US 20250067463A1

(19) United States

(12) Patent Application Publication

SATO

(10) Pub. No.: US 2025/0067463 A1

(43) Pub. Date: Feb. 27, 2025

(54) **AIR CONDITIONER**

(71) Applicant: **Mitsubishi Electric Corporation**, Tokyo (JP)

(72) Inventor: **Masanori SATO**, Tokyo (JP)

(21) Appl. No.: **18/719,185**

(22) PCT Filed: **Jan. 21, 2022**

(86) PCT No.: **PCT/JP2022/002222**

§ 371 (c)(1),
(2) Date: **Jun. 12, 2024**

Publication Classification

(51) **Int. Cl.**
F24F 11/83 (2006.01)

(52) **U.S. Cl.**
CPC *F24F 11/83* (2018.01)

(57) **ABSTRACT**

The air conditioner includes refrigerant, a refrigerant circuit, a heat exchanger, and a controller. The refrigerant circuit includes at least a compressor, a condenser and an evaporator, and is configured to circulate the refrigerant the refrigerant. The heat exchanger includes a first flow channel through which the refrigerant that has passed through the condenser flows and a second flow channel through which the refrigerant to be sucked into the compressor flows, and is configured to exchange heat between the refrigerant passing through the first flow channel and the refrigerant passing through the second flow channel. The controller controls the refrigerant circuit so as to cause the degree of superheat of the refrigerant flowing through an outlet of the evaporator to 5 degrees or less.

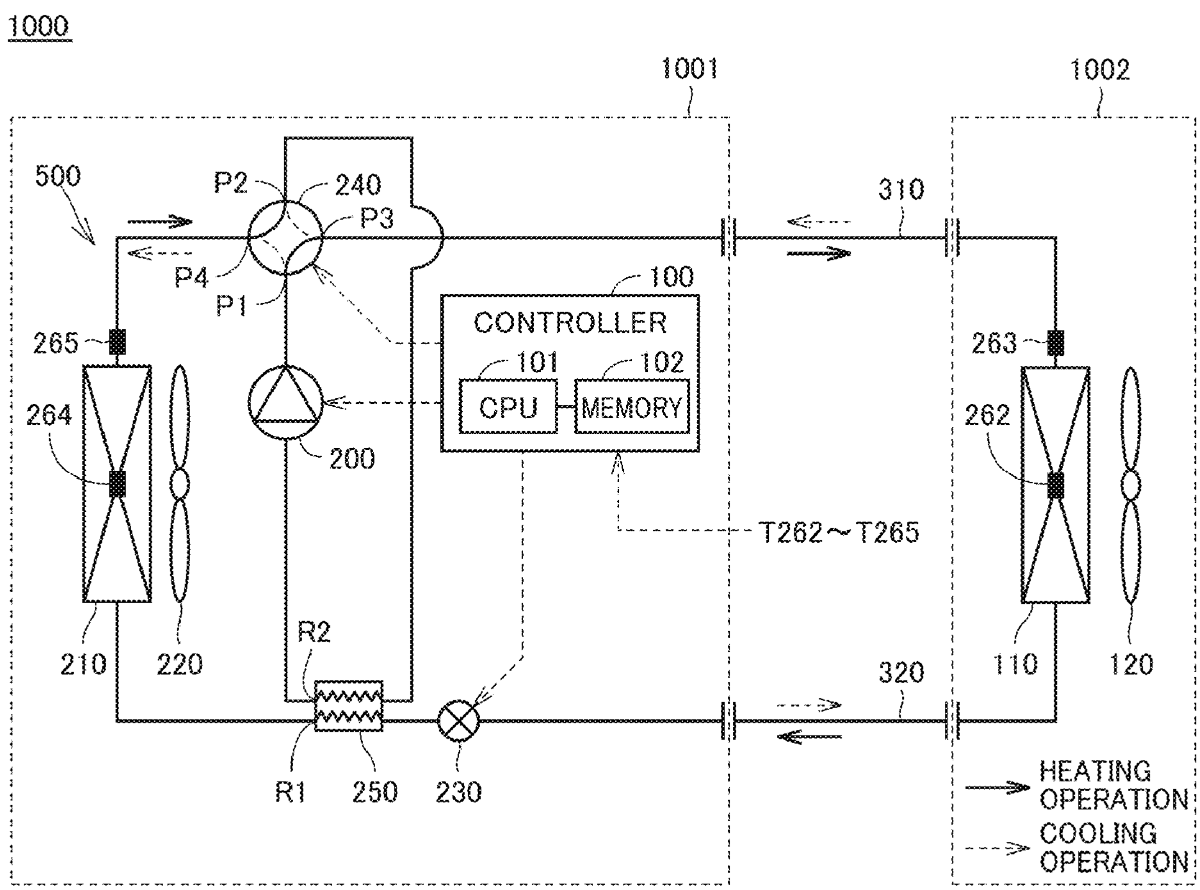


FIG.1

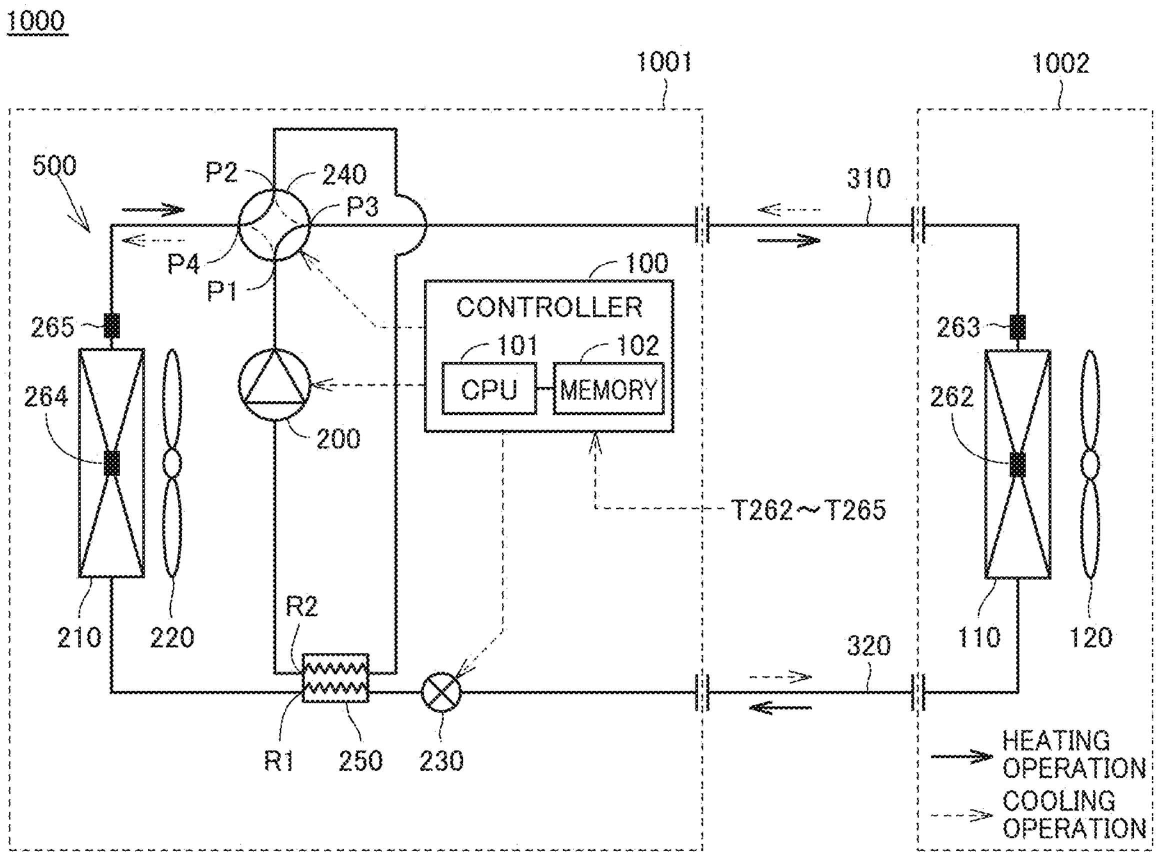

1000
1001
1002
500
P2
240
P3
P4
P1
310
100
CONTROLLER
101
102
CPU
MEMORY
265
263
264
262
200
T262~T265
R2
210
220
110
120
320
R1
250
230
HEATING OPERATION
COOLING OPERATION

FIG.2

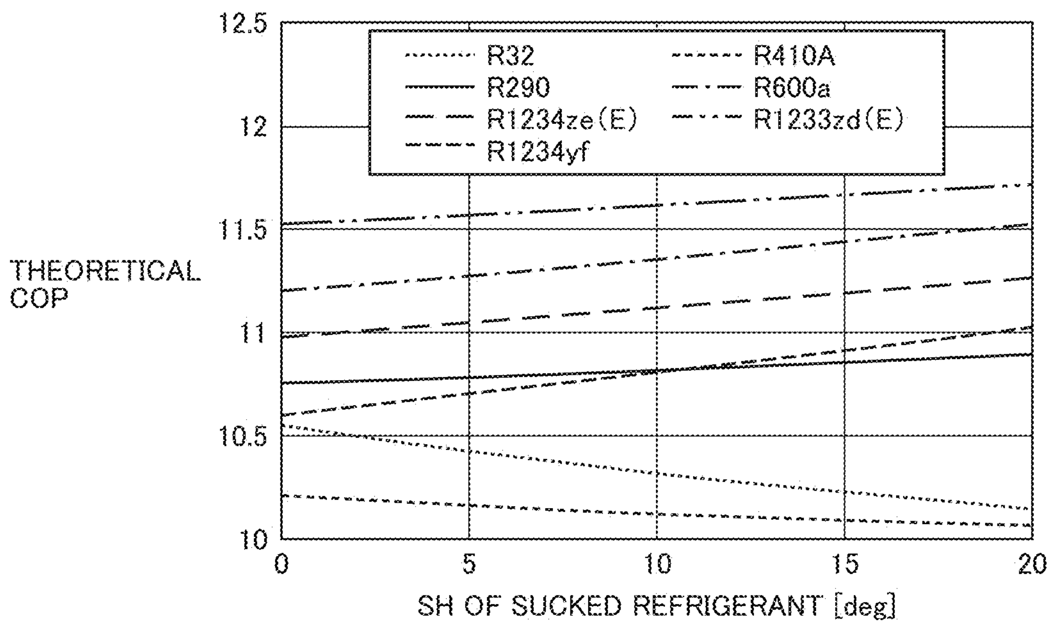

R32
R410A
R290
R600a
R1234ze(E)
R1233zd(E)
R1234yf
THEORETICAL COP
12.5
12
11.5
11
10.5
10
0
5
10
15
20
SH OF SUCKED REFRIGERANT [deg]

