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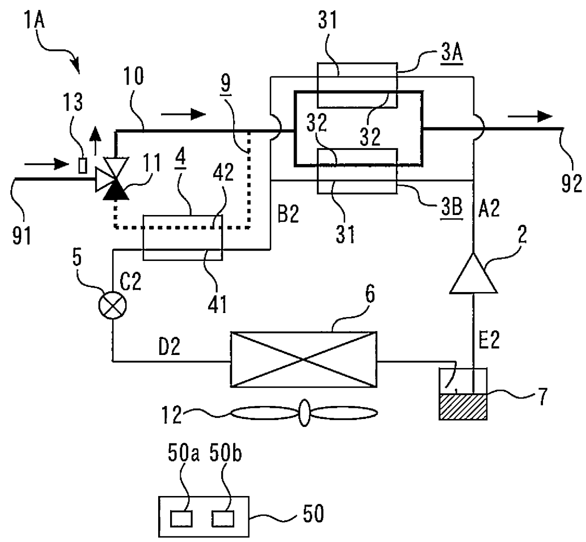
(54) REFRIGERATION CYCLE DEVICE

(57) The present invention has an object to provide a refrigeration cycle device capable of inhibiting a heat medium from heating refrigerant in a condenser when a temperature of the heat medium before heating is high. The refrigeration cycle device of the present invention includes: a first condenser; a second condenser having a smaller sectional area of a refrigerant flow path than the first condenser, for further condensing the refrigerant having passed through the first condenser; a heat medium path for passing a heat medium through the second condenser and the first condenser in this order; a second condenser bypass passage for bypassing the refrigerant flow path or the heat medium flow path in the second condenser; a flow path controlling element capable of varying a bypass rate of the refrigerant or the heat medium bypassing the second condenser; and control means for controlling the flow path controlling element so as to increase the bypass rate when a temperature of the heat medium before heat exchange with the refrigerant is high.

FIG. 9



EP 3 059 520 A1

Description

Technical Field

[0001] The present invention relates to a refrigeration cycle device for heating a heat medium with a condenser.

Background Art

[0002] Patent Literature 1 below discloses a heat pump type hot water supply device including: a refrigeration cycle circuit including a compressor, a four-way valve, a water heat exchanger (condenser), a pressure reducing device, and an air heat exchanger (evaporator) connected via a refrigerant pipe; and a water circuit including a pump, a water heat exchanger, and a hot water storage tank connected via a water pipe, hot water heated by the water heat exchanger in the refrigeration cycle circuit being stored in the hot water storage tank, wherein R410A or R407C is used as refrigerant for the refrigeration cycle circuit.

[0003] For an air conditioner using an R410A refrigerant, high pressure side design pressure is, for example, 4.25 MPa, which is converted into a saturation temperature of about 65°C. Any pressure described herein is absolute pressure. When the R410A refrigerant is used for a refrigeration cycle in a hot water supply device, design pressure needs to be 4.25 MPa as in the air conditioner so that components such as a compressor and a heat exchanger are common to those in the air conditioner.

[0004] In Patent Literature 1, when condensation pressure is 4.75 MPa, a saturation temperature is about 70°C, and a feed-water temperature is 5°C in use of the R410A refrigerant, output hot water temperature is about 85°C. On the other hand, if the design pressure of 4.25 MPa of the air conditioner as described above is an upper limit, a saturation temperature is about 65°C and an output hot water temperature is about 80°C. In this case, a refrigerant temperature at a condenser outlet is 10°C.

Citation List

Patent Literature

[0005]

Patent Literature 1: Japanese Patent Laid-Open No. 2002-89958
Patent Literature 2: Japanese Patent Laid-Open No. 2002-310498
Patent Literature 3: Japanese Patent Laid-Open No. 2007-232285
Patent Literature 4: Japanese Patent Laid-Open No. 2013-44441
Patent Literature 5: Japanese Patent Laid-Open No. 2009-222246
Patent Literature 6: Japanese Patent Laid-Open No. 2010-14374
Patent Literature 7: Japanese Patent Laid-Open No. 2001-82818

