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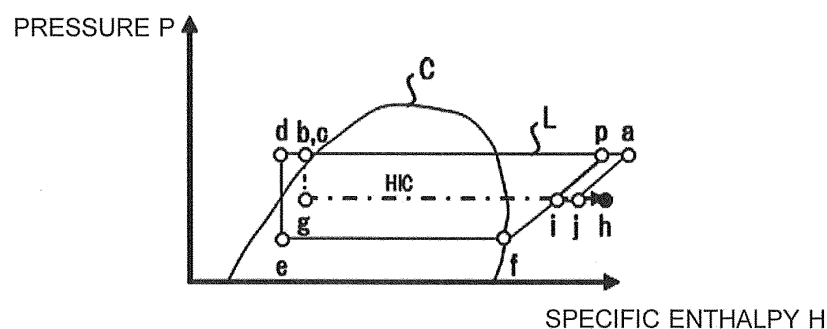
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(54) **VAPOR COMPRESSION REFRIGERATION CYCLE**

(57) A vapor compression refrigeration cycle that prevents decrease in a discharge temperature at an outlet of a compressor and improves operating efficiency in a field of warming and water heating is obtained. A vapor compression refrigeration cycle of the present invention is a vapor compression refrigeration cycle using HFO as refrigerant including: a compressor provided with an injection port for injecting circulating refrigerant; a pressure sensor and a temperature sensor, provided adjacent to the injection port, that measure pressure and a temperature; an expansion device that compresses and expands the refrigerant; and a controller that controls an

opening degree of the expansion device based on the pressure and the temperature obtained by the pressure sensor and the temperature sensor, in which the controller controls the expansion device for causing a portion, of a P-H diagram related to the vapor compression refrigeration cycle, corresponding to a compression process to be positioned outside a saturation curve composed of a saturated liquid line and a saturated vapor line, to be positioned below critical pressure, and to be positioned on a high enthalpy region side of the saturated vapor line.

FIG. 2



Description

Technical Field

[0001] The present invention relates to a refrigeration cycle of an intermediate injection type using HFO refrigerant, such as HFO-1234yF, HFO-1234ze or other low GWP refrigerants, and in particular, relates to improvement of operating efficiency thereof.

Background Art

[0002] Conventionally, for example, use of specified chlorofluorocarbons employed in water heaters or air conditioners has caused concern about destruction of the ozone layer or increased global warming, to be thereby subjected to regulations. In Europe, switching to the low GWP refrigerant is being pursued by, for example, F-gas Regulation. Based on such background, from now on, it is expected that the existing refrigerants, such as HFC or others, will be replaced with HFO refrigerants.

[0003] However, as compared to the conventional HFC refrigerant, the HFO refrigerant has the drawback that refrigerant at an outlet of a compressor has a low concentration, a low latent heat and a low discharge temperature. Therefore, when the refrigerant is used for the purpose of water heating or heating, reduction in COP, that is, reduction in operating efficiency was caused.

[0004] By the way, in regard to increase in operating efficiency of the refrigeration cycle, as a method thereof, improvement in refrigerant efficiency by the intermediate injection type is made, and improvement of COP (Coefficient of Performance) by an intermediate injection circuit in the refrigeration cycle caused thereby is achieved by improvement in compressor efficiency and improvement in cycle efficiency. In general, to improve the cycle efficiency, the refrigerant temperature is raised. When the refrigerant temperature is raised, a refrigerant circulation amount in a gas cooler side is increased, and thereby heating capacity and the COP are improved. As a conceivable specific method to achieve this is, for example, an intermediate injection port entrance is provided on a side surface of a compressor, and refrigerant in a heated gas state is injected from the part.

[0005] As to the intermediate injection type, a relationship between suction SH (superheat: a degree of superheat) of refrigerant in an intermediate injection port of a compressor and a discharge temperature of refrigerant at an outlet of the compressor is as shown in Fig. 12, and from this, it can be confirmed that the more the suction SH at the intermediate injection port of the compressor is, the more the discharge temperature of the refrigerant at the outlet of the compressor is, so that the coefficient of performance COP of the refrigeration cycle are improved with the increased suction SH at the intermediate injection port.

[0006] Patent Literature 1 shows a conventional example of this type, and a general outline of the conventional

vapor compression refrigeration cycle is as follows: a compressor having an injection port that serves as a refrigerant passage port at an intermediate part of a compression process that compresses sucked refrigerant and discharges it, a four-way valve, a heat-source side expansion device, a heat source side heat exchanger, a load-side expansion device and a load side heat exchanger, which are connected to form a circuit for circulating the refrigerant.

(Refer to Patent Literature 1)

[0007] The configuration of the vapor compression refrigeration cycle of the above Patent Literature 1 will be described with reference to Fig. 1 of Embodiment 1. Basically, the configuration is substantially similar to the above Patent Literature 1 except for a specific component (a liquid receiver 4), and therefore, Fig. 1 is used for describing the configuration of the conventional art.

[0008] The vapor compression refrigeration cycle is a vapor compression refrigeration cycle 100 including an intermediate injection port 16 for injecting a heated refrigerant into an inside of a compressor 1, with HFO as a new refrigerant, the vapor compression refrigeration cycle 100 being configured with a refrigeration cycle constituted by connecting, via refrigerant pipes, a compressor 1, a condenser 3 that is a heat exchanger on a high temperature side, an expansion valve 8 that is a second expanding device and an evaporator 7 that is a heat exchanger on a low temperature side, and a controller 18 that controls the refrigeration cycle, in which the compressor 1, the condenser 3, the expansion valve 8 and the evaporator 7 are connected with one another by the refrigerant pipes.

[0009] Moreover, by using a P-H diagram (pressure-specific enthalpy line diagram) in Fig. 11, operation of the vapor compression refrigeration cycle according to the above Patent Literature 1 will be described. In the P-H diagram, the vertical axis indicates the pressure P of the refrigerant, and the horizontal axis indicates the specific enthalpy H of the refrigerant, and in the Fig. 11, a saturation curve (C) composed of a saturated liquid line and a saturated vapor line and a P-H line (L) are depicted.

[0010] First, the P-H line (L) corresponding to the above Patent Literature 1 will be generally described: a general outline thereof is a combination of two geometries, a portion of a substantially non-isosceles trapezoid (that may be referred to as "the former"), in which an upper base is longer than a lower base, and a portion of a parallelogram (that may be referred to as "the latter") that is in contact with an upward-sloping portion to be contained in a portion of the non-isosceles trapezoid.

[0011] Hereinafter, in the former, the line p-d is configured with an upper base, the line d-e is configured with a short side, the line e-f is configured with a lower base (lower base < upper base) and the line f-p is configured with a long side. In the latter, the line a-p is configured with an upper base, the line p-i is configured with a right

side, the line i-j is configured with a lower base and the line j-a is configured with a left side. Note that b and c are overlapped and are the same point; therefore, represented as b, c.

