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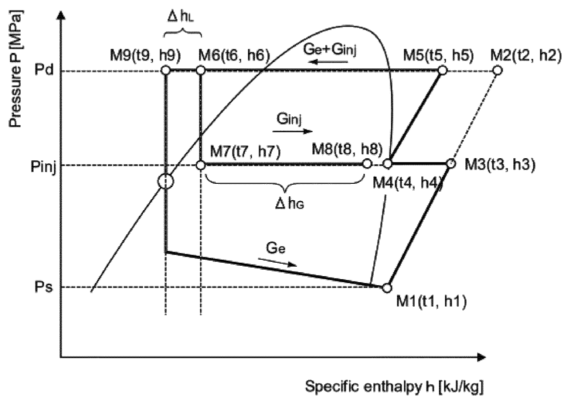
(54) AIR CONDITIONER

(57) An air conditioner according to an embodiment of the present invention includes: a refrigerant circuit; a first temperature detector; a first pressure detector; and a control device. The first temperature detector detects a temperature of a refrigerant that has flowed out from an inter-refrigerant heat exchanger and introduced into an intermediate-pressure portion of a compressor. The first pressure detector detects a pressure of the refrigerant that has flowed out from the inter-refrigerant heat exchanger and introduced into the intermediate-pressure portion of the compressor. The control device controls the degree of opening of an injection control valve such that a specific enthalpy of the refrigerant calculated on the basis of a refrigerant temperature detected by the first temperature detector and a refrigerant pressure detected by the first pressure detector is a specific enthalpy target value at which the degree of dryness of the refrigerant at a merging point between the injection pipe and the intermediate-pressure portion of the compressor is one.



$$\text{Injection ratio } \alpha = \frac{G_{inj}}{G_e} = \frac{\Delta h_L}{\Delta h_G} = \frac{h6-h9}{h8-h6} = \frac{h3-h4}{h4-h8}$$

FIG.3

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Description

Technical Field

[0001] The present invention relates to an air conditioner that includes a supercooling heat exchanger.

Background Art

[0002] A refrigeration circuit that includes a supercooling heat exchanger disposed at the outlet of a condenser for the purpose of improving the cooling capacity during high loads and has an injection path for injecting an intermediate-pressure two-phase refrigerant used to supercool the refrigerant flowing out from the outlet of the condenser into an intermediate-pressure portion of a compressor has been known.

[0003] For example, Patent Literature 1 discloses a refrigeration cycle apparatus that includes a flow control valve that reduces the pressure of part of a refrigerant condensed in a condenser, a supercooling heat exchanger that exchanges heat between the refrigerant whose pressure has been reduced by the flow control valve and the remaining refrigerant condensed in the condenser, a liquid outlet temperature sensor that detects the liquid temperature of the refrigerant flowing out from the outlet of a liquid side flow path that is the main stream of the supercooling heat exchanger, and an injection path for injecting the refrigerant flowing out from the outlet of a gas side flow path that is a tributary of the supercooling heat exchanger into an intermediate-pressure portion of the compressor. This refrigeration cycle apparatus is intended to improve the cooling capacity during high loads by adjusting the flow rate of the refrigerant in accordance with the refrigerant temperature detected by the liquid outlet temperature sensor.

Citation List

Patent Literature

[0004] Patent Literature 1: Japanese Patent Application Laid-open No. 2018-179383

Disclosure of Invention

Technical Problem

[0005] However, in the injection control based on the liquid outlet temperature of the supercooling heat exchanger, there is a possibility that the energy efficiency rating (EER) that is the value obtained by dividing the cooling capacity (output) by power consumption (input) decreases as compared with that during non-injection control.

[0006] That is, the advantage of the supercooling heat exchanger is that the pressure loss in an evaporator is reduced because the amount of a refrigerant flowing to the main stream side, the refrigeration capacity can be maintained even if the rotation speed of the compressor is reduced by increasing the evaporation temperature of the refrigerant in the evaporator (reducing the specific volume of the refrigerant sucked into the compressor), and the compression power can be suppressed. Meanwhile, during injection control, although the refrigeration capacity increases as compared with that during non-injection control, the condensation pressure increases as the flow rate of the refrigerant in high stages of the compressor increases in the case where the density of the injected refrigerant is high, and thus, the power of the compressor that rotates at a predetermined rotation speed tends to increase as compared with that during non-injection control. Therefore, although the refrigeration capacity can be maximized by simply monitoring the liquid outlet temperature of the supercooling heat exchanger, the decrease in the energy efficiency rating (EER) cannot be suppressed in some cases.

[0007] In view of the circumstances as described above, it is an object of the present invention to provide an air conditioner that is capable of suppressing the decrease in the energy efficiency rating (EER) while maximizing the refrigeration capacity.

Solution to Problem

[0008] An air conditioner according to an embodiment of the present invention includes: a refrigerant circuit; a first temperature detector; a first pressure detector; and a control device.

[0009] The refrigerant circuit includes a compressor, an outdoor heat exchanger, an indoor heat exchanger, a pressure reducer that is disposed between the outdoor heat exchanger and the indoor heat exchanger, a four-way valve that switches a flow direction of a refrigerant that is a non-azeotropic mixed refrigerant discharged from the compressor, an injection control valve that reduces a pressure of a first refrigerant that is part of a refrigerant condensed in the outdoor heat exchanger during a cooling operation, an inter-refrigerant heat exchanger that exchanges heat between the first refrigerant whose pressure has been reduced by the injection control valve and a second refrigerant that is a remainder of the refrigerant condensed in the outdoor heat exchanger, and an injection pipe that guides the first refrigerant that has flowed out from the inter-refrigerant heat exchanger to an intermediate-pressure portion of the compressor

[0010] The first temperature detector detects a temperature of the first refrigerant that has flowed out from the inter-refrigerant heat exchanger and introduced into the intermediate-pressure portion of the compressor.

[0011] The first pressure detector detects a pressure of the first refrigerant that has flowed out from the inter-refrigerant heat exchanger and introduced into the intermediate-pressure portion of the compressor.

[0012] The control device controls the degree of opening of the injection control valve such that a specific enthalpy of the first refrigerant calculated on the basis of a refrigerant temperature detected by the first temperature detector and a refrigerant pressure detected by the first pressure detector is a specific enthalpy target value at which the degree of dryness of a third refrigerant that is a refrigerant at a merging point between the injection pipe and the intermediate-pressure portion of the compressor is one.

[0013] The air conditioner may further include: a second temperature detector that detects a temperature of a refrigerant sucked into the compressor; and a second pressure detector that detects a pressure of a refrigerant sucked into the compressor.

[0014] The control device may calculate a refrigerant temperature in the intermediate-pressure portion of the compressor on the basis of a refrigerant temperature detected by the second temperature detector, a refrigerant pressure detected by the first pressure detector, and a refrigerant pressure detected by the second pressure detector.

[0015] The control device may calculate a specific enthalpy corresponding to the degree one of dryness of the third refrigerant on the basis of a specific enthalpy of a refrigerant in the intermediate-pressure portion of the compressor calculated on the basis of the refrigerant pressure detected by the first pressure detector and the refrigerant temperature in the intermediate-pressure portion of the compressor.

[0016] The control device may calculate the specific enthalpy target value of the first refrigerant on the basis of a specific enthalpy of the refrigerant in the intermediate-pressure portion of the compressor and the specific enthalpy corresponding to the degree one of dryness of the third refrigerant.

[0017] The air conditioner may further include: a third pressure detector that detects a pressure of a refrigerant discharged from the compressor; a third temperature detector that detects a temperature of a refrigerant flowing out from the outdoor heat exchanger during the cooling operation; and a fourth temperature detector that detects a temperature of the second refrigerant flowing out from the inter-refrigerant heat exchanger during the cooling operation.

[0018] The control device may calculate a specific enthalpy of the refrigerant flowing out from the outdoor heat exchanger during the cooling operation on the basis of a pressure detected by the second pressure detector and a temperature detected by the third temperature detector.

[0019] The control device may calculate a specific enthalpy of the second refrigerant flowing out from the inter-refrigerant heat exchanger during the cooling operation on the basis of the pressure detected by the second pressure detector and a temperature detected by the fourth temperature detector.

[0020] The control device may calculate an injection ratio that is a ratio of the first refrigerant to the second refrigerant on the basis of the specific enthalpy of the first refrigerant, the specific enthalpy of the refrigerant flowing out from the outdoor heat exchanger during the cooling operation, and the specific enthalpy of the second refrigerant flowing out from the inter-refrigerant heat exchanger during the cooling operation.

[0021] The control device may control the degree of opening of the injection control valve such that the injection ratio is a target injection ratio when the specific enthalpy of the first refrigerant is the specific enthalpy target value.

[0022] The compressor may be a scroll compressor. Advantageous Effects of Invention

[0023] In accordance with the present invention, it is possible to suppress the decrease in the energy efficiency rating (EER) while maximizing the refrigeration capacity.

Brief Description of Drawings

[0024]

[Fig. 1] Fig. 1 is a diagram showing a refrigerant circuit of an air conditioner according to an embodiment of the present invention.

[Fig. 2] Fig. 2 is a block diagram showing a configuration of an outdoor unit control device in the air conditioner.

[Fig. 3] Fig. 3 is a Mollier diagram of the refrigerant circuit during a cooling operation.