Summary of Invention

Technical Problem

[0006] A feed-water temperature to a condenser in a heat pump type hot water supply device is usually similar to an outside air temperature. However, the feed-water temperature is about 50°C or higher when hot water reduced in temperature by thermal dissipation in a hot water storage tank is reheated, or when hot water heated by a condenser is circulated to a heat exchanger for heating bathtub water. If an upper limit of a refrigerant saturation temperature in the condenser is about 65°C, refrigerant at a condenser outlet is brought into a gas-liquid two-phase state or a gas state when the feed-water temperature is high. If the refrigerant at the condenser outlet is brought into the gas-liquid two-phase state or the gas state, an average flow speed of the refrigerant in the condenser increases to increase pressure loss of the refrigerant. If the pressure loss reduces a refrigerant temperature, a part where the refrigerant temperature is lower than the feed-water temperature may be created in the condenser. In that case, in the part where the refrigerant temperature is lower than the water temperature, the refrigerant draws heat from the water, thereby reducing efficiency of the condenser heating the water.

[0007] The present invention is achieved to solve the problems described above, and has an object to provide a refrigeration cycle device capable of inhibiting a heat medium from heating refrigerant in a condenser when a temperature of the heat medium before heating is high.

Solution to Problem

[0008] A refrigeration cycle device of the invention includes: a compressor configured to compress refrigerant; a first condenser including a refrigerant flow path and a heat medium flow path, the first condenser being configured to condense the refrigerant compressed by the compressor; a second condenser including a refrigerant flow path having a small sectional area as compared with the refrigerant flow path in the first condenser, and a heat medium flow path, the second condenser being configured to further condense the refrigerant having passed through the first condenser; an evaporator configured to evaporate the refrigerant; a heat medium path configured to allow a liquid heat medium subjected to heat exchange with the refrigerant to pass through the second condenser and the first condenser in this order; a second condenser bypass passage configured to bypass the refrigerant flow path or the heat medium flow path in the second condenser; a flow path controlling element capable of varying a bypass rate that is a flow rate of the refrigerant or the heat medium flowing through the second condenser bypass passage; and control means for controlling an operation of the flow path controlling element so that the bypass rate in a case where an entry heat medium temperature that is a temperature of the heat medium before heat exchange with the refrigerant is higher than a reference temperature is larger than the bypass rate in a case where the entry heat medium temperature is lower than the reference temperature.

Advantageous Effects of Invention

[0009] According to the refrigeration cycle device of the present invention, a condenser is divided into a first condenser and a second condenser, a second condenser bypass passage for bypassing a refrigerant flow path or a heat medium flow path in the second condenser is provided, and when a temperature of a heat medium before heating is high, a flow rate of the refrigerant or heat medium bypassing the second condenser is increased to inhibit the heat medium from heating the refrigerant in the condenser.

Brief Description of Drawings

[0010]

Figure 1 is a configuration diagram of a refrigeration cycle device according to embodiment 1 of the present invention.
Figure 2 is a perspective view showing a part of a heat exchanger that constitutes a first condensers and a second condenser.
Figure 3 is a configuration diagram of a hot water storage type hot water supply system including the refrigeration cycle device of embodiment 1 and a tank unit.
Figure 4 is a flowchart showing a control operation in the refrigeration cycle device of embodiment 1.
Figure 5 shows a low temperature water input operation of the refrigeration cycle device of embodiment 1.
Figure 6 shows an example of changes in temperature of refrigerant and water in the first condensers and the second condenser in the low temperature water input operation of the refrigeration cycle device of embodiment 1.
Figure 7 shows a P-h diagram of the low temperature water input operation of the refrigeration cycle device of embodiment 1.
Figure 8 shows an example of a relationship between an outside air temperature and a feed-water temperature in the low temperature water input operation.
Figure 9 shows a high temperature water input operation of the refrigeration cycle device of embodiment 1.
Figure 10 shows an example of changes in temperature of refrigerant and water in the first condensers in the high temperature water input operation of the refrigeration cycle device of embodiment 1.
Figure 11 is a P-h diagram of the high temperature water input operation of the refrigeration cycle device of embodiment 1.
Figure 12 shows an example of a relationship between positions and temperatures of refrigerant and water in the first condensers and the second condenser of the refrigeration cycle device of embodiment 1.
Figure 13 shows a comparison between compressor discharge temperatures of an R410A refrigerant and an R32 refrigerant.
Figure 14 shows a relationship between a ratio between the numbers of refrigerant flow paths and a magnitude of refrigerant pressure loss in the first condenser.
Figure 15 shows a low temperature water input operation of a refrigeration cycle device according to embodiment 2 of the present invention.
Figure 16 shows a high temperature water input operation of the refrigeration cycle device of embodiment 2.
Figure 17 shows a middle temperature water input operation of a refrigeration cycle device according to embodiment 3 of the present invention.
Figure 18 is a flowchart showing a control operation of the refrigeration cycle device of embodiment 3.
Figure 19 shows a relationship between a feed-water temperature and a bypass percentage in the middle temperature water input operation of the refrigeration cycle device of embodiment 3.

