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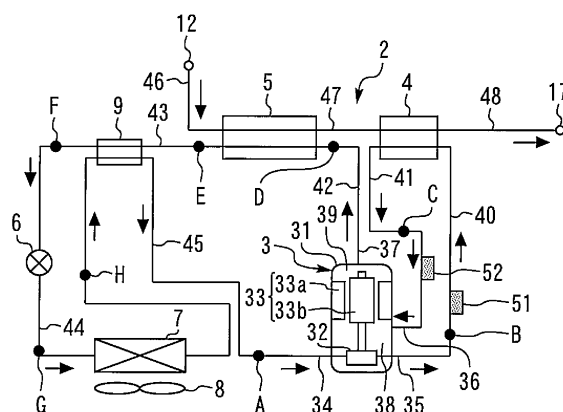
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(54) **HEAT PUMP SYSTEM**

(57) The present invention has an object to provide a heat pump system that prevents accumulation of a liquid refrigerant inside a sealed container in a low-heating operation. The heat pump system of the present invention includes a compressor having a first intake passage for leading a low-pressure refrigerant to a compressing element in the sealed container, a first discharge passage for discharging a high-pressure refrigerant compressed in a compressing element, a second intake passage for guiding the high-pressure refrigerant that has exchanged heat after being discharged from the first discharge passage to the internal space of the sealed container without compression, and a second discharge passage for discharging the high-pressure refrigerant in the internal space of the sealed container, a first heat exchanger for heating an object fluid by heat of the high-pressure refrigerant discharged from the first discharge passage, a second heat exchanger for heating the object fluid by heat of the high-pressure refrigerant discharged from the second discharge passage, and control means for performing a high-heating operation and the low-heating operation with a heating power smaller than that of the high-heating operation, and the control means controls, in the low-heating operation, so that a state of the refrigerant in the second intake passage is in a superheated gas state.

FIG. 2



Description

Field

5 **[0001]** The present invention relates to a heat pump system for heating an object fluid. Background
[0002] The following Patent Literature 1 discloses a water-heating cycle device as follows. The water-heating cycle device includes a compressor, a gas cooler, an expansion valve, and an evaporator. The compressor has a compressing element and an electric actuating element in a sealed container. The compressor includes a suction pipe directly leading a low-pressure refrigerant into the compressing element, a discharge pipe for discharging a high-pressure refrigerant compressed in the compressing element to the outside of the sealed container without releasing it into the sealed container, a refrigerant re-introducing pipe for re-introducing the refrigerant after being discharged from the discharge pipe and heat-exchanged into the sealed container, and a refrigerant re-discharge pipe for discharging the refrigerant led into the sealed container by the refrigerant re-introducing pipe and having passed through the electric actuating element to the outside of the sealed container. The gas cooler allows heat exchange therein between a water pipe through which the water for hot-water supply flows and a refrigerant pipe through which the compressed refrigerant flows for raising the temperature of water in the water pipe by the refrigerant in the refrigerant pipe. In the refrigerant pipe, the refrigerant pipe on a high temperature side connected to the discharge pipe exchanges heat with an outlet side of the water pipe of the gas cooler and the refrigerant pipe on a low temperature side connected to the refrigerant re-discharge pipe exchanges heat with an inlet side of the water pipe of the gas cooler.

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20 **[0003]** The following Patent Literature 2 discloses a heat-pump water heater as follows. The heat-pump water heater has a heat pump cycle in which a compressor, a water-refrigerant heat exchanger, an expansion valve, and an evaporator are connected annularly. The heat-pump water heater has a hot-water storing operation mode and a hot-water filling operation mode. In the hot-water filling operation mode, the heat pump cycle is operated, and water supplied from a hot water tank and water flowing out of a water-refrigerant heat exchanger of the heat pump cycle are mixed by a hot-water filling mixing valve, supplies the mixed water to a bathtub, and sets a hot-water temperature lower than that in the hot-water storing operation mode.

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Citation List

Patent Literature

[0004]

[PTL 1] JP 2006-132427 A

[PTL 2] JP 2011-21828 A

Summary

Technical Problem

40 **[0005]** When the device in Patent Literature 1 is operated in the hot-water filling operation mode as in Patent Literature 2, a discharge pressure of the compressor falls below a critical point, and the refrigerant is likely to be condensed to a gas-liquid two-phase state in the high-temperature side refrigerant pipe and the gas-liquid two-phase refrigerant flows into the sealed container. The discharge pipe is mounted on an upper part of the sealed container. If the gas-liquid two-phase refrigerant flows into the sealed container, gas-liquid separation occurs in the sealed container, and only a gas refrigerant flows out of the discharge pipe. As a result, a liquid refrigerant is accumulated in the sealed container, and the entire refrigerating cycles runs short of the refrigerant. In addition, if the liquid refrigerant collecting in the sealed container is heated by heat of the compressing element or the electric actuating element, evaporation of the refrigerant mixed with refrigerator oil causes the refrigerator oil to bubble, and the refrigerator oil flows out of the discharge pipe with the refrigerant. As a result, there is a concern that refrigerator oil in the sealed container runs short.

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50 **[0006]** The present invention has been made in order to solve the aforementioned problem, and an object of the present invention is to provide a heat pump system that prevents accumulation of a liquid refrigerant inside the sealed container in a low-heating operation.

Solution to Problem

55 **[0007]** A heat pump system of the invention includes: a compressor including a sealed container, a compressing element provided inside the sealed container, a first intake passage configured to lead low-pressure refrigerant sucked

from an outside of the sealed container to the compressing element without releasing the low-pressure refrigerant into an internal space of the sealed container, a first discharge passage configured to discharge high-pressure refrigerant compressed in the compressing element to the outside of the sealed container without releasing the high-pressure refrigerant to the internal space of the sealed container, a second intake passage configured to guide high-pressure refrigerant, the high-pressure refrigerant having exchanged heat after being discharged from the first discharge passage, to the internal space of the sealed container without compression, and a second discharge passage configured to discharge high-pressure refrigerant in the internal space of the sealed container to the outside of the sealed container without compression; a first heat exchanger configured to heat an object fluid by heat of the high-pressure refrigerant discharged from the first discharge passage; a second heat exchanger configured to heat the object fluid by the heat of the high-pressure refrigerant discharged from the second discharge passage; an expansion unit configured to expand high-pressure refrigerant having passed through the second heat exchanger to low-pressure refrigerant; an evaporator configured to evaporate the low-pressure refrigerant having passed through the expansion unit; and control means configured to perform a high-heating operation and a low-heating operation. The low-heating operation has a smaller total heating amount in the first heat exchanger and the second heat exchanger than the high-heating operation. The control means is configured to control, in the low-heating operation, so that a state of the refrigerant in the second intake passage is in a superheated gas state.

Advantageous Effect of Invention

[0008] According to the heat pump system of the present invention, accumulation of the liquid refrigerant inside the sealed container in the low-heating operation can be prevented.

Brief Description of Drawings

[0009]

Fig. 1 is a configuration diagram illustrating a heat pump system according to an embodiment 1 of the present invention.

Fig. 2 is a configuration diagram illustrating a heat pump unit included in the heat pump system according to the embodiment 1 of the present invention.

Fig. 3 is a pressure-enthalpy diagram of a high-heating operation in the heat pump system according to the embodiment 1 of the present invention.

Fig. 4 is a diagram illustrating an example of temperature changes of refrigerant and water in a first heat exchanger and a second heat exchanger in the high-heating operation in the heat pump system according to the embodiment 1 of the present invention.

Fig. 5 is a flowchart illustrating a control operation of a controller in the high-heating operation in the heat pump system according to the embodiment 1 of the present invention.

Fig. 6 is a pressure-enthalpy diagram of a low-heating operation in the heat pump system according to the embodiment 1 of the present invention.

Fig. 7 is a diagram illustrating an example of temperature changes of refrigerant and water in the first heat exchanger and the second heat exchanger in the low-heating operation in the heat pump system according to the embodiment 1 of the present invention.

Fig. 8 is a flowchart illustrating a control operation of the controller in the low-heating operation in the heat pump system according to the embodiment 1 of the present invention.

Fig. 9 is a flowchart illustrating a control operation of a controller in a low-heating operation in a heat pump system according to an embodiment 2 of the present invention.

Fig. 10 is a flowchart illustrating a control operation of a controller in a low-heating operation in a heat pump system according to an embodiment 3 of the present invention.

Fig. 11 is a configuration diagram illustrating a heat pump unit included in the heat pump system according to an embodiment 4 of the present invention.

Fig. 12 is a flowchart illustrating a control operation of a controller in a low-heating operation in the heat pump system according to the embodiment 4 of the present invention.

Description of Embodiments

[0010] Embodiments of the present invention will be described below by referring to the attached drawings. Elements common in each of the figures are given the same reference numerals and duplicated explanation will be omitted. The present invention is assumed to include any combination of each of the embodiments described below.

Embodiment 1.

[0011] Fig. 1 is a configuration diagram illustrating a heat pump system according to an embodiment 1 of the present invention. As illustrated in Fig. 1, the heat pump system 1 of this embodiment 1 has a heat pump unit 2 for heating water, a hot water tank 10, and a controller 50. The hot water tank 10 stores water by forming temperature stratification with an upper side at a high temperature and a lower side at a low temperature. A lower part of the hot water tank 10 and an inlet 12 of the heat pump unit 2 are connected to each other via an inlet conduit 11. In the middle of the inlet conduit 11, a pump 13 is installed. To an upper part of the hot water tank 10, one end of an upper conduit 14 is connected. The other end side of the upper conduit 14 branches into two parts, one of which is connected to a first inlet of a hot-water feeding mixing valve 15 and the other to a first inlet of a bath mixing valve 16, respectively. An outlet 17 of the heat pump unit 2 is connected to a middle position of the upper conduit 14 via an outlet conduit 18. Details of the heat pump unit 2 will be described later. In this embodiment 1, a case in which an object fluid to be heated is water will be described, but an object fluid in the present invention may be fluids other than water such as brine, an antifreezing liquid and the like.

[0012] To the lower part of the hot water tank 10, a feed-water pipe 19 for supplying water from a water source such as waterworks system is connected. In the middle of the feed-water pipe 19, a pressure-reducing valve 20 for reducing a water-source pressure to a predetermined pressure is installed. Inflow of the water from the feed-water pipe 19 keeps the hot water tank 10 in a full state at all times. From the feed-water pipe 19 between the hot water tank 10 and the pressure reducing valve 20, a feed-water pipe 21 branches. A downstream side of the feed-water pipe 21 branches into two parts, which are connected to a second inlet of the hot-water feeding mixing valve 15 and a second inlet of the bath mixing valve 16, respectively. An outlet of the hot-water feeding mixing valve 15 is connected to a hot-water tap 23 via a hot-water pipe 22. In the hot-water pipe 22, hot-water flowrate detecting means 24 and a hot-water temperature sensor 25 are installed. An outlet of the bath mixing valve 16 is connected to a bathtub 27 via a bath pipe 26. In the bath pipe 26, an opening/closing valve 28 and a bath temperature sensor 29 are installed. In the outlet conduit 18 in the vicinity of the outlet 17 of the heat pump unit 2, a heat-pump outlet temperature sensor 30 for detecting a heat-pump outlet temperature which is the temperature of water coming out of the heat pump unit 2 is installed. The heat-pump outlet temperature sensor 30 may be provided in a pipe (a water channel 48 which will be described later) inside the heat pump unit 2. In the following explanation, the temperature of the water flowing into the heat pump unit 2 will be referred to as a "heat-pump inlet temperature".