FIG.3

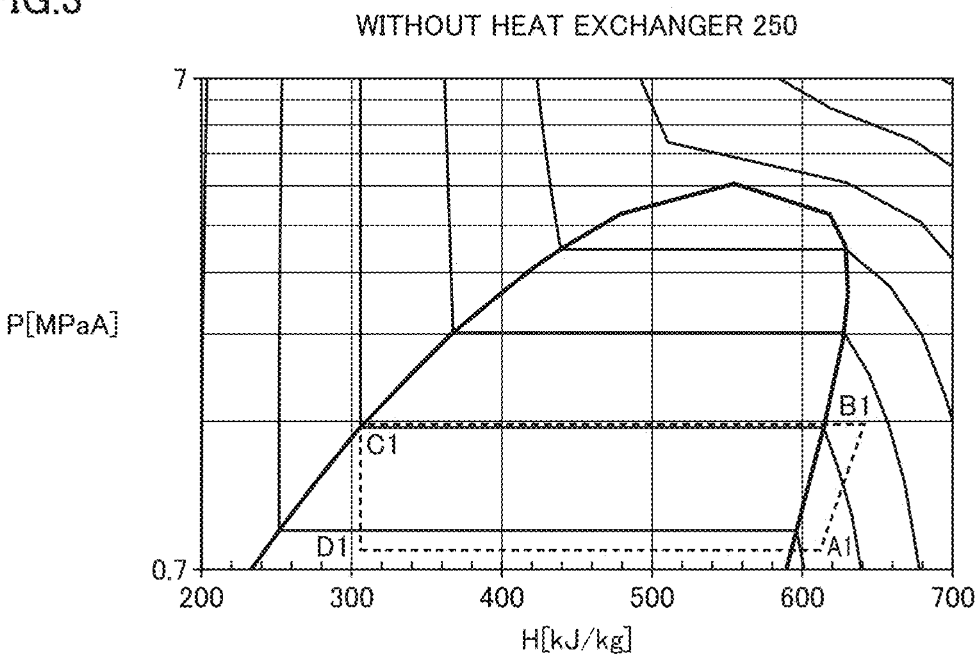

WITHOUT HEAT EXCHANGER 250
7
0.7
P[MPaA]
200
300
400
500
600
700
H[kJ/kg]
B1
C1
D1
A1

FIG.4

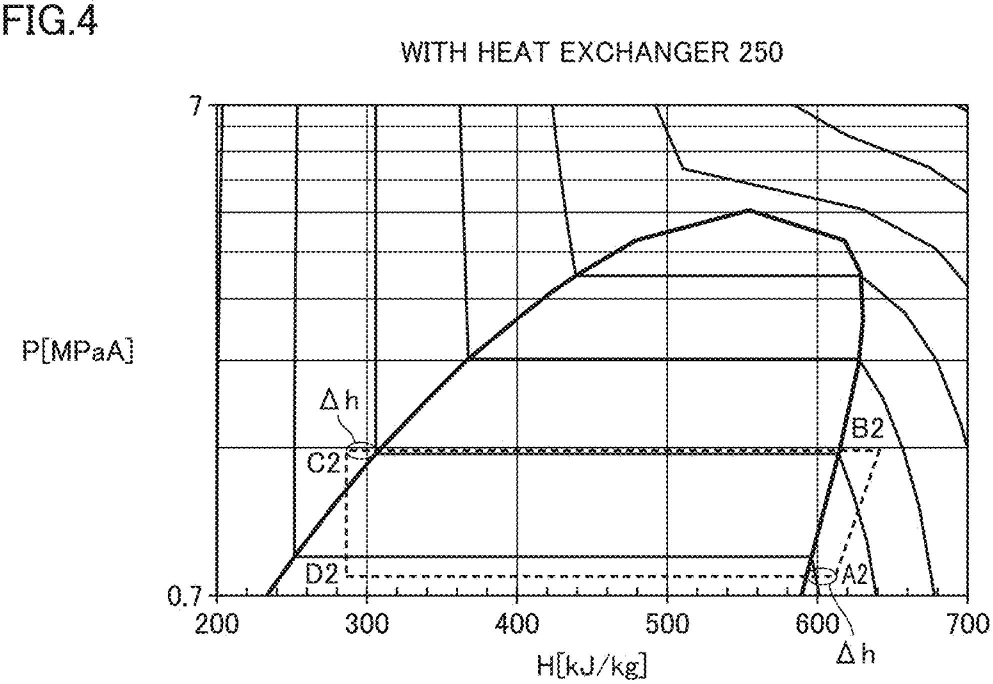

WITH HEAT EXCHANGER 250
7
0.7
P[MPaA]
Δh
B2
C2
D2
A2
Δh
200
300
400
500
600
700
H[kJ/kg]

FIG.5

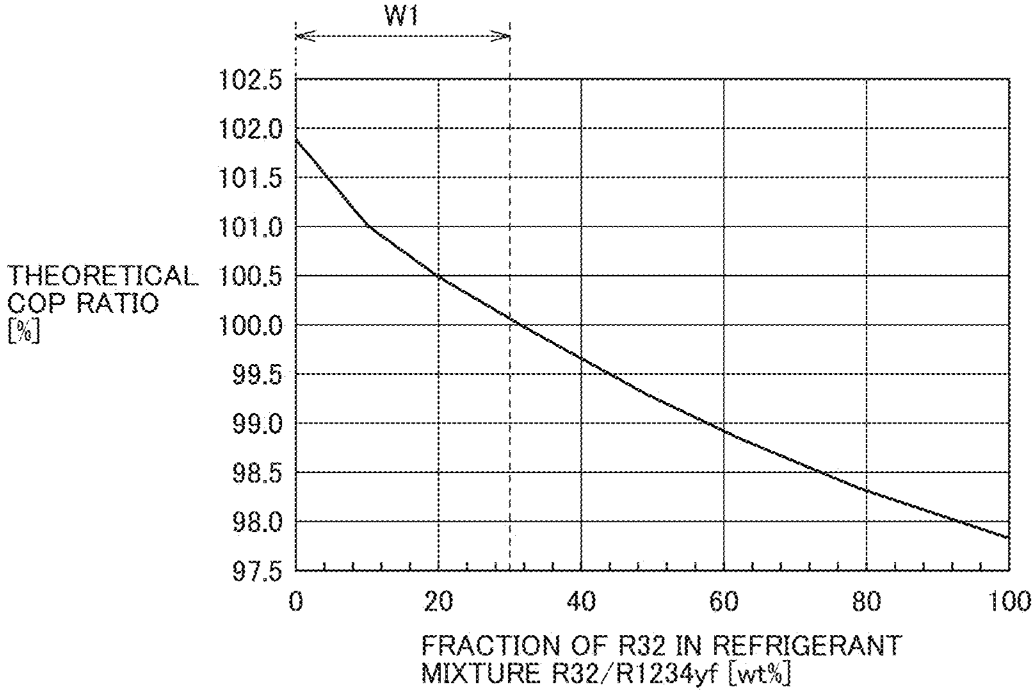

W1
THEORETICAL COP RATIO [%]
102.5
102.0
101.5
101.0
100.5
100.0
99.5
99.0
98.5
98.0
97.5
0
20
40
60
80
100
FRACTION OF R32 IN REFRIGERANT MIXTURE R32/R1234yf [wt%]

FIG.6

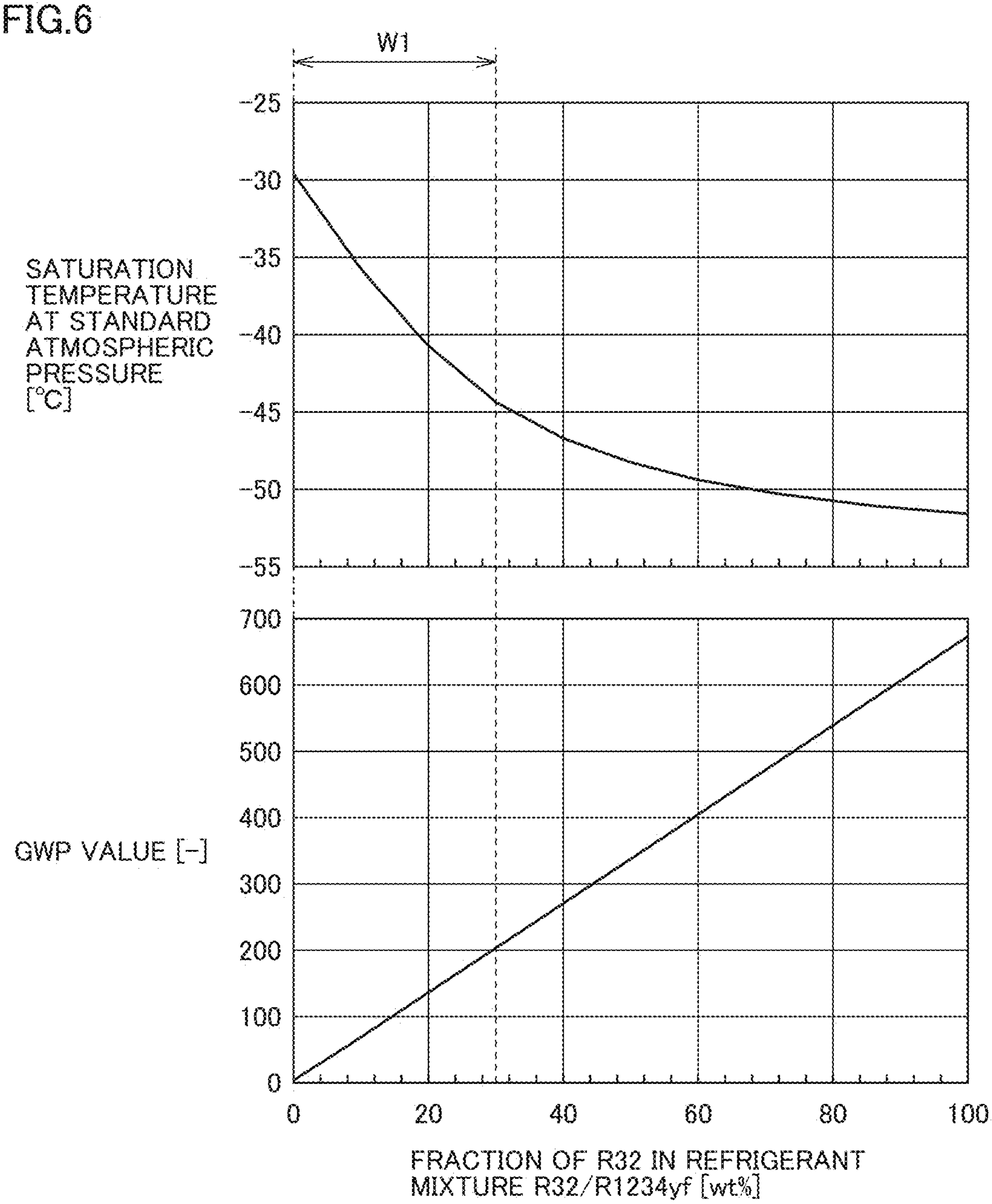

W1
SATURATION TEMPERATURE AT STANDARD ATMOSPHERIC PRESSURE [°C]
-25
-30
-35
-40
-45
-50
-55
GWP VALUE [-]
700
600
500
400
300
200
100
0
0
20
40
60
80
100
FRACTION OF R32 IN REFRIGERANT MIXTURE R32/R1234yf [wt%]