Description of Embodiments

[0011] Now, with reference to the drawings, embodiments of the present invention will be described. Throughout the drawings, like components are denoted by like reference numerals, and overlapping descriptions will be omitted. The present invention includes any combinations of embodiments described below.

Embodiment 1

[0012] Figure 1 is a configuration diagram of a refrigeration cycle device according to embodiment 1 of the present invention. As shown in Figure 1, a refrigeration cycle device 1A of this embodiment 1 includes a refrigerant circuit including a compressor 2, first condensers 3A, 3B, a second condenser 4, an expansion valve 5, an evaporator 6, and an evaporator 7 connected by a refrigerant piping. The refrigeration cycle device 1A further includes a heat medium path 9, a second condenser bypass passage 10, a flow path switching valve 11, a blower 12 for blowing air into the evaporator 6, an entry heat medium temperature sensor 13, and a control device 50 for controlling an operation of the refrigeration cycle device 1A. The refrigeration cycle device 1A of this embodiment 1 functions as a heat pump for heating a liquid heat medium. Although the heat medium in this embodiment 1 is water, the heat medium in the present invention may be antifreeze, brine, or the like. Although the refrigeration cycle device 1A of this embodiment 1 is used as a hot water supply device, the refrigeration cycle device according to the present invention may be used for heating a heat medium for applications other than hot water supply (such as an indoor heating). In the description below, for simplicity of description, specific enthalpy [kJ/kg] is simply referred to as enthalpy.

[0013] The two first condensers 3A, 3B have the same configuration and are connected in parallel. The first condensers 3A, 3B each include a refrigerant flow path 31 and a heat medium flow path 32. The second condenser 4 includes a refrigerant flow path 41 and a heat medium flow path 42. The compressor 2 compresses a low pressure refrigerant gas into a high pressure refrigerant gas. The high pressure refrigerant gas compressed by the compressor 2 is divided to flow into the refrigerant flow path 31 in the first condenser 3A and the refrigerant flow path 31 in the first condenser 3B. Streams of the high pressure refrigerant having passed through the first condensers 3A, 3B merge and flow into the refrigerant flow path 41 in the second condenser 4. The first condensers 3A, 3B function as one condenser. In the present invention, the first condensers 3A, 3B may be integrated.

[0014] The expansion valve 5 is a pressure reducing device for reducing pressure of and expanding the high pressure refrigerant. An opening of the expansion valve 5 is preferably changeable. The high pressure refrigerant having passed through the refrigerant flow path 41 in the second condenser 4 is reduced in pressure and expanded by the expansion valve 5 into a low pressure refrigerant. The low pressure refrigerant flows into the evaporator 6.

[0015] The evaporator 6 is a heat exchanger for exchanging heat between refrigerant and air. The evaporator 6 causes the refrigerant to absorb heat from outside air blown in by the blower 12. A heat source of the evaporator 6 in this embodiment 1 is outside air. However, the heat source of the evaporator in the present invention is not limited to the outside air, but may be, for example, waste heat, underground heat, groundwater, solar hot water or the like. Also, in the present invention, a fluid cooled by the evaporator may be used for an indoor cooling or the like.