[Fig. 4] Fig. 4 is a flowchart showing an example of a procedure of processing executed by the outdoor unit control device.

[Fig. 5] Fig. 5 is a Mollier diagram of a refrigerant circuit in the existing air conditioner. Mode(s) for Carrying Out the Invention

[0025] An embodiment of the present invention will be described below with reference to the drawings.

[Refrigerant circuit of air conditioner]

[0026] Fig. 1 is a diagram showing a refrigerant circuit of the air conditioner 1 according to an embodiment of the present invention.

[0027] As shown in Fig. 1, the air conditioner 1 according to this embodiment includes an outdoor unit 2 installed outdoors and an indoor unit 3 that is installed indoors and connected to the outdoor unit 2 by a liquid pipe 4 and a gas pipe 5. In detail, a liquid side shutoff valve 25 of the outdoor unit 2 and a liquid pipe connection portion 33 of the indoor unit 3 are connected to each other by the liquid pipe 4. Further, a gas side shutoff valve 26 of the outdoor unit 2 and a gas pipe connection portion 34 of the indoor unit 3 are connected to each other by the gas pipe 5. In this way, a refrigerant circuit 10 of the air conditioner 1 is formed.

(Refrigerant)

[0028] As a refrigerant, a non-azeotropic mixed refrigerant is used. In this embodiment, an HFO non-azeotropic mixed refrigerant is used. As a non-azeotropic mixed refrigerant, a mixed refrigerant containing an R1234yf refrigerant and an R32 refrigerant is used.

(Refrigerant circuit of outdoor unit)

[0029] The outdoor unit 2 will be described first.

[0030] The outdoor unit 2 includes a compressor 21, a four-way valve 22, an outdoor heat exchanger 23, an accumulator 24, the liquid side shutoff valve 25 to which the liquid pipe 4 is connected, the gas side shutoff valve 26 to which the gas pipe 5 is connected, an outdoor fan 27, an outdoor expansion valve 28 (pressure reducer) located on the downstream side of an indoor expansion valve 35 (pressure reducer) provided in the indoor unit 3 during a heating operation, an injection control valve 29, and an inter-refrigerant heat exchanger 82. Then, these devices excluding the outdoor fan 27 are connected to each other by respective refrigerant pipes described below to form an outdoor unit refrigerant circuit 10a that is part of the refrigerant circuit 10.

[0031] Note that the indoor expansion valve 35 may be disposed in the outdoor unit 2 (e.g., between the liquid side shutoff valve 25 and the inter-refrigerant heat exchanger 82). Here, for convenience, the indoor expansion valve 35 will also be mentioned in the description of the outdoor unit 2.

[0032] The compressor 21 is a variable-capacity compressor whose operating capacity can be changed by controlling the rotation speed with an inverter (not shown). In this embodiment, the compressor 21 is a two-stage compressor that includes a low-stage compression portion and a high-stage compression portion, and a scroll compressor or a rotary compressor is adopted, for example. The refrigerant discharge side of the compressor 21 is connected to a port a of the four-way valve 22 by a discharge pipe 61. Further, the refrigerant suction side of the compressor 21 is connected to a port c of the four-way valve 22 by a suction pipe 66 via the accumulator 24.

[0033] The four-way valve 22 is a valve for switching the flowing direction of a refrigerant, and includes four ports a, b, c, and d. The port a is connected to the refrigerant discharge side of the compressor 21 by the discharge pipe 61 as described above. The port b is connected to one refrigerant inlet/outlet of the outdoor heat exchanger 23 by a refrigerant pipe 62. The port c is connected to the refrigerant suction side of the compressor 21 by the suction pipe 66 as described above. Then, the port d is connected to the gas side shutoff valve 26 by an outdoor unit gas pipe 64.

[0034] The outdoor heat exchanger 23 exchanges heat between a refrigerant and the outside air taken into the outdoor unit 2 by rotation of the outdoor fan 27. The one refrigerant inlet/outlet of the outdoor heat exchanger 23 is connected to the port b of the four-way valve 22 by the refrigerant pipe 62 as described above, and the other refrigerant inlet/outlet is connected to the liquid side shutoff valve 25 by an outdoor unit liquid pipe 63. The outdoor heat exchanger 23 functions as a condenser during a cooling operation and functions as an evaporator during a heating operation by switching the four-way valve 22.

[0035] The indoor expansion valve 35 and the outdoor expansion valve 28 and are each an electronic expansion valve to be driven by a pulse motor (not shown). Specifically, the degree of opening thereof is adjusted by the number of pulses to be applied to the pulse motor. During a heating operation, the indoor expansion valve 35 is fully open, and the outdoor expansion valve 28 is adjusted such that the discharge temperature of the compressor 21 is a predetermined target value (value at which the refrigerant sucked into the compressor 21 is in an appropriate state) or such that the degree of suction superheat (suction SH) of the refrigerant sucked into the compressor 21 is a predetermined target value (value at which the refrigerant sucked into the compressor 21 is in an appropriate state). Further, during a cooling operation or a dehumidification operation, the indoor expansion valve 35 is adjusted such that the degree of suction superheat(suction SH) of the refrigerant sucked into the compressor 21 is a predetermined target value and the outdoor expansion valve 28 is fully open.

[0036] The outdoor fan 27 is formed of a resin material and is disposed in the vicinity of the outdoor heat exchanger 23. The outdoor fan 27 has its center connected to the rotation shaft of a fan motor (not shown). When the fan motor rotates, the outdoor fan 27 rotates. By the rotation of the outdoor fan 27, the outside air is taken into the outdoor unit 2 from a suction port (not shown) of the outdoor unit 2, and the outside air that has exchanged heat with the refrigerant in the outdoor heat exchanger 23 is released to the outside of the outdoor unit 2 from an air outlet (not shown) of the outdoor unit 2.

[0037] The outdoor unit refrigerant circuit 10a includes an injection pipe 65. The injection pipe 65 is branched from the outdoor unit liquid pipe 63 between the outdoor expansion valve 28 and the inter-refrigerant heat exchanger 82, and is connected to an intermediate-pressure portion 21a between the low-stage compression portion and the high-stage compression portion of the compressor 21 via the injection control valve 29 and the inter-refrigerant heat exchanger 82.

[0038] The injection pipe 65 is part of an injection circuit that causes part of the refrigerant condensed in the outdoor heat exchanger 23 during a cooling operation to flow into the compressor 21, and is a refrigerant pipe that guides the refrigerant whose pressure has been reduced by the injection control valve 29 to the intermediate-pressure portion 21a of the compressor 21. In the middle of the injection pipe 65, the inter-refrigerant heat exchanger 82 that exchanges heat between a refrigerant flowing through the injection pipe 65 (first refrigerant) and a refrigerant flowing through the outdoor unit liquid pipe 63, which is the remainder of the refrigerant condensed in the outdoor heat exchanger 23 (second refrigerant), is provided.

[0039] The inter-refrigerant heat exchanger 82 is a supercooling heat exchanger that exchanges heat between the refrigerant flowing through the outdoor unit liquid pipe 63 and the refrigerant flowing through the injection pipe 65 during a cooling operation. In the inter-refrigerant heat exchanger 82, the outdoor unit liquid pipe 63 forms the main stream of the inter-refrigerant heat exchanger 82 (liquid side flow path), and the injection pipe 65 forms a tributary of the inter-refrigerant heat exchanger 82 (gas side flow path).

[0040] The injection control valve 29 is an electronic expansion valve that is driven by a pulse motor (not shown), and is provided in the middle of the pipe between the inter-refrigerant heat exchanger 82 and a branch point 30 between the outdoor unit liquid pipe 63 and the injection pipe 65. The injection control valve 29 is provided as a means for opening/closing the injection pipe 65 and a means for adjusting the flow rate of the refrigerant in the injection pipe 65.

[0041] In addition to the configurations described above, the outdoor unit 2 is provided with various sensors. As shown in Fig. 1, the discharge pipe 61 is provided with a discharge pressure sensor 71 that detects a pressure of the refrigerant discharged from the compressor 21, and a discharge temperature sensor 73 that detects a temperature of a refrigerant discharged from the compressor 21. The suction pipe 66 is provided with a suction pressure sensor 72 that detects a pressure of the refrigerant sucked into the compressor 21, and a suction temperature sensor 74 that detects a temperature of the refrigerant sucked into the compressor 21. The outdoor unit liquid pipe 63 is provided with an outdoor-unit-liquid-pipe temperature sensor 77c that detects a temperature of the refrigerant flowing into the outdoor expansion valve 28 during a cooling operation, and an inter-refrigerant heat exchange liquid pipe temperature sensor 43 that detects a temperature of the refrigerant flowing out from the inter-refrigerant heat exchanger 82 during a cooling operation.

[0042] An outdoor heat exchange intermediate temperature sensor 75 that detects an outdoor heat exchange temperature that is the temperature of the outdoor heat exchanger 23 is provided in substantially the middle of a refrigerant path (not shown) of the outdoor heat exchanger 23. Then, an outside air temperature sensor 76 that detects a temperature of the outside air flowing into the outdoor unit 2, i.e., the outside air temperature is provided in the vicinity of a suction port (not shown) of the outdoor unit 2.