[0013] The controller 50 is control means composed of a microcomputer or the like, for example. The controller 50 includes a storage unit including an ROM (Read Only Memory), an RAM (Random Access Memory), a non-volatile memory and the like, a central processing unit (CPU) for executing calculation processing on the basis of a program stored in the storage unit, an input/output port for inputting/outputting an external signal to the CPU, a timer for counting time and the like. The controller 50 is electrically connected to various actuators and sensors included in the heat pump system 1, respectively. The controller 50 is connected to an operation unit 60, capable of mutual communication. A user can set a hot water temperature, a bathtub hot-water amount, a bathtub hot-water temperature and the like and preset time to fill hot water in the bathtub and the like using a timer by operating the operation unit 60. The controller 50 controls an operation of the heat pump system 1 by controlling an operation of each of the actuators in accordance with the program stored in the storage unit on the basis of information detected by each of the sensors and instruction information from the operation unit 60.

[0014] Subsequently, the hot-water storing operation will be described. The hot-water storing operation is an operation of increasing a hot-water storage amount and a heat accumulation amount in the hot water tank 10. In the hot-water storing operation, the controller 50 operates the heat pump unit 2 and the pump 13. In the hot-water storing operation, low-temperature water led out of the lower part of the hot water tank 10 through the pump 13 is fed to the heat pump unit 2 through the inlet conduit 11, is heated in the heat pump unit 2 and made into high-temperature water. This high-temperature water flows through the outlet conduit 18 and the upper conduit 14 and flows into the upper part of the hot water tank 10. By means of this hot-water storing operation, the high-temperature water accumulates in the hot water tank 10 from the upper side.

[0015] In the hot-water storing operation, the controller 50 controls the heat-pump outlet temperature detected by the heat-pump outlet temperature sensor 30 to be in the range of approximately 65 to 90°C, for example. By controlling the pump 13 so that a flowrate of the water flowing through the heat pump unit 2 rises, the heat-pump outlet temperature lowers. By controlling the pump 13 to lower the flowrate of the water flowing through the heat pump unit 2, the heat-pump outlet temperature rises. In the hot-water storing operation, the controller 50 controls the heat pump unit 2 to perform a high-heating operation. The high-heating operation of the heat pump unit 2 is an operation in which a heating power of the heat pump unit 2 becomes a predetermined rated power.

[0016] Subsequently, a hot-water feeding operation will be described. The hot-water feeding is an operation for feeding hot water to a hot-water tap 23. When the user opens the hot-water tap 23, the water from the feed-water pipe 19 flows into a lower part inside the hot water tank 10 by the water-source pressure, and the high-temperature water in the upper part in the hot water tank 10 flows out into the upper conduit 14. In the hot-water feeding mixing valve 15, the low-

temperature water supplied from the feed-water pipe 21 and the high-temperature water supplied from the hot water tank 10 through the upper conduit 14 are mixed. This mixed water is emitted to the outside from the hot-water tap 23 through the hot-water pipe 22. At this time, when flow of the mixed water is detected by the hot-water flowrate detecting means 24, the controller 50 controls a mixing ratio of the hot-water feeding mixing valve 15 so that a hot water temperature detected by the hot-water temperature sensor 25 is brought to a hot-water temperature set value set by the operation unit 60 by the user in advance.

[0017] Subsequently, a hot-water filling operation will be described. The hot-water filling operation is an operation for accumulating hot water in the bathtub 27. When the user performs a start operation of the hot-water filling operation by the operation unit 60 or when the time set by the timer has come, the hot-water filling operation is started. In the hot-water filling operation, the controller 50 operates the heat pump unit 2 and the pump 13 and brings the opening/closing valve 28 into an open state. When the water from the feed-water pipe 19 flows into the lower part of the hot water tank 10 by the water-source pressure, the high-temperature water in the upper part of the hot water tank 10 flows out to the upper conduit 14. The low-temperature water led out through the pump 13 from the lower part of the hot water tank 10 is fed to the heat pump unit 2 through the inlet conduit 11 and is heated in the heat pump unit 2. The water heated in the heat pump unit 2 flows into the upper conduit 14 through the outlet conduit 18. The high-temperature water supplied from the hot water tank 10 and the water heated in the heat pump unit 2 merge with each other in the upper conduit 14 and are supplied to the bath mixing valve 16. In the bath mixing valve 16, the low-temperature water supplied from the feed-water pipe 21 and the hot water supplied through the upper conduit 14 are mixed. This mixed water passes through the bath pipe 26 and the opening/closing valve 28 and is discharged into the bathtub 27. At this time, the controller 50 controls the mixing ratio of the bath mixing valve 16 so that the hot water temperature detected by the bath temperature sensor 29 is brought to a bathtub temperature set value set by the user in the operation unit 60 in advance.

[0018] As described above, in the hot-water filling operation of this embodiment 1, the hot water is supplied to the bathtub 27 by using not only the high-temperature water stored in the hot water tank 10 but also the water heated in the heat pump unit 2 supplementarily. In the hot-water filling operation, the controller 50 controls the heat-pump outlet temperature detected by the heat-pump outlet temperature sensor 30 to be lower than the bathtub temperature set value. In the hot-water filling operation, the controller 50 controls the heat pump unit 2 to perform a low-heating operation. The low-heating operation of the heat pump unit 2 is an operation in which the heating power of the heat pump unit 2 is lower than the high-heating operation.

[0019] Here, a hot-water filling power will be described. The hot-water filling power is heat energy per unit time required when the bathtub is to be filled at a target bathtub temperature under predetermined conditions of a bathtub capacity, a feed-water temperature, and a hot-water filling flowrate. Assuming that the bathtub capacity is 180 L, the feed-water temperature is 9°C, the target bathtub temperature is 45°C, and the hot-water filling flowrate is at 10 to 20 L/min, the standard hot-water filling power is 25 to 50 kW, for example. The rated heating power of the heat pump unit 2 is approximately 4.5 to 9 kW, for example. The standard hot-water filling power cannot be satisfied only by the heating power of the heat pump unit 2. In order to satisfy the standard hot-water filling power, the high-temperature water stored in the hot water tank 10 needs to be used.

[0020] The heat pump unit 2 has a characteristic: the lower the heating power it is operated with, the higher its COP (Coefficient Of Performance) becomes. In addition, the heat pump unit 2 has a characteristic that the lower the heat-pump outlet temperature falls, the higher the COP rises. Thus, the COP of the low-heating operation is higher than the COP of the high-heating operation. Assuming that the COP of the low-heating operation is C1, the COP of the high-heating operation is C2, hot-water filling COP which is a substantial COP of the hot-water filling operation is C3, and a ratio of the heating power of the low-heating operation of the heat pump unit 2 to the hot-water filling power is R_{hp}, the hot-water filling COP is expressed by the following equation:

$$C3 = C1 \times R_{hp} + C2 \times (1 - R_{hp}) \dots (1)$$

[0021] As described above, the high-heating operation is performed in the hot-water storing operation. That is, the high-temperature water in the hot water tank 10 is generated in the high-heating operation. Thus, if the heat pump unit 2 is not operated and the hot water is supplied to the bathtub 27 by using only the high-temperature water stored in the hot water tank 10 during the hot-water filling operation, the hot-water filling COP equals to the COP of the high-heating operation. That is, assuming that it is R_{hp} = 0 in the aforementioned equation (1), it is C3 = C2. On the other hand, according to the hot-water filling operation of this embodiment 1, by performing the low-heating operation and by supplementarily using the water heated by the heat-pump unit 2, the hot-water filling COP can be made higher than the COP of the high-heating operation. That is, since it is R_{hp} > 0 in the aforementioned equation (1), it is C3 > C2. As described above, in this embodiment 1, by performing the hot-water filling operation involving the low-heating operation, the hot-water filling COP can be improved while the aforementioned hot-water filling power is ensured. In addition, by

operating the heat pump unit 2 in the hot-water filling operation, a part of the heat energy required for hot-water filling can be covered. Thus, the heat accumulated amount required for the hot water tank 10 can be reduced. Thus, a heat-dissipation loss from the hot water tank 10 can be reduced, and energy efficiency as a whole can be further improved. Since the heat accumulated amount required for the hot water tank 10 can be reduced, a capacity of the hot water tank 10 can be reduced, and a size of the hot water tank 10 can be reduced.

[0022] Fig. 2 is a configuration diagram illustrating the heat pump unit 2 included in the heat pump system 1 according to the embodiment 1 of the present invention. As illustrated in Fig. 2, the heat pump unit 2 includes a refrigerant circuit connecting a compressor 3, a first heat exchanger 4, a second heat exchanger 5, an expansion valve 6, and an evaporator 7 by refrigerant pipes. The first heat exchanger 4 and the second heat exchanger 5 are heat exchangers for heating water by heat of the refrigerant. The evaporator 7 is composed of an air-refrigerant heat exchanger that exchanges heat between the air and the refrigerant. The heat pump unit 2 further includes an air blower 8 for blowing air to the evaporator 7 and a high-low pressure heat exchanger 9 that exchanges heat between a high-pressure refrigerant and a low-pressure refrigerant. In this embodiment 1, carbon dioxide is used as the refrigerant. The evaporator 7 in the present invention is not limited to that exchanging heat between the air and the refrigerant but may include those exchanging heat between the refrigerant and underground water, solar-heat hot water or the like, for example.

[0023] The compressor 3 includes a sealed container 31, a compressing element 32 and an electric actuating element 33 provided inside the sealed container 31, a first intake passage 34, a first discharge passage 35, a second intake passage 36, and a second discharge passage 37. The compressing element 32 is arranged below the electric actuating element 33. An internal space 38 between the compressing element 32 and the electric actuating element 33 and an internal space 39 above the electric actuating element 33 are provided inside the sealed container 31. The first intake passage 34 does not emit the low-pressure refrigerant sucked into the compressor 3 into the internal spaces 38 or 39 of the sealed container 31 but directly leads this low-pressure refrigerant to the compressing element 32. The compressing element 32 compresses the low-pressure refrigerant to convert it into high-pressure refrigerant. The compressing element 32 is driven by the electric actuating element 33. The electric actuating element 33 is a motor having a stator 33a and a rotor 33b. The compressing element 32 discharges the compressed high-pressure refrigerant into the first discharge passage 35. The first discharge passage 35 does not emit this high-pressure refrigerant into the internal space 38 or 39 of the sealed container 31 but directly discharges this high-pressure refrigerant to the outside of the sealed container 31. The high-pressure refrigerant discharged from the first discharge passage 35 passes through a refrigerant channel 40 and flows into the first heat exchanger 4. The high-pressure refrigerant cooled by water in the first heat exchanger 4 passes through a refrigerant channel 41 and the second intake passage 36 and is sucked into the compressor 3 again.

[0024] An outlet of the second intake passage 36 is located in the internal space 38 between the electric actuating element 33 and the compressing element 32. The second intake passage 36 does not compress the high-pressure refrigerant sucked into the compressor 3 again but discharges this high-pressure refrigerant into the internal space 38 between the electric actuating element 33 and the compressing element 32. An inlet of the second discharge passage 37 is located in the internal space 39 above the electric actuating element 33. The high-pressure refrigerant emitted from the outlet of the second intake passage 36 into the internal space 38 between the electric actuating element 33 and the compressing element 32 passes through a gap between the rotor 33b and the stator 33a of the electric actuating element 33 or the like and reaches the internal space 39 above the electric actuating element 33. At this time, the electric actuating element 33 whose temperature has risen high is cooled by the high-pressure refrigerant, and the high-pressure refrigerant is heated by the heat of the electric actuating element 33. The second discharge passage 37 does not compress the high-pressure refrigerant in the internal space 39 above the electric actuating element 33 but discharges this high-pressure refrigerant to the outside of the sealed container 31.