[0012] Moreover, a portion of a dot-and-dash line drawn inside the substantially non-isosceles trapezoid corresponds to the intermediate injection. The portion of the dot-and-dash line is configured with the line b-g parallel to the line d-e and the line g-h parallel to the line e-f. Moreover, i, j are positioned on an extension of the line g-h.

[0013] Here, a to j positioned in Fig. 11 correspond to a to j of the refrigeration cycle in Fig. 1, respectively. In Fig. 1, each of a to j indicates a position in configuration of the refrigeration cycle; however, in Fig. 11, a to j indicate the states of the refrigeration cycle corresponding to those positions.

[0014] In Fig. 11, "a" (hereafter the quotation may be omitted) indicates the state of refrigerant corresponding to the outlet of the compressor 1; the pressure is P_a , and the specific enthalpy is H_a at "a".

[0015] During the process from a to b, the refrigerant changes from the state a to the state b positioned in the left direction in parallel to the horizontal axis along the solid line. This corresponds to, when the refrigerant passes through the condenser 3, rejecting heat from the refrigerant and reduction in enthalpy due to condensation from a gas state to a liquid state. In the state b (c), as compared to the state a, the pressure P_b is not changed, that is, $P_b (P_c) = P_a$, the specific enthalpy is reduced to $H_b (H_c)$.

[0016] Then, c indicates the state of the refrigerant flowing through the entrance of the second expansion device 8, and c is plotted at the same position as b. Here, in the P-H diagram in Fig. 11, the reason why b and c are plotted at the same position is that, though positions of the refrigerant flowing through the refrigerant pipe are different in the configuration shown in Fig. 1, states of the refrigerant (pressure, specific entropy) are the same on the P-H diagram shown in Fig. 11.

[0017] During the process from c to d, the state c changes to the state d positioned in the left direction parallel to the horizontal axis along the solid line. This is because, when the refrigerant passes through the inside heat exchanger 5, a primary refrigerant pipe connected to the evaporator 7 and a auxirially refrigerant pipe (to be described later) connected to the injection port 16 exchange heat with each other. In other words, the primary refrigerant pipe discharges heat, and the auxirially refrigerant pipe absorbs heat. The refrigerant flowing through the primary refrigerant pipe corresponds to reduction in the enthalpy thereof. In the state d, as compared to the state c, the pressure P_d is not changed, that is, $P_d = P_c$, and the specific enthalpy is reduced to H_d .

[0018] During the process from d to e, the refrigerant changes from the state d to the state e positioned in the lower direction in parallel to the vertical axis along the solid line. This corresponds to reduction in the pressure

due to expansion of the refrigerant at the time when the refrigerant passes through the first expansion device 6,. In the state e, as compared to the state d, the specific enthalpy is not changed, that is, $H_e = P_a$, and the pressure is reduced to P_e .

[0019] During the period from e to f, the refrigerant changes from the state e to the state f positioned in the right direction in parallel to the horizontal axis along the solid line. This corresponds to absorption of heat by the refrigerant evaporating from the liquid state to the gas state and increase in enthalpy at the time when the refrigerant passes through the evaporator 7. In the state f, as compared to the state e, the pressure P_f is not changed, that is, $P_f = P_e$, and the specific enthalpy is increased to H_f .

[0020] During the period from f to i, the refrigerant changes from the state f to the state i positioned in the upper right direction along the upper-right sloping linear solid line. This corresponds to increase in the pressure and the enthalpy of the refrigerant due to condensation of the refrigerant to the intermediate pressure inside the compressor 1 by the compressor 1. In the state i, the pressure P_f is increased to P_i and the specific enthalpy H_f is increased to H_i .

[0021] Similarly, during the process from b (c) corresponding to the intermediate injection to g, the refrigerant changes from the state b (c) to the state g positioned in the lower direction parallel to the vertical axis along the dot-and-dash line. This corresponds to reduction in the pressure due to expansion of the refrigerant, when the refrigerant passes through the second expansion device 8,. In the state g as compared to the state c, the specific enthalpy is not changed, that is, $H_g = H_c$, and the pressure is reduced to P_g .

[0022] During the process from g to h, the refrigerant changes from the state g to the state h positioned in the right direction parallel to the horizontal axis along the dot-and-dash line. This corresponds to absorption of heat by the refrigerant by means of heat exchange, and increase in enthalpy of the refrigerant when the refrigerant passes through the inside heat exchanger 5. In the state h, as compared to the state g, the pressure P_h is not changed, that is, $P_h = P_g$, and the specific enthalpy is increased to H_h . Here, h is on the saturation curve (C) or inside the line C.

[0023] During the process from h, i to j, the states h, i and j are on the same line and the refrigerant changes from the state h toward the state j in the right direction parallel to the horizontal axis along the dot-and-dash line, and contrary to this, the refrigerant changes from the state i toward j to the state j positioned in the left direction parallel to the horizontal axis along the dot-and-dash line. This corresponds to averaging of enthalpy of the refrigerant in total as a result of mixing of two lines of refrigerant, the refrigerant supplied via the primary refrigerant pipe and the refrigerant supplied via the auxirially refrigerant pipe and thermal average thereof. In the state j, as compared to the state i, the pressure P_j is not changed,

that is, $P_j = P_h = P_i$, and the specific enthalpy H_j is an average value of H_h and H_i .

[0024] During the process from j to a, the refrigerant changes from the state j to the state a positioned in the upper right direction along the upper-right sloping linear solid line. The compressor 1 compresses thereinside the refrigerant from intermediate pressure to high pressure, and thereby, the pressure and the enthalpy of the refrigerant are increased. In the state a, the pressure P_a has been increased from the pressure P_j and the specific enthalpy H_a has been increased from the pressure H_j .

[0025] This brings about the state in which the pressure is P_a and the specific enthalpy is H_a and the refrigerant returns to the original state a of refrigerant flowing at the outlet of the compressor 1, to thereby complete 1 cycle.

[0026] Note that, as is clear from the above description, in the above P-H line (L) diagram, there exist two compression processes, namely, the first compression process during the period from f to i and the second compression process during the period from j to a, and a portion commonly corresponding to the two compression processes is simply referred to as a compression process.

Citation List

Patent Literature

[0027] Patent Literature 1: Japanese Unexamined Patent Application Publication No. 2013-15264 (Paragraph 0014, Fig. 1)

Summary of Invention

Technical Problem

[0028] However, in the conventional intermediate injection type shown in Fig. 11, the refrigerant injected from the injection port 16 is not provided with sufficient suction SH. In other words, in Fig. 11, h is on the curve of the saturated vapor line, whereas, j is slightly on the right side of the saturated vapor line and on the left side of the compression process of the P-H diagram related to the vapor compression refrigeration cycle. Therefore, the portion of the P-H diagram corresponding to the compression process does not exist at a proper position on the P-H diagram. In other words, this means that the controller 18 does not provide sufficient suction SH to the refrigerant inside the compressor 1 and cannot control the expansion valve 8 to cause j to be positioned on the right side of the saturated vapor line.

