications, a greater effect can be obtained. Of course, the present invention can be employed suitably also for a refrigeration cycle apparatus for a single application.

## Claims

**1.** A refrigeration cycle apparatus comprising:

a positive displacement low-pressure compressor for pre-compressing a refrigerant,
a high-pressure compressor for further compressing the refrigerant that has been pre-compressed in the low-pressure compressor;
an intermediate-pressure flow path serially connecting the low-pressure compressor and the high-pressure compressor so as to allow the refrigerant that has been pre-compressed in the low-pressure compressor to be delivered to the high-pressure compressor;
a radiator for cooling the refrigerant that has been compressed in the high-pressure compressor;
a positive displacement expander for recovering power by allowing the refrigerant to expand, the expander being coaxially coupled to the low-pressure compressor for power transmission and configured to allow the entire amount of the refrigerant that has been cooled in the radiator to pass through itself,
a gas-liquid separator for separating the refrigerant that has been expanded in the expander into gas refrigerant and liquid refrigerant;
an evaporator for allowing the liquid refrigerant that has been separated in the gas-liquid separator to evaporate;
an expansion valve with variable opening degree, the expansion valve being provided on a flow path between a liquid refrigerant outlet of the gas-liquid separator and an inlet of the evaporator;
a reciprocating flow path connecting the intermediate-pressure flow path and the gas-liquid separator so as to allow switching between a first circulation state in which the refrigerant stored in the gas-liquid separator is introduced into an inlet of the high-pressure compressor without passing through the evaporator and the low-pressure compressor and a second circulation state in which a part of the refrigerant that has been pre-compressed in the low-pressure compressor flows back to the gas-liquid separator and
a controller for regulating refrigerant flow rate in the reciprocating flow path in each state of the first circulation and the second circulation by controlling the opening degree of the expansion valve.

**2.** The refrigeration cycle apparatus according to claim 1, wherein
the low-pressure compressor and the expander each have a constant cylinder volume, and
the cylinder volume of the low-pressure compressor is greater than the cylinder volume of the expander.

**3.** The refrigeration cycle apparatus according to claim 1 or 2, wherein
the controller includes:

a means for calculating a target intermediate pressure at which theoretical recovery power of the expander at an arbitrary optimal high pressure of a refrigeration cycle and theoretical compression work of the low-pressure compressor at the arbitrary optimal high pressure are equal; and
a means for controlling the opening degree of the expansion valve so that an actual pressure inside the gas-liquid separator approaches the calculated target intermediate pressure.

**4.** The refrigeration cycle apparatus according to claim 3, further comprising:

a first temperature sensor for detecting an inlet refrigerant temperature of the expander;
a second temperature sensor for detecting a refrigerant evaporation temperature of the evaporator; and
a third temperature sensor for detecting an inlet refrigerant temperature of the low-pressure compressor, wherein the controller calculates the arbitrary optimal high pressure and the target intermediate pressure based on the detection results of the first to the third temperature sensors.

**5.** The refrigeration cycle apparatus according to claim 4, further comprising:

a fourth temperature sensor for detecting a refrigerant temperature in the gas-liquid separator, wherein
the means for controlling the opening degree of the expansion valve calculates the actual pressure inside the gas-liquid separator based on a detection result of the fourth temperature sensor.

**6.** The refrigeration cycle apparatus according to any one of claims 1 to 5, wherein assuming that a suction volume of the expander is $V_{ex}$, a suction volume of the low-pressure compressor is $V_{lc}$, a degree of dryness of a discharge refrigerant of the expander is $Q_{exo}$, a suction refrigerant density of the expander is $\rho_{exi}$ and a suction refrigerant density of the low-pressure compressor is $\rho_{lci}$, a relationship expressed by the following formula 1 is satisfied:

$$(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci}) \leq (V_{lc}/V_{ex}) \ ...(1).$$

**7.** The refrigeration cycle apparatus according to any one of claims 1 to 5, wherein assuming that a suction volume of the expander is $V_{ex}$, a suction volume of the low-pressure compressor is $V_{lc}$, a suction refrigerant density of the expander is $\rho_{exi}$ and a suction refrigerant density of the low-pressure compressor is $\rho_{lci}$, a relationship expressed by the following formula 2 is satisfied:

$$(V_{lc}/V_{ex}) \leq (\rho_{exi}/\rho_{lci}) \ ...(2).$$

**8.** The refrigeration cycle apparatus according to any one of claims 1 to 5, wherein assuming that a suction volume of the expander is $V_{ex}$, a suction volume of the low-pressure compressor is $V_{lc}$, a degree of dryness of a discharge refrigerant of the expander is $Q_{exo}$, a suction refrigerant density of the expander is $\rho_{exi}$ and a suction refrigerant density of the low-pressure compressor is $\rho_{lci}$,
a relationship expressed by the following formula 3 is satisfied:

$$(1 - Q_{exo}) \times (\rho_{exi}/\rho_{lci}) \leq (V_{lc}/V_{ex}) \leq (\rho_{exi}/\rho_{lci}) \ ...(3).$$

**9.** The refrigeration cycle apparatus according to any one of claims 1 to 8, further comprising:

an expander injection path connecting a suction path of the expander and an expander injection port opening into an expansion chamber of the expander; and
an expander injection valve provided on the expander injection path, wherein
the expander is capable of changing its suction volume by controlling the expander injection valve.

**10.** The refrigeration cycle apparatus according to any one of claims 1 to 9, wherein the low-pressure compressor and the expander are disposed in a common closed casing.

**11.** A multi-functional heat pump system comprising:

a heat pump water heater with a water-heating function for supplying heated water to a tap and/or a floor-heating function for heating indoor space by circulating heated water in a pipe running throughout a floor of a house; and
an air-conditioner configured to adjust an indoor temperature by heat exchange between indoor air and a refrigerant, wherein
the refrigeration cycle apparatus according to any one of claims 1 to 10 is used as a common refrigeration cycle apparatus for the water heater and the air-conditioner.