[0025] The high-pressure refrigerant discharged from the second discharge passage 37 passes through a refrigerant channel 42 and flows into the second heat exchanger 5. The high-pressure refrigerant cooled by water in the second heat exchanger 5 passes through a refrigerant channel 43 and reaches the expansion valve 6. The expansion valve 6 is an expansion unit for expanding the high-pressure refrigerant to convert it into low-pressure refrigerant. The low-pressure refrigerant expanded by the expansion valve 6 passes through a refrigerant channel 44 and flows into the evaporator 7. In the evaporator 7, the low-pressure refrigerant is heated by heat exchange with the outside air led by the air blower 8 and is evaporated. The low-pressure refrigerant having passed through the evaporator 7 passes through a refrigerant channel 45 and reaches the first intake passage 34 of the compressor 3 and is sucked into the compressor 3. The high-low pressure heat exchanger 9 exchanges heat between the high-pressure refrigerant in the middle of the refrigerant channel 43 and the low-pressure refrigerant in the middle of the refrigerant channel 45.

[0026] In the following explanation, a pressure of the refrigerant discharged from the compressing element 32 will be referred to as a "compressing element discharge pressure", a pressure of the refrigerant sucked into the compressing element 32 as a "compressing element suction pressure", the temperature of the refrigerant discharged from the compressing element 32 as a "compressing element discharge temperature", and the temperature of the refrigerant sucked into the compressing element 32 as a "compressing element suction temperature". The pressure of the high-pressure refrigerant discharged from the first discharge passage 35 is equal to the compressing element discharge pressure. The

high-pressure refrigerant discharged from the first discharge passage 35 has its pressure lowered by a pressure loss to the second intake passage 36 through the first heat exchange 4. Thus, the pressure of the high-pressure refrigerant in the internal space 38 of the sealed container 31 is slightly lower than the pressure of the high-pressure refrigerant discharged from the first discharge passage 35, that is, the compressing element discharge pressure.

[0027] The heat pump unit 2 further includes a water channel 46 for leading the water having flowed in from the inlet 12 to the second heat exchanger 5, a water channel 47 for leading the water having passed through the second heat exchanger 5 to the first heat exchanger 4, and a water channel 48 for leading the water having passed through the first heat exchanger 4 to the outlet 17. In the heating operation, the water having flowed in from the inlet 12 passes through the water channel 46 and flows into the second heat exchanger 5 and is heated by the heat of the refrigerant in the second heat exchanger 5. The water heated in the second heat exchanger 5 flows into the first heat exchanger 4 and is further heated by the heat of the refrigerant in the first heat exchanger 4. The water further heated in the first heat exchanger 4 reaches the outlet 17 through the water channel 48 and flows to the outlet conduit 18.

[0028] In the first discharge passage 35 or the refrigerant channel 40, a discharge temperature sensor 51 for detecting the compressing element discharge temperature is provided. In the second intake passage 36 or the refrigerant channel 41, a refrigerant temperature sensor 52 for detecting a refrigerant temperature in the second intake passage 36 is provided.

[0029] Subsequently, the high-heating operation in the hot-water storing operation will be further described. As described above, in the high-heating operation, the controller 50 controls the temperature of the water coming out of the first heat exchanger 4, that is, the heat-pump outlet temperature to approximately 65 to 90°C, for example. Fig. 3 is a pressure-enthalpy diagram of the high-heating operation. Reference characters A to H in Fig. 3 correspond to reference characters A to H in Fig. 2. As illustrated in Fig. 3, the refrigerant is compressed in the compressing element 32 to a pressure exceeding a critical pressure (A → B). The high-pressure refrigerant in the super critical state is cooled by the first heat exchanger 4 (B → C). The state of the high-pressure refrigerant sucked into the internal space 38 of the sealed container 31 from the second intake passage 36 is C in Fig. 3. This high-pressure refrigerant is heated by the heat of the electric actuating element 33 while it reaches the internal space 39 (C → D). The state of the high-pressure refrigerant discharged from the second discharge passage 37 is D in Fig. 3. This high-pressure refrigerant is cooled in the second heat exchanger 5 (D → E). After that, the high-pressure refrigerant is further cooled in the high-low pressure heat exchanger 9 (E → F). The high-pressure refrigerant having passed through the high-low pressure heat exchanger 9 has its pressure reduced by the expansion valve 6 and is made into a low-pressure refrigerant (F → G). This low-pressure refrigerant is evaporated in the evaporator 7 (G → H). The low-pressure refrigerant evaporated in the evaporator 7 is heated in the high-low pressure heat exchanger 9 (H → A). The heat-pump outlet temperature of 65 to 90°C in such high-heating operation is sufficiently higher than the critical temperature of carbon dioxide which is the refrigerant. In the high-heating operation, the compressing element discharge pressure and the pressures of the refrigerant inside the first heat exchanger 4, the sealed container 31, and the second heat exchanger 5 are pressures exceeding the critical pressure.

[0030] Fig. 4 is a diagram illustrating an example of temperature changes of the refrigerant and the water in the first heat exchanger 4 and the second heat exchanger 5 in the high-heating operation. Reference characters B to E in Fig. 4 correspond to B to E in Figs. 2 and 3. The lateral axis in Fig. 4 expresses positions inside the first heat exchanger 4 and the second heat exchanger 5 by a ratio of stream lengths. That is, the lateral axis in Fig. 4 expresses a ratio of a water stream length from the water inlet of the second heat exchanger 5, assuming the entire length of the water streams of the first heat exchanger 4 and the second heat exchanger 5 is 1. Or the lateral axis in Fig. 4 expresses a ratio of the refrigerant stream length from the refrigerant outlet of the second heat exchanger 5, assuming the entire length of the refrigerant streams of the first heat exchanger 4 and the second heat exchanger 5 is 1. In the example illustrated in Fig. 4, the temperature of the water flowing into the second heat exchanger 5, that is, the heat-pump inlet temperature is approximately 9°C, and the temperature of the water flowing out of the first heat exchanger 4, that is the heat-pump outlet temperature is approximately 65°C. The temperature of the refrigerant flowing into the first heat exchanger 4, that is, the compressing element discharge temperature is approximately 85°C.

[0031] Fig. 5 is a flowchart illustrating the control operation of the controller 50 in the high-heating operation. In the high-heating operation, the controller 50 controls each of the actuators as follows. The controller 50 controls the compressor 3 so that the heating power of the heat pump unit 2 is brought to the rated power (Step S1). Here, the heating power is a water heating amount per time of the total of the first heat exchanger 4 and the second heat exchanger 5. The controller 50 can control the heating power by controlling the capacity of the compressor 3. The controller 50 can control the capacity of the compressor 3 by controlling a driving speed, a driving frequency and the like of the compressor 3. In addition, the controller 50 controls a water flowrate using the pump 13 so that the heat-pump outlet temperature detected by the heat-pump outlet temperature sensor 30 is brought to a predetermined heating temperature set value within a range of 65 to 90°C. In addition, the controller 50 controls an air blowing amount of the air blower 8 in accordance with required evaporation capacity. The evaporating capacity is a heat amount absorbed by the refrigerant from the air in the evaporator 7.

[0032] In the high-heating operation, the controller 50 controls the refrigerant flowrate using the expansion valve 6 so

that the compressing element discharge temperature matches a target value. The lower the refrigerant flowrate is made by decreasing an opening degree of the expansion valve 6, the higher the compressing element discharge temperature becomes. The compressing element discharge temperature can be detected by the discharge temperature sensor 51 provided at B in Figs. 2 and 6. The controller 50 stores a table defining a relation between parameters such as the heat-pump outlet temperature, the outside air temperature, the heating power and the like and the target value of the compressing element discharge temperature. The target value of the compressing element discharge temperature is determined so that the maximum COP can be obtained in accordance with the parameters such as the heat-pump outlet temperature, the outside air temperature, the heating power and the like. The controller 50 determines the target value of the compressing element discharge temperature on the basis of the parameters such as the heat-pump outlet temperature, the outside air temperature, the heating power and the like and the table. Then, the controller 50 determines whether or not the compressing element discharge temperature matches the target value (Step S2). If the compressing element discharge temperature matches the target value at Step S2, the controller 50 returns to Step S1. If the compressing element discharge temperature does not match the target value at Step S2, the controller 50 controls the refrigerant flowrate using the expansion valve 6 so that the compressing element discharge temperature matches the target value (Step S3). By executing the control as above, the COP during the high-heating operation can be made sufficiently high.

[0033] In the following explanation, degrees of superheat of the refrigerant coming out of the evaporator 7 will be referred to as an "evaporator outlet superheat degrees". The compressing element discharge temperature and the evaporator outlet superheat degrees have correlation. Thus, in the high-heating operation, the controller 50 may control the refrigerant flowrate using the expansion valve 6 so that the evaporator outlet superheat degrees matches the target value instead of the aforementioned Steps S2 and S3. The lower the refrigerant flowrate is made by decreasing the opening degree of the expansion valve 6, the larger the degrees of superheat of the outlet of the evaporator 7 becomes. The evaporator outlet superheat degrees can be detected as a temperature difference between two temperature sensors by providing the temperature sensors at G and H in Figs. 2 and 6, respectively, for example. In this case, the controller 50 stores a table defining a relation between the parameters such as the heat-pump outlet temperature, the outside air temperature, the heating power and the like and a target value of the evaporator outlet superheat degrees. The target value of the evaporator outlet superheat degrees is determined so that the maximum COP can be obtained in accordance with the parameters such as the heat-pump outlet temperature, the outside air temperature, the heating power and the like. The controller 50 determines the target value of the evaporator outlet superheat degrees on the basis of the parameters such as the heat-pump outlet temperature, the outside air temperature, the heating power and the like and the aforementioned table. Then, the controller 50 determines whether or not the evaporator outlet superheat degrees matches the target value instead of the aforementioned Step S2. If the evaporator outlet superheat degrees matches the target value, the controller 50 returns to Step S1. If the evaporator outlet superheat degrees does not match the target value, the controller 50 controls the refrigerant flowrate using the expansion valve 6 so that the evaporator outlet superheat degrees matches the target value instead of the aforementioned Step S3. By executing control as above, the COP during the high-heating operation can be made sufficiently high.

[0034] Subsequently, the low-heating operation in the hot-water filling operation will be further described. In the low-heating operation, the controller 50 controls the heat-pump outlet temperature to approximately 20 to 30°C, for example. Fig. 6 is a pressure-enthalpy diagram of the low-heating operation. Reference characters A to H in Fig. 6 correspond to reference characters A to H in Fig. 2. As illustrated in Fig. 6, the refrigerant is compressed in the compressing element 32 to a pressure not more than a critical pressure (A → B). The high-pressure refrigerant after being compressed in the compressing element 32 (B in Fig. 6) is in a superheated gas state. This high-pressure refrigerant in the superheated gas state is cooled by the first heat exchanger 4 (B → C). The high-pressure refrigerant (C in Fig. 6) cooled in the first heat exchanger 4 is also in the superheated gas state. This high-pressure refrigerant is heated by the heat of the electric actuating element 33 while it reaches the internal space 39 (C → D). The state of the high-pressure refrigerant discharged from the second discharge passage 37 is D in Fig. 6. This high-pressure refrigerant is condensed by being cooled in the second heat exchanger 5 and liquefied (D → E). After that, the high-pressure refrigerant is further cooled in the high-low pressure heat exchanger 9 (E → F). The high-pressure refrigerant having passed through the high-low pressure heat exchanger 9 has its pressure reduced by the expansion valve 6 and is made into a low-pressure refrigerant (F → G). This low-pressure refrigerant is evaporated in the evaporator 7 (G → H). The low-pressure refrigerant evaporated in the evaporator 7 is heated in the high-low pressure heat exchanger 9 (H → A). The heat-pump outlet temperature of 20 to 30°C in this low-heating operation is lower than the critical temperature of carbon dioxide which is the refrigerant. In the low-heating operation, the compressing element discharge pressure and the pressures of the refrigerant inside the first heat exchanger 4, the sealed container 31, and the second heat exchanger 5 are pressures not more than the critical pressure.