[0029] Therefore, operation by the conventional injection method naturally has its own limit to obtain heat for water heating, such as, a need to raise the temperature of circulation water on the second side up to a predetermined temperature (for example, 60 degrees C) by heating by use of heat generated from the condenser 3, which is a heat exchanger on the high temperature side.

[0030] In this manner, since the discharge temperature

at the outlet of the compressor sharply decreases, the refrigeration cycle using HFO refrigerant as new refrigerant instead of the conventional HFC, there is a natural limit in water heating capacity, and accordingly, when the injection amount is determined in accordance with the conventional suggestion, it is impossible to obtain sufficient performance improvement effect and reliability improvement effect by intermediate injection.

[0031] The present invention has been made to overcome the above-described problem, and, even when the HFO refrigerant that tends to decrease in the discharge temperature at the outlet side of the compressor is used, decrease in the discharge temperature of the refrigerant is prevented, to thereby obtain highly efficient refrigeration cycle for water heating and warming.

Solution to Problem

[0032] A vapor compression refrigeration cycle of one aspect of the present invention is a vapor compression refrigeration cycle using HFO as refrigerant, the vapor compression refrigeration cycle being characterized by including: a compressor provided with an injection port for injecting refrigerant circulating in the vapor compression refrigeration cycle; a pressure sensor and a temperature sensor, provided adjacent to the injection port, that measure pressure and a temperature; an expansion device that performs opening and closing operation for compressing and expanding the refrigerant; and a controller that controls an opening degree of the expansion device based on the pressure and the temperature obtained by the pressure sensor and the temperature sensor, wherein the controller controls the expansion device for causing a portion, of a P-H diagram related to the vapor compression refrigeration cycle, corresponding to a compression process to be positioned outside a saturation curve composed of a saturated liquid line and a saturated vapor line, to be positioned below critical pressure, and to be positioned on a high enthalpy region side of the saturated vapor line.

Advantageous Effects of Invention

[0033] The vapor compression refrigeration cycle related to one aspect of the present invention is able to maintain the temperature of the refrigerant injected into the inside of the compressor of the intermediate injection type at a proper superheat state inside the compressor, and even when the HFO refrigerant is used, able to prevent decrease in the discharge temperature at the outlet of the compressor caused by injection into the inside of the compressor, and eventually, to improve operating efficiency (COP) of the refrigeration cycle.

Brief Description of Drawings

[0034]

[Fig. 1] Fig. 1 is a circuit diagram of a vapor compression refrigeration cycle 100 showing Embodiment 1 of the present invention.

[Fig. 2] Fig. 2 is a P-H diagram showing a state of refrigerant related to Embodiment 1.

[Fig. 3] Fig. 3 is a P-H diagram showing an image of rising in a discharge temperature when heated refrigerant is supplied to an intermediate injection port 16.

[Fig. 4] Fig. 4 is a flow diagram of opening degree control of a second expansion device 8 related to Embodiment 1.

[Fig. 5] Fig. 5 is a circuit diagram of a vapor compression refrigeration cycle 200 when an air-to-refrigerant heat exchanger 10 is used related to Embodiment 2.

[Fig. 6] Fig. 6 is a block diagram showing an overview of flow of a signal related to Embodiment 2.

[Fig. 7] Fig. 7 is a circuit diagram of a vapor compression refrigeration cycle 300 when heat discharged from a control board is used related to a modified example of Embodiment 2.

[Fig. 8] Fig. 8 is a circuit diagram of a vapor compression refrigeration cycle 400 when heat discharged from a compressor 1 is used related to Embodiment 3.

[Fig. 9] Fig. 9 is a block diagram showing an overview of flow of a signal related to Embodiment 3.

[Fig. 10] Fig. 10 is a circuit diagram of a vapor compression refrigeration cycle 400 when gas refrigerant injection inside a liquid receiver 4 is used related to Embodiment 4.

[Fig. 11] Fig. 11 is a P-H diagram showing a state of refrigerant related to a conventional injection circuit 9.

[Fig. 12] Fig. 12 shows a relationship between suction SH and COP (Coefficient of Performance) in the intermediate injection port 16.

Description of Embodiments

Embodiment 1

[0035] Fig. 1 shows the entire vapor compression refrigeration cycle 100 related to Embodiment 1 of the invention. Hereinafter, a refrigeration cycle for water heating placed as a home water heater or a water heater for a building will be described as an example.

[0036] Prior to specific description of a configuration in detail, first, the vapor compression refrigeration cycle 100 and main circuits constituting the main circuits will be described.

[0037] Hereinafter, Embodiment 1 will be specifically described based on Fig. 1.

[0038] The vapor compression refrigeration cycle 100 includes: a compressor 1 that compresses refrigerant; a four-way valve 2 that changes the flowing direction of the refrigerant; a condenser 3; a liquid receiver 4, which is a

high-pressure container for storing high-pressure liquid refrigerant; an inside heat exchanger 5 that exchanges heat; an evaporator 7; a first expansion device 6 and a second expansion device 8 that expand the refrigerant; and an injection circuit 9 that heats the refrigerant injected from an injection port 16.

[0039] Moreover, in the proximity of the entrance of the injection port 16, a pressure sensor 13 and a temperature sensor 14 for measuring pressure and temperature of the refrigerant are provided, and further, a controller 18 that controls an opening degree of the second expansion device 8 is provided. Then, the controller 18 has a function of controlling the opening degree of the second expansion device 8 based on information of the refrigerant obtained by the pressure sensor 13 and the temperature sensor 14.

[0040] The liquid receiver 4 is provided on the way of a route of a pipe connecting an output side of the condenser 3 and an inlet side of the second expansion device 8, and the refrigerant in a gas-liquid two-phase state flow accumulated in the liquid receiver 4 is branched from a primary refrigerant pipe, to be described later. Then, of gas-liquid two-phase flow accumulated in the liquid receiver 4, the refrigerant in one state flows into the inlet on the upper side of the inside heat exchanger 5, and the other one flows into the inlet of the lower side of the inside heat exchanger 5, via the second expansion device 8.

[0041] Inside the heat exchanger 5, two lines of refrigerant pipes, namely, the primary refrigerant pipe that connects the liquid receiver 4 and the expansion device 6 and the auxirially refrigerant pipe that connects the liquid receiver 4 and the second expansion device 6 via the second expansion device 8, are arranged closely, and heat is exchanged between the refrigerant flowing through the primary refrigerant pipe and the refrigerant flowing through the auxirially refrigerant pipe with each other. As a result, heat is rejected from the primary refrigerant pipe to the auxirially refrigerant pipe and heat from the auxirially refrigerant pipe is absorbed by the primary refrigerant pipe, and accordingly, the temperature of the refrigerant flowing through the primary refrigerant pipe is decreased and the temperature of the refrigerant flowing through the auxirially refrigerant pipe is raised.