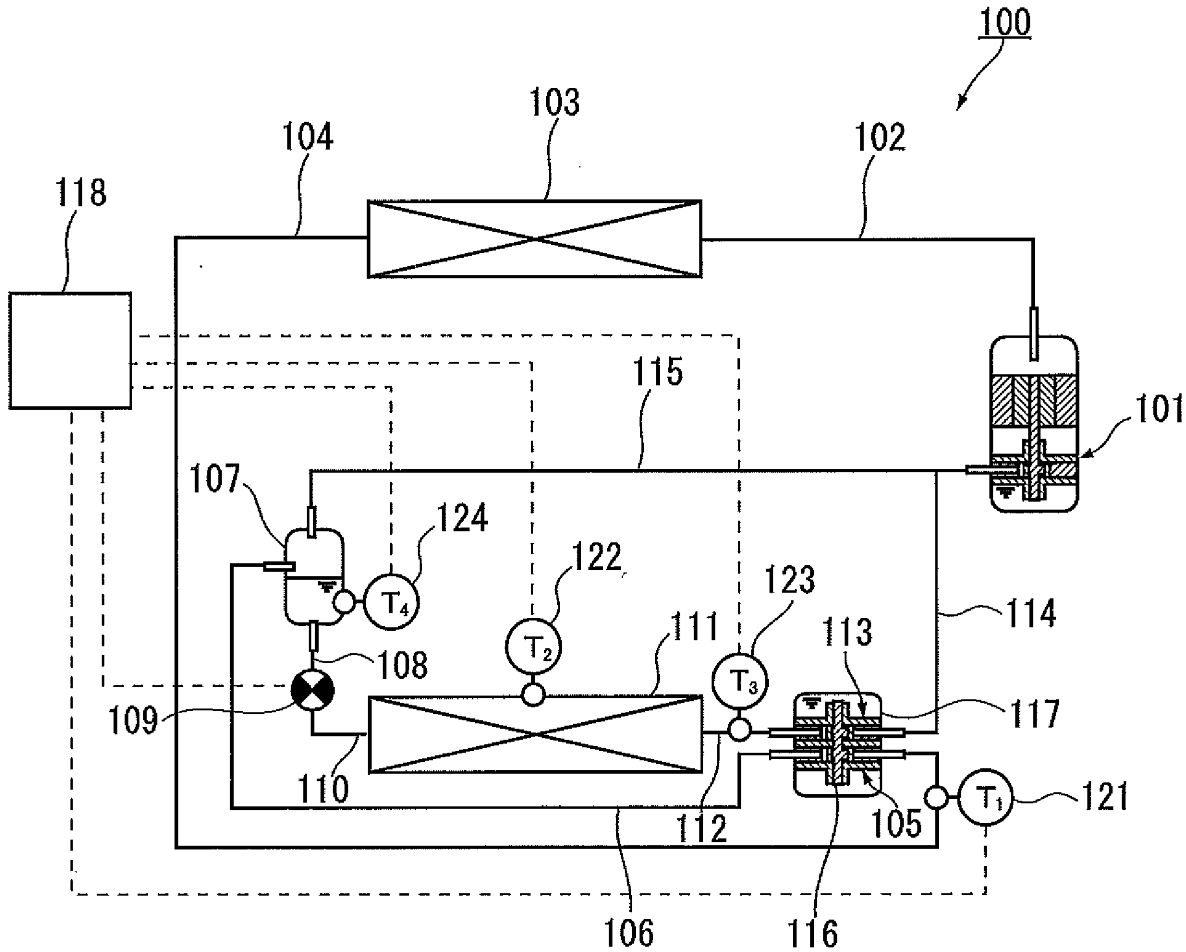

100
103
104
102
118
115
101
107
124
122
123
111
113
114
108
109
117
110
121
112
105
106
116
T4
T2
T3
T1

FIG.1

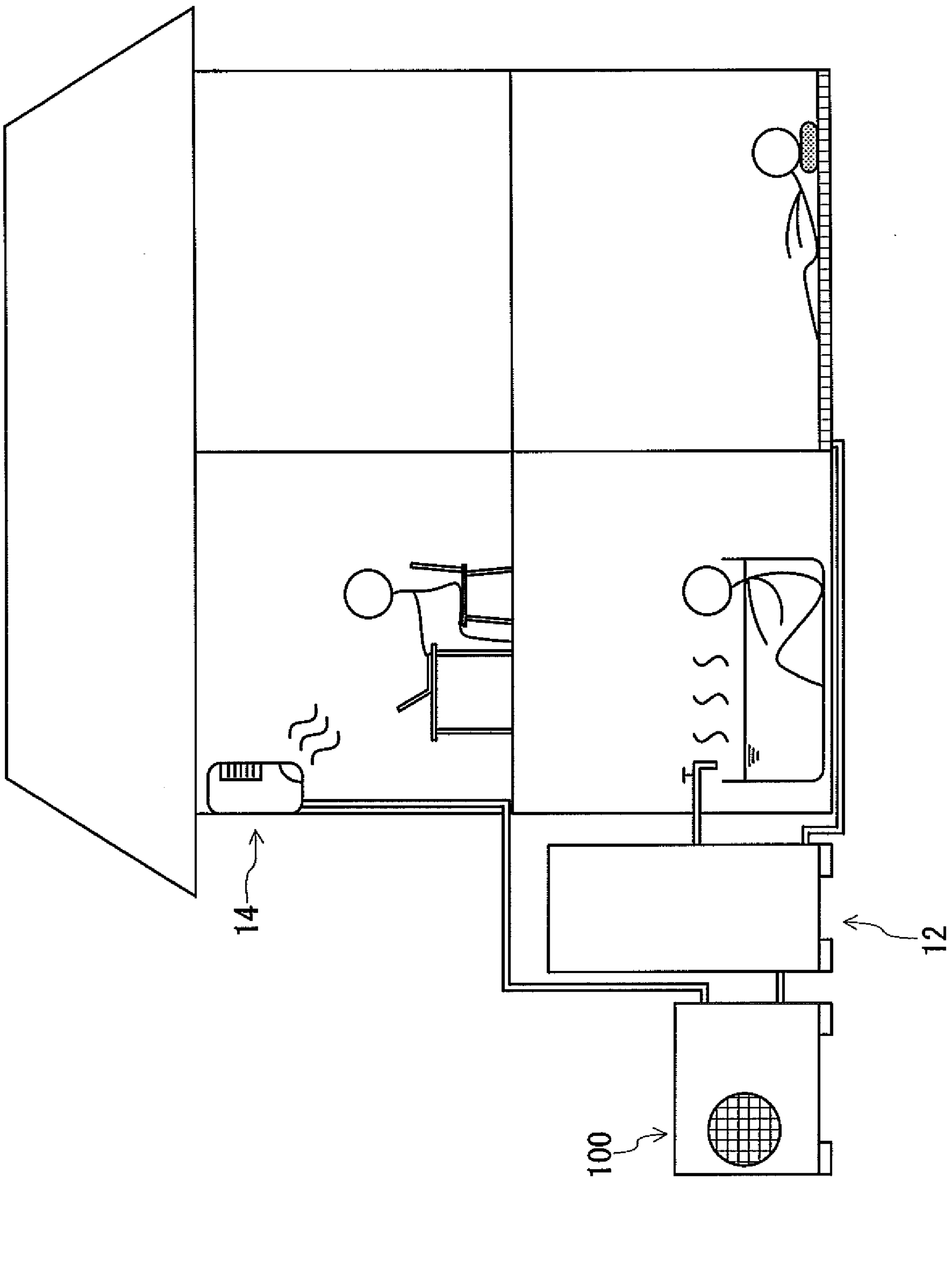



FIG.2

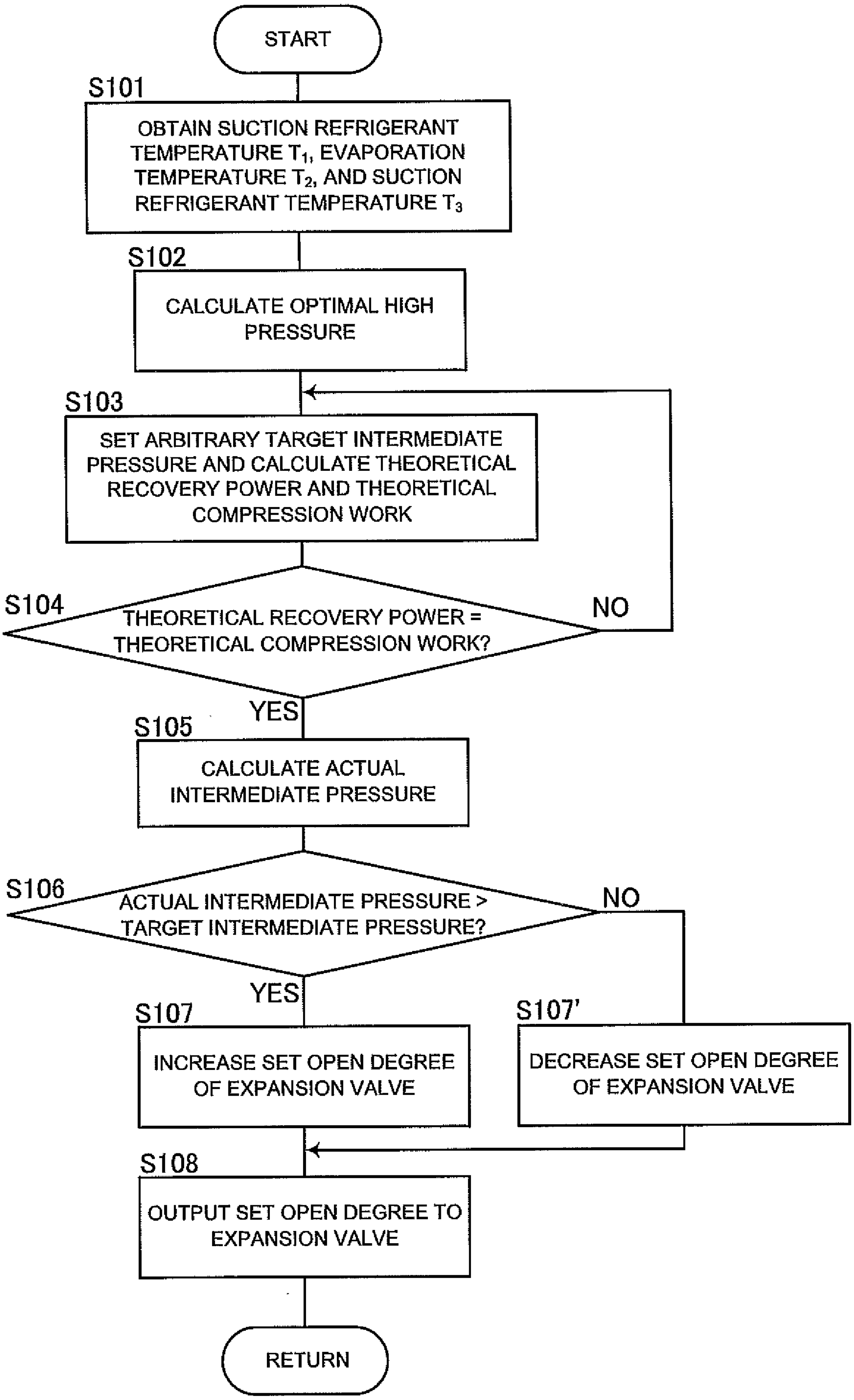
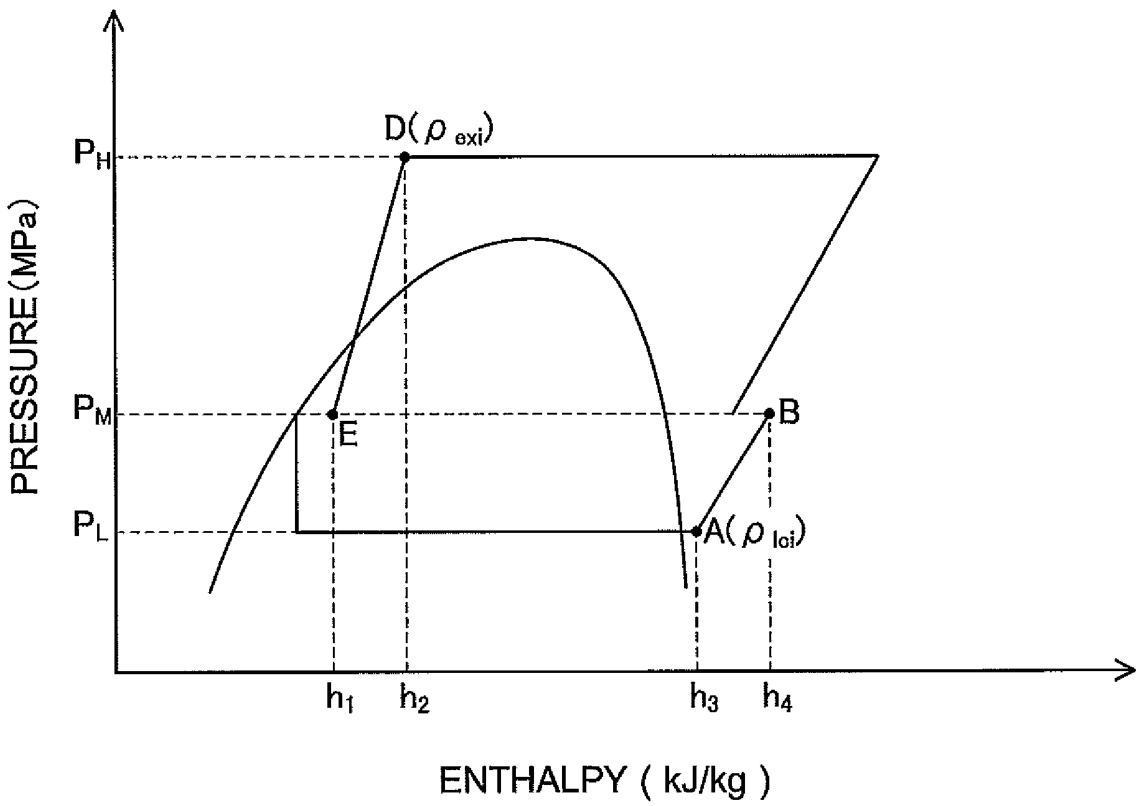