[0035] Fig. 7 is a diagram illustrating an example of the temperature changes of the refrigerant and water in the first heat exchanger 4 and the second heat exchanger 5 in the low-heating operation. Reference characters B to E in Fig. 7 correspond to B to E in Figs. 2 and 6. A meaning of the lateral axis in Fig. 7 is the same as the lateral axis in Fig. 4. In

the example illustrated in Fig. 7, the temperature of the water flowing into the second heat exchanger 5, that is, the heat-pump inlet temperature is approximately 9°C, and the heat-pump outlet temperature is approximately 25°C. The temperature of the refrigerant flowing into the first heat exchanger 4, that is, the compressing element discharge temperature is approximately 45°C. A condensation saturation temperature of the refrigerant in the second heat exchanger 5 is approximately 22°C. In the following explanation, the condensation saturation temperature of the refrigerant in the second heat exchanger 5 will be referred to simply as a "condensation saturation temperature". In addition, an evaporation saturation temperature of the refrigerant in the evaporator 7 will be referred to simply as an "evaporation saturation temperature".

[0036] Fig. 8 is a flowchart illustrating a control operation of the controller 50 in the low-heating operation. In the low-heating operation, the controller 50 controls each of the actuators as follows. The controller 50 controls the compressor 3 so that the heating power of the heat pump unit 2 is brought to a power lower than the heating power in the high-heating operation, that is, a power lower than the rated power (Step S11). At this Step S11, the controller 50 controls the capacity of the compressor 3 to be lower than in the high-heating operation. For example, the driving speed, the driving frequency and the like of the compressor 3 are made lower than those in the high-heating operation. The refrigerant flowrate in the low-heating operation is lower than the refrigerant flowrate in the high-heating operation. In addition, the controller 50 controls a water flowrate using the pump 13 so that the heat-pump outlet temperature detected by the heat-pump outlet temperature sensor 30 is brought to a predetermined heating temperature set value within a range of 20 to 30°C. The water flowrate in the low-heating operation is higher than the water flowrate in the high-heating operation. In addition, the controller 50 controls the air blowing amount of the air blower 8 in accordance with a required evaporation capacity.

[0037] In the low-heating operation, the controller 50 controls the refrigerant flowrate using the expansion valve 6 so that the state of the refrigerant flowing into the internal space 38 of the sealed container 31, that is, the state of the refrigerant in the second intake passage 36 becomes a superheated gas state. At this time, if the opening degree of the expansion valve 6 is decreased, the refrigerant flowrate lowers, and the degrees of superheat of the refrigerant in the second intake passage 36 increases. The degrees of superheat is a difference between the temperature of a superheated gas (that is, a superheated vapor) and the temperature of a saturated vapor. If the degrees of superheat is larger than zero, the state of the refrigerant becomes the superheated gas state.

[0038] In this embodiment 1, in the low-heating operation, the controller 50 estimates the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 by a method described later (Step S12). The controller 50 compares the estimated degrees of superheat SH_{si} with a reference value α (Step S13). The reference value α is a predetermined value not less than zero. If the degrees of superheat SH_{si} is larger than the reference value α at Step S 13, it can be determined that the degrees of superheat SH_{si} is sufficiently large, and the state of the refrigerant in the second intake passage 36 can be reliably maintained in the superheated gas state. In this case, the routine returns to Step S11. On the other hand, if the degrees of superheat SH_{si} is at the reference value α or less at Step S 13, it can be determined that the degrees of superheat SH_{si} is not sufficient. In this case, the controller 50 raises the degrees of superheat SH_{si} by decreasing the opening degree of the expansion valve 6 (Step S14). As a result, the state of the refrigerant in the second intake passage 36 can be reliably maintained in the superheated gas state.

[0039] Subsequently, the method of estimating the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 in the low-heating operation will be described. The degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 can be calculated by the following equation from a condensation saturation temperature T_c and a refrigerant temperature Ts₂ in the second intake passage 36:

$$SH_{si} = Ts_2 - T_c \dots (2)$$

[0040] The method of estimating the condensation saturation temperature from the heat-pump outlet temperature will be described by referring to Fig. 7. As illustrated in Fig. 7, as compared with a region in which the refrigerant is a superheated gas and a region in which the refrigerant is an overcooled liquid, a heat-transfer coefficient of the refrigerant is higher in a region in which the refrigerant is in a gas-liquid two phase, whereby heat transfer from the refrigerant to the water is promoted. Thus, most of the region where the water temperature rises is a gas-liquid two phase region. In the low-heating operation, a water temperature change from the heat-pump inlet temperature to the heat-pump outlet temperature is smaller than that in the high-heating operation. In addition, in the low-heating operation, the water flowrate is higher than that in the high-heating operation, and the heat-transfer coefficient of the water is higher. As a result, in the low-heating operation, a temperature difference between the refrigerant and the water at a pinch point becomes smaller than that in the high-heating operation. The pinch point is a point where the refrigerant temperature and the water temperature are the closest to each other. In the superheated gas region, inclination of the temperature change of the refrigerant with respect to a stream direction is the larger if it is the closer to B (the refrigerant inlet of the first heat

exchanger 4) in Fig. 7 and the smaller if it is the closer to the pinch point. Thus, the inclination of the temperature change of the water with respect to a stream direction is also smaller in the vicinity of the pinch point. Thus, a water temperature T_{wp} at the pinch point and a water temperature T_{wgcli} at a water inlet of the first heat exchanger 4 can be considered to be substantially equal. That is, the following equation holds true:

$$T_{wp} \approx T_{wgcli} \dots (3)$$

[0041] A heat exchange rate Q_{gc1} in the first heat exchanger 4 can be calculated by a temperature difference between the inlet/outlet of the refrigerant. That is, the following equation holds true:

$$Q_{gc1} = Gr \times C_{pr} \times (T_{d1} - T_{s2}) \dots (4)$$

where reference character Gr denotes a refrigerant flowrate, reference character C_{pr} denotes a constant pressure specific heat of the refrigerant, and reference character T_{d1} denotes a compressing element discharge temperature. The refrigerant flowrate Gr can be estimated from the capacity of the compressor 3, the outside air temperature and the like. The compressing element discharge temperature T_{d1} can be detected by the discharge temperature sensor 51.

[0042] In addition, the heat exchange rate Q_{gc1} in the first heat exchanger 4 can be also calculated from the temperature difference between the refrigerant and the water. That is, the following equation holds true:

$$Q_{gc1} = A_{gc1} \times K_{gc1} \times \Delta T_{gc1} \dots (5)$$

where reference character A_{gc1} denotes a heat transfer area of the first heat exchanger 4, reference character K_{gc1} denotes a overall heat-transfer coefficient of the first heat exchanger 4, and reference character ΔT_{gc1} denotes a temperature difference between the refrigerant and the water in the first heat exchanger 4. The value of A_{gc1} is a fixed value. The value of K_{gc1} can be considered to be a substantially constant value. The value of ΔT_{gc1} can be calculated by the following equation, assuming that it is an arithmetic mean temperature difference:

$$\Delta T_{gc1} = (T_{d1} + T_{s2})/2 - (T_{wo} + T_{wgcli})/2 \dots (6)$$

where, reference character T_{wo} denotes a heat-pump outlet temperature. The arithmetic mean temperature difference can be easily calculated by the aforementioned equation (6). Alternatively, a logarithmic mean temperature difference may be also used as ΔT_{gc1} . In that case, ΔT_{gc1} can be calculated more accurately.

[0043] By making the aforementioned equations (4), (5) and (6) into simultaneous equations, the water temperature T_{wgcli} at the water inlet of the first heat exchanger 4 can be estimated on the basis of the compressing element discharge temperature T_{d1} , the refrigerant temperature T_{s2} in the second intake passage 36, and the heat-pump outlet temperature T_{wo} . By considering the T_{wgcli} estimated as above to be equal to the water temperature T_{wp} at the pinch point, the water temperature T_{wp} at the pinch point can be acquired. The condensation saturation temperature T_c can be calculated by the following equation:

$$T_c = T_{wp} + \Delta T_p \dots (7)$$

where reference character ΔT_p denotes a temperature difference between the refrigerant and the water at the pinch point. The temperature difference ΔT_p between the refrigerant and the water at the pinch point is approximately 1 to 3°C. By substituting the condensation saturation temperature T_c calculated by the aforementioned equation (7) in the aforementioned equation (2), the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 can be acquired. At Step S12 in Fig. 8, the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 can be estimated on the basis of the compressing element discharge temperature T_{d1} , the refrigerant temperature T_{s2} in the second intake passage 36, and the heat-pump outlet temperature T_{wo} as described above.

[0044] In addition, the method of estimating the water temperature T_{wgcli} at the water inlet of the first heat exchanger 4 may be as follows instead of the aforementioned method. The heat exchange rate Q_{gc1} in the first heat exchanger 4 can be calculated by the temperature difference between the inlet/outlet of the water. That is, the following equation holds true:

$$Q_{gc1} = G_w \times C_{pw} \times (T_{wo} - T_{wgcli}) \dots (8)$$

where reference character G_w denotes a water flowrate and reference character C_{pw} denotes a specific heat of water.

The water flowrate G_w can be estimated from the driving speed of the pump 13. Alternatively, the water flowrate G_w may be detected by a flowrate sensor. By making either one of the aforementioned equations (4) and (5) and the aforementioned equation (8) into simultaneous equations, the water temperature T_{wgcli} at the water inlet of the first heat exchanger 4 can be estimated on the basis of the compressing element discharge temperature T_{d1} , the refrigerant temperature T_{s2} in the second intake passage 36, and the heat-pump outlet temperature T_{wo} . By considering T_{wgcli} estimated as above to be equal to the water temperature T_{wp} at the pinch point and by using the aforementioned equations (7) and (2), the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 can be estimated.

[0045] In this embodiment 1, the refrigerant temperature T_{s2} in the second intake passage 36 can be detected by the refrigerant temperature sensor 52. However, the refrigerant temperature T_{s2} in the second intake passage 36 may be detected by a temperature sensor provided on an outer surface of the sealed container 31 or the like instead of the refrigerant temperature sensor 52.

[0046] In addition, the refrigerant temperature T_{s2} in the second intake passage 36 may be estimated as follows on the basis of the compressing element discharge temperature T_{d1} and the heat-pump outlet temperature T_{wo} without providing the refrigerant temperature sensor 52. The heat exchange rate Q_{gc1} in the first heat exchanger 4 can be considered to be in proportion to the temperature difference between the compressing element discharge temperature T_{d1} and the heat-pump outlet temperature T_{wo} . Thus, assuming that its proportional coefficient is F , the following equation holds true:

$$Q_{gc1} = F \times (T_{d1} - T_{wo}) \dots (9)$$

By making the aforementioned equations (4) and (9) into simultaneous equations, the refrigerant temperature T_{s2} in the second intake passage 36 can be estimated on the basis of the compressing element discharge temperature T_{d1} and the heat-pump outlet temperature T_{wo} . By making the aforementioned equations (8) and (9) into simultaneous equations, the water temperature T_{wgcli} at the water inlet of the first heat exchanger 4 can be estimated on the basis of the compressing element discharge temperature T_{d1} and the heat-pump outlet temperature T_{wo} . By considering T_{wgcli} estimated as above to be equal to the water temperature T_{wp} at the pinch point and by using the aforementioned equations (7) and (2), the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 can be estimated.