FIG.7

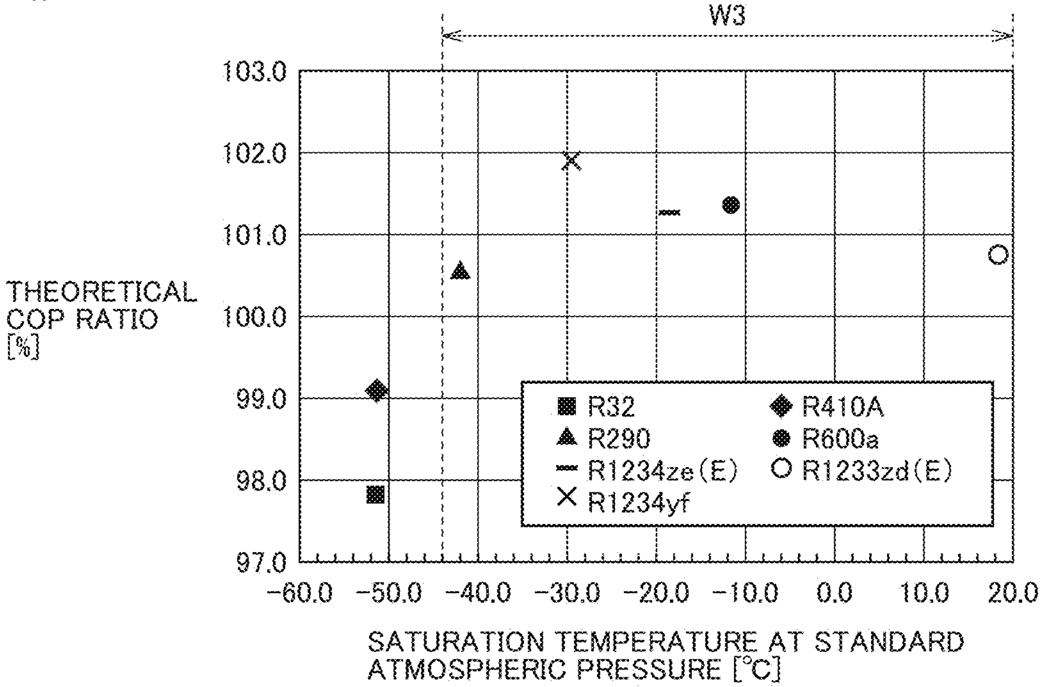

W3
103.0
102.0
101.0
100.0
99.0
98.0
97.0
THEORETICAL COP RATIO [%]
R32
R410A
R290
R600a
R1234ze(E)
R1233zd(E)
R1234yf
−60.0
−50.0
−40.0
−30.0
−20.0
−10.0
0.0
10.0
20.0
SATURATION TEMPERATURE AT STANDARD ATMOSPHERIC PRESSURE [°C]

FIG.8

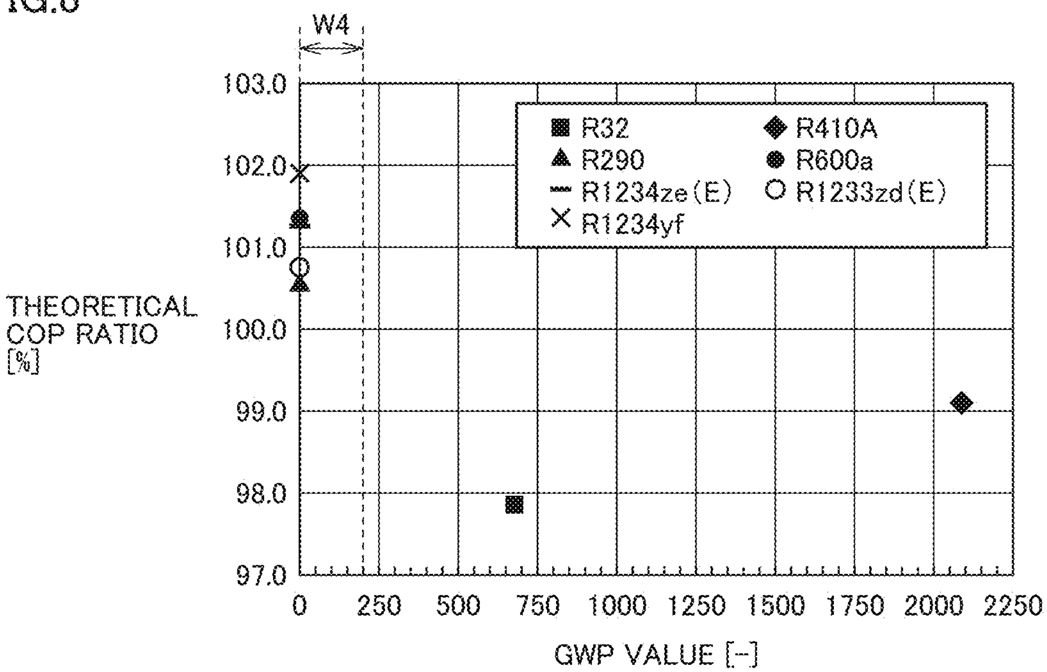

W4
103.0
102.0
101.0
100.0
99.0
98.0
97.0
THEORETICAL COP RATIO [%]
R32
R410A
R290
R600a
R1234ze(E)
R1233zd(E)
R1234yf
0
250
500
750
1000
1250
1500
1750
2000
2250
GWP VALUE [−]

FIG.9

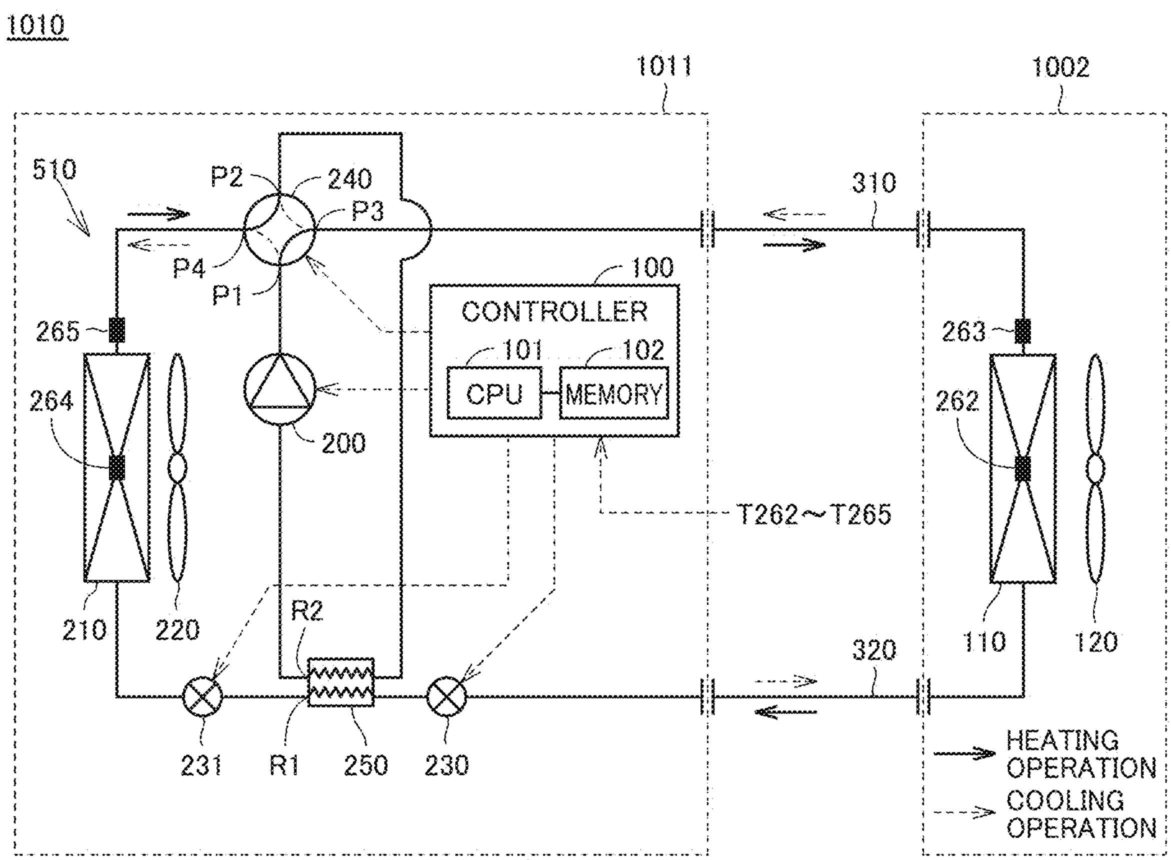

1010
1011
1002
510
P2
240
P3
P4
P1
310
100
CONTROLLER
101
102
CPU
MEMORY
265
263
264
262
200
T262～T265
210
220
R2
320
110
120
231
R1
250
230
HEATING OPERATION
COOLING OPERATION