START
S101
OBTAIN SUCTION REFRIGERANT TEMPERATURE T1, EVAPORATION TEMPERATURE T2, AND SUCTION REFRIGERANT TEMPERATURE T3
S102
CALCULATE OPTIMAL HIGH PRESSURE
S103
SET ARBITRARY TARGET INTERMEDIATE PRESSURE AND CALCULATE THEORETICAL RECOVERY POWER AND THEORETICAL COMPRESSION WORK
S104
THEORETICAL RECOVERY POWER = THEORETICAL COMPRESSION WORK?
NO
YES
S105
CALCULATE ACTUAL INTERMEDIATE PRESSURE
S106
ACTUAL INTERMEDIATE PRESSURE > TARGET INTERMEDIATE PRESSURE?
NO
YES
S107
INCREASE SET OPEN DEGREE OF EXPANSION VALVE
S107'
DECREASE SET OPEN DEGREE OF EXPANSION VALVE
S108
OUTPUT SET OPEN DEGREE TO EXPANSION VALVE
RETURN

FIG.3



PRESSURE (MPa)
$P_H$
$P_M$
$P_L$
D( $\rho_{exi}$)
E
B
A( $\rho_{lci}$)
$h_1$
$h_2$
$h_3$
$h_4$
ENTHALPY ( kJ/kg )