[0042] As the first expansion device 6 and the second expansion device 8, expansion valves are used to decompress the refrigerant in the primary refrigerant pipe and the auxirially refrigerant pipe, respectively. The injection circuit 9 includes the auxirially refrigerant pipe and heats the refrigerant injected from the injection port 16 into the inside of the compressor 1.

[0043] The injection port 16 is formed on a side surface of the compressor 1 to inject the refrigerant being the heated gas state into the inside of the compressor 1. Moreover, the pressure sensor 13 and the temperature sensor 14 are provided adjacent to the injection port 16 to measure the pressure and temperature of the refrigerant. Moreover, at the inlet of the inside heat exchanger

5, a temperature sensor 17 is provided. Further, an outside-air temperature sensor 15 is disposed close to the condenser 3 to measure the air temperature around the condenser 3.

[0044] Then, the controller 18 is connected to the second expansion device 8 via a communication line or others and controls opening and closing operation of the second expansion device 8 based on information of the refrigerant obtained by the pressure sensor 13 and the temperature sensor 14. For example, the controller 18 controls the opening degree of the second expansion device 8 so that the refrigerant injected from the compressor injection port is heated vapor with a degree of superheat of 20 degrees C or more and a pressure higher than a low pressure at the compressor by a difference of no less than 0.35 times a difference between a high pressure and the low pressure in a compression process of the compressor.

[0045] Note that the condenser 3 corresponds to the heat exchanger on the high temperature side, and the high heat obtained in the condenser 3 is used as a high temperature heat source on the second side. Similarly, the evaporator 7 corresponds to the heat exchanger on the low temperature side, and the low heat obtained in the evaporator 7 is used as a low temperature heat source on the second side.

[0046] First, a sketch of the P-H diagram (L) of the embodiment will be described.

[0047] Though operation of the vapor compression refrigeration cycle according to the present invention will be described later, since a general outline thereof is the same as the conventional P-H diagram except that the states h and j shown in Fig. 11 are shifted to the right side (shifted to a high enthalpy region side), description thereof is omitted, and points of difference from the conventional art will be mainly described with reference to Fig. 2.

[0048] The sketch of the P-H line (L) is the same in the point that it is a combination of two geometries, a portion of a substantially non-isosceles trapezoid (former), in which an upper base is longer than a lower base, and a portion on the right of a parallelogram (latter) that is in contact with an upward-sloping portion of the non-isosceles trapezoid; however, there is a difference that the latter is outside of the former.

[0049] Moreover, although the point that a to j positioned in Fig. 2 (the states of the refrigerant) correspond to a to j in the refrigeration cycle in Fig. 1 (positions on the configuration of the refrigeration cycle), respectively, is the same, there is a difference in the point that the operation of the vapor compression refrigeration cycle according to the present invention corresponds to Fig. 2.

[0050] Consideration will be given by dividing the operating period of the vapor compression refrigeration cycle according to the present invention into three periods; a first period (a period from a to g), a second period (a period from g to h) and a third period (a period from h to a). Of these, the first period and the third period are the

same as before. Here, description will be given with a focus on the second period (the period from g to h), which is different from before.

[0051] During the process from g to h, the refrigerant changes from the state g to the state h positioned in the right direction in parallel to the horizontal axis along the dot-and-dash line. This corresponds to absorption of heat by the refrigerant by means of heat exchange, and increase in enthalpy of the refrigerant at the time when the refrigerant passes through the inside heat exchanger 5. The state h corresponds to, as compared to the state g, constancy in the pressure P_h , that is, $P_h = P_g$, and increase in the specific enthalpy H_h . Here, h is on the outside of the saturation curve (C).

[0052] In other words, during the period from g to h, the refrigerant changes from the state g in the right direction parallel to the horizontal axis along the dot-and-dash line greater than that described in Fig. 11. This indicates that the refrigerant absorbs more heat than that described in Fig. 11 by heat exchange when the refrigerant passes through the inside heat exchanger 5, and the refrigerant having absorbs heat is supplied from the injection port 16, to thereby further increase enthalpy of the refrigerant inside the compressor 1.

[0053] Further, to describe specifically, the states h and j are shifted to the high enthalpy region side (the right side) of the conventional P-H diagram described by using Fig. 11 before. In other words, it can be found that the specific enthalpy of the refrigerant is increased inside the compressor 1 in Embodiment 1 of the present invention.

[0054] As described above, of the P-H diagram in Fig. 2 related to the vapor compression refrigeration cycle 100, both of the portions corresponding to those two compression process exist in a region satisfying the following three conditions.

Condition 1: positioned outside the saturation curve (C) composed of the saturated liquid line and the saturated vapor line.

Condition 2: positioned below critical pressure. (Put another way, the states h and j on the P-H diagram (L) are below the states a and p.)

Condition 3: positioned sufficiently on the high enthalpy region side of the saturated vapor line. (Put another way, the states h and j on the P-H diagram (L) are on the right side of the state i.)

[0055] In other words, this is nothing other than the controller 18 that provides sufficient suction SH to the refrigerant inside the compressor 1 and is able to control the expansion valve 8 to cause the point j to be positioned on the right side of the saturated vapor line.

[0056] Moreover, a portion of a line a-p from the point a to the point p indicates the rise ΔH of the discharge temperature of the refrigerant (see Fig. 3).

[0057] Adjustment in refrigerant temperature by control of the opening degree of the second expansion device 8 in the vapor compression refrigeration cycle 100 can

be achieved by carrying out processing procedures shown in Fig. 4 (hereinafter, step S1 to step S5). Description will be given with reference to Fig. 4.

[0058] When an operation command of the vapor compression refrigeration cycle 100 turns ON, the outside-air temperature sensor 15 detects air temperature around the condenser 3 (step S1).

[0059] The controller 18 detects the pressure value of the refrigerant from the pressure sensor 13, and obtains the temperature of the refrigerant from the temperature sensor 15. Thereafter, based on the pressure value obtained from the pressure sensor 13, the controller 18 computes a saturation temperature of the refrigerant at the pressure (step S2).

[0060] When computation of the saturation temperature of the refrigerant is completed at the above step S2, the controller 18 compares the saturation temperature with the temperature obtained by the temperature sensor 17 placed at the inlet of the inside heat exchanger 5, to thereby calculate the degree of superheat SH of the refrigerant (step S3). Here, the degree of superheat SH is a difference in temperature between the temperature at the inlet side of the inside heat exchanger 5 and the temperature at the outlet side thereof.

[0061] Next, the degree of superheat SH is compared with a target value of degree of superheat SHs, which is a preset target value, and based on the result of comparison, the controller 18 determines the opening degree of the second expansion device 8.