[0047] The aforementioned theory is based on the premise that the refrigerant in the second intake passage 36 is in the superheated gas state, when the heat exchange rate Q_{gc1} in the first heat exchanger 4 is to be calculated. Thus, assuming that the refrigerant in the second intake passage 36 enters the gas-liquid two-phase state, the heat exchange rate Q_{gc1} in the first heat exchanger 4 is estimated to be smaller than the actuality. Estimation of the heat exchange rate Q_{gc1} in the first heat exchanger 4 smaller than the actuality leads to estimation of the temperature difference ΔT_{gc1} between the refrigerant and the water in the first heat exchanger 4 smaller than the actuality. The estimation of the temperature difference ΔT_{gc1} between the refrigerant and the water in the first heat exchanger 4 smaller than the actuality leads to estimation of the water temperature T_{wgcli} at the water inlet of the first heat exchanger 4 higher than the actuality. The estimation of the water temperature T_{wgcli} at the water inlet of the first heat exchanger 4 higher than the actuality leads to estimation of the condensation saturation temperature T_c higher than the actuality. The degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 is calculated by the aforementioned equation (2). Thus, the estimation of the condensation saturation temperature T_c higher than the actuality leads to estimation of the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 smaller than the actuality. Thus, by executing control such that the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 estimated conforming to the aforementioned theory becomes larger than zero, the state of the refrigerant in the second intake passage 36 can be reliably controlled to the superheated gas state.

[0048] At Step S12 in Fig. 8, instead of the aforementioned method, the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 may be estimated on the basis of an evaporation saturation temperature T_e , the compressing element suction temperature T_{s1} , and the compressing element discharge temperature T_{d1} as follows. A compression process in the compressing element 32 is assumed to be polytropic change and a polytropic index is n , the following equation holds true. The polytropic index n can be considered to be a constant value determined from a physical property value of the refrigerant and the compressor efficiency:

$$Td1 = Ts1 \times (Pd1/Ps1)^{(n-1)/n} \dots (10)$$

where reference character Pd1 is the compressing element discharge pressure. The compressing element suction temperature Ts1 can be detected by providing a temperature sensor at A in Figs. 2 and 6. The compressing element discharge temperature Td1 can be detected by the discharge temperature sensor 51 provided at B in Figs. 2 and 6.

[0049] The evaporation saturation temperature Te can be detected by providing a temperature sensor at G in Figs. 2 and 6. An evaporation saturation pressure Pe can be calculated from a relation between a saturation temperature and a saturation pressure on the basis of the evaporation saturation temperature Te. By ignoring a pressure loss in the evaporator 7, the compressing element suction pressure Ps1 can be considered to be equal to the evaporation saturation pressure Pe. In addition, the compressing element suction pressure Ps1 may be calculated by subtracting a constant value of the pressure loss from the evaporation saturation pressure Pe. By substituting the compressing element suction pressure Ps1 acquired as above in the aforementioned equation (10), the compressing element discharge pressure Pd1 can be calculated on the basis of the evaporation saturation temperature Te, the compressing element suction temperature Ts1, and the compressing element discharge temperature Td1. On the basis of the compressing element discharge pressure Pd1, the condensation saturation temperature Tc can be calculated from the relation between the saturation temperature and the saturation pressure. By substituting the condensation saturation temperature Tc calculated as above in the aforementioned equation (2), the degrees of superheat SHsi of the refrigerant in the second intake passage 36 can be estimated.

[0050] According to the heat pump system 1 of this embodiment 1, the following effects can be obtained:

(1) In the hot-water storing operation, that is, in the high-heating operation, the compressing element discharge pressure Pd1 becomes a super-critical pressure, and the refrigerant in the first heat exchanger 4 and the second heat exchanger 5 is not condensed. Thus, the refrigerant flowrate can be controlled using the expansion valve 6 so that the COP is maximized.

(2) In the hot-water filling operation, that is, in the low-heating operation, inflow of the gas-liquid two-phase refrigerant into the internal space 38 of the sealed container 31 can be reliably prevented by executing control such that the state of the refrigerant in the second intake passage 36 is in the superheated gas state. As a result, the refrigerant flowrate can be controlled properly while reliability is improved. As a result, an efficient operation can be realized.

(3) The degrees of superheat SHsi of the refrigerant in the second intake passage 36 can be estimated on the basis of detected temperatures of the temperature sensors such as the heat-pump outlet temperature sensor 30, the discharge temperature sensor 51, the refrigerant temperature sensor 52 and the like. Thus, such control can be executed that the state of the refrigerant in the second intake passage 36 in the low-heating operation becomes the superheated gas state without using an expensive pressure sensor.

(4) Since the inflow of the gas-liquid two-phase refrigerant into the sealed container 31 can be reliably prevented in the low-heating operation, the liquid refrigerant is not accumulated in the sealed container 31. If the liquid refrigerant accumulated in the sealed container 31 is heated by the compressing element 32 or the electric actuating element 33, the liquid refrigerant mixed with the refrigerator oil is evaporated, whereby the refrigerator oil bubbles. If the refrigerator oil bubbles, the refrigerator oil is mixed with the gas refrigerant, and the refrigerator oil flows out of the second discharge passage 37 with the gas refrigerant. As a result, the refrigerator oil in the sealed container 31 runs short, and defective lubrication in a sliding portion of the compressor 3 occurs. In addition, the refrigerator oil remains in the second heat exchanger 5, which reduces heat transfer of the refrigerant and lowers performances. On the other hand, according to this embodiment 1, since the liquid refrigerant is not accumulated in the sealed container 31, these troubles can be reliably prevented.

[0051] The present invention is not limited to the aforementioned configuration but may be configured as follows, for example. By providing a pressure sensor for detecting the compressing element discharge pressure Pd1, the degrees of superheat of the refrigerant in the second intake passage 36 may be controlled on the basis of a value of the pressure sensor. By providing a temperature sensor in the water channel 47 connecting the first heat exchanger 4 and the second heat exchanger 5, the water temperature Twgc1i at the water inlet of the first heat exchanger 4 may be detected by the temperature sensor. By providing a temperature sensor at a middle point of the refrigerant channel of the second heat exchanger 5, the condensation saturation temperature Tc may be directly detected by the temperature sensor.

[0052] The advantageous effects of the present invention are exerted particularly markedly when the refrigerant (carbon dioxide, for example) is used which causes the compressing element discharge pressure in the high-heating operation to be brought to a pressure exceeding the critical pressure as in this embodiment 1. However, the present invention can be also applied when a refrigerant which causes the compressing element discharge pressure in the high-heating operation to be brought to the critical pressure or less is used. Such refrigerants include R410A, R32, R22, R407C,

propane, propylene, HFO-1234yf, HFO-1234ze or a mixed refrigerant of them.

[0053] A typical design method using the refrigerant which causes the compressing element discharge pressure in the high-heating operation to be brought to the critical pressure or less is a method in which a ratio in size (heat exchange rate) between the first heat exchanger 4 and the second heat exchanger 5 is designed so that the refrigerant in the second intake passage 36 becomes a superheated gas in accordance with a condition in the high-heating operation operating at a rated power. However, even if it is designed as above, the heat-pump outlet temperature is low in the low-heating operation, the compressing element discharge pressure lowers, and an enthalpy difference in the first heat exchanger 4 and the second heat exchanger 5 decreases and thus, the refrigerant in the second intake passage 36 can enter the gas-liquid two-phase state easily. On the other hand, by applying the present invention, the effect similar to the above can be obtained. Thus, even if the refrigerant which causes the compressing element discharge pressure in the high-heating operation to be brought to the critical pressure or less is used, it is significant to apply the invention of the present application.

Embodiment 2.

[0054] Subsequently, an embodiment 2 of the present invention will be described by referring to Fig. 9, but a difference from the aforementioned embodiment 1 will be mainly described, and the same portions or corresponding portions are given the same reference numerals and explanation will be omitted. Fig. 9 is a flowchart illustrating a control operation of the controller 50 in the low-heating operation of the heat pump system 1 according to the embodiment 2 of the present invention. Since Step S11 to Step S13 in Fig. 9 are similar to those in the embodiment 1, explanation will be omitted. In this embodiment 2, if the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 is at the reference value α or less at Step S13, the controller 50 raises the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 by controlling the compressor 3 (by increasing the driving speed of the compressor 3, for example) so that the capacity of the compressor 3 increases (Step S15). As a result, in this embodiment 2, the state of the refrigerant in the second intake passage 36 can be maintained in the superheated gas state more reliably. Since this embodiment 2 is similar to the embodiment 1 except the aforementioned matter, further explanation will be omitted.

Embodiment 3.

[0055] Subsequently, an embodiment 3 of the present invention will be described by referring to Fig. 10, but a difference from the aforementioned embodiment 1 will be mainly described, and the same or corresponding portions are given the same reference numerals and explanation will be omitted. Fig. 10 is a flowchart illustrating a control operation of the controller 50 in the low-heating operation of the heat pump system 1 according to the embodiment 3 of the present invention. Since Step S11 to Step S13 in Fig. 10 are similar to those in the embodiment 1, explanation will be omitted. In this embodiment 3, if the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 is at the reference value α or less at Step S13, the controller 50 raises the degrees of superheat SH_{si} of the refrigerant in the second intake passage 36 by controlling the pump 13 so that the water flowrate lowers (Step S16). If the water flowrate lowers, the compressing element discharge pressure rise. Since a saturated vapor line in the pressure-enthalpy diagram is inclined to upper left, the higher the pressure rises, the smaller the enthalpy of the saturated gas becomes. As a result, the degrees of superheat SH_{si} increases. According to this embodiment 3, the state of the refrigerant in the second intake passage 36 can be maintained in the superheated gas state more reliably. Since this embodiment 3 is similar to the embodiment 1 except the aforementioned matter, further explanation will be omitted.

Embodiment 4.

[0056] Subsequently, an embodiment 4 of the present invention will be described by referring to Figs. 11 and 12, but a difference from the aforementioned embodiment 1 will be mainly described, and the same or corresponding portions are given the same reference numerals and explanation will be omitted. Fig. 11 is a configuration diagram illustrating the heat pump unit 2 included in the heat pump system 1 according to the embodiment 4 of the present invention. As illustrated in Fig. 11, the heat pump unit 2 of this embodiment 4 further includes a bypass channel 55 for causing the refrigerant to flow from the first discharge passage 35 to the second intake passage 36 without passing through the first heat exchanger 4 and a bypass valve 56 provided in this bypass channel 55 in addition to the configuration similar to that of the embodiment 1. The bypass channel 55 is a channel bypassing the refrigerant channel of the first heat exchanger 4. The bypass valve 56 is a channel control element capable of varying a rate of the refrigerant passing through the bypass channel 55. In the state where the bypass valve 56 is closed, all the refrigerant in the superheated gas state discharged from the first discharge passage 35 passes through the first heat exchanger 4. That is, it is the state similar to the embodiment 1. On the other hand, in the state where the bypass valve 56 is opened, the refrigerant in the superheated gas state discharged from the first discharge passage 35 flows through the refrigerant channel of the first

heat exchanger 4 and the bypass channel 55, separately. Then, the refrigerant having passed through the refrigerant channel of the first heat exchanger 4 and the refrigerant having passed through the bypass channel 55 merge with each other and flow to the second intake passage 36.