FIG.4

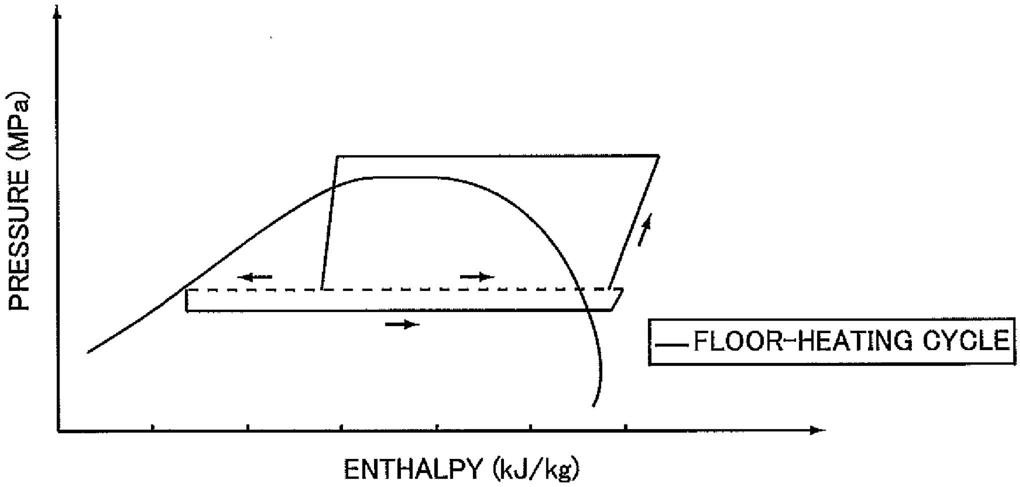



FIG.5A

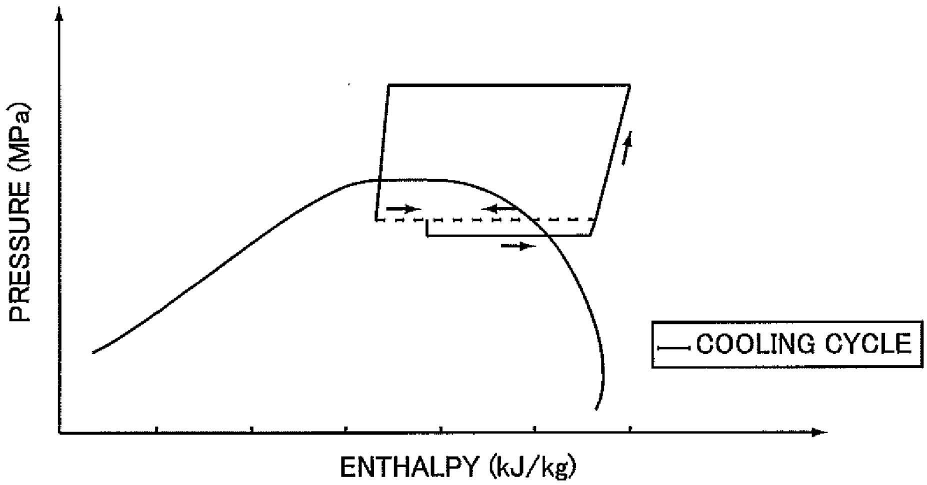



FIG.5B

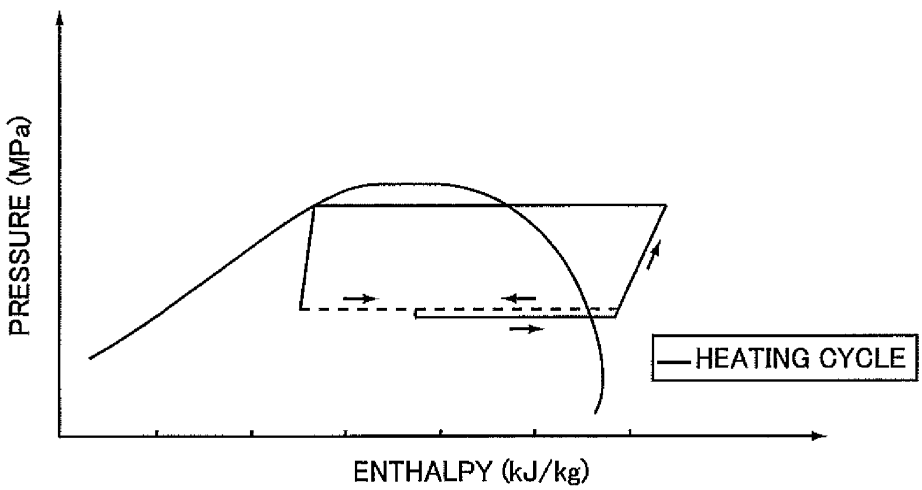



FIG.5C

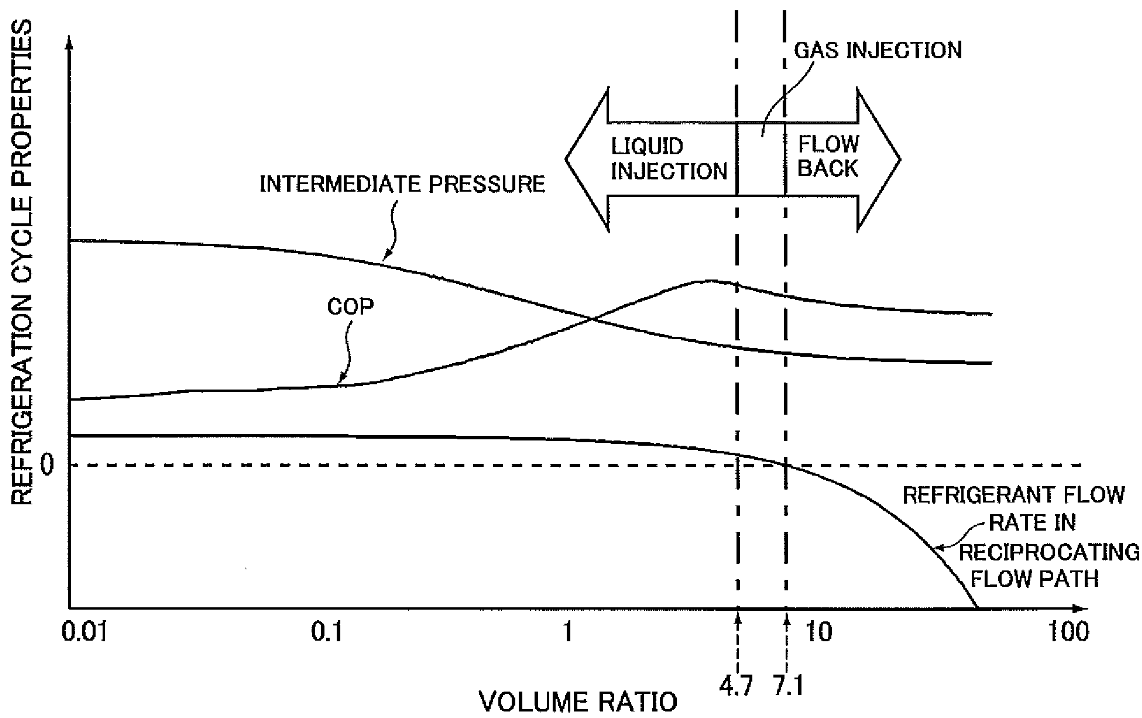

REFRIGERATION CYCLE PROPERTIES
GAS INJECTION
LIQUID INJECTION
FLOW BACK
INTERMEDIATE PRESSURE
COP
0
REFRIGERANT FLOW RATE IN RECIPROCATING FLOW PATH
0.01
0.1
1
10
100
4.7
7.1
VOLUME RATIO

FIG.6A

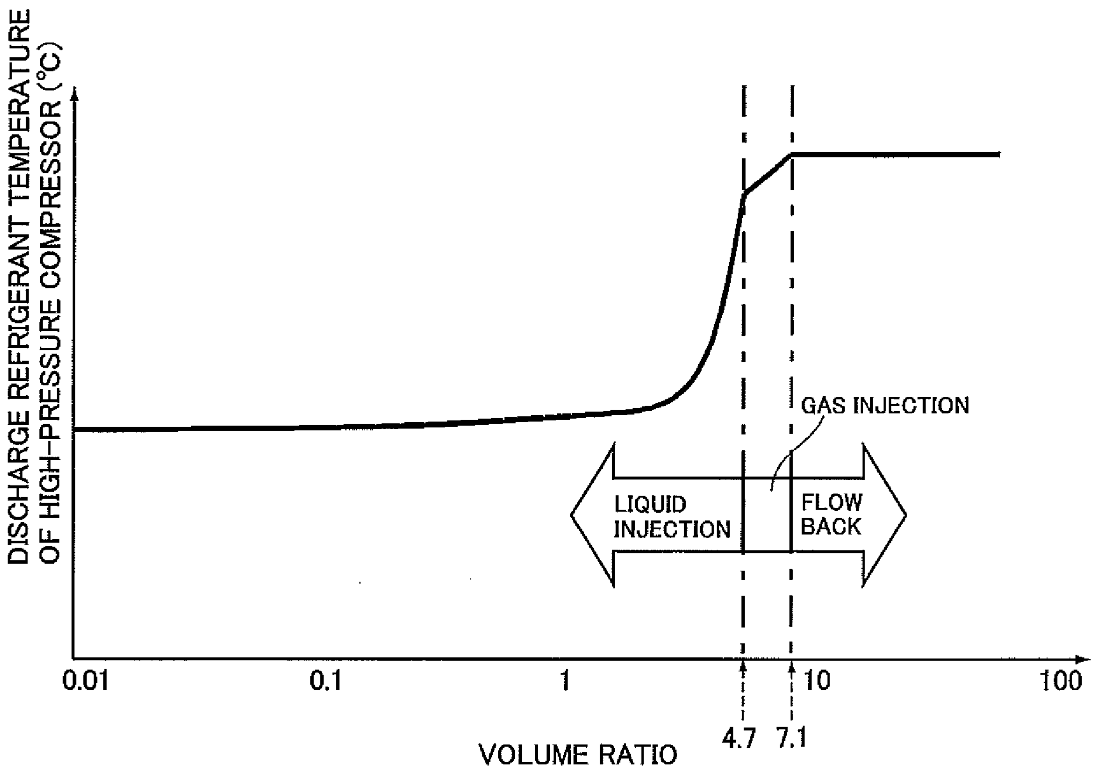

DISCHARGE REFRIGERANT TEMPERATURE OF HIGH-PRESSURE COMPRESSOR (°C)
GAS INJECTION
LIQUID INJECTION
FLOW BACK
0.01
0.1
1
10
100
4.7
7.1
VOLUME RATIO

FIG.6B

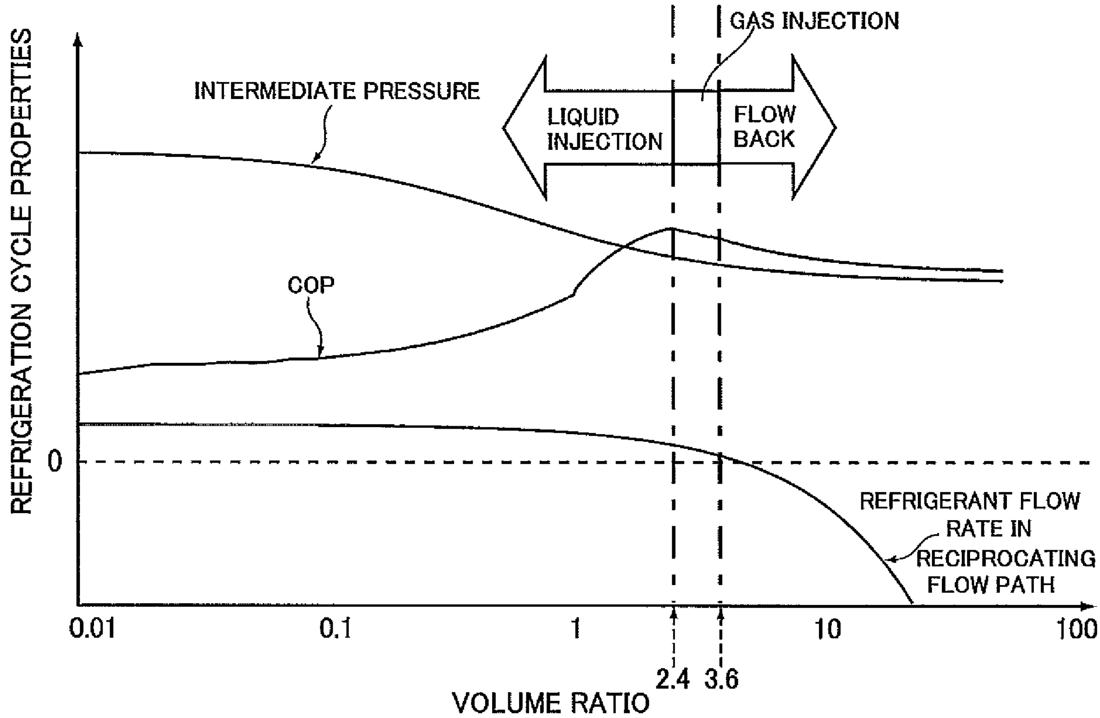

REFRIGERATION CYCLE PROPERTIES
GAS INJECTION
INTERMEDIATE PRESSURE
LIQUID INJECTION
FLOW BACK
COP
0
REFRIGERANT FLOW RATE IN RECIPROCATING FLOW PATH
0.01
0.1
1
10
100
2.4
3.6
VOLUME RATIO