[0043] The injection pipe 65 is provided with an injection temperature sensor 41 that detects a temperature of the refrigerant (first refrigerant) that has flowed out from the inter-refrigerant heat exchanger 82 and introduced into the intermediate-pressure portion 21a the compressor 21 during a cooling operation, and an injection pressure sensor 42 that detects a pressure of the refrigerant (first refrigerant) that has flowed out from the inter-refrigerant heat exchanger 82 and introduced into the intermediate-pressure portion 21a of the compressor 21.

[0044] Further, the outdoor unit 2 is provided with an outdoor unit control device 200. The outdoor unit control device 200 is mounted on a control board housed in an electrical equipment box (not shown) of the outdoor unit 2. As shown in Fig. 2, the outdoor unit control device 200 includes a CPU 210, a storage unit 220, a communication unit 230, and a sensor input unit 240. The outdoor unit control device 200 corresponds to the control device in the present invention.

[0045] The storage unit 220 includes a flash memory and stores a control program of the outdoor unit 2, a detection value corresponding to a detection signal from various sensors, and control states of the compressor 21, the outdoor fan 27, and the like. Further, although not shown, a rotation speed table that determines the rotation speed of the compressor 21 in accordance with the required capacity received from the indoor unit 3 is stored in the storage unit 220 in advance.

[0046] The communication unit 230 is an interface that communicates with the indoor unit 3. The sensor input unit 240 takes in detection results of the various sensors of the outdoor unit 2 and outputs them to the CPU 210.

[0047] The CPU 210 takes in the above-mentioned detection results of the various sensors of the outdoor unit 2 via the sensor input unit 240. Further, the CPU 210 takes in a control signal transmitted from the indoor unit 3 via the communication unit 230. The CPU 210 performs drive control of the compressor 21 and the outdoor fan 27 on the basis of the taken-in detection result, control signal, and the like. Further, the CPU 210 performs switch control of the four-way valve 22 on the basis of the taken-in detection result and control signal. Further, the CPU 210 performs adjustment of the degree of opening of the indoor expansion valve 35 and the outdoor expansion valve 28, control of opening/closing of the injection control valve 29, and adjustment of the degree of opening on the basis of the taken-in detection result and control signal.

(Refrigerant circuit of indoor unit)

[0048] Next, the indoor unit 3 will be described with reference to Fig. 1.

[0049] The indoor unit 3 includes an indoor heat exchanger 31, an indoor fan 32, the liquid pipe connection portion 33 to which the other end of the liquid pipe 4 is connected, the gas pipe connection portion 34 to which the other end of the gas pipe 5 is connected, and the indoor expansion valve 35 disposed between the indoor heat exchanger 31 and the liquid pipe connection portion 33. Then, these devices excluding the indoor fan 32 are connected to each other by respective refrigerant pipes described below in detail to form an indoor unit refrigerant circuit 10b that is part of the refrigerant circuit 10.

[0050] The indoor heat exchanger 31 exchanges heat between a refrigerant and the indoor air taken into the indoor unit 3 from a suction port (not shown) of the indoor unit 3 by rotation of the indoor fan 32 described below. One refrigerant inlet/outlet of the indoor heat exchanger 31 is connected to the liquid pipe connection portion 33 by an indoor unit liquid pipe 67. The other refrigerant inlet/outlet of the indoor heat exchanger 31 is connected to the gas pipe connection portion 34 by an indoor unit gas pipe 68. The indoor heat exchanger 31 functions as an evaporator in the case where the indoor unit 3 performs a cooling operation and functions as a condenser in the case where the indoor unit 3 performs a heating operation.

[0051] The indoor fan 32 is formed of a resin material and is disposed in the vicinity of the indoor heat exchanger 31. The indoor fan 32 is caused to rotate by a fan motor (not shown) to take in the indoor air from the suction port (not shown) of the indoor unit 3 into the indoor unit 3, and blows out the indoor air that has exchanged heat with a refrigerant in the indoor heat exchanger 31 from an air outlet (not shown) of the indoor unit 3 into the room.

[0052] In addition to the configurations described above, the indoor unit 3 is provided with various sensors. An indoor heat exchange intermediate temperature sensor 77a that detects a temperature of a refrigerant in the indoor heat exchanger 31 is provided in the middle of the flow path in the indoor heat exchanger 31. The indoor unit gas pipe 68 is provided with an indoor unit gas pipe temperature sensor 77b that detects a temperature of a refrigerant flowing into the indoor expansion valve 35 during a heating operation, and a gas side temperature sensor 78 that detects a temperature of a refrigerant flowing out from the indoor heat exchanger 31 or flowing into the indoor heat exchanger 31. Then, a room temperature sensor 79 that detects the temperature of the indoor air flowing into the indoor unit 3, i.e., the room temperature, is provided in the vicinity of the suction port (not shown) of the indoor unit 3.

[Overview of operation of refrigerant circuit]

[0053] Next, an overview of an operation of the air conditioner 1 according to this embodiment will be described.

(Heating operation)

[0054] A flow of a refrigerant during a heating operation will be described first.

[0055] When performing a heating operation, as shown in Fig. 1, the CPU 210 switches the four-way valve 22 to the state shown by the solid line, i.e., such that the port a and the port d of the four-way valve 22 communicate with each other and the port b and the port c communicate with each other. As a result, in the refrigerant circuit 10, a refrigerant circulates in the direction indicated by the solid line, and a heating cycle in which the outdoor heat exchanger 23 functions as an evaporator and the indoor heat exchanger 31 functions as a condenser is obtained.

[0056] Further, when performing a heating operation, the CPU 210 fully opens the indoor expansion valve 35 and fully closes the injection control valve 29. Further, the CPU 210 adjusts the degree of opening of the outdoor expansion valve 28 such that the discharge temperature of the compressor 21 is a predetermined target value or the degree of suction superheat (suction SH) of the refrigerant sucked into the compressor 21 is a predetermined target value.

[0057] A high-temperature and high-pressure gas refrigerant discharged from the compressor 21 flows into the four-way valve 22 through the discharge pipe 61. The high-temperature and high-pressure gas refrigerant that has flowed into the port a of the four-way valve 22 flows into the gas pipe 5 via the gas side shutoff valve 26 from the port d of the four-way valve 22 through the outdoor unit gas pipe 64. The high-temperature and high-pressure gas refrigerant flowing through the gas pipe 5 flows into the indoor unit 3 via the gas pipe connection portion 34.

[0058] The high-temperature and high-pressure gas refrigerant that has flowed into the indoor unit 3 flows into the indoor heat exchanger 31 through the indoor unit gas pipe 68, exchanges heat with the indoor air taken into the indoor unit 3 by rotation of the indoor fan 32, and is condensed to become a high-temperature and high-pressure liquid refrigerant. In this way, the indoor heat exchanger 31 functions as a condenser and the indoor air that has exchanged heat with the refrigerant in the indoor heat exchanger 31 is blown out from the air outlet (not shown) into the room, thereby heating the room in which the indoor unit 3 is installed.

[0059] The high-temperature and high-pressure liquid refrigerant that has flowed out from the indoor heat exchanger 31 flows through the indoor unit liquid pipe 67, passes through the indoor expansion valve 35 in the fully open state, and flows into the liquid pipe 4 via the liquid pipe connection portion 33. The high-temperature and high-pressure liquid refrigerant that has flowed into the outdoor unit 2 via the liquid side shutoff valve 25 through the liquid pipe 4 flows through the outdoor unit liquid pipe 63 and is reduced in pressure when passing through the outdoor expansion valve 28 to become a low-temperature and low-pressure gas-liquid two-phase refrigerant. As described above, during a heating operation, the degree of opening of the outdoor expansion valve 28 is adjusted such that the discharge temperature of the compressor 21 is a predetermined target value or the degree of suction superheat (suction SH) of the refrigerant sucked into the compressor 21 is a predetermined target value.

[0060] Note that the degree of opening of the outdoor expansion valve 28 during a heating operation does not necessarily need to be in a fully open state. For example, the degree of opening may be adjusted such that the degree of supercooling (SC) of the refrigerant after flowing out of the indoor heat exchanger 31 is a predetermined target value.

[0061] The low-temperature and low-pressure gas-liquid two-phase refrigerant that has passed through the outdoor expansion valve 28 and flowed into the outdoor heat exchanger 23 exchanges heat with the outside air taken into the outdoor unit 2 by rotation of the outdoor fan 27 and evaporates to become a low-temperature and low-pressure gas refrigerant. The low-temperature and low-pressure gas refrigerant that has flowed out from the outdoor heat exchanger 23 into the refrigerant pipe 62 is sucked into the compressor 21 through the port b and the port c of the four-way valve 22 and the suction pipe 66 and is compressed again.

[Cooling operation]

[0062] Next, a flow of a refrigerant during a cooling operation will be described.

[0063] When performing a cooling operation, as shown in Fig. 1, the CPU 210 switches the four-way valve 22 to the state shown by the broken line, i.e., such that the port a and the port b of the four-way valve 22 communicate with each other and the port c and the port d communicate with each other. As a result, a refrigerant circulates in the direction indicated by the broken line in the refrigerant circuit 10, and a cooling cycle in which the outdoor heat exchanger 23 functions as a condenser and the indoor heat exchanger 31 functions as an evaporator is obtained.