[0062] As a result of the comparison of the degree of superheat SH and the target value of degree of superheat SHs, when the target value of degree of superheat SHs is larger than the degree of superheat SH (target value of degree of superheat SHs > degree of superheat SH), control is exercised to reduce the opening degree of the second expansion device 8, whereas, when the target value of degree of superheat SHs is smaller than the degree of superheat SH (target value of degree of superheat SHs < degree of superheat SH), control is exercised to increase the opening degree of the second expansion device 8 (step S4).

[0063] Here, in step S4, the controller 18 determines the opening degree of the second expansion device 8 and directs the determination result to the second expansion device 8.

[0064] In the above step S4, after carrying out the opening degree control of the second expansion device 8, the controller 18 determines whether or not further control is required. In other words, whether or not continuous operation of the vapor compression refrigeration cycle 100 is required is determined, and when the continuous operation is further required, the process proceeds to YES to return to S1, whereas, when the continuous operation is not required, the process proceeds to NO, to thereby finish the opening degree control of the second expansion device 8 by the controller 18 (step S5).

[0065] Here, step S5 is a step to determine whether or not the opening degree of the second expansion device

8 is computed again.

[0066] Hereinafter, the processes of the above steps S1 to S5 are repeatedly carried out.

5 Embodiment 2

[0067] In a vapor compression refrigeration cycle 200 showing Embodiment 2 of the invention, since the system configuration, the P-H diagram and the control flow are substantially the same as those of Embodiment 1, description thereof is omitted, and different points from Embodiment 1 will be mainly described with reference to Fig. 5.

[0068] Embodiment 2 newly provides an air-to-refrigerant heat exchanger 10, which is a second heating unit, close to the auxirially refrigerant pipe communicating to the injection port 16, and accordingly, carries out heating of the refrigerant injected into the inside of the compressor 1 from the injection port 16 by use of two types of heating units, namely, the inside heat exchanger 5 and the air-to-refrigerant heat exchanger 10.

[0069] Therefore, it becomes easy to add the degree of superheat SH to the HFO refrigerant flowing through the auxirially refrigerant pipe communicating to the injection port 16, and accordingly, as compared to Embodiment 1, it is possible to prevent decrease in the discharge temperature at the outlet side of the compressor, and to realize further efficient refrigeration cycle for water heating and warming.

[0070] Further, in Embodiment 2, in addition to use of the two types of heating units, a fan is included in the air-to-refrigerant heat exchanger 10, which is the second heating unit, to perform rotation speed control by an inverter. Therefore, it is possible to obtain a sufficient heat exchanging amount while designing the heat exchanger compactly, and to stably supply the superheat refrigerant of a desired temperature from the injection port 16 of the compressor. In Embodiment 2, the states h and j are shifted to the high enthalpy region side (the right side) of the P-H diagram in Embodiment 1.

[0071] Moreover, the refrigeration cycle related to Embodiment 2 is able to omit installation of the fan by increasing the heat exchanging area of the air-to-refrigerant heat exchanger 10.

[0072] Moreover, in addition to the air-to-refrigerant heat exchanger 10, discharged heat from a control board may be used as a heat source (refer to Fig. 7). In this case, effect of adding further degree of superheat SH is expected.

[0073] Note that, as shown in Fig. 6, the flow of the control signal, when the controller 18 controls the second expansion device 8 and the air-to-refrigerant heat exchanger 10 based on the detected values of the refrigerant detected by the pressure sensor 13 and the temperature sensor 14, is shown.

Embodiment 3

[0074] In a vapor compression refrigeration cycle 300 showing Embodiment 3 of the invention, since the system configuration, the P-H diagram and the control flow are substantially the same as those of Embodiment 1, description thereof is omitted, and different points from Embodiment 2 will be mainly described with reference to Fig. 8.

[0075] In Embodiment 3, a circulation heat exchanger 11, which is a second heating unit, is provided at the same position as the air-to-refrigerant heat exchanger 10 of Embodiment 2 to replace thereof. In other words, close to the auxirially refrigerant pipe communicating to the injection port 16, the circulation heat exchanger 11, which is the second heating unit that preforms heat exchange with liquid (air or brine) recovering the discharged heat of the compressor 1, and a pump 12 are newly provided, and accordingly, heating of the refrigerant injected into the inside of the compressor 1 from the injection port 16 by use of two types of heating units, namely, the inside heat exchanger 5 and the circulation heat exchanger 11, is carried out.

[0076] Here, the circulation heat exchanger 11 heats the refrigerant flowing through the auxirially refrigerant pipe by utilizing the liquid (air or brine) recovering the discharged heat from the compressor 1 close to the auxirially refrigerant pipe. The pump 12 is disposed at the middle of the pipe connecting the compressor 1 and the circulation heat exchanger 11 to circulate the liquid (air or brine) recovering the discharged heat from the compressor 1.

[0077] Therefore, it becomes easy to add the degree of superheat SH to the HFO refrigerant flowing through the auxirially refrigerant pipe communicating to the injection port 16, and accordingly, as compared to Embodiment 1, it is possible to prevent decrease in the discharge temperature at the outlet side of the compressor, and to realize further efficient refrigeration cycle for water heating and warming. In Embodiment 3, the states h and j are shifted to the high enthalpy region side (the right side) of the P-H diagram in Embodiment 1.

[0078] Note that, in the above, by performing heat exchange by directly winding the injection circuit 9 around the compressor 1 as a modified example of Embodiment 3, it is possible to omit the circulation heat exchanger 11, the pump 12, and a liquid circuit configured with water (or brine) attached thereto. Here, though the modified example of Embodiment 3 is a liquid-winding type, the heat source thereof also remains the liquid-heating type.

[0079] Note that, as shown in Fig. 9, the flow of the control signal, when the controller 18 controls the second expansion device 8, the circulation heat exchanger 11 and the pump 12 based on the detected values of the refrigerant detected by the pressure sensor 13 and the temperature sensor 14, is shown.

Embodiment 4

[0080] In a vapor compression refrigeration cycle 500 showing Embodiment 4 of the invention, since the system configuration, the P-H diagram and the control flow are substantially the same as those of Embodiment 1, description thereof is omitted, and different points from Embodiment 1 will be mainly described with reference to Fig. 10.

[0081] Embodiment 4 adds an outflow path for gas refrigerant so that, of the refrigerant in the state of gas-liquid two-phase flow accumulated in the liquid receiver 4, a gas portion in a gas phase state that has become heated vapor (gas refrigerant) flows into an inlet side of the second expansion device 8 from an upper portion of the liquid receiver 4, to thereby improve the heat exchange efficiency in the inside heat exchanger 5, which is a single heating unit.