[0057] Fig. 12 is a flowchart illustrating the control operation of the controller 50 in the low-heating operation of the heat pump system 1 according to the embodiment 4 of the present invention. Since Step S11 to Step S13 in Fig. 12 are similar to those in the embodiment 1, explanation will be omitted. In this embodiment 4, if the degrees of superheat SHsi of the refrigerant in the second intake passage 36 is at the reference value α or less at Step S13, the controller 50 raises the degrees of superheat SHsi of the refrigerant in the second intake passage 36 by controlling the bypass valve 56 so that an opening degree of the bypass valve 56 increases (Step S17). That is, at Step S17, if the bypass valve 56 is closed, the bypass valve 56 is opened to a predetermined opening degree, while if the bypass valve 56 is already open, the opening degree of the bypass valve 56 is increased. By increasing the opening degree of the bypass valve 56, a ratio of the refrigerant passing through the bypass channel 55 rises. Since the refrigerant in the superheated gas state having passed through the bypass channel 55 has not been subjected to heat exchange, its temperature is higher than that of the refrigerant having passed through the refrigerant channel of the first heat exchanger 4. Thus, by increasing the opening degree of the bypass valve 56 and by raising the ratio of the refrigerant passing through the bypass channel 55, the degrees of superheat SHsi of the refrigerant in the second intake passage 36 rises. By means of such configuration, the state of the refrigerant in the second intake passage 36 can be maintained in the superheated gas state more reliably in this embodiment 4. Since this embodiment 4 is similar to the embodiment 1 except the aforementioned matter, further explanation will be omitted.

Reference Signs List

[0058]

25	1	heat pump system
	3	compressor
	4	first heat exchanger
	5	second heat exchanger
	6	expansion valve
30	7	evaporator
	8	air blower
	9	high-low pressure heat exchanger
	10	hot water tank
	11	inlet conduit
35	12	inlet
	13	pump
	14	upper conduit
	15	hot-water feeding mixing valve
	16	bath mixing valve
40	17	outlet
	18	outlet conduit
	19	feed-water pipe
	20	pressure-reducing valve
	21	feed-water pipe
45	22	hot-water pipe
	23	hot-water tap
	24	hot-water flowrate detecting means
	25	hot-water temperature sensor
	26	bath pipe
50	27	bathtub
	28	opening/closing valve
	29	bath temperature sensor
	30	heat-pump outlet temperature sensor
	31	sealed container
55	32	compressing element
	33	electric actuating element
	33a	stator
	33b	rotor

34	first intake passage
35	first discharge passage
36	second intake passage
37	second discharge passage
5 38, 39	internal space
40, 41, 42, 43, 44, 45	refrigerant channel
46, 47, 48	water channel
50	controller
51	discharge temperature sensor
10 52	refrigerant temperature sensor
55	bypass channel
56	bypass valve
60	operation unit

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Claims

1. A heat pump system comprising:

20 a compressor including a sealed container, a compressing element provided inside the sealed container, a first intake passage configured to lead low-pressure refrigerant sucked from an outside of the sealed container to the compressing element without releasing the low-pressure refrigerant into an internal space of the sealed container, a first discharge passage configured to discharge high-pressure refrigerant compressed in the compressing element to the outside of the sealed container without releasing the high-pressure refrigerant to the

25 internal space of the sealed container, a second intake passage configured to guide high-pressure refrigerant, the high-pressure refrigerant having exchanged heat after being discharged from the first discharge passage, to the internal space of the sealed container without compression, and a second discharge passage configured to discharge high-pressure refrigerant in the internal space of the sealed container to the outside of the sealed container without compression;

30 a first heat exchanger configured to heat an object fluid by heat of the high-pressure refrigerant discharged from the first discharge passage;

a second heat exchanger configured to heat the object fluid by the heat of the high-pressure refrigerant discharged from the second discharge passage;

35 an expansion unit configured to expand high-pressure refrigerant having passed through the second heat exchanger to low-pressure refrigerant;

an evaporator configured to evaporate the low-pressure refrigerant having passed through the expansion unit; and

control means configured to perform a high-heating operation and a low-heating operation, the low-heating operation having a smaller total heating amount in the first heat exchanger and the second heat exchanger than the high-heating operation,

40 the control means being configured to control, in the low-heating operation, so that a state of the refrigerant in the second intake passage is in a superheated gas state.

45 **2.** The heat pump system according to claim 1, wherein a pressure of the refrigerant discharged from the compressing element is brought to a pressure exceeding a critical pressure in the high-heating operation, and wherein a pressure of the refrigerant discharged from the compressing element is brought to a pressure at the critical pressure or less in the low-heating operation.

50 **3.** The heat pump system according to claim 1 or 2, wherein the control means is configured to control, in the high-heating operation, a refrigerant flowrate using the expansion unit so that a temperature of the refrigerant discharged from the compressing element or degrees of superheat of the refrigerant coming out of the evaporator is brought to a target value.

55 **4.** The heat pump system according to any one of claims 1 to 3, wherein the control means is configured to:

estimate, in the low-heating operation, degrees of superheat of the refrigerant in the second intake passage based on a temperature of the refrigerant discharged from the compressing element, a temperature of the refrigerant in the second intake passage, and a temperature of the object fluid coming out of the first heat

exchanger; and

control, based on the estimated value, at least one of an opening degree of the expansion unit, a capacity of the compressor, and a flowrate of the object fluid.

- 5 **5.** The heat pump system according to any one of claims 1 to 3, wherein the control means is configured to:

estimate, in the low-heating operation, degrees of superheat of the refrigerant in the second intake passage based on a temperature of the refrigerant sucked into the compressing element, a temperature of the refrigerant discharged from the compressing element, and an evaporation saturation temperature in the evaporator; and
10 control, based on the estimated value, at least one of an opening degree of the expansion unit, a capacity of the compressor, and a flowrate of the object fluid.

- 6.** The heat pump system according to any one of claims 1 to 3, further comprising:

15 a bypass channel configured to cause the refrigerant to flow to the second intake passage from the first discharge passage without passing through the first heat exchanger; and
a channel control element capable of varying a rate of the refrigerant passing through the bypass channel,

wherein the control means is configured to control, in the low-heating operation, the rate of the refrigerant passing through the bypass channel using the channel control element so that the state of the refrigerant in the second
20 intake passage is in the superheated gas state.

- 7.** The heat pump system according to any one of claims 1 to 6, further comprising a hot water tank,
wherein the control means is configured to:

25 perform the high-heating operation in a hot-water storing operation for causing water heated in the first heat exchanger and the second heat exchanger to flow into the hot water tank; and
perform the low-heating operation in a hot-water filling operation for supplying the water heated in the first heat exchanger and the second heat exchanger to a bathtub.
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FIG. 1