[0064] Further, when performing a cooling operation, the CPU 210 fully opens the outdoor expansion valve 28 and adjusts the degree of opening of the indoor expansion valve 28 such that the degree of suction superheat (suction SH) of the refrigerant sucked into the compressor 21 is a predetermined target value. Further, the CPU 210 adjusts the degree of opening of the injection control valve 29 such that a target injection ratio is obtained as described below in detail.

[0065] The high-temperature and high-pressure gas refrigerant discharged from the compressor 21 flows into the four-way valve 22 through the discharge pipe 61. The high-temperature and high-pressure gas refrigerant that has flowed into the port a of the four-way valve 22 flows into the outdoor heat exchanger 23 from the port b of the four-way valve 22 through the refrigerant pipe 62, exchanges heat with the outside air taken into the outdoor unit 2 by rotation of the outdoor fan 27, and is condensed to become a high-temperature and high-pressure liquid refrigerant.

[0066] The high-temperature and high-pressure liquid refrigerant that has flowed out from the outdoor heat exchanger 23 passes through the outdoor expansion valve 28 in the fully open state, and is branched into a refrigerant flowing through the outdoor unit liquid pipe 63, which forms the main stream of the inter-refrigerant heat exchanger 82, and a refrigerant flowing through the injection pipe 65, which forms a tributary (gas side flow path) of the inter-refrigerant heat exchanger 82 via the injection control valve 29. The high-temperature and high-pressure liquid refrigerant flowing through the outdoor unit liquid pipe 63 flows into the liquid pipe 4 via the liquid side shutoff valve 25. The high-temperature and high-pressure liquid refrigerant flowing through the injection pipe 65 is reduced in pressure by the injection control valve 29 and then exchanges heat with a refrigerant flowing through the main stream of the inter-refrigerant heat exchanger 82 in the inter-refrigerant heat exchanger 82 to become an intermediate pressure gas-liquid two-phase refrigerant and be introduced into the intermediate-pressure portion 21a of the compressor 21.

[0067] The refrigerant that has flowed into the indoor unit 3 via the liquid pipe connection portion 33 through the liquid pipe 4 flows through the indoor unit liquid pipe 67 and is reduced when passing through the indoor expansion valve 35. As described above, during a cooling operation, the degree of opening of the indoor expansion valve 35 is adjusted such that the degree of suction superheat (suction SH) of the refrigerant sucked into the compressor 21 is a predetermined target value.

[0068] The low-temperature and low-pressure gas-liquid two-phase refrigerant that has passed through the indoor expansion valve 35 and flowed into the indoor heat exchanger 31 exchanges heat with the indoor air taken into the indoor unit 3 by rotation of the indoor fan 32, and evaporates to become a low-temperature and low-pressure gas refrigerant. In this way, the indoor heat exchanger 31 functions as an evaporator, and the indoor air that has exchanged heat with a refrigerant in the indoor heat exchanger 31 is blown out from the air outlet (not shown) into the room, thereby cooling the room in which the indoor unit 3 is installed.

[0069] The low-temperature and low-pressure gas refrigerant that has flowed out from the indoor heat exchanger 31 flows through the indoor unit gas pipe 68 and flows into the gas pipe 5 via the gas pipe connection portion 34. The low-temperature and low-pressure gas refrigerant that has flowed into the outdoor unit 2 via the gas side shutoff valve 26 through the gas pipe 5 flows through the port d and the port c of the four-way valve 22 and the suction pipe 66, is sucked into the compressor 21, and is compressed again.

[Regarding injection control]

[0070] The air conditioner 1 according to this embodiment includes the inter-refrigerant heat exchanger 82 as a supercooling heat exchanger for the purpose of improving the cooling capacity during high loads, the injection pipe 65 that injects the intermediate-pressure gas-liquid two-phase refrigerant used for supercooling into the intermediate-pressure portion 21a of the compressor 21, and the injection control valve 29 that controls the flow rate of the refrigerant flowing into the injection pipe 65.

[0071] Fig. 3 is a Mollier diagram of the refrigerant circuit 10 during a cooling operation. Operating points M1 to M9 in the figure are as follows:

M1: between the indoor heat exchanger 31 and the compressor 21,
M2: between the compressor 21 and the outdoor heat exchanger 23 during non-injection control (when the injection control valve 29 is fully closed),
M3: the intermediate-pressure portion 21a of the compressor 21,
M4: the merging point between the intermediate-pressure portion 21a of the compressor 21 and the injection pipe 65,
M5: between the compressor 21 and the outdoor heat exchanger 23 during injection control,
M6: between the outdoor unit liquid pipe 63 and the inter-refrigerant heat exchanger 82,
M7: between the injection control valve 29 and the inter-refrigerant heat exchanger 82,
M8: between the inter-refrigerant heat exchanger 82 and the intermediate-pressure portion 21a of the compressor 21, and
M9: between the inter-refrigerant heat exchanger 82 and the indoor expansion valve 35.

[0072] The injection control is different from the non-injection control in the following points.

(1) The temperature of the refrigerant compressed in the compressor 21 decreases during compression when part of the refrigerant (first refrigerant) in the condensation process flows into the intermediate-pressure portion 21a of the compressor 21 via the injection pipe 65 in a two-phase state during the compression process in the compressor 21 (between the points M1 to M3 to M4 o M5), the discharge temperature (t5) of the compressor 21 decreases as compared with the discharge temperature (t2) of the compressor 21 in the case where a refrigerant is not circulated through the injection pipe 65 (during non-injection control).

(2) The refrigerant becomes a liquid state when passing through the condensation process (between the points M5 to M6) and then is supercooled by exchanging heat with the refrigerant flowing between the points M7 and M8 of the injection pipe 65 by the inter-refrigerant heat exchanger 82 between the points M6 to M9 .

(3) The refrigerant that has branched from the points M5 to M6 and flowed into the injection pipe 65 is reduced in pressure between the points M6 to M7 via the injection control valve 29 and becomes a two-phase state, and then, the degree of dryness thereof increases by exchanging heat with the refrigerant flowing between the points M6 to M9 by the inter-refrigerant heat exchanger 82 between the points M7 to M8, and the refrigerant is injected into the intermediate-pressure portion 21a of the compressor 21 from the point M8.

[0073] As a result, the amount of the refrigerant flowing through the tributary (gas side flow path) of the inter-refrigerant heat exchanger 82 (Ginj) becomes larger than zero, while the amount of the refrigerant flowing through the main stream (liquid side flow path) of the inter-refrigerant heat exchanger 82 (Ge) decreases. As a result, the pressure loss in the evaporator (indoor heat exchanger 31) becomes smaller than that during non-injection control. As a result, since the evaporation temperature of the refrigerant in the evaporator increases (the specific volume of the refrigerant sucked into the compressor decreases), it is possible to suppress the compression power while maintaining the refrigeration capacity even if the rotation speed of the compressor is reduced. For this reason, it is possible to improve the cooling capacity during high loads and suppress the compression power.

[0074] However, in the case where the density of the injected refrigerant is high, since the condensation pressure increases as the flow rate of the refrigerant in the high stages of the compressor 21 increases, the power of the compressor 21 that rotates at the same rotation speed increases as compared with that during non-injection control, and the decreases in the energy efficiency rating (EER) that is a value obtained by dividing the cooling capacity (output) by power consumption (input) cannot be suppressed.

[0075] In this regard, in this embodiment, the degree of opening of the injection control valve 29 is controlled such that a specific enthalpy (h8) of the refrigerant (first refrigerant) injected into the intermediate-pressure portion 21a of the compressor 21 via the injection pipe 65 is a specific enthalpy target value at which the degree of dryness of a refrigerant (third refrigerant) in the merging point(point M4 in Fig. 3) between the injection pipe 65 and the intermediate-pressure portion 21a of the compressor 21 is one.

[0076] The above third refrigerant is a refrigerant after the above first refrigerant joins together with the refrigerant in the intermediate-pressure portion 21a of the compressor 21 (refrigerant discharged from the low-stage compression portion), and this joined refrigerant is compressed in the high-stage compression portion of the compressor 21.

[0077] As a result, even in the case where the density of the injected refrigerant is high, the increase in the compression load in the intermediate-pressure portion 21a of the compressor 21 is suppressed, ant thus, it is possible to suppress the decrease in the energy efficiency rating (EER) while maximizing the refrigeration capacity.

[Details of injection control]

[0078] Details of injection control will be described below. Fig. 4 is a flowchart showing an example of a procedure of processing executed by the outdoor unit control device 200 (CPU 210).

(Start condition)

[0079] When a cooling operation is started, the CPU 210 determines whether or not the start condition of injection control is satisfied (Step 101). Although the start condition of injection control is not particularly limited, examples thereof include the following two conditions.

(1) Rotation speed of compressor 21

[0080] The second condition is whether or not the rotation speed of the compressor 21 is a predetermined threshold value or more. The purpose is to prevent the injection amount from becoming excessive, because the time the injection port is open (the time it communicates with the intermediate-pressure portion 21a) is relatively long when the compressor 21 operates at low speed. The above redetermined threshold value is, for example, 50 rps.

(2) Degree of supercooling at condenser outlet

[0081] The third condition is whether or not the degree of supercooling (SC) of the refrigerant at the outlet of the condenser (outdoor heat exchanger 23) is a predetermined threshold value or more. The purpose is to introduce a refrigerant in a liquid state to the injection control valve 29 to ensure the flow rate. The above predetermined threshold value is, for example, 8°C.