FIG.7A

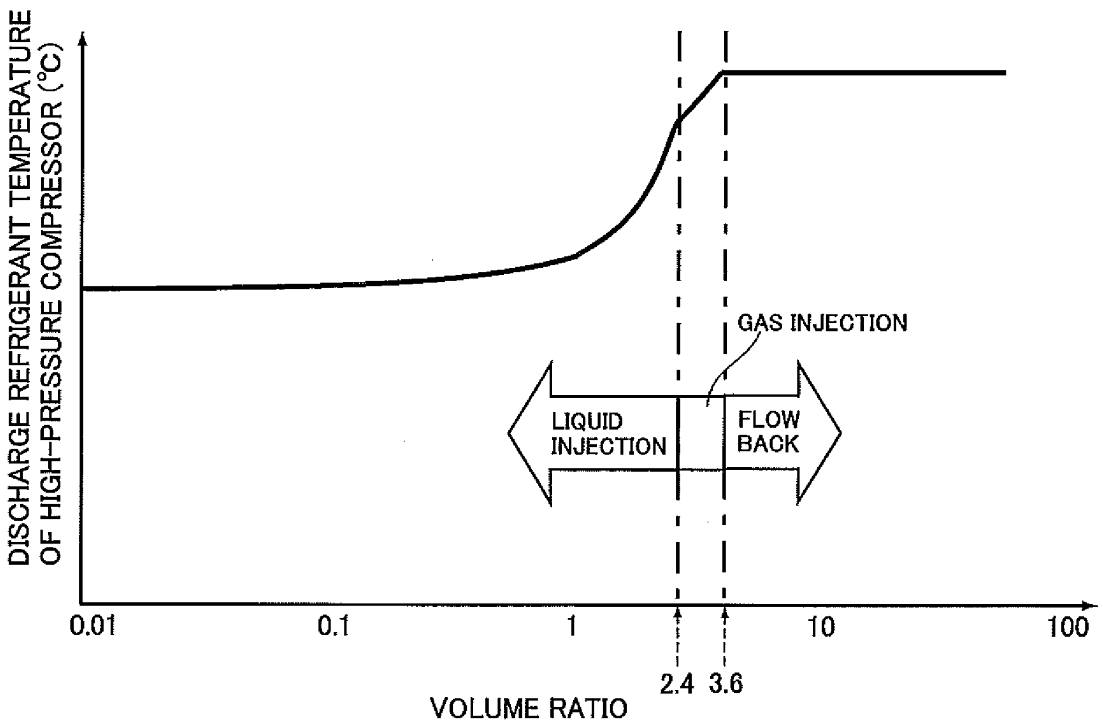

DISCHARGE REFRIGERANT TEMPERATURE OF HIGH-PRESSURE COMPRESSOR (°C)
GAS INJECTION
LIQUID INJECTION
FLOW BACK
0.01
0.1
1
10
100
2.4
3.6
VOLUME RATIO

FIG.7B

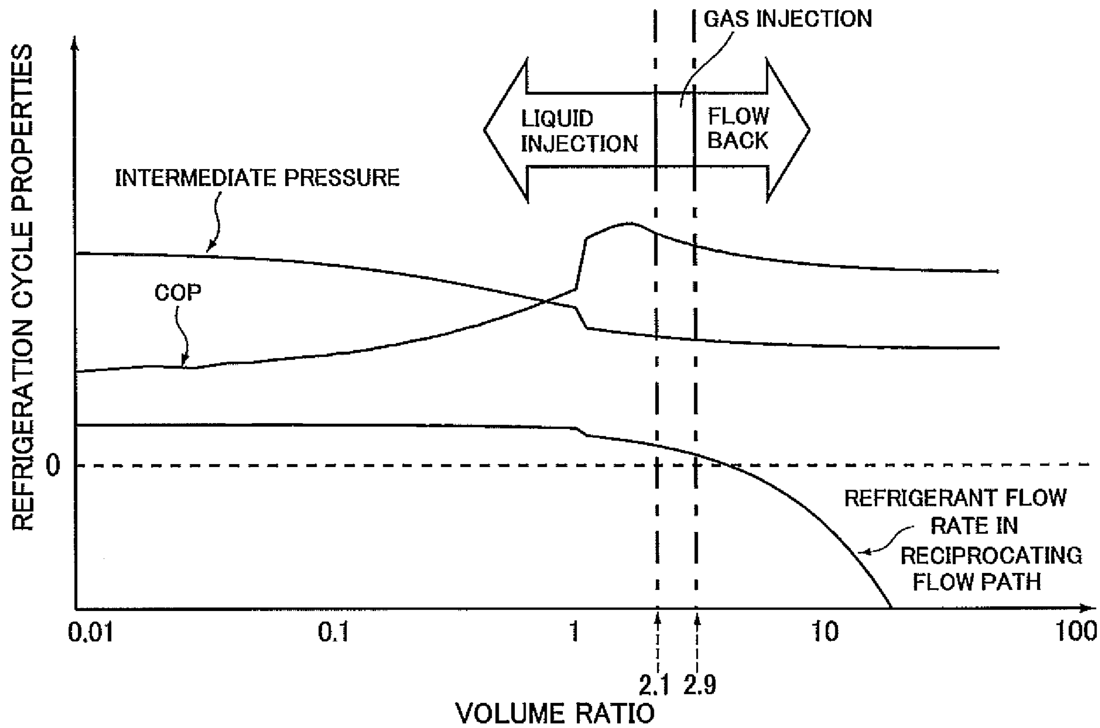

GAS INJECTION
LIQUID INJECTION
FLOW BACK
INTERMEDIATE PRESSURE
COP
REFRIGERATION CYCLE PROPERTIES
0
REFRIGERANT FLOW RATE IN RECIPROCATING FLOW PATH
0.01
0.1
1
10
100
2.1
2.9
VOLUME RATIO

FIG.8A

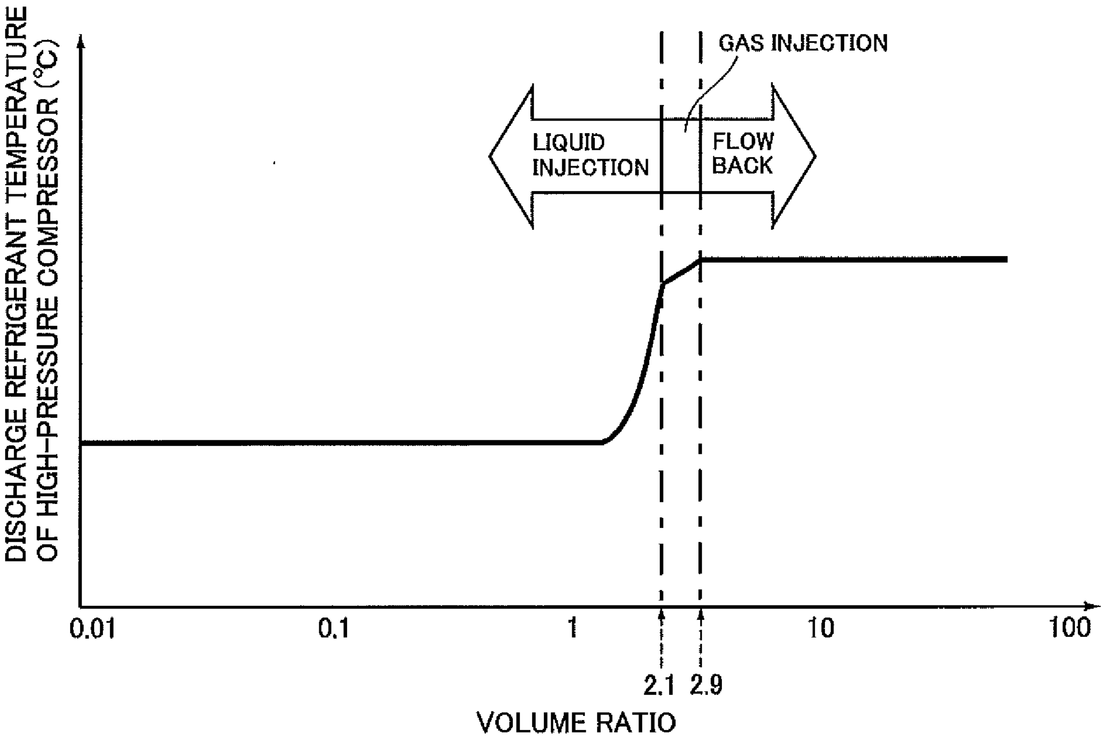

GAS INJECTION
LIQUID INJECTION
FLOW BACK
DISCHARGE REFRIGERANT TEMPERATURE OF HIGH-PRESSURE COMPRESSOR (°C)
0.01
0.1
1
10
100
2.1
2.9
VOLUME RATIO