FIG.10

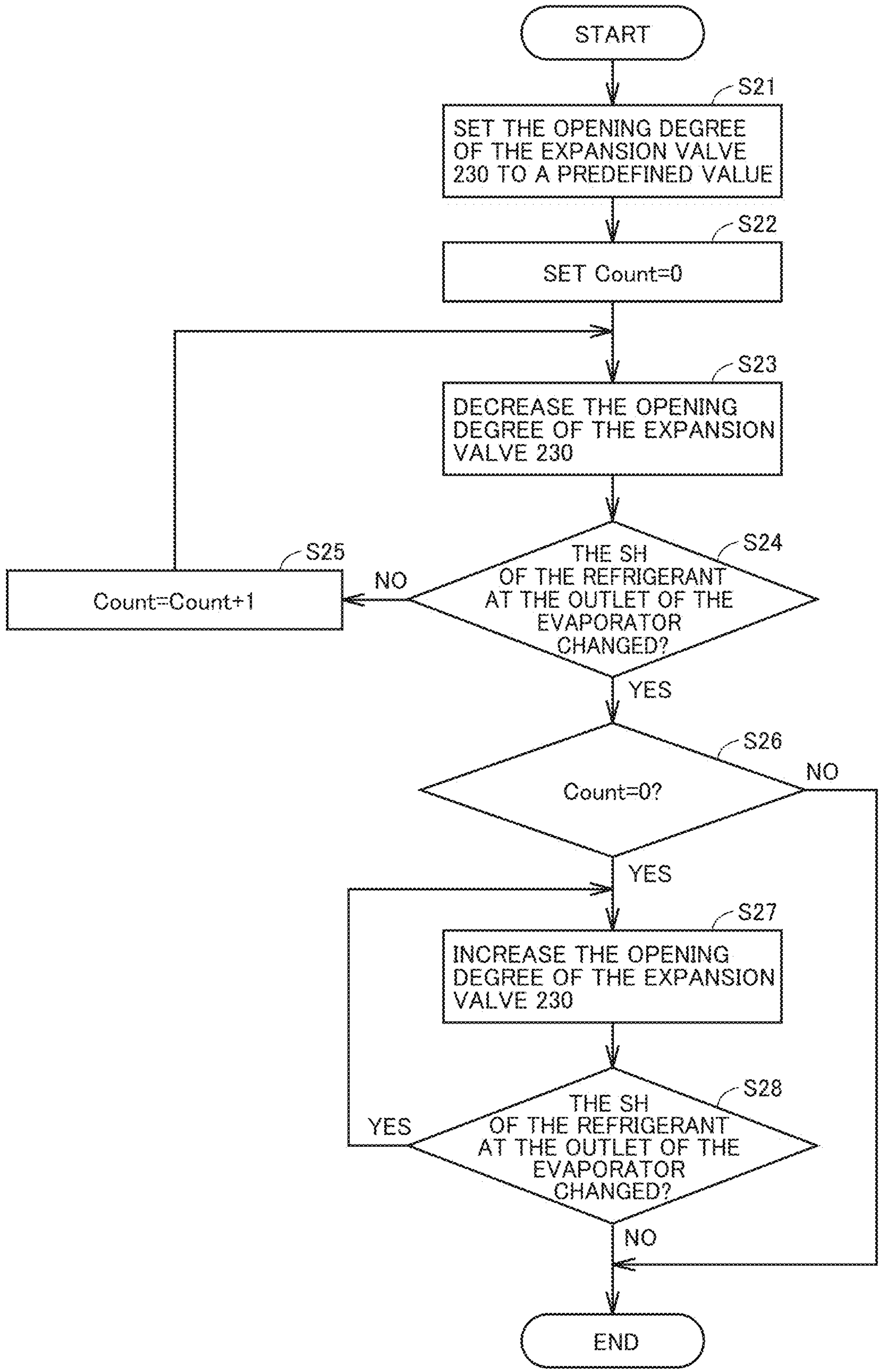

START
S21
SET THE OPENING DEGREE OF THE EXPANSION VALVE 230 TO A PREDEFINED VALUE
S22
SET Count=0
S23
DECREASE THE OPENING DEGREE OF THE EXPANSION VALVE 230
S24
THE SH OF THE REFRIGERANT AT THE OUTLET OF THE EVAPORATOR CHANGED?
NO
S25
Count=Count+1
YES
S26
Count=0?
NO
YES
S27
INCREASE THE OPENING DEGREE OF THE EXPANSION VALVE 230
S28
THE SH OF THE REFRIGERANT AT THE OUTLET OF THE EVAPORATOR CHANGED?
YES
NO
END

FIG.11

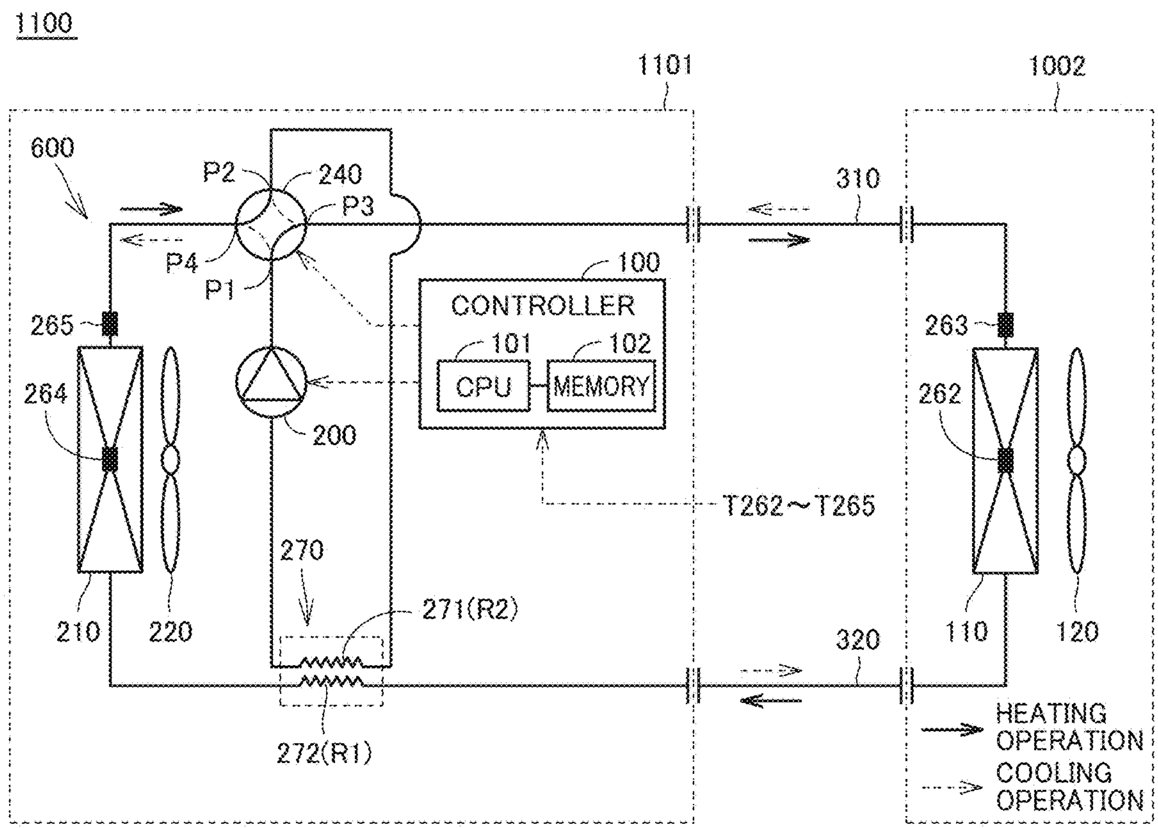


FIG.12

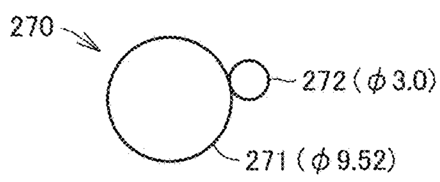


FIG.13

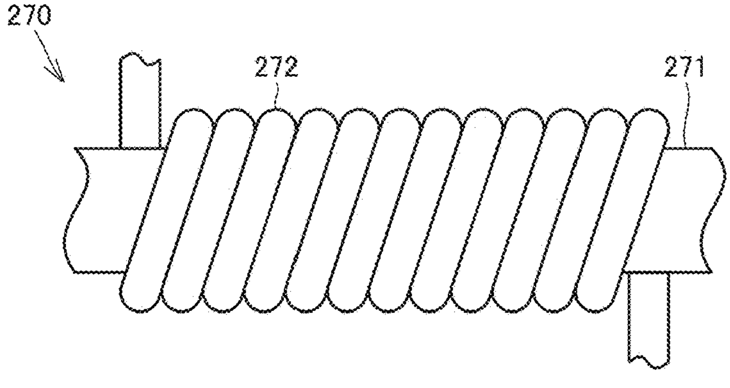
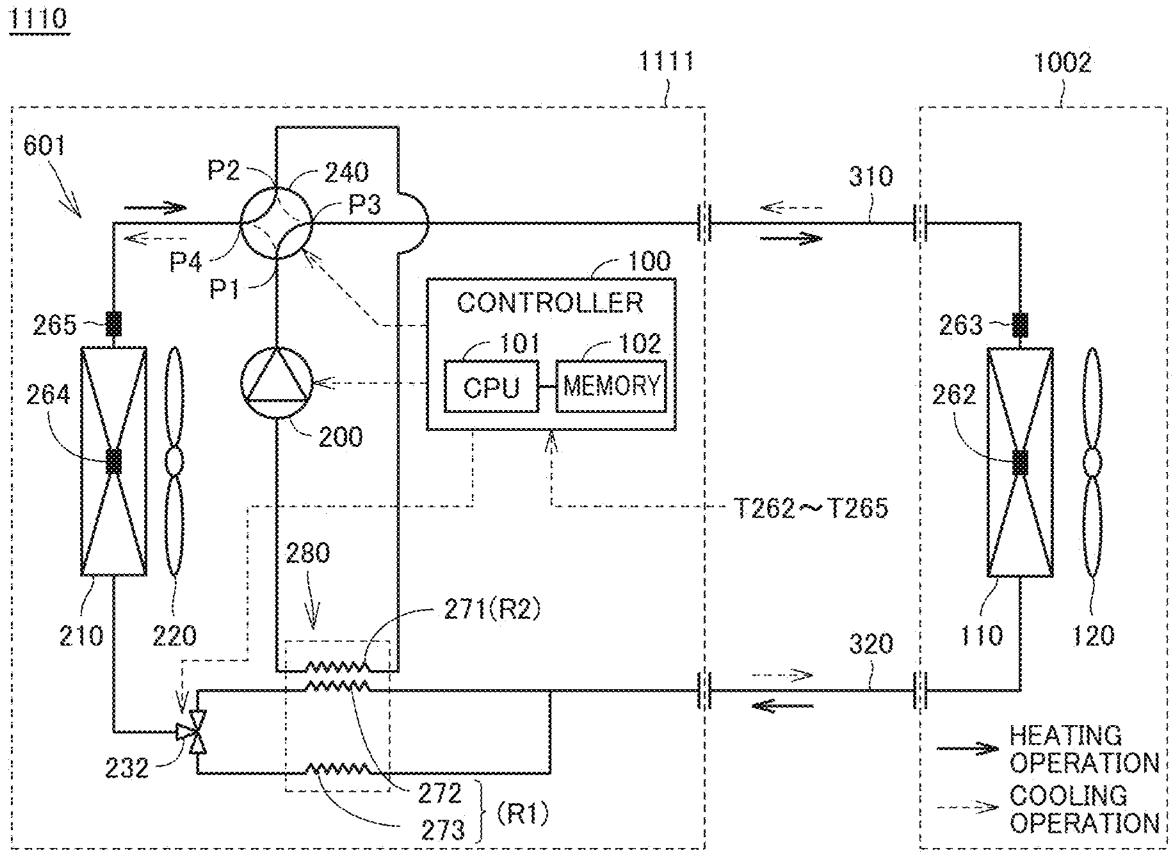
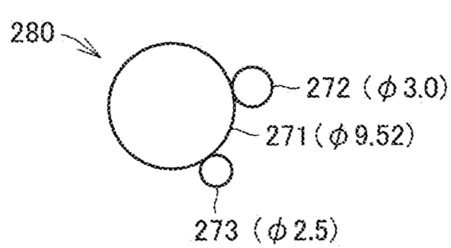


FIG.14



1110
1111
1002
601
P2
240
P3
P4
P1
310
100
CONTROLLER
101
102
CPU
MEMORY
265
263
264
262
200
T262～T265
280
271(R2)
210
220
110
120
320
232
272
273
(R1)
HEATING OPERATION
COOLING OPERATION

FIG.15



280
272 (φ3.0)
271 (φ9.52)
273 (φ2.5)

AIR CONDITIONER

CROSS-REFERENCE TO RELATED APPLICATION

[0001] This application is a U.S. National Stage Application of PCT/JP2022/002222 filed on Jan. 21, 2022, the contents of which are incorporated herein by reference.