[0082] The degree of supercooling at a condenser is calculated by, for example, the following formula.

(Degree of supercooling at a condenser) = (high-pressure liquid saturation temperature) - (supercooling liquid temperature)

[0083] The high-pressure liquid saturation temperature can be obtained by converting the detection value (corresponding to the pressure Pd in Fig. 3) of the discharge pressure sensor 71 installed in the discharge pipe 61 into the saturation temperature as described above.

[0084] As the supercooling liquid temperature, the detection value (corresponding to the temperature t6 at the point M6 in Fig. 3) of the outdoor-unit-liquid-pipe temperature sensor 77c installed at the outlet of the outdoor heat exchanger 23 can be used.

[0085] In this embodiment, injection control is started in the case where both the above two conditions are met. Note that the present invention is not limited thereto, and injection control may be started in the case where at least one of the above two conditions is met or another condition may be adopted.

[0086] In the case where it is determined that the start condition of injection control is satisfied (Yes in Step 101), the CPU 210 adjusts the initial degree of opening of the injection control valve 29 (Step 102). The initial degree of opening of the injection control valve 29 is not particularly limited as long as it is the degree of opening capable of preventing liquid compression due to an excessive injection amount, and is set to, for example, the degree of opening corresponding to 10% of the maximum pulse output (when fully open).

[0087] Subsequently, the CPU 210 calculates an actual injection ratio α (Step 103).

[0088] As shown in Fig. 3, the actual injection ratio α is defined by the following formula.

$$\text{Injection ratio } \alpha = Ginj / Ge = \Delta h_L / \Delta h_G$$
$$= (h6 - h9) / (h8 - h6)$$
$$= (h3 - h4) / (h4 - h8)$$

[0089] Here, Ginj represents the flow rate of the refrigerant (first refrigerant) flowing through the tributary (gas side flow path) of the inter-refrigerant heat exchanger 82, i.e., the injection pipe 65, and Ge represents the flow rate of the refrigerant (second refrigerant) flowing through the main stream (liquid side flow path) of the inter-refrigerant heat exchanger 82, i.e., the outdoor unit liquid pipe 63. That is, the actual injection ratio means the flow rate ratio (Ginj / Ge) of the flow rate of the above first refrigerant to the flow rate of the above second refrigerant achieved by the current degree of opening of the injection control valve 29.

[0090] Further, Δh_L represents a specific enthalpy difference of the refrigerant in the main stream of the inter-refrigerant heat exchanger 82 before and after heat exchange with the tributary, and Δh_G represents a specific enthalpy difference of the refrigerant in the tributary of the inter-refrigerant heat exchanger 82 before and after heat exchange with the main stream.

[0091] Further, h6 represents a specific enthalpy of the refrigerant at the outlet of the outdoor heat exchanger 23, h8 represents a specific enthalpy of the refrigerant at the outlet of the main stream (liquid side flow path) of the inter-refrigerant heat exchanger 82, and h represents a specific enthalpy of the refrigerant at the outlet of the tributary (gas side flow path) of the inter-refrigerant heat exchanger 82 (see Fig. 3).

[0092] The specific enthalpy can be calculated by referring to the pressure and temperature of the refrigerant.

[0093] For example, when calculating h6, the detection value (corresponding to Pd in Fig. 3) of the discharge pressure sensor 71 installed in the discharge pipe 61 can be used as the pressure. The detection value (corresponding to t6 in Fig. 3) of the outdoor-unit-liquid-pipe temperature sensor 77c installed at the outlet of the outdoor heat exchanger 71 can be used as the temperature.

[0094] When calculating h9, the detection value (corresponding to d in Fig. 3) of the discharge pressure sensor 71 installed in the discharge pipe 61 can be used as the pressure. The detection value (corresponding to t9 in Fig. 3) of the main-stream-side outlet of the inter-refrigerant heat exchanger 82 of the inter-refrigerant heat exchange liquid pipe temperature sensor 43 can be used as the temperature.

[0095] Then, when calculating h8, the detection value (corresponding to Pinj in Fig. 3) of the injection pressure sensor 42 installed in the injection pipe 65 can be used as the pressure, and the detection value (corresponding to t8 in Fig. 3) of the injection temperature sensor 41 installed in the injection pipe 65 can be used as the temperature.

[0096] Subsequently, the CPU 210 calculates a target injection ratio αtgt (Step 104).

[0097] The target injection ratio αtgt is represented as follows.

$$\text{Target injection ratio } \alpha tgt = Ginj / Ge = \Delta h_L / \Delta h_G$$
$$= (h6 - h9) / (h8^* - h6)$$
$$= (h3 - h4^*) / (h4^* - h8^*) \quad \text{(Formula 1)}$$

[0098] In the formula (2), h8* represents a target value (specific enthalpy target value) of the specific enthalpy (h8) of the refrigerant injected into the intermediate-pressure portion 21a of the compressor 21 via the injection pipe 65, which is a specific enthalpy at which the degree of dryness of the refrigerant at the merging point between the injection pipe 65 and the intermediate-pressure portion 21a of the compressor 21 is one. That is, the target injection ratio is an injection ratio at which the specific enthalpy (h8) of the refrigerant (first refrigerant) injected into the intermediate-pressure portion 21a of the compressor 21 via the injection pipe 65 is the specific enthalpy target value h8*.

[0099] Further, h4* represents a specific enthalpy corresponding to the degree one of dryness of the refrigerant (third refrigerant) at the merging point (M4) between the injection pipe 65 and the intermediate-pressure portion 21a of the compressor 21. h4* can be represented as follows.

$$h4^* = (h3 + \alpha tgt \cdot h8^*) / (1 + \alpha tgt)$$

[0100] When expanded,

$$h8^* = \{(1 + \alpha tgt)h4^* - h3\} / \alpha tgt \quad \text{(formula 2)}$$

[0101] The target injection ratio αtgt and the specific enthalpy target value h8* can be obtained as a simultaneous solution of the (formula 1) and the (formula 2).

[0102] h3 can be calculated using the detection value (Pinj) of the injection pressure sensor 42 and the low-stage discharge temperature (corresponding to t3 in Fig. 3) of the compressor 21.

[0103] The low-stage discharge temperature t3 can be calculated as a temperature of the compression process using the low-stage suction temperature(the detection value (corresponding to t1) of the suction temperature sensor 74) and the low-stage suction pressure (the detection value (corresponding to Ps) of the suction pressure sensor 72) of the compressor 21, the detection value (corresponding to Pinj) of the injection pressure sensor 42, and the polytropic constant n during actual compression as shown in the following formula (A). Ps and Pinj each represent the absolute pressure.

[Math. 1]

$$t_3 = (t_1 + 273.15)\left(\frac{P_{inj}}{P_s}\right)^{\frac{n-1}{n}} - 273.15 \quad \cdots (A)$$

[0104] The polytropic constant n during actual compression corresponds to an adiabatic index during ideal adiabatic compression, and is a variable that depends on the specifications of the compressor 21, the detection value (Pd) of the discharge pressure sensor 71, the low-stage suction pressure (Ps), the type of refrigerant, and the like. In this embodiment, the polytropic constant n during actual compression is determined by a compressor unit performance experiment under a predetermined load and refrigerant temperature.

[0105] h4 can be calculated from the following formula (b) as the energy (specific enthalpy × flow rate) balance taking into account the inflow from the side of the injection pipe 65 and the side of the low-stage discharge portion of the compressor 21 with respect to the merging point (point M4 in Fig. 3) between the intermediate-pressure portion 21a of the compressor 21 and the injection pipe 65.

[Math. 2]

$$G_{inj} \cdot h_8 + G_e \cdot h_3 \quad (G_e + G_{inj})\, h_4$$

$$\therefore \quad h_4 = \frac{h_3 + \alpha \cdot h_8}{1 + \alpha} \quad \left(\alpha = \frac{G_{inj}}{G_e}\right) \quad \cdots (B)$$

[0106] Therefore, h4* can be calculated by replating "h8" with h8* and "α" with αtgt in the formula (B).

[0107] Subsequently, the CPU 210 calculates Δα (αtgt - α), which is the difference obtained by subtracting the actual injection ratio α from the target injection ratio αtgt, and determines whether or not Δα is -β or more and β or less (Steps 105 and 106).

[0108] β represents a threshold value of the deviation of the control deviation amount (Δα = α - αtgt), which is the difference between the target value of the injection ratio (target injection ratio αtgt) and the current value (actual injection ratio α), and is defined as β = Δα / α. The value of the threshold value β is not particularly limited, can be arbitrarily set in accordance with the control period, the size of the injection port of the compressor 21, and the like, and is 2% in this embodiment.

[0109] In the case where the results is -β ≤ Δα ≤ β, (Yes in both Steps 105 and 106), the degree of opening of the injection control valve 29 is maintained at the current degree of opening (Step 107) assuming that the specific enthalpy (h8) of the refrigerant (first refrigerant) injected into the intermediate-pressure portion 21a of the compressor 21 via the injection pipe 65 is equal to or substantially equal to the specific enthalpy target value h8* that is the control target value.