FIG.8B

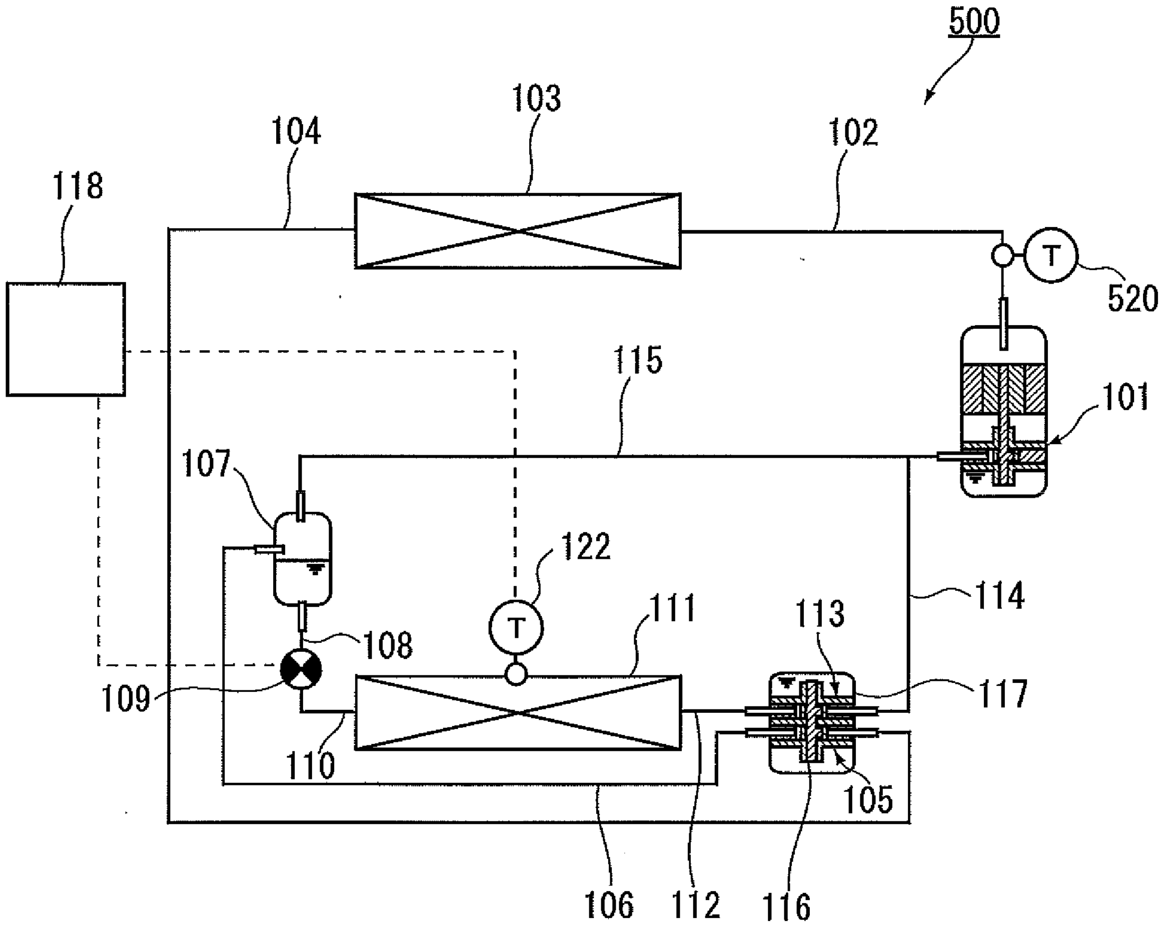

500
103
104
102
118
520
115
101
107
122
114
111
113
108
117
109
110
105
106
112
116

FIG.9

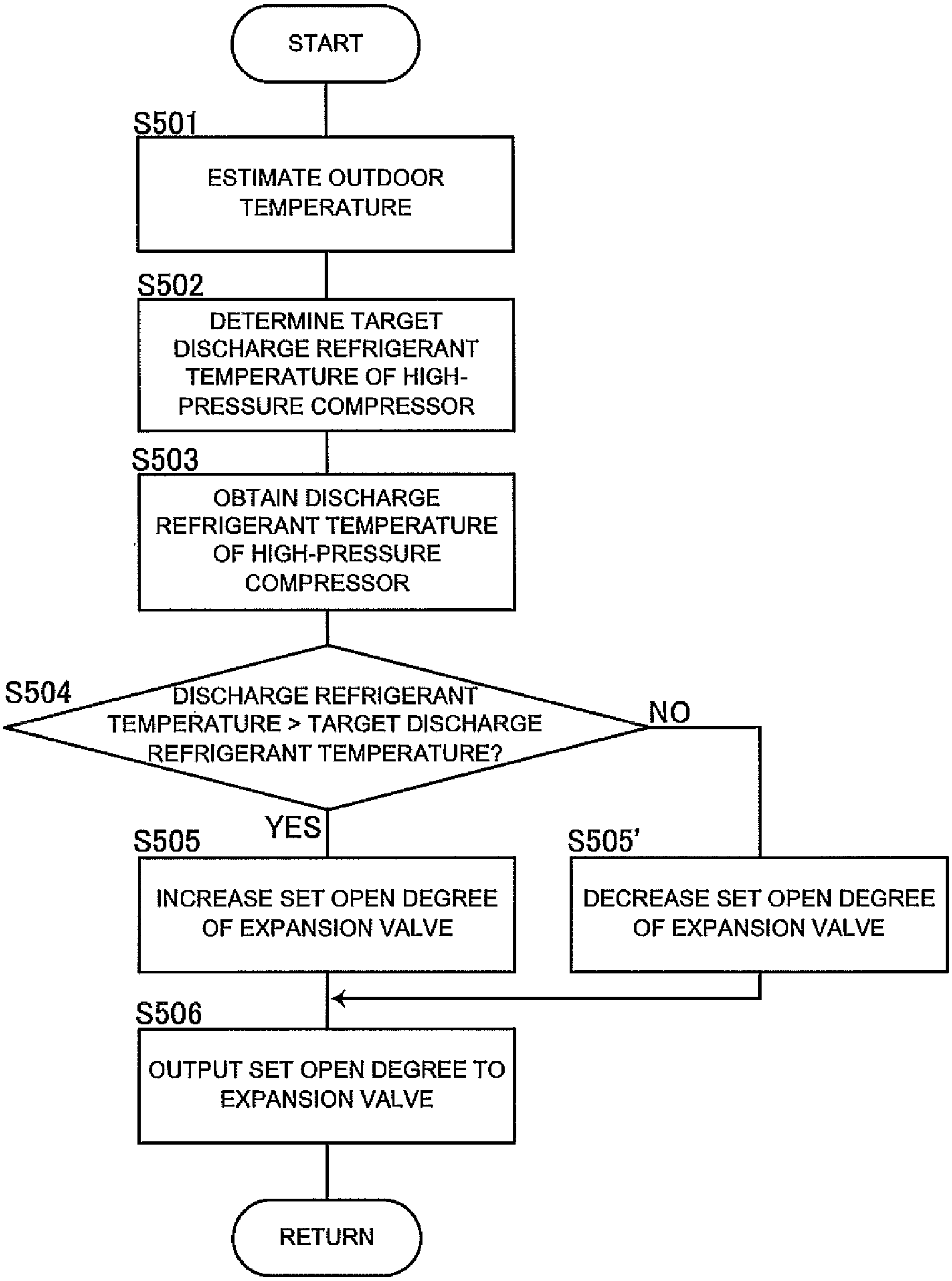

START
S501
ESTIMATE OUTDOOR TEMPERATURE
S502
DETERMINE TARGET DISCHARGE REFRIGERANT TEMPERATURE OF HIGH-PRESSURE COMPRESSOR
S503
OBTAIN DISCHARGE REFRIGERANT TEMPERATURE OF HIGH-PRESSURE COMPRESSOR
S504
DISCHARGE REFRIGERANT TEMPERATURE > TARGET DISCHARGE REFRIGERANT TEMPERATURE?
YES
NO
S505
INCREASE SET OPEN DEGREE OF EXPANSION VALVE
S505'
DECREASE SET OPEN DEGREE OF EXPANSION VALVE
S506
OUTPUT SET OPEN DEGREE TO EXPANSION VALVE
RETURN