[0082] In regard to the configuration of Embodiment 4, a difference in configuration from Embodiment 1 to Embodiment 3 is the point that the outflow path for the gas refrigerant is newly and additionally provided to the upper portion of the liquid receiver 4. One end of the refrigerant pipe is in contact with the proximity of a liquid level in the upper portion of the liquid receiver 4, and the other end is connected to the inlet side of the second expansion device 8. Then, of the refrigerant in the state of gas-liquid two-phase flow accumulated in the liquid receiver 4, the gas portion in a gas phase state that has become heated vapor flows into the inlet side of the second expansion device 8, and then flows into the entrance of the upper side of the inside heat exchanger 5, via the second expansion device 8, whereas, the liquid portion flows into the entrance of the lower side of the inside heat exchanger 5. Thereafter, inside the inside heat exchanger 5, heat exchange is carried out between the refrigerant flowing through the auxirially refrigerant pipe and the refrigerant flowing through the primary refrigerant pipe.

[0083] Therefore, in the inside heat exchanger 5, it becomes easy to add the degree of superheat SH to the HFO refrigerant flowing through the auxirially refrigerant pipe communicating to the injection port 16, and accordingly, as compared to Embodiment 1, it is possible to prevent decrease in the discharge temperature at the outlet side of the compressor, and to realize further efficient refrigeration cycle for water heating and warming. In Embodiment 4, the states h and j are shifted to the high enthalpy region side (the right side) of the P-H diagram in Embodiment 1 to Embodiment 3.

[0084] As described above, in the present invention, even when the HFO refrigerant apt to have low discharge temperature is used, the temperature of the refrigerant subjected to intermediate injection into the compressor is maintained at a proper superheat state, and thereby it is possible to prevent decrease in the discharge temperature at the outlet of the compressor, and eventually, to improve the operating efficiency (COP) of the refrigeration cycle.

[0085] By the way, in the above description, an example was used, in which a single condenser 3 and a single evaporator 7 were connected; however, plural condensers or evaporators may be connected.

[0086] Moreover, in the above description, description was given with an example using operation of a heater; however, it goes without saying that, even in the case in which this is replaced with operation of cooling, the similar effect can be obtained. Note that the present system is merely an example, and the present invention is not limited to this system configuration.

[0087] Moreover, the vapor compression refrigeration cycle of the present invention may be adopted not only to the field of using water heating, but also to other fields.

[0088] It goes without saying that, within a scope not deviating from the gist of this invention, techniques disclosed in the respective embodiments may be appropriately combined.

Reference Signs List

[0089] 1 compressor 2 four-way valve 3 condenser4 liquid receiver 5 inside heat exchanger 6 first expansion device 7 evaporator8 second expansion device 9 injection circuit 10 air-to-refrigerant heat exchanger 11 circulation heat exchanger 12 pump 13 pressure sensor 14 temperature sensor 15 outside-air temperature sensor 16 injection port 17 temperature sensor 18 controller 100, 200, 300, 400, 500 vapor compression refrigeration cycle

Claims

1. A vapor compression refrigeration cycle using HFO as refrigerant, the vapor compression refrigeration cycle comprising:

a compressor having an injection port for injecting refrigerant circulating in the vapor compression refrigeration cycle;

a pressure sensor provided adjacent to the injection port and configured to measure a pressure;

a temperature sensor provided adjacent to the injection port and configured to measure a temperature;

an expansion device configured to open and close to decompress the refrigerant; and

a controller configured to control an opening degree of the expansion device based on the pressure and the temperature obtained by the pressure sensor and the temperature sensor, the controller being configured to control the expansion device to achieve a refrigeration cycle of the vapor compression refrigeration cycle represented by a P-H diagram in which a portion of the diagram representing a compression proc-

ess is positioned outside a saturation curve composed of a saturated liquid line and a saturated vapor line, and positioned below a critical pressure, and positioned more to a side of higher enthalpy than the saturated vapor line.

2. The vapor compression refrigeration cycle of claim 1, further comprising an air-to-refrigerant heat exchanger provided adjacent to a refrigerant pipe communicating to the injection port, wherein the refrigerant injected to the injection port is heated by using the air-to-refrigerant heat exchanger.

3. The vapor compression refrigeration cycle of claim 1, further comprising a circulation heat exchanger provided adjacent to a refrigerant pipe communicating to the injection port, wherein the refrigerant injected to the injection port is heated by using the circulation heat exchanger.

4. The vapor compression refrigeration cycle of any of claims 1 to 3, further comprising a liquid receiver connected to an inlet side of the expansion device, and an outflow path for gas refrigerant from the liquid receiver to cause a gas portion in a gas phase state, of the refrigerant in a gas-liquid two-phase state, accumulated in the liquid receiver, the gas portion being heated vapor, to flow into the inlet side of the expansion device.

5. The vapor compression refrigeration cycle of any of claims 1 to 4, wherein the controller is configured to control the opening degree of the expansion device to cause the refrigerant injected to the injection port to be heated vapor having a degree of superheat of 20 degrees C or more, and a pressure higher than a low pressure at the compressor by a difference of no less than 0.35 times a difference between a high pressure and the low pressure in a compression process of the compressor.