[0110] Meanwhile, in the case where Δα is larger than the threshold value β (No in Step 105), control of loosening the degree of opening of the injection control valve 29 by a predetermined amount (increasing the degree of opening by a predetermined amount) is executed (Step 108). This allows the actual injection ratio α to approach the target ratio αtgt.

[0111] Meanwhile, in the case where Δα is smaller than the threshold value -β (No in Step 106), control of narrowing the degree of opening of the injection control valve 29 by a predetermined amount (decreasing the degree of opening by a predetermined amount) is executed (Step 109). This allows the actual injection ratio α to approach the target ratio αtgt.

[0112] Subsequently, the CPU 210 determines whether or not the end condition of injection control is satisfied (Step 110). In this embodiment, in the case where at least one of the above-mentioned two start conditions of injection control is no longer applicable, it is determined that the end condition of injection control is satisfied.

[0113] In the case where the end condition of injection control is not satisfied (No in Step110), the processing proceeds to Step 103, where the degree of opening of the injection control valve 29 is controlled again such that the actual injection ratio α is the target injection ratio αtgt. Meanwhile, in the case where the end condition of injection control is satisfied (Yes in Step 110), the CPU 210 closes the injection control valve 29 and ends the injection control (Step 111).

[0114] By repeatedly executing the above processing, the degree of opening of the injection control valve 29 is controlled such that the specific enthalpy (h8) of the refrigerant (first refrigerant) injected into the intermediate-pressure portion 21a of the compressor 21 via the injection pipe 65 is the specific enthalpy target value (h8*) at which the degree of dryness of the refrigerant (third refrigerant) at the merging point between the injection pipe 65 and the intermediate-pressure portion 21a of the compressor 21 is one. The degree of dryness of the third refrigerant being one means that all of the refrigerant in the intermediate-pressure portion 21a of the compressor 21 is a gas refrigerant. For this reason, even in the case where the density of the injected refrigerant is high, the increase in the compression load in the intermediate-pressure portion 21a of the compressor 21 is suppressed, ant thus, it is possible to suppress the decrease in the energy efficiency rating (EER) while maximizing the refrigeration capacity.

[0115] Further, in this embodiment, since a non-azeotropic mixed refrigerant is used as the refrigerant, it is possible to appropriately calculate a specific enthalpy of the refrigerant in the two-phase region and appropriately control the degree of opening of the injection control valve 29 such that a target injection ratio is obtained. That is, the temperature of the non-azeotropic mixed refrigerant changes with the specific enthalpy due to the temperature gradient. The degree of dryness of the refrigerant in the intermediate-pressure portion 21a of the compressor 21 can be calculated using the fact that the degree of dryness of the refrigerant is determined by two variables of a temperature and a pressure.

[0116] Further, in the existing technology (see, for example, Patent Literature 1) in which the degree of opening of the injection control valve is adjusted in accordance with the liquid-side outlet temperature of the inter-refrigerant heat exchanger in order to improve the refrigeration capacity, the refrigerant suction point (corresponding to M4) in the high stage of the compressor shifts to the left of the saturated vapor line as shown in Fig. 5, and the degree of dryness of the refrigerant at this point cannot be set to one. For this reason, the high stage load of the compressor increases, which has been a factor in reducing the energy efficiency rating.

[0117] On the other hand, in accordance with this embodiment, since the degree of opening of the injection control valve 29 is adjusted sch that the degree of dryness of the refrigerant in the intermediate-pressure portion 21a of the compressor 21 is one, the refrigerant suction point (M4) in the high stage of the compressor 21 is located on the saturated vapor line, which suppresses the high stage load of the compressor 21. Therefore, it is possible to improve the energy efficiency rating as compared with the case of Fig. 5.

[0118] Further, in accordance with this embodiment, since a scroll compressor is used as the compressor 21 and thus, the injection port is wider and the internal pressure of the compressor is also lower than those of a rotary compressor and the like, it is possible to increase the injection amount. As a result, it is possible to stably perform target injection control regardless of the rotation speed of the compressor 21.

Reference Signs List

[0119]

1 air conditioner
2 outdoor unit
3 indoor unit
10 refrigerant circuit
21 compressor
21a intermediate-pressure portion
22 four-way valve
23 outdoor heat exchanger
28 outdoor expansion valve
29 injection control valve
41 injection temperature sensor (first temperature detector)
42 injection pressure sensor (first pressure detector)
65 injection pipe
72 suction pressure sensor (second pressure detector)
74 suction temperature sensor (second temperature detector)
71 discharge pressure sensor (third pressure detector)
77c outdoor-unit-liquid-pipe temperature sensor (third temperature detector)
200 outdoor unit control device

Claims

1. An air conditioner, comprising:

   a refrigerant circuit that includes a compressor, an outdoor heat exchanger, an indoor heat exchanger, a pressure reducer that is disposed between the outdoor heat exchanger and the indoor heat exchanger, a four-way valve that switches a flow direction of a refrigerant that is a non-azeotropic mixed refrigerant discharged from the compressor, an injection control valve that reduces a pressure of a first refrigerant that is part of a refrigerant condensed in the outdoor heat exchanger during a cooling operation, an inter-refrigerant heat exchanger that exchanges heat between the first refrigerant whose pressure has been reduced by the injection control valve and a second refrigerant that is a remainder of the refrigerant condensed in the outdoor heat exchanger, and an injection pipe that guides the first refrigerant that has flowed out from the inter-refrigerant heat exchanger to an intermediate-pressure portion of the compressor;
   a first temperature detector that detects a temperature of the first refrigerant that has flowed out from the inter-refrigerant heat exchanger and introduced into the intermediate-pressure portion of the compressor;
   a first pressure detector that detects a pressure of the first refrigerant that has flowed out from the inter-refrigerant heat exchanger and introduced into the intermediate-pressure portion of the compressor; and
   a control device that controls the degree of opening of the injection control valve such that a specific enthalpy of the first refrigerant calculated on a basis of a refrigerant temperature detected by the first temperature detector and a refrigerant pressure detected by the first pressure detector is a specific enthalpy target value at which the degree of dryness of a third refrigerant that is a refrigerant at a merging point between the injection pipe and the intermediate-pressure portion of the compressor is one.

2. The air conditioner according to claim 1, further comprising:

   a second temperature detector that detects a temperature of a refrigerant sucked into the compressor; and
   a second pressure detector that detects a pressure of a refrigerant sucked into the compressor,
   the control device

      calculating a refrigerant temperature in the intermediate-pressure portion of the compressor on a basis of a refrigerant temperature detected by the second temperature detector, a refrigerant pressure detected by the first pressure detector, and a refrigerant pressure detected by the second pressure detector,
      calculating a specific enthalpy corresponding to the degree one of dryness of the third refrigerant on a basis of a specific enthalpy of a refrigerant in the intermediate-pressure portion of the compressor calculated on a basis of the refrigerant pressure detected by the first pressure detector and the refrigerant temperature in the intermediate-pressure portion of the compressor, and
      calculating the specific enthalpy target value of the first refrigerant on a basis of a specific enthalpy of the refrigerant in the intermediate-pressure portion of the compressor and the specific enthalpy corresponding to the degree one of dryness of the third refrigerant.

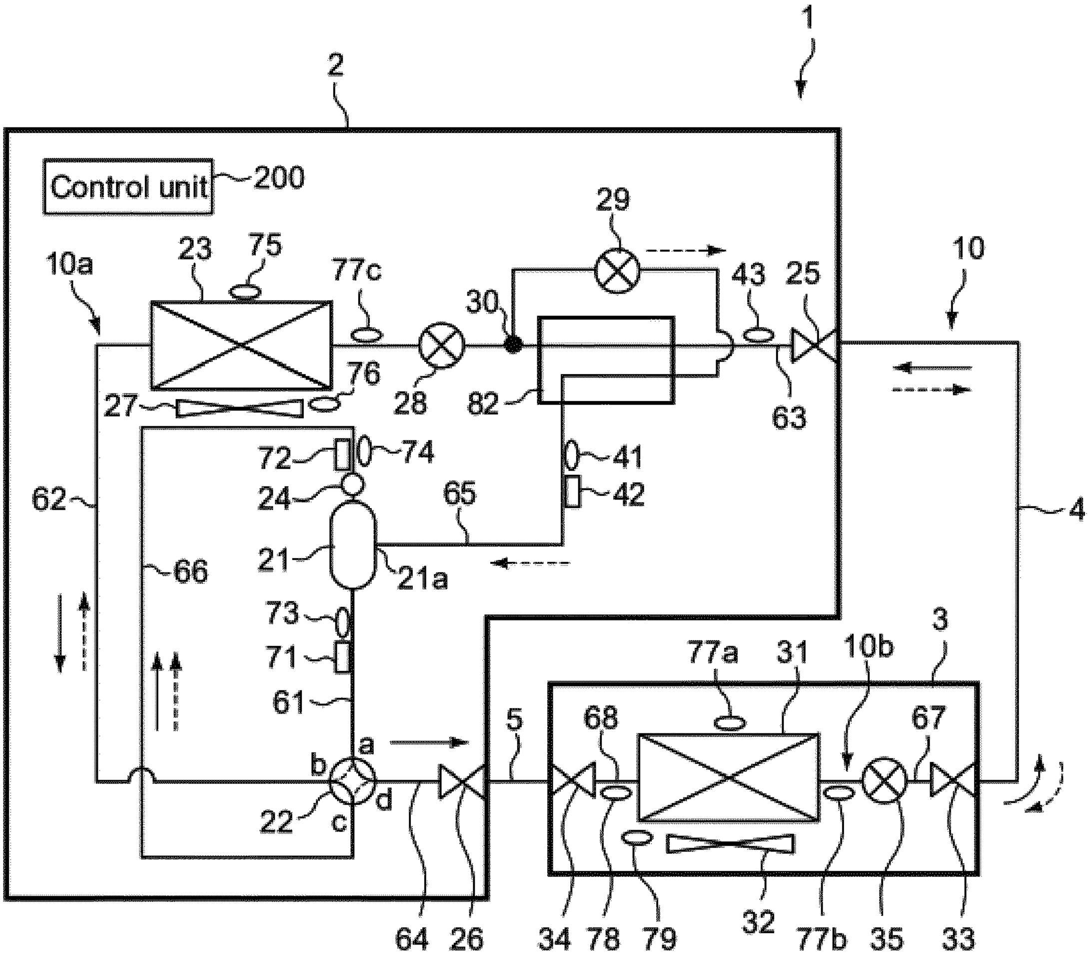
3. The air conditioner according to claim 1 or 2, further comprising:

   a third pressure detector that detects a pressure of a refrigerant discharged from the compressor;
   a third temperature detector that detects a temperature of a refrigerant flowing out from the outdoor heat exchanger during the cooling operation; and
   a fourth temperature detector that detects a temperature of the second refrigerant flowing out from the inter-refrigerant heat exchanger during the cooling operation,
   the control device

      calculating a specific enthalpy of the refrigerant flowing out from the outdoor heat exchanger during the cooling operation on a basis of a pressure detected by the second pressure detector and a temperature detected by the third temperature detector,
      calculating a specific enthalpy of the second refrigerant flowing out from the inter-refrigerant heat exchanger during the cooling operation on a basis of the pressure detected by the second pressure detector and a temperature detected by the fourth temperature detector, and
      calculating an injection ratio that is a ratio of the first refrigerant to the second refrigerant on a basis of the specific enthalpy of the first refrigerant, the specific enthalpy of the refrigerant flowing out from the outdoor heat exchanger during the cooling operation, and the specific enthalpy of the second refrigerant flowing out

from the inter-refrigerant heat exchanger during the cooling operation.

4. The air conditioner according to claim 3, wherein
the control device controls the degree of opening of the injection control valve such that the injection ratio is a target injection ratio when the specific enthalpy of the first refrigerant is the specific enthalpy target value.


1
2
Control unit
200
29
10a
23
75
77c
30
43
25
10
76
27
28
82
63
72
74
41
24
65
42
62
4
21
21a
66
73
3
77a
31
10b
71
61
68
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b
d
22
c
64
26
34
78
79
32
77b
35
33

FIG.1

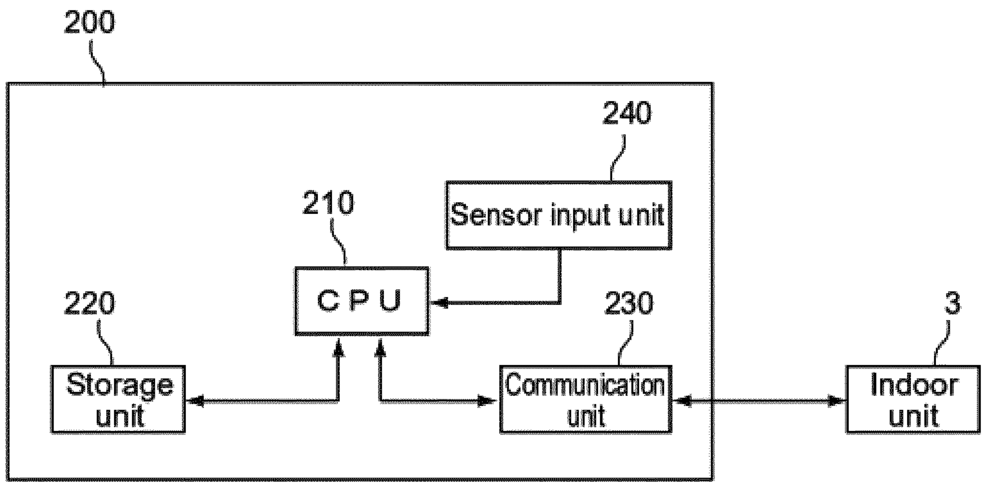
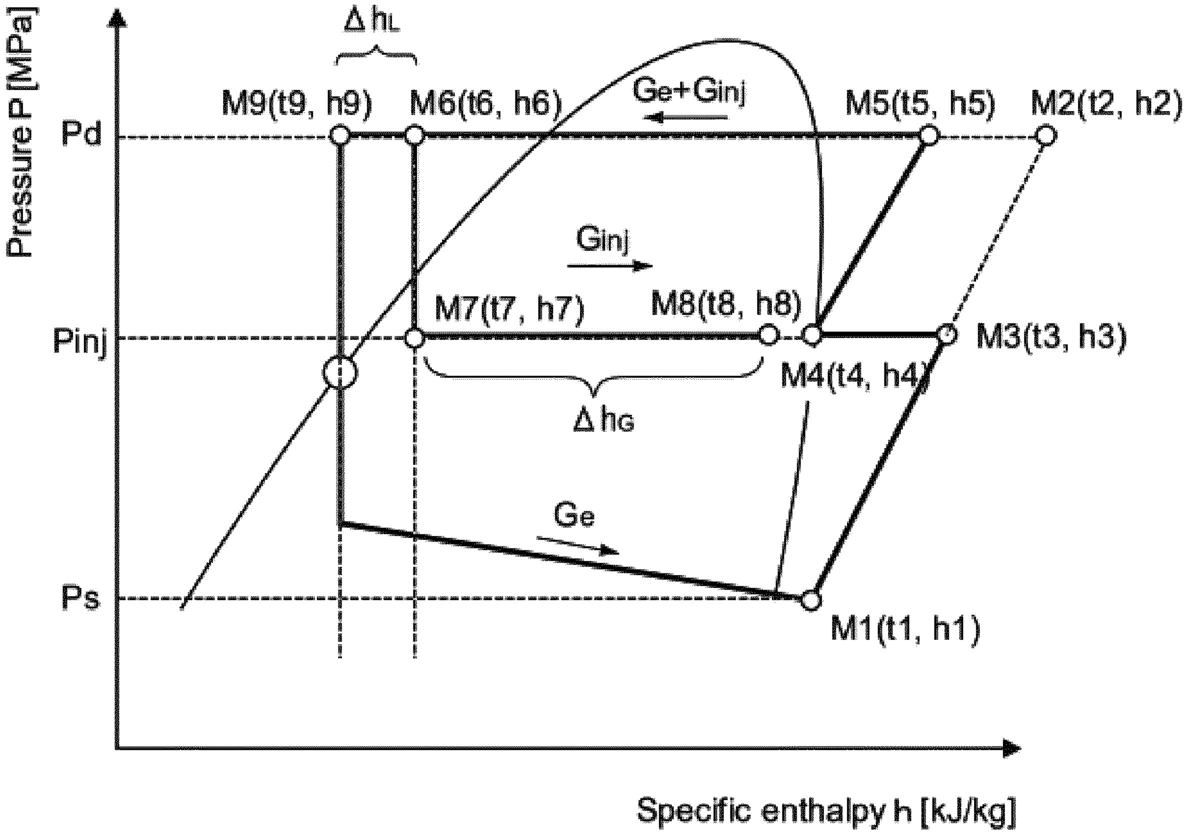

200
240
210
Sensor input unit
CPU
220
230
3
Storage unit
Communication unit
Indoor unit

FIG.2



$$\text{Injection ratio } \alpha = \frac{G_{inj}}{G_e} = \frac{\Delta h_L}{\Delta h_G}$$