FIG.10

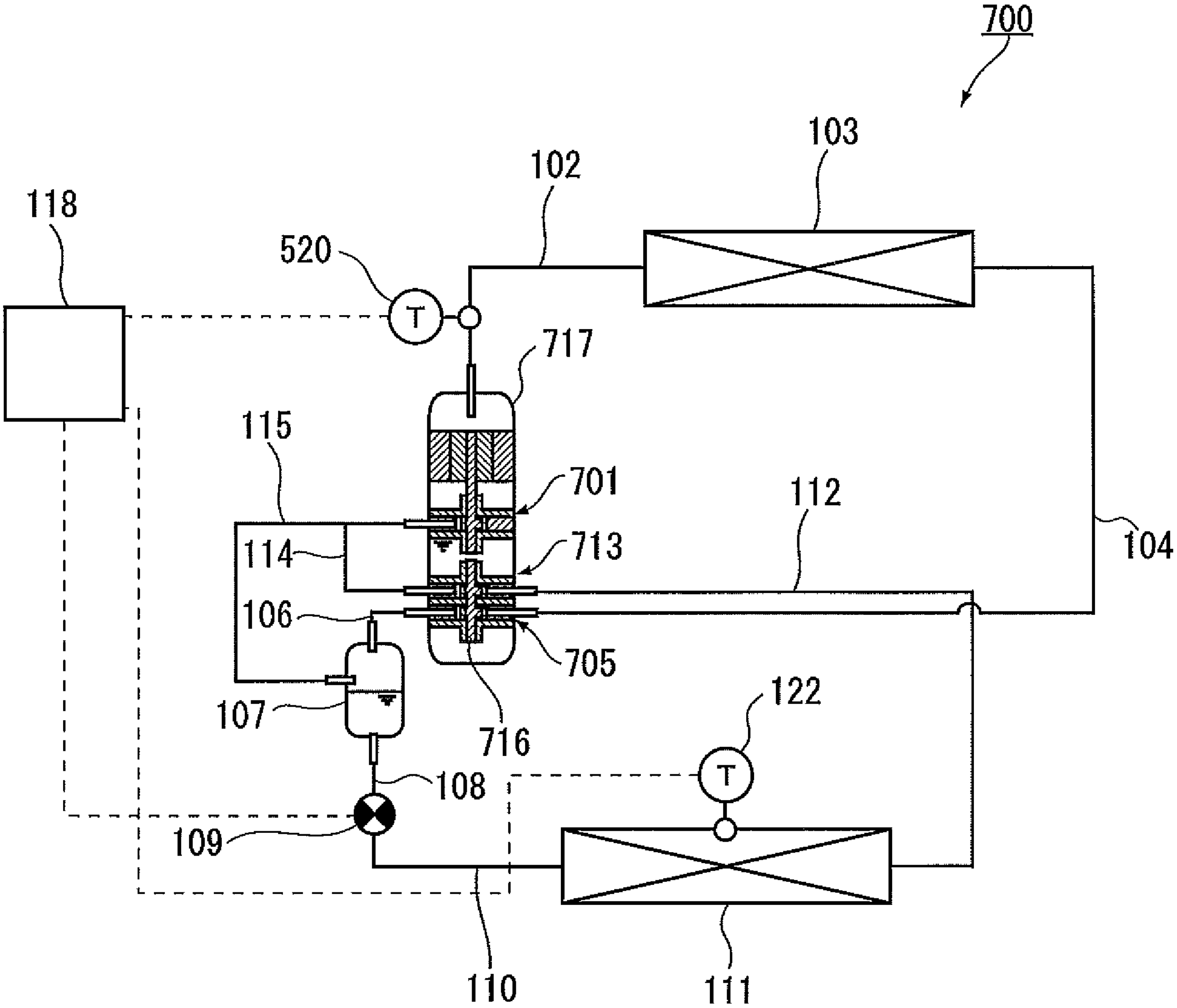

700
103
102
118
520
717
115
701
112
114
713
104
106
705
122
107
716
108
109
110
111