FIG. 1

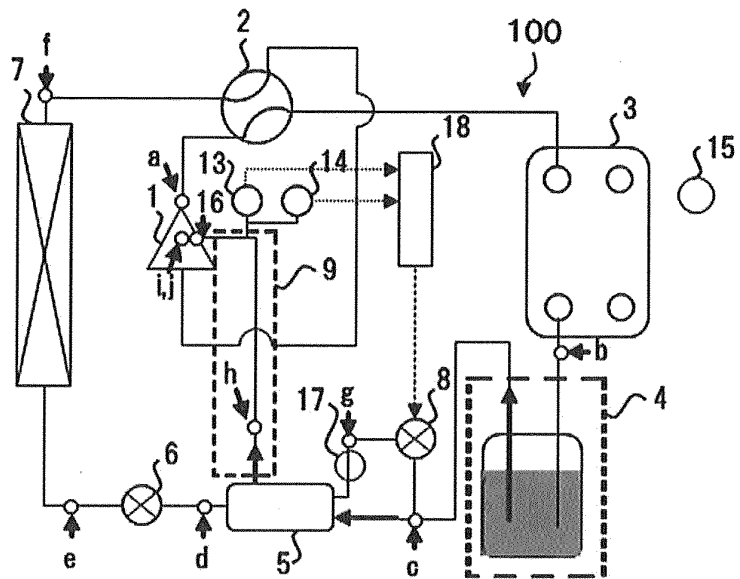


FIG. 2

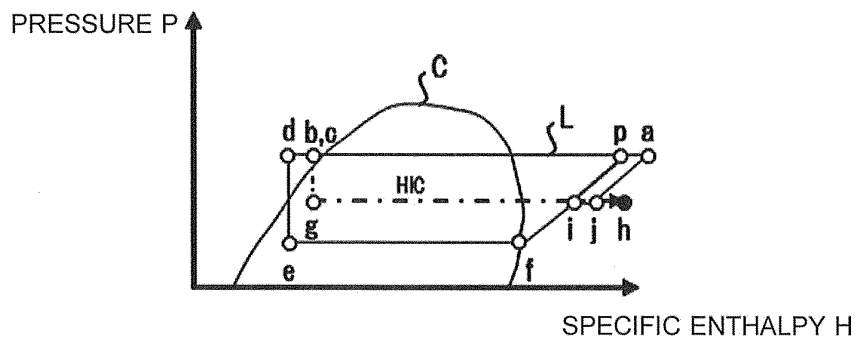


FIG. 3

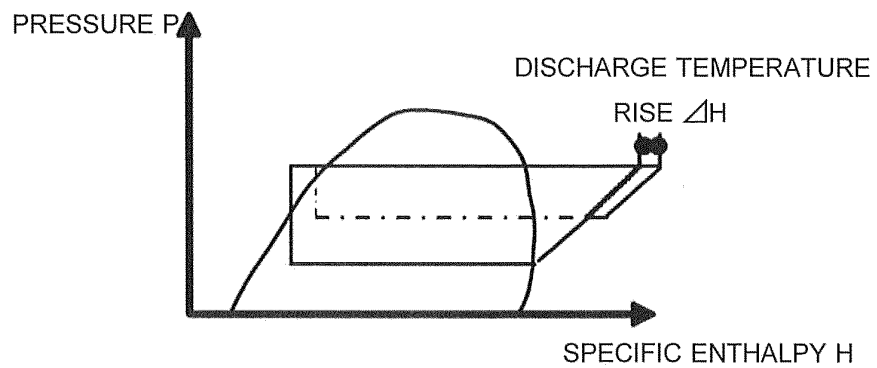


FIG. 4

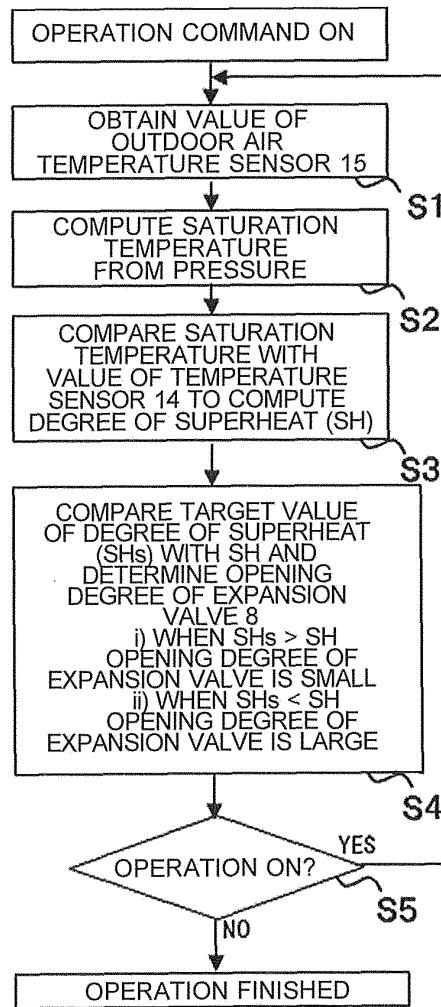


FIG. 5

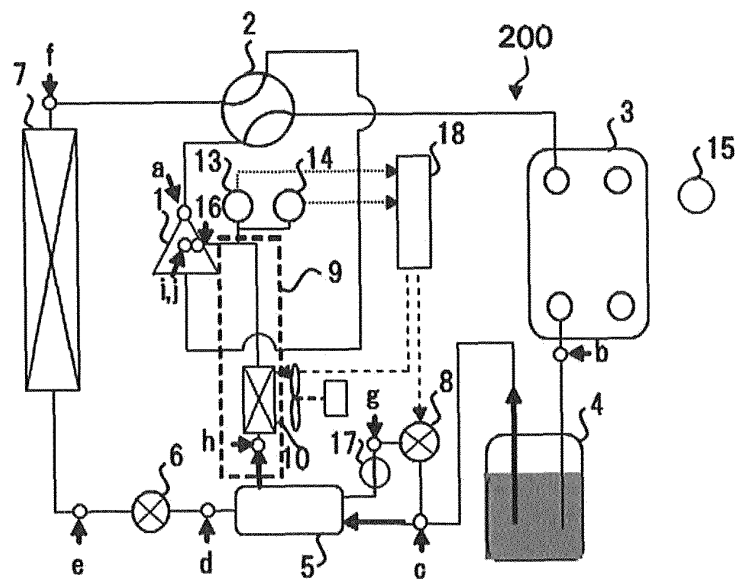


FIG. 6

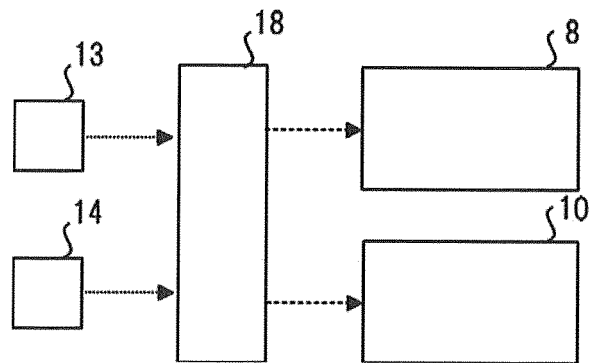


FIG. 7

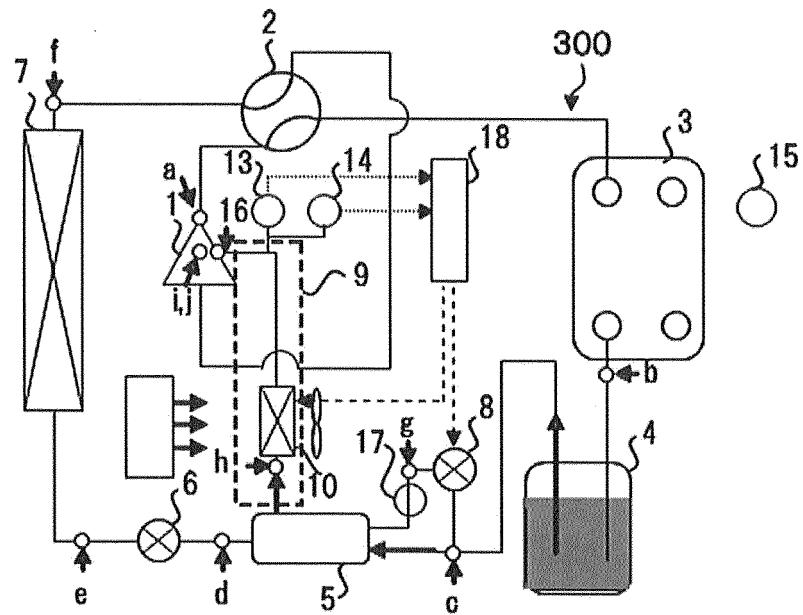


FIG. 8

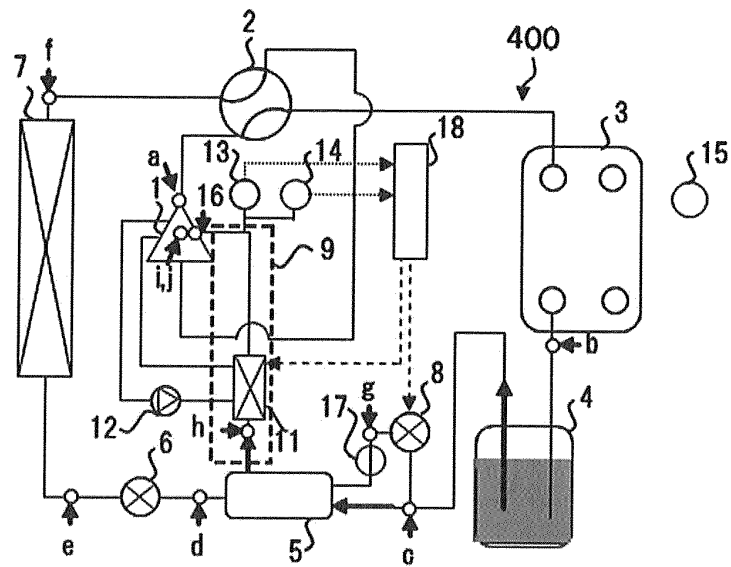


FIG. 9

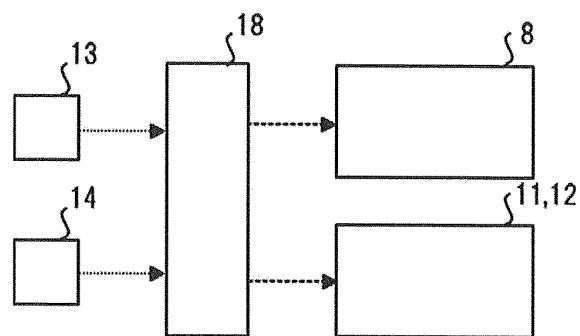


FIG. 10

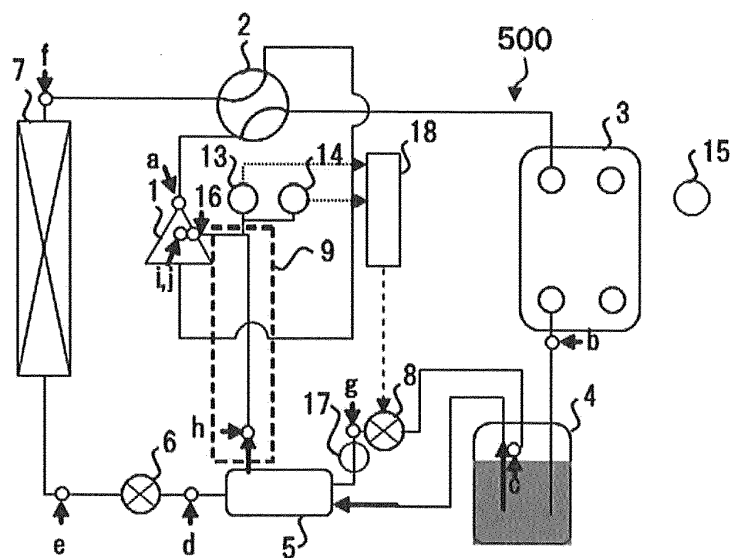


FIG. 11

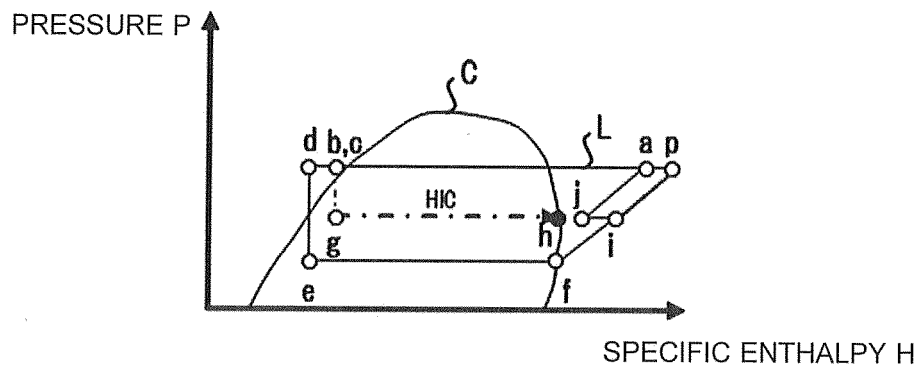


FIG. 12

SUCTION SH [°C]	20.1	25.41	32.93
DISCHARGE TEMPERATURE [°C]	60.7	62.6	64.5
HEATING AMOUNT [kcal]	22.5	22.5	22.5
C O P	3.433	3.456	3.474

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/002558

A. CLASSIFICATION OF SUBJECT MATTER

F25B1/00(2006.01)i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F25B1/00

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

Jitsuyo Shinan Koho 1922-1996 Jitsuyo Shinan Toroku Koho 1996-2014

Kokai Jitsuyo Shinan Koho 1971-2014 Toroku Jitsuyo Shinan Koho 1994-2014

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 2010-156536 A (Daikin Industries, Ltd.), 15 July 2010 (15.07.2010), paragraphs [0053], [0081] to [0084], [0091] to [0099]; fig. 1 to 4 & US 2011/0232325 A1 & EP 2357427 A1 & WO 2010/064427 A1 & CN 102227599 A	1-5
Y	JP 7-120076 A (Mitsubishi Electric Corp.), 12 May 1995 (12.05.1995), paragraphs [0029] to [0030]; fig. 7 (Family: none)	1-5
Y	JP 2006-23002 A (Mitsubishi Electric Corp.), 26 January 2006 (26.01.2006), claim 3; paragraphs [0021], [0036] to [0039]; fig. 4, 9 (Family: none)	2-5

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

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Date of the actual completion of the international search

15 July, 2014 (15.07.14)

Date of mailing of the international search report

22 July, 2014 (22.07.14)

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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/002558

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
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A	JP 55-85853 A (Tokyo Shibaura Electric Co., Ltd.), 28 June 1980 (28.06.1980), page 5, upper left column, lines 1 to 18; fig. 1, 15 & US 4326868 A & GB 2037965 A	1-5
A	JP 2009-270822 A (Mitsubishi Electric Corp.), 19 November 2009 (19.11.2009), paragraph [0012]; fig. 1 (Family: none)	1-5

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