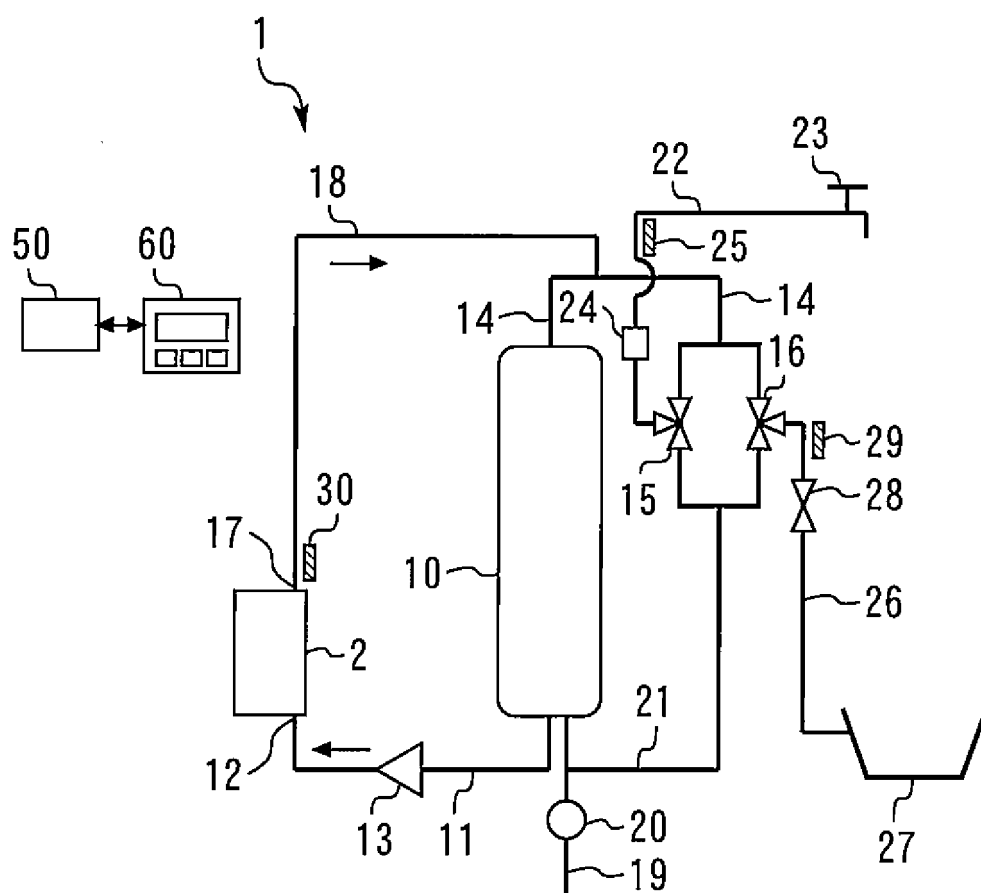


FIG. 2

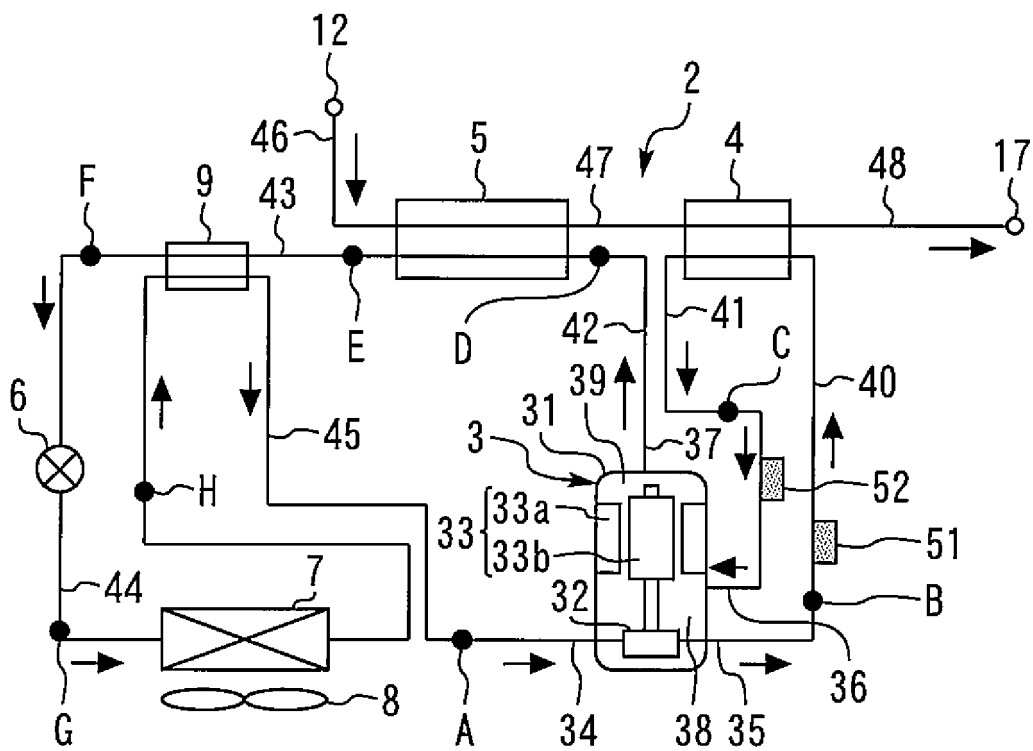


FIG. 3

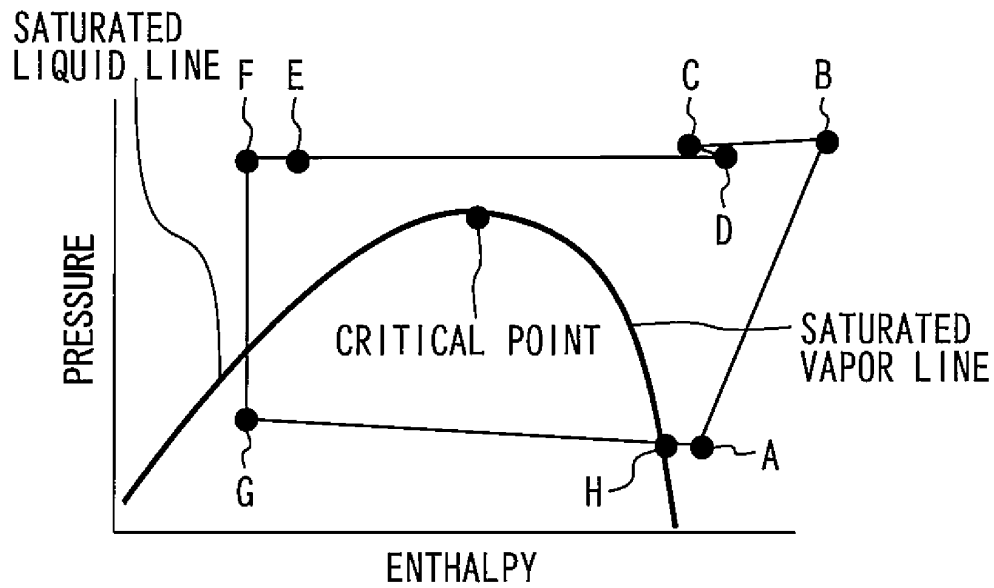


FIG. 4

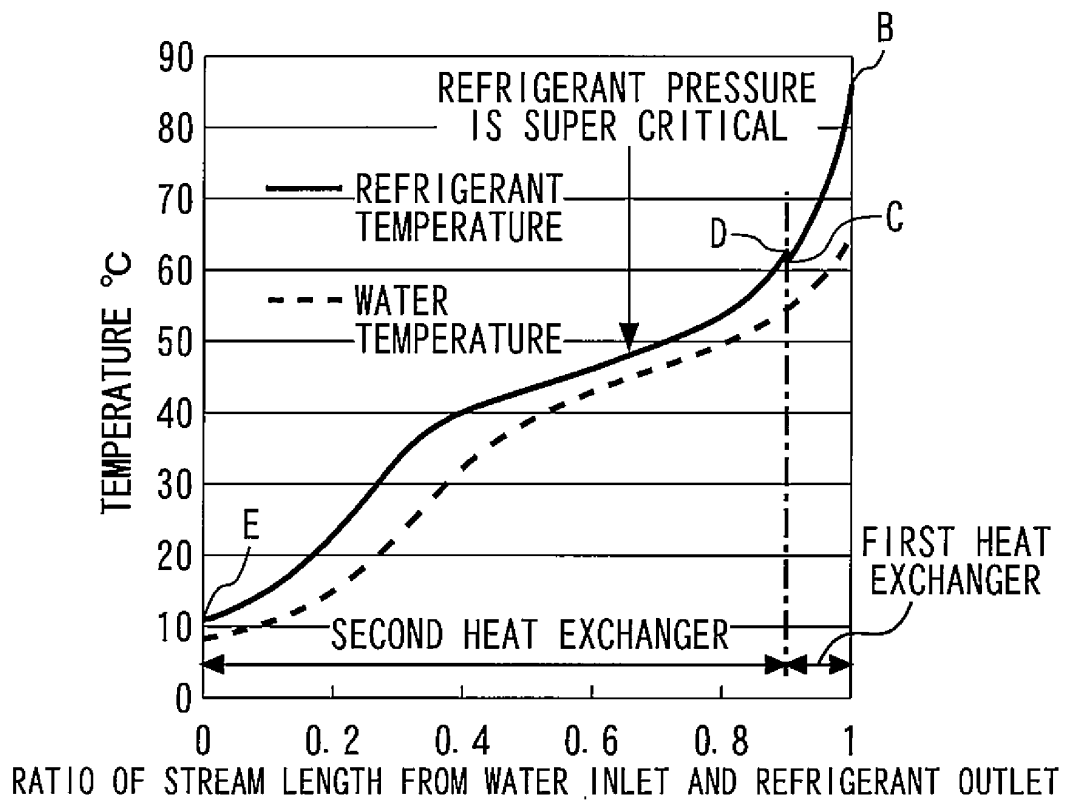


FIG. 5

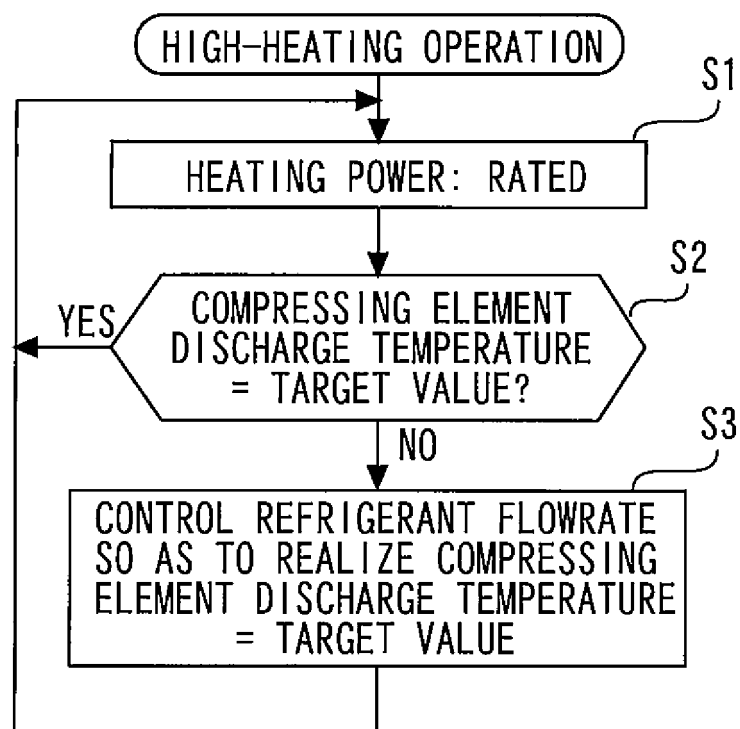


FIG. 6

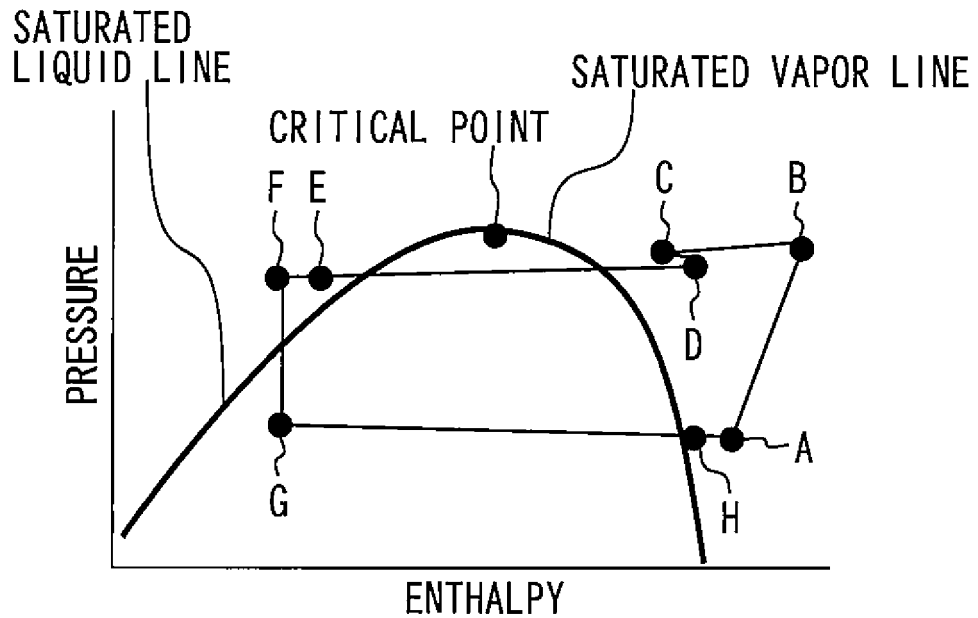


FIG. 7

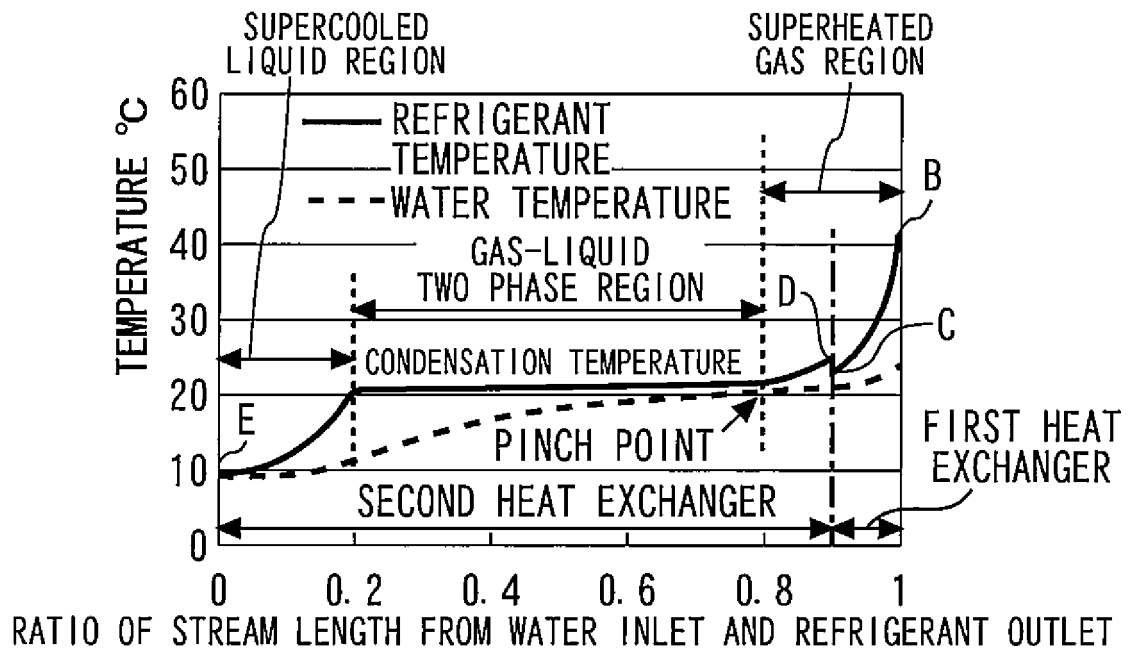


FIG. 8

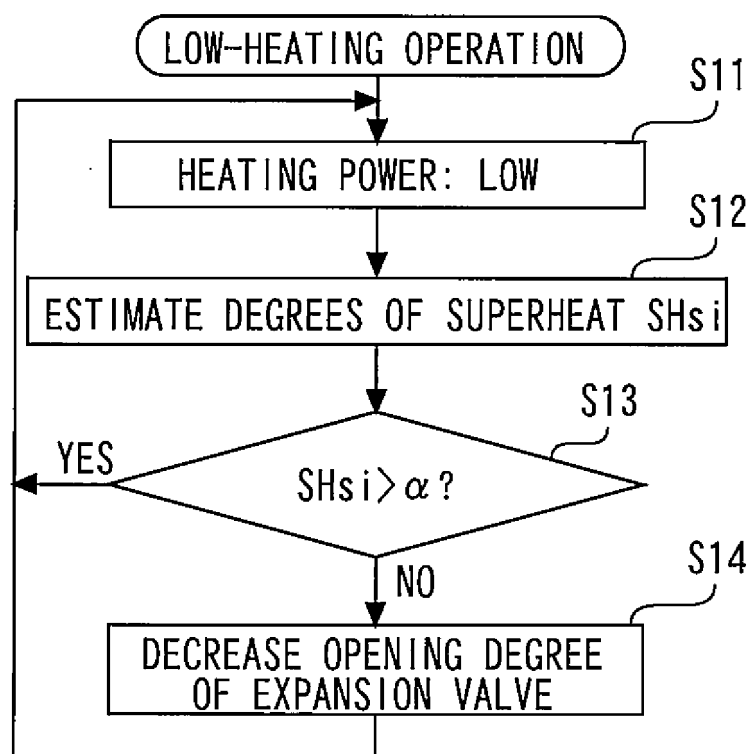


FIG. 9

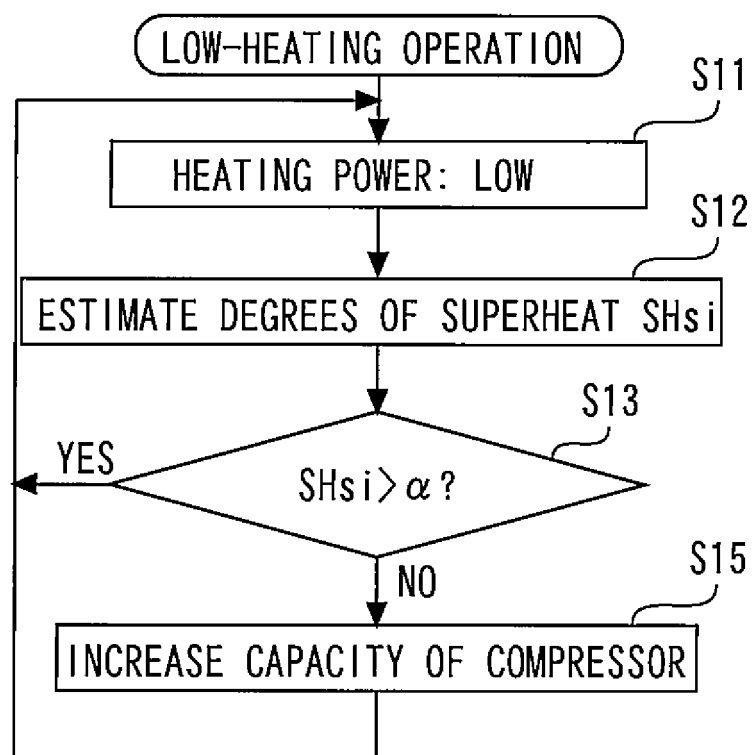


FIG. 10

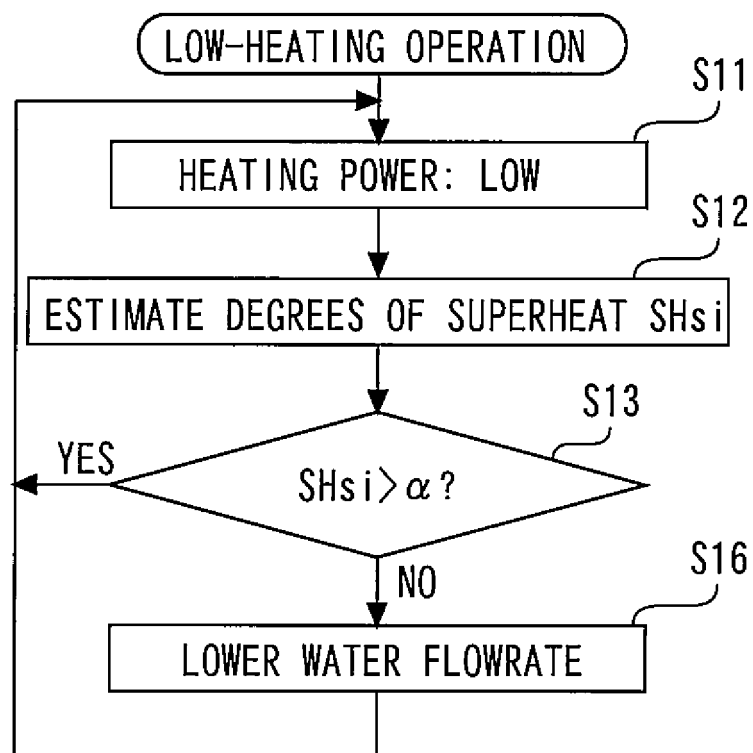


FIG. 11

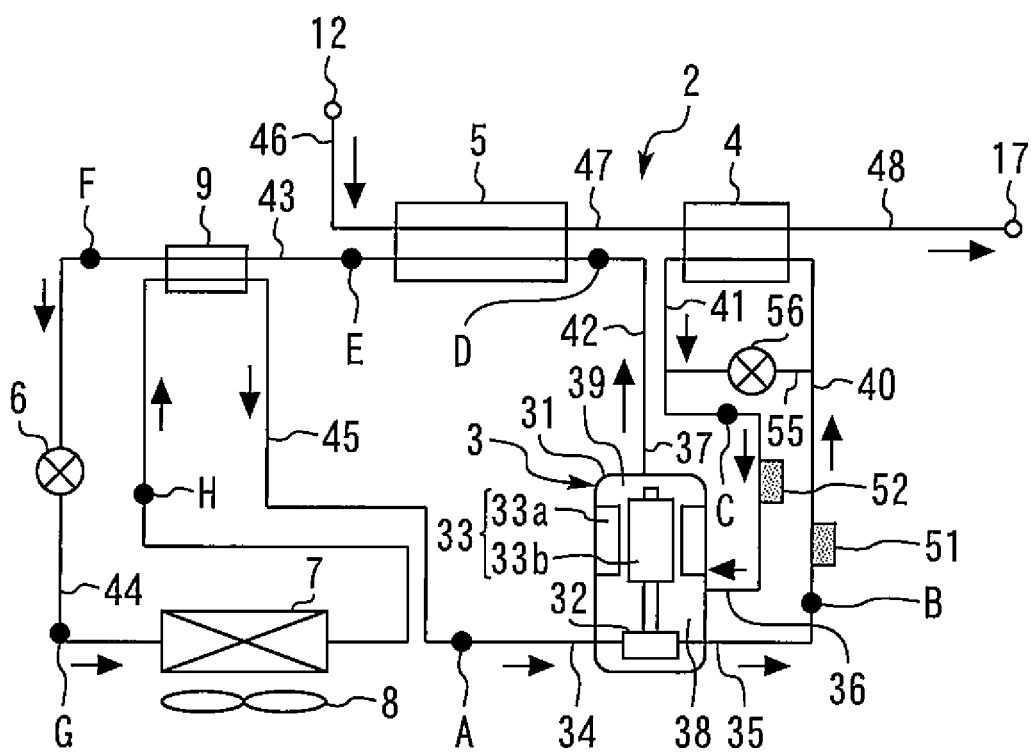
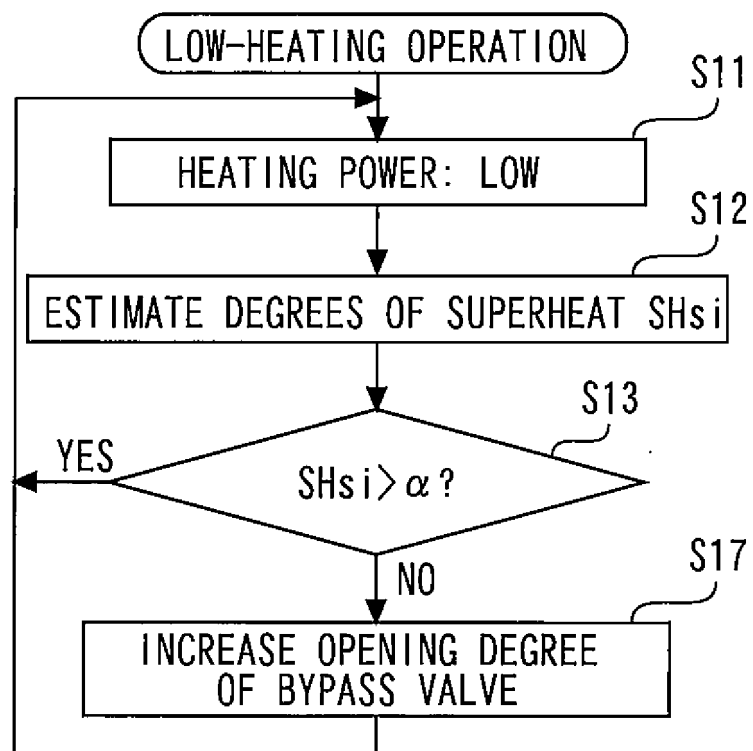


FIG. 12



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/056149

A. CLASSIFICATION OF SUBJECT MATTER

F25B1/00(2006.01)i, F24H1/00(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, F24H1/00

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 A	JP 2006-132427 A (Mitsubishi Electric Corp.), 25 May 2006 (25.05.2006), paragraphs [0013] to [0017]; fig. 1, 2 (Family: none)	1-3, 6-7 4-5
Y A	JP 55-34540 Y2 (Daikin Industries, Ltd.), 15 August 1980 (15.08.1980), column 2, lines 5 to 29; fig. 1 (Family: none)	1-3, 6-7 4-5
Y A	JP 51-2325 Y2 (Sanyo Electric Co., Ltd.), 23 January 1976 (23.01.1976), column 1, line 18 to column 2, line 7; drawings (Family: none)	1-3, 6-7 4-5



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

21 May, 2014 (21.05.14)