FIG.11

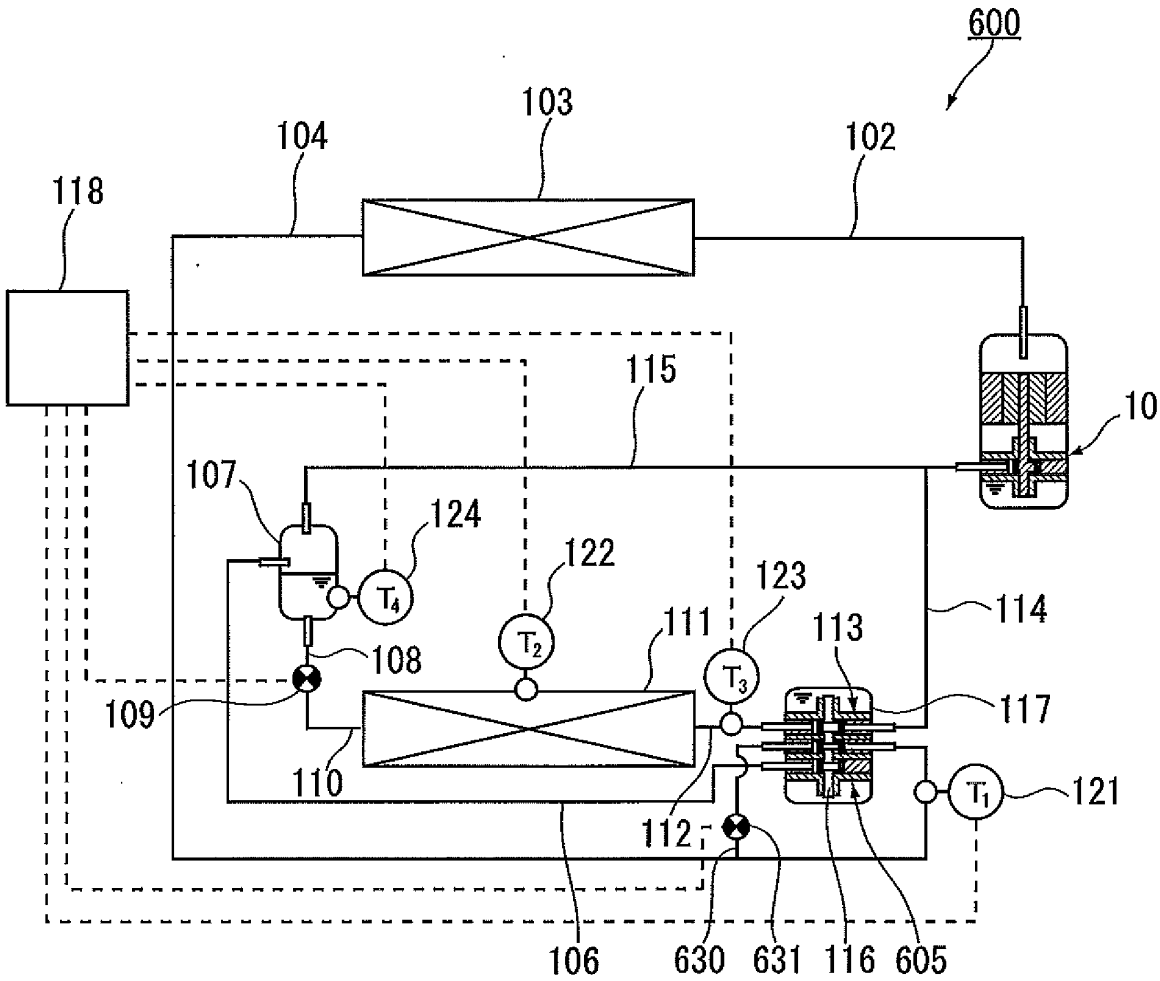

600
103
104
102
118
115
101
107
124
122
123
111
113
114
108
109
T4
T2
T3
T1
117
110
121
112
106
630
631
116
605

FIG.12A

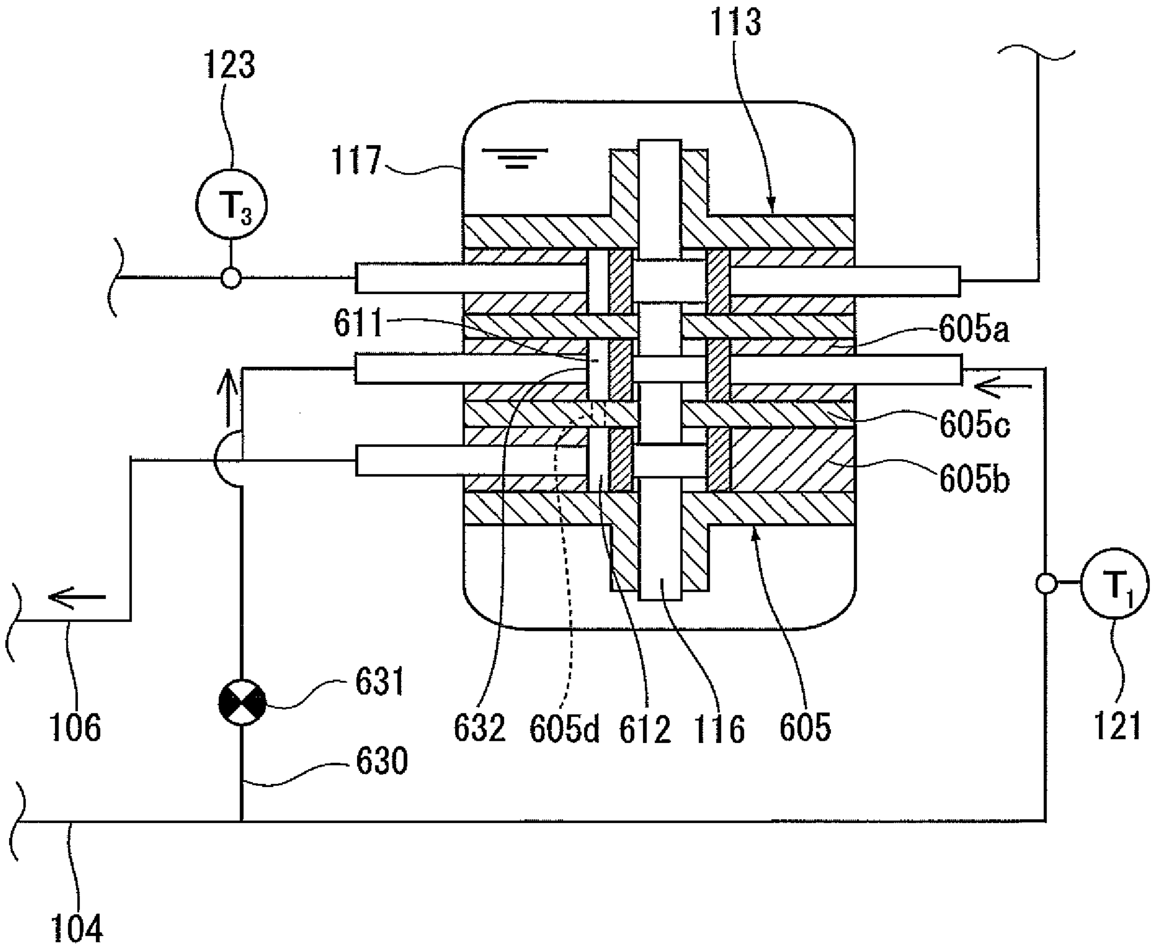

113
123
T3
117
611
605a
605c
605b
T1
106
631
632
605d
612
116
605
121
630
104

FIG.12B

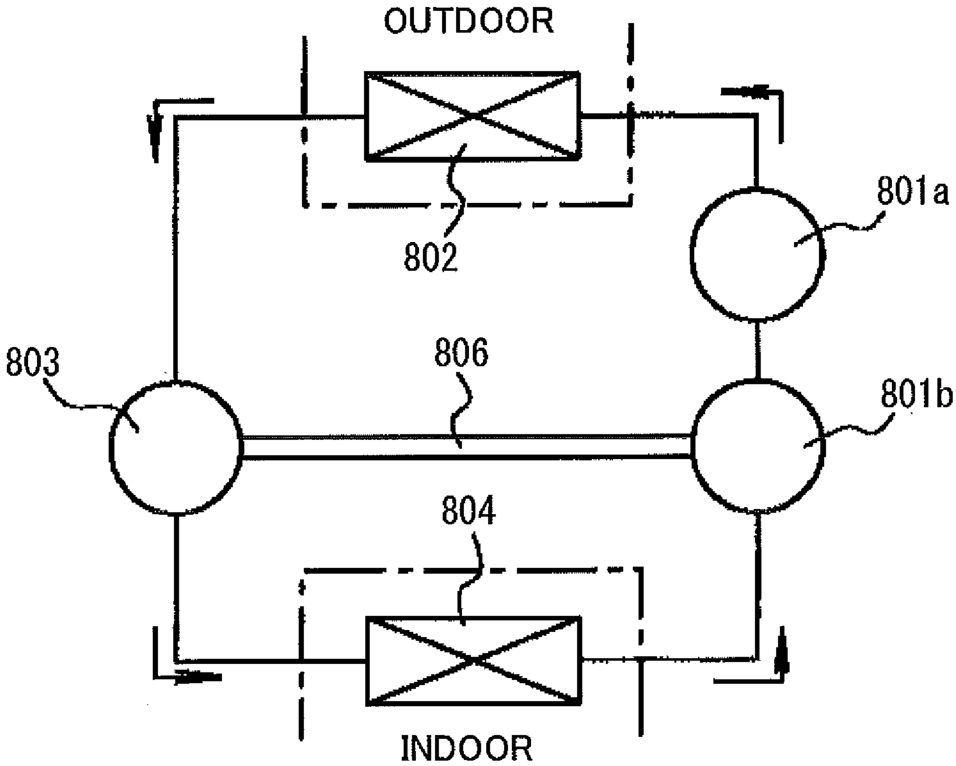



FIG.13

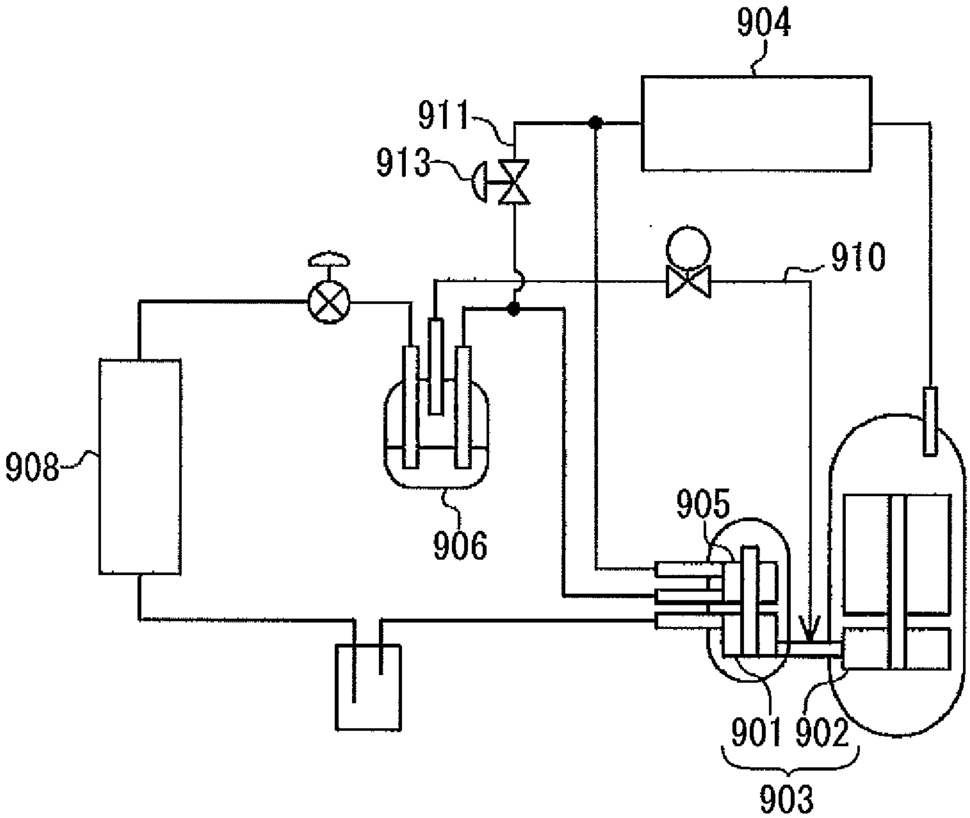



FIG.14

**INTERNATIONAL SEARCH REPORT**

International application No.
PCT/JP2009/002443

A. CLASSIFICATION OF SUBJECT MATTER
*F25B1/10*(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)
F25B1/10, 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 | 1996-2009 |
| Kokai Jitsuyo Shinan Koho | 1971-2009 | Toroku Jitsuyo Shinan Koho | 1994-2009 |

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

| Category* | Citation of document, with indication, where appropriate, of the relevant passages | Relevant to claim No. |
|---|---|---|
| X<br>Y<br>A | JP 2005-300031 A (Matsushita Electric Industrial Co., Ltd.),<br>27 October, 2005 (27.10.05),<br>Par. Nos. [0038] to [0080]; Figs. 1 to 7<br>(Family: none) | 1,6-8,11<br>2,9,10<br>3-5 |
| Y | WO 2007/029493 A1 (TGK Co., Ltd.),<br>15 March, 2007 (15.03.07),<br>Full text; Figs. 1 to 11<br>(Family: none) | 2,10 |
| Y | JP 2006-23004 A (Daikin Industries, Ltd.),<br>26 January, 2006 (26.01.06),<br>Claim 1<br>& WO 2006/004047 A1 & EP 1780478 A1<br>& US 2007-251245 A1 | 9 |

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

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"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

| Date of the actual completion of the international search | Date of mailing of the international search report |
|---|---|
| 10 June, 2009 (10.06.09) | 23 June, 2009 (23.06.09) |

| Name and mailing address of the ISA/ | Authorized officer |
|---|---|
| Japanese Patent Office | |
| Facsimile No. | Telephone No. |

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## REFERENCES CITED IN THE DESCRIPTION

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### Patent documents cited in the description


- JP 2003307358 A **[0006]**
- JP 2006071257 A **[0006]**