$$= \frac{h6 - h9}{h8 - h6} = \frac{h3 - h4}{h4 - h8}$$

FIG.3

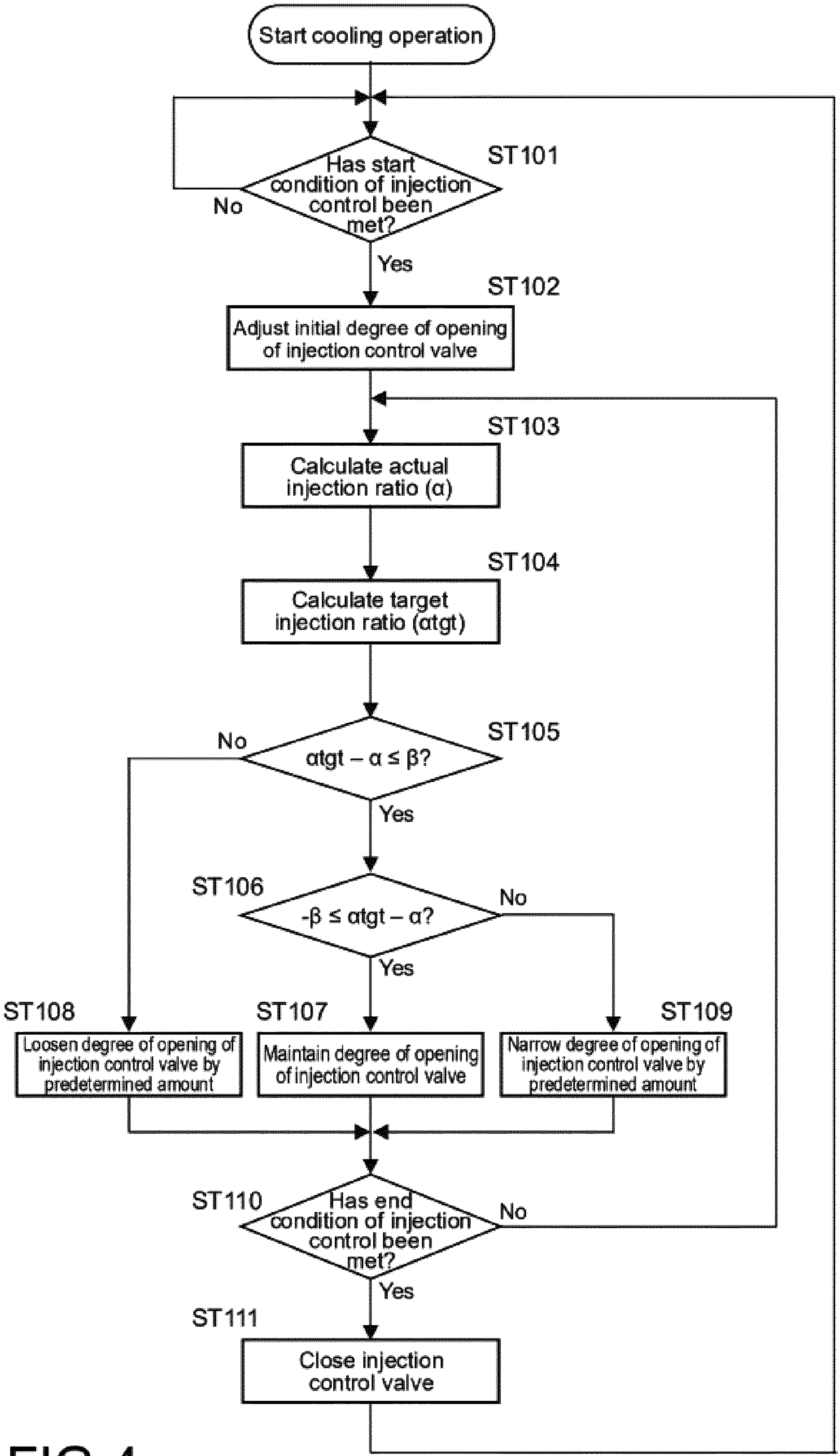

Start cooling operation
Has start condition of injection control been met?
ST101
No
Yes
ST102
Adjust initial degree of opening of injection control valve
ST103
Calculate actual injection ratio (α)
ST104
Calculate target injection ratio (αtgt)
ST105
αtgt – α ≤ β?
No
Yes
ST106
-β ≤ αtgt – α?
No
Yes
ST108
Loosen degree of opening of injection control valve by predetermined amount
ST107
Maintain degree of opening of injection control valve
ST109
Narrow degree of opening of injection control valve by predetermined amount
ST110
Has end condition of injection control been met?
No
Yes
ST111
Close injection control valve

FIG.4

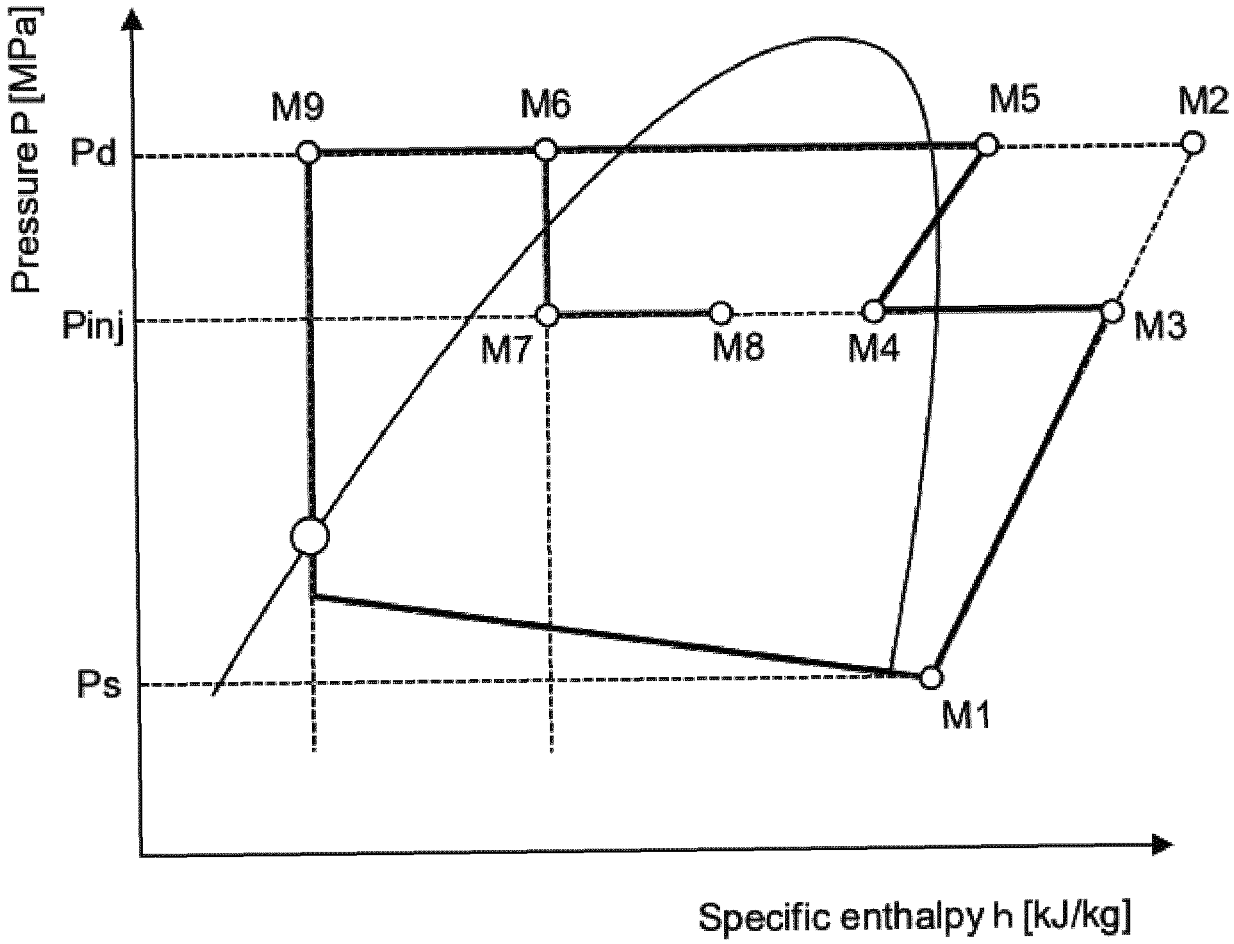

Pressure P [MPa]
Pd
Pinj
Ps
M9
M6
M5
M2
M7
M8
M4
M3
M1
Specific enthalpy h [kJ/kg]

FIG.5

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2023/011779

A. CLASSIFICATION OF SUBJECT MATTER

F25B 1/00(2006.01)i; ***F25B 1/10***(2006.01)i
FI: F25B1/00 331E; F25B1/00 304H; F25B1/00 311A; F25B1/00 396B; F25B1/10 R

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)

F25B1/00-49/04

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Published examined utility model applications of Japan 1922-1996
Published unexamined utility model applications of Japan 1971-2023
Registered utility model specifications of Japan 1996-2023
Published registered utility model applications of Japan 1994-2023

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 2004-183913 A (MITSUBISHI ELECTRIC CORP) 02 July 2004 (2004-07-02) paragraphs [0011]-[0014], [0018], [0026]-[0027], fig. 1, 4	1-4
A	WO 2013/001572 A1 (MITSUBISHI ELECTRIC CORP) 03 January 2013 (2013-01-03) paragraphs [0161]-[0179], fig. 14-16	1-4
A	WO 2015/001613 A1 (MITSUBISHI ELECTRIC CORP) 08 January 2015 (2015-01-08) paragraphs [0010]-[0019], [0043]-[0045], fig. 1-3	1-4

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

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
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"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"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
05 June 2023	**13 June 2023**

Name and mailing address of the ISA/JP	Authorized officer
Japan Patent Office (ISA/JP) **3-4-3 Kasumigaseki, Chiyoda-ku, Tokyo 100-8915** **Japan**	Telephone No.

Form PCT/ISA/210 (second sheet) (January 2015)

INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/JP2023/011779

Patent document cited in search report	Publication date (day/month/year)	Patent family member(s)	Publication date (day/month/year)
JP 2004-183913 A	02 July 2004	(Family: none)	
WO 2013/001572 A1	03 January 2013	US 2014/0083123 A1 paragraphs [0189]-[0208], fig. 14-16 EP 2728277 A1	
WO 2015/001613 A1	08 January 2015	GB 2530453 A paragraphs [0010]-[0019], [0043]-[0045], fig. 1-3	

Form PCT/ISA/210 (patent family annex) (January 2015)

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2018179383 A **[0004]**