Date of mailing of the international search report

03 June, 2014 (03.06.14)

Name and mailing address of the ISA/
Japanese Patent Office

Authorized officer

Facsimile No.

Telephone No.

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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2014/056149

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y A	JP 7-294025 A (Nihon Kentetsu Co., Ltd.), 10 November 1995 (10.11.1995), paragraphs [0004] to [0008]; fig. 8 to 11 (Family: none)	1-3, 6-7 4-5
Y	JP 2005-134070 A (Matsushita Electric Industrial Co., Ltd.), 26 May 2005 (26.05.2005), paragraphs [0038] to [0047] (Family: none)	3, 6-7
Y	JP 2011-21828 A (Mitsubishi Electric Corp.), 03 February 2011 (03.02.2011), paragraphs [0009] to [0010]; fig. 1 (Family: none)	7
A	JP 4-73553 A (Sanyo Electric Co., Ltd.), 09 March 1992 (09.03.1992), entire text; all drawings (Family: none)	1-7
A	JP 2009-133547 A (Mitsubishi Electric Corp.), 18 June 2009 (18.06.2009), entire text; all drawings & US 2010/0205987 A1 & EP 2196745 A1 & WO 2009/069524 A1 & CN 101842645 A	1-7
A	JP 2014-16041 A (Denso Corp.), 30 January 2014 (30.01.2014), entire text; all drawings (Family: none)	1-7

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REFERENCES CITED IN THE DESCRIPTION

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- JP 2006132427 A [0004]
- JP 2011021828 A [0004]