TECHNICAL FIELD

[0002] The present disclosure relates to an air conditioner.

BACKGROUND

[0003] Due to the European refrigerant regulations and other requirements, a refrigerant used in a refrigeration cycle of an air conditioner is required to have a low GWP (global warming potential).

[0004] WO 2020/144764 discloses an example of an air conditioner that uses such a refrigerant with a low GWP to improve the coefficient of performance. In order to increase efficiency, the air conditioner includes an internal heat exchanger.

PATENT LITERATURE

[0005] PTL 1: WO 2020/144764

[0006] However, even if a refrigerant has a low GWP, if it is used as a single refrigerant, the pressure of the refrigerant may become too low to deteriorate the energy-saving performance. For this reason, it is considered to mix the refrigerant with other refrigerants. In such a case, the problem is how to mix the refrigerants so as to improve the final coefficient of performance of the air conditioner.

SUMMARY

[0007] The present disclosure has been made to solve the aforementioned problem, and an object of the present disclosure is to provide an air conditioner with an improved coefficient of performance.

[0008] The present disclosure relates to an air conditioner. The air conditioner includes a refrigerant, a refrigerant circuit, a heat exchanger, and a controller. The refrigerant circuit includes at least a compressor, a condenser and an evaporator, and is configured to circulate the refrigerant. The heat exchanger includes a first flow channel through which the refrigerant that has passed through the condenser flows and a second flow channel through which the refrigerant to be sucked into the compressor flows, and is configured to exchange heat between the refrigerant passing through the first flow channel and the refrigerant passing through the second flow channel. The controller is configured to control the refrigerant circuit so as to bring a degree of superheat of the refrigerant flowing through an outlet of the evaporator to 5 degrees or less.

[0009] The air conditioner according to the present disclosure can ensure the degree of superheat of the sucked refrigerant without decreasing the heat exchange performance of the evaporator. This improves the coefficient of performance of the air conditioner that uses an internal heat exchanger to exchange heat between the refrigerant that has passed through the condenser and the refrigerant to be sucked into the compressor.

BRIEF DESCRIPTION OF DRAWINGS

[0010] FIG. **1** is a diagram illustrating a configuration of an air conditioner **1000** according to a first embodiment;

[0011] FIG. **2** is a diagram illustrating a comparison of theoretical coefficients of performance (hereinafter referred to as "theoretical COP") of the air conditioner when various types of refrigerant are used;

[0012] FIG. **3** is a PH diagram of a refrigeration cycle apparatus that uses R290 as a refrigerant and is not provided with an internal heat exchanger;

[0013] FIG. **4** is a PH diagram of a refrigeration cycle apparatus that uses R290 as a refrigerant and is provided with an internal heat exchanger;

[0014] FIG. **5** is a graph illustrating the relationship between the fraction of R32 in a refrigerant mixture R32/R1234yf and the theoretical COP ratio;

[0015] FIG. **6** is a graph illustrating the relationship between the fraction of R32 in the refrigerant mixture R32/R1234yf and the saturation temperature at standard atmospheric pressure and the GWP value;

[0016] FIG. **7** is a graph illustrating the relationship between the saturation temperature of various refrigerants at standard atmospheric pressure and the theoretical COP ratio;

[0017] FIG. **8** is a graph illustrating the relationship between the GWP values of various refrigerants and the theoretical COP ratios;

[0018] FIG. **9** is a diagram illustrating a modification of the air conditioner illustrated in FIG. **1**;

[0019] FIG. **10** is a flowchart illustrating the control of the expansion valve **230**;

[0020] FIG. **11** is a diagram illustrating a configuration of an air conditioner according to a second embodiment;

[0021] FIG. **12** is a cross-sectional view illustrating a specific example of a pressure-reducing heat exchanger **270**;

[0022] FIG. **13** is a diagram illustrating another specific example of the pressure-reducing heat exchanger **270**;

[0023] FIG. **14** is a diagram illustrating a configuration of a modification of the air conditioner illustrated in FIG. **11**; and

[0024] FIG. **15** is a cross-sectional view illustrating a configuration example of a pressure-reducing heat exchanger **280**.

DETAILED DESCRIPTION

[0025] Hereinafter, embodiments of the present disclosure will be described in detail with reference to the drawings. In the following description, a plurality of embodiments will be described, and appropriate combinations of components described in the respective embodiments are also originally intended. In the drawings, the same or corresponding parts are denoted by the same reference numerals, and the description thereof will not be repeated. In the following drawings, the dimensions of each component may be different from the actual ones.

First Embodiment

[0026] FIG. **1** is a diagram illustrating a configuration of an air conditioner **1000** according to a first embodiment. The air conditioner **1000** illustrated in FIG. **1** includes a refrigerant circuit **500**, an internal heat exchanger **250**, and a controller **100**.

[0027] The refrigerant circuit **500** includes at least a compressor **200**, an outdoor heat exchanger **210**, an expansion valve **230** and an indoor heat exchanger **110**, and is configured to circulate the refrigerant. The refrigerant used in the present embodiment has a GWP value or a saturation temperature at standard atmospheric pressure in a certain range to be described later in detail.

[0028] In the example of FIG. **1**, the refrigerant circuit **500** includes a compressor **200**, an outdoor heat exchanger **210**, an outdoor blower **220**, an expansion valve **230**, a four-way valve **240**, an indoor heat exchanger **110**, and an indoor blower **120**. The four-way valve **240** has ports P**1** to P**4**. A linear expansion valve (LEV), for example, can be used as the expansion valve **230**.

[0029] The refrigerant circuit **500** is divided into an outdoor unit **1001** and an indoor unit **1002**. The outdoor unit **1001** includes a compressor **200**, a four-way valve **240**, an outdoor heat exchanger **210**, an outdoor blower **220**, an expansion valve **230**, a controller **100**, and an internal heat exchanger **250**. The indoor unit **1002** includes an indoor heat exchanger **110** and an indoor blower **120**. The outdoor unit **1001** and the indoor unit **1002** are connected by a pipe **310** and a pipe **320**.

[0030] The compressor **200** is configured to change the operating frequency according to a control signal received from the controller **100**. Specifically, the compressor **200** incorporates an inverter-controlled drive motor with variable rotation speed, and the rotational speed of the drive motor changes as the operating frequency is changed. The output of the compressor **200** is adjusted by changing the operating frequency of the compressor **200**. The compressor **200** may be any type of compressors such as a rotary compressor, a reciprocating compressor, a scroll compressor, or a screw compressor.

[0031] The four-way valve **240** is controlled by a control signal received from the controller **100** to switch the operation mode of the air conditioner to a cooling operation mode or a heating operation mode. In the cooling operation mode, as illustrated by the broken lines, the port P**1** communicates with the port P**4**, and the port P**2** communicates with the port P**3**. In the heating operation mode, as illustrated by the solid lines, the port P**1** communicates with the port P**3**, and the port P**2** communicates with the port P**4**. When the compressor **200** is operated in the cooling operation mode, the refrigerant is circulated in the refrigerant circuit in the direction indicated by the broken arrows. When the compressor **200** is operated in the heating operation mode, the refrigerant is circulated in the refrigerant circuit in the direction indicated by the solid arrows.

[0032] The internal heat exchanger **250** includes a flow channel R**1** and a flow channel R**2**. During the cooling operation, high-pressure high-temperature refrigerant that has passed through the condenser (the outdoor heat exchanger **210**) flows through the flow channel R**1**. During the cooling operation, low-pressure and low-temperature refrigerant to be sucked into the compressor **200** flows through the flow channel R**2**. During the cooling operation, the internal heat exchanger **250** exchanges heat between the high-pressure high-temperature refrigerant that has passed through the condenser (the outdoor heat exchanger **210**) and the low-pressure low-temperature refrigerant to be sucked into the compressor **200**.

[0033] The air conditioner **1000** further includes temperature sensors **262** to **265**. The temperature sensor **262** is disposed in the indoor heat exchanger **110**, and is configured to measure a refrigerant temperature T**262** which is an evaporation temperature of the refrigerant during the cooling operation or a condensation temperature of the refrigerant during the heating operation. The temperature sensor **263** is disposed in a pipe that connects the indoor heat exchanger **110** to the port P**3** of the four-way valve **240**, and is configured to measure a temperature T**263** of the refrigerant. The temperature sensor **264** is disposed in the outdoor heat exchanger **210**, and is configured to measure a refrigerant temperature T**264** which is a condensation temperature of the refrigerant during the cooling operation or an evaporation temperature of the refrigerant during the heating operation. The temperature sensor **265** is disposed in a pipe connecting the outdoor heat exchanger **210** and the port P**4** of the four-way valve **240**, and is configured to measure a temperature T**265** of the refrigerant.

[0034] The controller **100** controls the opening degree of the expansion valve **230** so as to adjust the SH (the degree of superheat) of the refrigerant at an outlet of the evaporator according to the outputs from the temperature sensors **262** to **265**.

[0035] The controller **100** includes a CPU (Central Processing Unit) **101**, a memory **102** (such as a ROM (Read Only Memory) or a RAM (Random Access Memory)), an input/output buffer (not shown), and the like. The CPU **101** loads programs stored in the ROM into the RAM or the like and executes the programs. The programs stored in the ROM are programs that describe the processing procedure of the controller **100**. The controller **100** controls each device in the air conditioner **1000** in accordance with these programs. This control is not limited to being processed by software, but may be processed by dedicated hardware (electronic circuit).

[0036] As described in the above, the refrigerant circuit of the air conditioner **1000** includes the compressor **200**, the outdoor heat exchanger **210**, the outdoor blower **220**, the expansion valve **230**, the four-way valve **240**, the internal heat exchanger **250**, the indoor heat exchanger **110**, and the indoor blower **120**. During the cooling operation, the internal heat exchanger **250** exchanges heat between the high-pressure refrigerant flowing out from the outdoor heat exchanger **210** through the flow channel R**1** and the low-pressure refrigerant flowing through the flow channel R**2** into the compressor **200**.

[0037] Next, various types of refrigerants to be used in the air conditioner having the configuration illustrated in FIG. **1** will be examined. FIG. **2** is a diagram illustrating a comparison of theoretical coefficients of performance (hereinafter referred to as "theoretical COP") of the air conditioner when various types of refrigerants are used. The results of FIG. **2** are obtained through calculations by setting the supercooling degree (SC) to 0 degrees, the evaporation temperature (ET) to 17° C., the condensation temperature (CT) to 40° C. and the compressor efficiency to 1.

[0038] As illustrated in FIG. **2**, the theoretical COP of a hydrofluorocarbon refrigerant (hereinafter referred to as "HFC refrigerant") such as R32 and R410A decreases as the degree of superheat (SH) of the refrigerant to be sucked into the compressor increases. On the other hand, the theoretical COP of a hydrocarbon refrigerant (hereinafter referred to as "HC refrigerant") such as R290 or R600a increases as the degree of superheat (SH) of the sucked refrigerant increases. Similar to the HC refrigerant, the theoretical COP of a hydrofluoroolefin refrigerant (hereinafter referred to as "HFO refrigerant") such as R1234ze(E), R1233zd(E) or

R1234yf also increases as the degree of superheat (SH) of the sucked refrigerant increases.

[0039] This is because when the degree of superheat (SH) of the refrigerant to be sucked into the compressor increases, the increase ratio of the evaporator enthalpy difference is greater than the decrease ratio of the refrigerant circulation amount in the case of the HC refrigerant, the HFO refrigerant and other refrigerants than in the case of the HFC refrigerant.

[0040] Therefore, in an air conditioner that uses the HC refrigerant or the HFO refrigerant, it is theoretically possible to make the air conditioner operate with high performance by controlling the degree of superheat of the sucked refrigerant. However, as the degree of superheat (SH) of the refrigerant to be sucked into the compressor increases, the proportion of the gas refrigerant in the evaporator increases, which decreases the performance of the evaporator, and whereby the evaporation temperature decreases, resulting in a problem that the actual coefficient of performance (hereinafter referred to as "actual COP") of the air conditioner will decrease.

[0041] In order to solve this problem, according to the air conditioner of the present embodiment, in addition to controlling the degree of superheat at the outlet of the evaporator to 5 degrees or less and using an internal heat exchanger, by using a low GWP refrigerant that satisfies certain conditions, it is possible to make the air conditioner operate with high performance while preventing the performance of the evaporator from decreasing.

[0042] Next, how the PH diagram changes depending on the presence or absence of an internal heat exchanger will be described with reference to FIGS. **3** to **4**.

[0043] FIG. **3** is a PH diagram of a refrigeration cycle apparatus that uses R290 as a refrigerant and is not provided with an internal heat exchanger. FIG. **4** is a PH diagram of a refrigeration cycle apparatus that uses R290 as a refrigerant and is provided with an internal heat exchanger.

[0044] The result in the case where the internal heat exchanger is not provided as illustrated in FIG. **3** was calculated by assuming the degree of superheat (SH) of the sucked refrigerant is 10 degrees, the degree of supercooling (SC) of the refrigerant at the outlet of the condenser is 0 degrees, the evaporation temperature (ET) is 17° C., the condensation temperature (CT) is 40° C., and the compressor efficiency is 1. On the other hand, the result in the case where the internal heat exchanger is provided as illustrated in FIG. **4** was calculated by assuming that the capacity of the evaporator is the same, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator is controlled at 0 degrees by using the expansion valve **230**, and the low-temperature and low-pressure refrigerant at the outlet of the evaporator increases by 10° C. as the internal heat exchanger **250** exchanges heat (Δh) with the high-temperature and high-pressure refrigerant at the outlet of the condenser. In other words, the degree of superheat (SH) of the sucked refrigerant is 10 degrees. Since the capacity of the compressor is the same in both FIG. **3** and FIG. **4**, the evaporation temperature (ET) is the same regardless of the refrigerant temperature at the outlet of the evaporator.

[0045] The theoretical COP is equal to 10.82 in both the configuration where the internal heat exchanger is provided and the configuration where the internal heat exchanger is not provided. However, in the configuration where the internal heat exchanger is not provided, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator is equal to the degree of superheat of the refrigerant to be sucked into the compressor. Therefore, when the degree of superheat of the sucked refrigerant is 10 degrees, the degree of dryness of the refrigerant is equal to I before the outlet of the evaporator, and thereby, the performance of the evaporator decreases by an amount corresponding to a decrease in the heat exchange performance caused by the portion of the gas refrigerant. Therefore, the evaporation temperature ET becomes lower than 17° C., and the actual COP becomes lower than that of the configuration where the internal heat exchanger is provided (evaporator SH=0 degrees). By providing an internal heat exchanger in the air conditioner that uses R290 as the refrigerant illustrated in FIGS. **3** and **4**, it is possible to improve the performance by increasing the degree of superheat (SH) of the refrigerant to be sucked into the compressor without reducing the actual COP.

[0046] Generally, a refrigerant is required to have such a characteristic that the theoretical COP increases as the superheating degree (SH) of the refrigerant to be sucked into the compressor increases. In other words, generally a refrigerant is required to improve the actual COP when an internal heat exchanger is provided. As described with reference to FIG. **2**, in the case of an HFC refrigerant, the theoretical COP decreases when the degree of superheat (SH) of the refrigerant to be sucked into the compressor increases, but in the case of an HFO refrigerant, the theoretical COP increases when the degree of superheat (SH) of the refrigerant to be sucked into the compressor increases.

[0047] Currently, an HFO refrigerant such as R1234yf is used in a car air conditioner or the like. However, in a residential air conditioner or the like which is used in an airtight and well-insulated space and is configured to pursue better energy saving performance, in order to prevent the evaporation temperature from becoming excessively low with respect to an indoor temperature, it has been studied to use R32, which is an HFC refrigerant used in a residential air conditioner, as a mixture with R1234yf.

[0048] The following study examines that when a refrigerant mixture composed of R32 (an HFC refrigerant) and R1234yf (an HFO refrigerant) is used and the superheating degree (SH) of the refrigerant to be sucked into the compressor is ensured, to what extent an increase in the fraction of R32 will decrease the theoretical COP.

[0049] FIG. **5** is a graph illustrating the relationship between the fraction of R32 in the refrigerant mixture R32/R1234yf and the theoretical COP ratio. The theoretical COP ratio is the ratio of the theoretical COP in the case where the internal heat exchanger is provided to the theoretical COP in the case where the internal heat exchanger is not provided. In other words, the theoretical COP ratio is (the theoretical COP of the configuration with the internal heat exchanger)/(the theoretical COP of the configuration without the internal heat exchanger).

[0050] FIG. **5** shows the calculation result when the temperature at the inlet of the internal heat exchanger changes by 10 degrees due to heat exchange. As can be seen from FIG. **5**, when the mass fraction of R32 exceeds 30%, the theoretical COP of the configuration where the internal heat exchanger is provided will not increase. In other words, the effective range **W1** of the mass fraction of R32 in the case where the internal heat exchanger is provided is 0% to 30%.

[0051] FIG. **6** is a graph illustrating the relationship between the fraction of R32 in the refrigerant mixture R32/R1234yf and the saturation temperature at standard atmospheric pressure and the GWP value. The saturation temperature of a refrigerant depends on the dryness of the refrigerant at the same pressure but with a temperature gradient. The results illustrated in FIG. **6** represent the saturation temperature at the standard atmospheric pressure when the dryness is 0.5.

[0052] The effective range W**1** (0% to 30%) of the mass fraction of R32 in the case where the internal heat exchanger is provided, which is obtained in FIG. **5**, is also applied to FIG. **6**. Thus, if a refrigerant has a saturated temperature of −44.4° C. or more at the standard atmospheric pressure, it is effective in the case where the internal heat exchanger is provided. Similarly, if a refrigerant has a GWP of 205 or less, it is effective in the case where the internal heat exchanger is provided.

[0053] FIG. **7** is a graph illustrating the relationship between the saturation temperature of various refrigerants at standard atmospheric pressure and the theoretical COP ratio. FIG. **8** is a graph illustrating the relationship between the GWP values of various refrigerants and the theoretical COP ratio.

[0054] At the time of calculating the theoretical COP ratio, the COP in the case where the internal heat exchanger is not provided was calculated by assuming the degree of superheat (SH) of the sucked refrigerant is 10 degrees, the degree of supercooling (SC) is 0 degrees, the evaporation temperature (ET) is 17° C., the condensation temperature (CT) is 40° C., and the compressor efficiency is 1. On the other hand, the COP in the case where the internal heat exchanger is provided was calculated by assuming that the capacity of the evaporator is the same, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator is controlled at 0 degrees by using the expansion valve, and the internal heat exchanger exchanges an amount of heat of 10° C. at the inlet. As can be seen from FIGS. **7** and **8**, the saturation temperature range W**3** and the GWP value range W**4** which are obtained by applying the range W**1** to FIG. **6** and effective in the case where the internal heat exchanger is provided are generally true for various refrigerants except for R32 and R410A.

[0055] Therefore, the refrigerant suitably used in the present embodiment has such a characteristic that the saturated temperature at the standard atmospheric pressure is −44.4° C. or more or the GWP is 205 or less. The developer or the user of the air conditioner may use this characteristic as an indicator to select a refrigerant.

[0056] In the configuration illustrated in FIG. **1**, the internal heat exchanger **250** is configured to operate mainly during the cooling operation, it also may be configured to operate mainly during the heating operation. Alternatively, as illustrated in the following modifications, an expansion valve may be further provided.

[0057] FIG. **9** is a diagram illustrating a modification of the air conditioner illustrated in FIG. **1**. The air conditioner **1010** illustrated in FIG. **9** includes a refrigerant circuit **510** instead of the refrigerant circuit **500**. The refrigerant circuit **510** further includes a second expansion valve **231** inside the outdoor unit **1011** as compared with the refrigerant circuit **500**. An electronic expansion valve can be used as the expansion valve **231**. The expansion valve **231** is connected between the first flow channel R**1** of the internal heat exchanger **250** and the outdoor heat exchanger **210**.

[0058] The controller **100** fully opens the second expansion valve **231** and controls the superheat (SH) of the refrigerant at the outlet of the evaporator with the expansion valve **230** during the cooling operation, and fully opens the expansion valve **230** and controls the superheat (SH) of the refrigerant at the outlet of the evaporator with the second expansion valve **231** during the heating operation. This makes it possible to improve the efficiency of heat exchange in the internal heat exchanger **250** during both the cooling operation and the heating operation.

[0059] The internal heat exchanger **250** may be any heat exchanger as long as it exchanges heat between the high-pressure refrigerant that has passed through the condenser and the low-pressure refrigerant to be sucked into the compressor, and for example, it may be a double-tube heat exchanger composed of an inner pipe and an outer pipe, or may be a heat exchanger in which the high-pressure pipe and the low-pressure pipe are brazed and brought into contact with each other by soldering or the like to perform heat exchange.

[0060] Although the air conditioner illustrated in FIGS. **1** and **9** is provided with a four-way valve, the air conditioner may not be provided with a four-way valve and thereby is used exclusively for cooling.

[0061] In FIGS. **1** and **9**, the flow of the refrigerant in the heating operation mode is indicated by a solid line, and the flow of the refrigerant in the cooling operation mode is indicated by a broken line. Similar to a general air conditioner, the controller **100** changes the frequency of the compressor **200** so as to bring the indoor temperature to a target (set) temperature.

[0062] The following describes the control of the expansion valve **230** illustrated in FIG. **1** and the control of the expansion valve **230** when the expansion valve **231** is fixed to fully open as illustrated in FIG. **9**. FIG. **10** is a flowchart illustrating the control of the expansion valve **230**.

[0063] Firstly, in step S**21**, the controller **100** sets the opening degree of the expansion valve **230** to a predefined value. After a certain period of time has elapsed, in step S**22**, the controller **100** initializes the variable Count to 0. Thereafter, in step S**23**, the controller **100** decreases the opening degree of the expansion valve **230** by a certain value. After a certain period of time has elapsed, in step S**24**, the controller **100** determines whether or not the degree of superheat (SH) of the refrigerant at the outlet of the evaporator has changed.

[0064] During the cooling operation, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator is calculated by subtracting the evaporation temperature of the refrigerant obtained by the temperature sensor **262** from the refrigerant temperature at the outlet of the evaporator obtained by the temperature sensor **263**. During the heating operation, the degree of refrigerant superheat (SH) at the outlet of the evaporator is calculated by subtracting the evaporation temperature of the refrigerant obtained by the temperature sensor **264** from the refrigerant temperature at the outlet of the evaporator obtained by the temperature sensor **265**.

[0065] If the determination result is NO in step S**24**, in other words, if the degree of superheat (SH) of the refrigerant at the outlet of the evaporator is not changed, since the state of the refrigerant at the outlet of the evaporator is not changed from the gas-liquid two-phase state, the controller **100** adds 1 to the variable Count in step S**25**, returns the procedure to step S**23**, and decreases the opening degree of the expansion valve **230** by a certain value.

[0066] By repeating the process of steps S**23** to S**25**, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator can be brought to an optimum state where the degree of superheat is substantially zero. On the other hand, if the determination result is YES in step S**24** (when the refrigerant at the outlet of the evaporator is superheated gas), the controller **100** proceeds the procedure to step S**26** to determine whether or not the variable Count is 0.

[0067] If the determination result is NO in step S**26**, in other words, if the variable Count is not 0, since the degree of superheat (SH) of the refrigerant at the outlet of the evaporator has been appropriately controlled in steps S**23** to S**25**, the controller **100** ends the procedure of this flowchart. On the other hand, if the determination result is YES in step S**26**, since the variable Count is 0, the procedure passes step S**24** once without going through step S**25**. Since the refrigerant at the outlet of the evaporator is in a superheated gas state at the default opening degree in step S**21**, and the degree of superheat (SH) of the refrigerant is further increased as a result of the process in step S**23** from that state, the state cannot be regarded as an appropriate state. Thus, the controller **100** increases the opening degree of the expansion valve **230** by a certain value in step S**27**. After a certain period of time has elapsed, in step S**28**, the controller **100** determines whether or not the degree of superheat (SH) of the refrigerant at the outlet of the evaporator has changed.

[0068] If the determination result is NO in step S**28**, in other words, if the degree of superheat (SH) of the refrigerant at the outlet of the evaporator has changed, since there is a change in the degree of superheat (SH) of the refrigerant at the outlet of the evaporator, the process returns to step S**27** where the controller **100** increases the opening degree of the expansion valve **230** by a certain value. By repeating the process of steps S**27** to S**28**, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator can be brought to an optimum state where the degree of superheat is substantially zero. On the other hand, if the determination result is YES in step S**28**, in other words, if there is no change in the degree of superheat (SH) of the refrigerant at the outlet of the evaporator, it can be determined that the refrigerant at the outlet of the evaporator is in a gas-liquid two-phase state (the degree of superheat is 0), and the heat exchange efficiency of the evaporator is good, the controller **100** ends the procedure of this flowchart.

[0069] When a certain period of time has elapsed, the procedure of the flowchart in FIG. **10** is performed again, whereby the opening degree of the expansion valve **230** is set to bring the degree of superheat of the refrigerant at the outlet of the evaporator to substantially zero. In this case, the opening degree of the expansion valve **230** determined in a previous procedure may be used as the predefined value in step S**21**.

[0070] The control of the expansion valve **230** has been described above. In the flowchart illustrated in FIG. **10**, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator is controlled to approach the target value (zero), however, the target value of the degree of superheat (SH) of the refrigerant discharged from the compressor **200** or the target value of the temperature of the refrigerant discharged from the compressor **200** corresponding to the degree of superheat (SH) of the refrigerant at the outlet of the evaporator may be determined in advance, and the degree of superheat (SH) or the temperature of the refrigerant discharged from the compressor may be controlled to be the target value. Also, the expansion valve **231** in the configuration of FIG. **9** during the heating operation may be controlled in the same manner as in FIG. **10** with the expansion valve **230** fixed to fully open.

[0071] The method for calculating the degree of superheat (SH) of the refrigerant at the outlet of the evaporator described in the flowchart of FIG. **10** was based on temperature sensors. In this method, when there is a pressure loss between temperature sensors or when there is a temperature gradient in the refrigerant, the degree of superheat (SH) of the refrigerant changes. When the refrigerant has a larger pressure loss, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator obtained by this method becomes smaller, and when the refrigerant has a larger temperature gradient, the degree of superheat (SH) of the refrigerant at the outlet of the evaporator obtained by this method becomes larger. However, in real life, if the degree of superheat (SH) is about 5 degrees, the performance of the evaporator does not drop that much, and thus, the degree of superheat (SH) may be controlled to 5 degrees or less to tolerate some error.

[0072] As described in the above, according to the air conditioner of the first embodiment, it is possible to ensure the degree of superheat (SH) of the refrigerant to be sucked into the compressor without deteriorating the performance of the evaporator even when a low GWP refrigerant is used, which makes it possible to improve the COP of the air conditioner.

Second Embodiment

[0073] FIG. **11** is a diagram illustrating a configuration of an air conditioner according to a second embodiment. Hereinafter, the differences from the configuration of FIG. **1** will be described. The air conditioner **1100** illustrated in FIG. **11** includes a refrigerant circuit **600** instead of the refrigerant circuit **500** illustrated in FIG. **1**. The refrigerant circuit **600** includes a pressure-reducing heat exchanger **270** inside the outdoor unit **1101** instead of the internal heat exchanger **250** and the expansion valve **230** in the refrigerant circuit **500**.

[0074] In other words, in the air conditioner **1100**, the internal heat exchanger **250** and the expansion valve **230** are replaced by a pressure-reducing heat exchanger **270** which solely performs pressure reduction and heat exchange of the high-pressure refrigerant. The pressure-reducing heat exchanger **270** includes a low-pressure pipe **271** through which low-pressure refrigerant flows and a first medium-pressure pipe **272** through which medium-pressure refrigerant flows. The inner diameter of the first medium-pressure pipe **272** is configured to be smaller than that of the low-pressure pipe **271**. Further, the inner diameter of the first medium-pressure pipe **272** is configured to be smaller than that of the pipes connected to both ends of the medium-pressure pipe **272** so as to reduce the pressure of the high-pressure refrigerant flowing out from the condenser. The first low-pressure pipe **271** and the first medium-pressure pipe **272** are in contact with each other so as to exchange heat. Specifically, the two pipes are brazed by solder or the like, and are brought into contact with each other so as to exchange heat between the two pipes.

[0075] FIG. **12** is a cross-sectional view illustrating a specific example of the pressure-reducing heat exchanger **270**. When the air conditioner is a room air conditioner, the diameter of the low-pressure pipe **271** is φ9.52, and the diameter of the first medium-pressure pipe **272** is φ3.0. The diameter of the low-pressure pipe **271** is set larger so as to reduce the effect of the pressure loss, and the diameter of the medium-pressure pipe **272** is set smaller so as to reduce the pressure from high pressure to low pressure.

[0076] FIG. **13** is a diagram illustrating another specific example of the pressure-reducing heat exchanger **270**. In the example illustrated in FIG. **13**, the first medium-pressure pipe **272** is spirally wound around the low-pressure pipe **271**. This increases the contact area of the first medium-pressure pipe **272** per unit length of the low-pressure pipe **271**, which improves the heat exchange efficiency. Accordingly, the length of the low-pressure pipe **271** for exchanging a required amount of heat in the pressure-reducing heat exchanger **270** can be made shorter than the length along which the medium-pressure pipe **272** is not wound. Since the length of the low-pressure pipe **271** is shortened, the pipe can be easily routed in a machine room. Further, since the pressure loss of the low-pressure pipe **271** is reduced by shortening the length, it is possible to reduce the diameter of the low-pressure pipe **271**.

[0077] In the configuration of FIG. **11**, the first medium-pressure pipe **272** is one pipe, and since one pipe can only form one fixed throttle, a plurality of pipes may be installed in parallel.

[0078] FIG. **14** is a diagram illustrating a configuration of a modification of the air conditioner illustrated in FIG. **11**. The air conditioner **1110** illustrated in FIG. **14** includes a refrigerant circuit **601** instead of the refrigerant circuit **600** illustrated in FIG. **11**. The refrigerant circuit **601** includes a pressure-reducing heat exchanger **280** inside the outdoor unit **1111** instead of the pressure-reducing heat exchanger **270** in the configuration of the refrigerant circuit **600**.

[0079] The pressure-reducing heat exchanger **280** further includes a second medium-pressure pipe **273** in addition to the low-pressure pipe **271** and the first medium-pressure pipe **272**. The medium-pressure pipe **272** and the medium-pressure pipe **273** have different diameters. The flow channel may be switched by the switching valve **232** so that the refrigerant flows through a pipe having an optimal inner diameter corresponding to the refrigerant circulation amount, or the pipe through which the refrigerant flows may be changed between the cooling operation and the heating operation. Preferably, the diameter of the medium-pressure pipe through which the refrigerant flows during the heating operation is smaller than the diameter of the medium-pressure pipe through which the refrigerant flows during the cooling operation.

[0080] The reason for this is that the temperature difference between the air to be heat-exchanged by the indoor heat exchanger **110** and the air to be heat-exchanged by the outdoor heat exchanger **210** is larger during the heating operation than during the cooling operation, and thereby, it is suitable to increase the throttle amount in the medium-pressure pipe during the heating operation than during the cooling operation. For example, under the standard cooling conditions defined by the Japan Industrial Standard (JIS), the air temperature at the inlet of the outdoor heat exchanger **210** is 35° C., and the air temperature at the inlet of the indoor heat exchanger **110** is 27° C., the temperature difference between the two air temperatures is 8° C. On the other hand, under the standard heating conditions, the air temperature at the inlet of the indoor heat exchanger **110** is 20° C., and the air temperature at the inlet of the outdoor heat exchanger **210** is 7° C., the temperature difference between the two air temperatures is 13° C.

[0081] FIG. **15** is a cross-sectional view illustrating a configuration example of the pressure-reducing heat exchanger **280**. As illustrated in FIG. **15**, when there are a plurality of medium-pressure pipes, the plurality of medium-pressure pipes are configured to have different diameters, and are disposed in contact with the low-pressure pipe to exchange heat.

[0082] Specifically, if the air conditioner is a room air conditioner, the diameter of the low-pressure pipe **271** is φ9.52, the diameter of the first intermediate-pressure pipe **272** is φ3.0, and the diameter of the second intermediate-pressure pipe **273** is φ2.5. The diameter of the low-pressure pipe **271** should be larger to reduce the influence of the pressure loss, and the diameters of the medium-pressure pipes **272** and **273** should be smaller to reduce the pressure from a high pressure to a low pressure.

[0083] As described with reference to FIG. **14**, it is preferable that the refrigerant flows through the medium-pressure pipe **273** having a diameter of 2.5 mm during the heating operation, and flows through the medium-pressure pipe **272** having a diameter of 3.0 mm during the cooling operation.

[0084] As described above, according to the air conditioner of the second embodiment, in addition to the effects exhibited by the air conditioner according to the first embodiment, since the temperature difference in the heat exchanger can always be ensured in either the cooling operation or the heating operation, the performance of the air conditioner can be improved in both the cooling operation and the heating operation. Further, since the expansion valve **230** is not provided, the air conditioner can be made cheaper.

Summary

[0085] Hereinafter, embodiments of the present invention will be summarized with reference to the drawings. Note that the units in parentheses correspond to those used in the cooling operation.

[0086] The present disclosure relates to an air conditioner. The air conditioner **1000** illustrated in FIG. **1** includes a refrigerant, a refrigerant circuit **500**, a heat exchanger **250**, and a controller **100**. The refrigerant circuit **500** includes at least a compressor **200**, a condenser (an outdoor heat exchanger **210**) and an evaporator (an indoor heat exchanger **110**), and is configured to circulate the refrigerant. The heat exchanger **250** includes a first flow channel R**1** through which the refrigerant that has passed through the condenser (the outdoor heat exchanger **210**) flows and a second flow channel R**2** through which the refrigerant to be sucked into the compressor **200** flows, and is configured to exchange heat between the refrigerant passing through the first flow channel R**1** and the refrigerant passing through the second flow channel R**2**. The controller **100** is configured to control the refrigerant circuit **500** so as to bring the degree of superheat of the refrigerant flowing through an outlet of the evaporator (the indoor heat exchanger **110**) to 5 degrees or less.

[0087] Preferably, the refrigerant circuit **500** illustrated in FIG. **1** further includes an expansion valve **230** configured to expand the refrigerant condensed in the condenser (the outdoor heat exchanger **210**). The controller **100** controls an opening degree of the expansion valve **230** to change the degree of superheat of the refrigerant flowing through the outlet of the evaporator (the indoor heat exchanger **110**).

[0088] More preferably, during the cooling operation, the condenser (the outdoor heat exchanger **210**) is configured to condense the refrigerant, and the evaporator (the indoor heat exchanger **110**) is configured to evaporate the refrigerant. The refrigerant circuit **500** further includes a four-way valve **240** that changes the flow direction of the refrigerant passing through the condenser (the outdoor heat exchanger **210**) and the evaporator (the indoor heat exchanger **110**) from the flow direction during the cooling operation so as to cause the condenser (the outdoor heat exchanger **210**) to evaporate the refrigerant and cause the evaporator (the indoor heat exchanger **110**) to condense the refrigerant during the heating operation. The first flow channel R**1** of the heat exchanger **250** is disposed between the condenser (the outdoor heat exchanger **210**) and the expansion valve **230**, and the second flow channel R**2** of the heat exchanger **250** is disposed between the four-way valve **240** and the inlet of the compressor **200**.

[0089] Preferably, as illustrated in FIGS. **11** and **12**, the pressure of the refrigerant passing through the first flow channel (the middle-pressure pipe **272**) is higher than the pressure of the refrigerant passing through the second flow channel (the low-pressure pipe **271**), and the inner diameter of the pipe for the first flow channel (the middle-pressure pipe **272**) is smaller than the inner diameter of the pipe for the second flow channel (the low-pressure pipe **271**).

[0090] More preferably, during the cooling operation, the condenser (the outdoor heat exchanger **210**) is configured to condense the refrigerant, and the evaporator (the indoor heat exchanger **110**) is configured to evaporate the refrigerant. The refrigerant circuit **600** further includes a four-way valve **240** that changes the flow direction of the refrigerant passing through the condenser (the outdoor heat exchanger **210**) and the evaporator (the indoor heat exchanger **110**) from the flow direction during the cooling operation so as to cause the condenser (the outdoor heat exchanger **210**) to evaporate the refrigerant and cause the evaporator (the indoor heat exchanger **110**) to evaporate the refrigerant during the heating operation. The first flow channel (the middle-pressure pipe **272**) of the pressure-reducing heat exchanger **270** is disposed between the condenser (the outdoor heat exchanger **210**) and the evaporator (the indoor heat exchanger **110**). The second flow channel (the low-pressure pipe **271**) of the pressure-reducing heat exchanger **270** is disposed between the four-way valve **240** and the inlet of the compressor **200**.

[0091] More preferably, as illustrated in FIGS. **14** and **15**, the first flow channel includes a first pipe (the medium-pressure pipe **272**) and a second pipe (the medium-pressure pipe **273**) which is provided in parallel with the first pipe (the medium-pressure pipe **272**) and has an inner diameter smaller than that of the first pipe (the medium-pressure pipe **272**). The air conditioner **1110** further includes a switching valve **232** to switch the refrigerant to flow through the first pipe (the middle-pressure pipe **272**) or the second pipe (the medium-pressure pipe **273**). The controller **100** controls the switching valve **232** to cause the refrigerant to flow through the first pipe (the medium-pressure pipe **272**) during the cooling operation and cause the refrigerant to flow through the second pipe (the medium-pressure pipe **273**) during the heating operation.

[0092] Preferably, the degree of superheat of the refrigerant flowing through the outlet of the evaporator (the indoor heat exchanger **110**) is 0 degrees, and the degree of superheat of the refrigerant to be sucked into the compressor **200** is greater than 0 degrees.

[0093] Preferably, the refrigerant used in the refrigerant circuit **500** has a global warming coefficient of 205 or less, or a saturation temperature of −44.4° C. or more at a standard atmospheric pressure.

[0094] Preferably, the refrigerant has a global warming coefficient of 205 or less and a saturation temperature of −44.4° C. or more at a standard atmospheric pressure.

[0095] Preferably, the refrigerant includes R32 and R1234yf, and a mass fraction of R32 in the refrigerant is 30% or less.

[0096] It should be understood that the embodiments disclosed herein are illustrative and non-restrictive in all respects. The scope of the present invention is defined by the terms of the claims rather than the description of the embodiments above, and is intended to include any modifications within the scope and meaning equivalent to the terms of the claims.

1. An air conditioner comprising:

a refrigerant;

a refrigerant circuit which includes at least a compressor, a condenser and an evaporator, and is configured to circulate the refrigerant;

a heat exchanger which includes a first flow channel through which the refrigerant that has passed through the condenser flows and a second flow channel through which the refrigerant to be sucked into the compressor flows; and

a controller which is configured to control the refrigerant circuit so as to bring a degree of superheat of the refrigerant flowing through an outlet of the evaporator to a target value of 5 degrees or less, wherein

the pressure of the refrigerant passing through the first flow channel is higher than the pressure of the refrigerant passing through the second flow channel,

an inner diameter of a pipe for the first flow channel is smaller than an inner diameter of a pipe for the second flow channel, and

the heat exchanger is configured to perform pressure reduction of the refrigerant passing through the first flow channel and exchange heat between the refrigerant passing through the first flow channel and the refrigerant passing through the second flow channel.

2-**4**. (canceled)

5. The air conditioner according to claim **1**, wherein

during a cooling operation, the condenser is configured to condense the refrigerant, and the evaporator is configured to evaporate the refrigerant,

the refrigerant circuit further includes a four-way valve that changes a flow direction of the refrigerant passing through the condenser and the evaporator from the flow direction during the cooling operation so as to cause the condenser to evaporate the refrigerant and cause the evaporator to condense the refrigerant during the heating operation,

the first flow channel of the heat exchanger is disposed between the condenser and the evaporator, and the second flow channel of the heat exchanger is disposed between the four-way valve and an inlet of the compressor.

6. The air conditioner according to claim **5**, wherein
the first flow channel includes a first pipe and a second pipe which is disposed in parallel with the first pipe and has an inner diameter smaller than that of the first pipe,
the air conditioner further includes a switching valve to switch the refrigerant to flow through the first pipe or the second pipe, and
the controller controls the switching valve to cause the refrigerant to flow through the first pipe during the cooling operation and cause the refrigerant to flow through the second pipe during the heating operation.

7. The air conditioner according to claim **1**, wherein
the degree of superheat of the refrigerant flowing through the outlet of the evaporator is 0 degrees, and the degree of superheat of the refrigerant to be sucked into the compressor is greater than 0 degrees.

8. The air conditioner according to claim **1**, wherein
the refrigerant has a global warming coefficient of 205 or less, or a saturation temperature of −44.4° C. or more at a standard atmospheric pressure.

9. The air conditioner according to claim **1**, wherein
the refrigerant has a global warming coefficient of 205 or less and a saturation temperature of −44.4° C. or more at a standard atmospheric pressure.

10. The air conditioner according to claim **1**, wherein
the refrigerant includes R32 and R1234yf, and
a mass fraction of R32 in the refrigerant is 30% or less.

* * * * *