[0016] The low pressure refrigerant having passed through the evaporator 6 flows into the accumulator 7. Out of the refrigerant having flowed into the accumulator 7, a refrigerant liquid is stored in the accumulator 7, while a refrigerant gas flows out of the accumulator 7 and is sucked into the compressor 2. In the refrigerant circuit as described above, generally, a section before the high pressure refrigerant compressed by the compressor 2 flows into the pressure reducing device is referred to as a "high pressure side", and a section before the low pressure refrigerant reduced in pressure by the pressure reducing device is sucked into the compressor 2 is referred to as a "low pressure side".

[0017] The heat medium path 9 allows water to pass through the heat medium flow path 42 in the second condenser 4 and the heat medium flow paths 32 in the first condensers 3A, 3B in this order. The heat medium path 9 connects a water inlet 91 and an inlet of the heat medium flow path 42 in the second condenser 4, connects an outlet of the heat medium flow path 42 in the second condenser 4 and inlets of the heat medium flow paths 32 in the first condensers 3A, 3B, and connects outlets of the heat medium flow paths 32 in the first condensers 3A, 3B and a water outlet 92. In the first condensers 3A, 3B, the refrigerant and the water form counter flows. In the second condenser 4, the refrigerant and the water form counter flows.

[0018] The second condenser bypass passage 10 bypasses the heat medium flow path 42 in the second condenser 4. The flow path switching valve 11 is a three-way valve. The flow path switching valve 11 is provided in a middle of the heat medium path 9 between the water inlet 91 and the inlet of the heat medium flow path 42 in the second condenser 4. One end of the second condenser bypass passage 10 is connected to the flow path switching valve 11, and the other end of the second condenser bypass passage 10 is connected in a middle of the heat medium path 9 between the outlet of the heat medium flow path 42 in the second condenser 4 and the inlets of the heat medium flow paths 32 in the first condensers 3A, 3B.

[0019] The flow path switching valve 11 can be switched between a state where all of water having flowed in from the water inlet 91 is allowed to flow to the heat medium flow path 42 in the second condenser 4, and a state where all of water having flowed in from the water inlet 91 is allowed to flow to the second condenser bypass passage 10. The flow path switching valve 11 may be able to change a rate of distribution of the water having flowed in from the water inlet 91 to the heat medium flow path 42 in the second condenser 4 and the second condenser bypass passage 10. In this embodiment 1, out of the total flow of water flowing in from the water inlet 91, a percentage of water flowing through the second condenser bypass passage 10 rather than the second condenser 4 is referred to as a "bypass percentage". In this embodiment 1, the flow path switching valve 11 corresponds to a flow path controlling element capable of varying a bypass rate that is a flow rate of water flowing through the second condenser bypass passage 10.

[0020] The entry heat medium temperature sensor 13 is provided in the middle of the heat medium path 9 between the water inlet 91 and the flow path switching valve 11. The entry heat medium temperature sensor 13 detects a temperature of a heat medium, that is, water before heat exchange with the refrigerant. Hereinafter, a temperature detected by the entry heat medium temperature sensor 13 is referred to as an "feed-water temperature".

[0021] The control device 50 is control means for controlling an operation of the refrigeration cycle device 1A. The compressor 2, the expansion valve 5, the flow path switching valve 11, the blower 12, and the entry heat medium temperature sensor 13 are electrically connected to the control device 50. Besides, actuators, sensors, a user interface device, or the like may be further connected to the control device 50. The control device 50 has a processor 50a and a memory 50b that stores a control program and data or the like. The control device 50 controls operations of the compressor 2, the expansion valve 5, the flow path switching valve 11, and the blower 12 according to the program stored in the memory 50b based on information detected by each sensor, instruction information from the user interface device, or the like, to control the operation of the refrigeration cycle device 1A.

[0022] In this embodiment 1, R32 is used as the refrigerant. An advantage of using R32 as the refrigerant will be described later.

[0023] Figure 2 is a perspective view showing a part of a heat exchanger that constitutes the first condensers 3A, 3B and the second condenser 4. As shown in Figure 2, a heat exchanger 60 includes one twisted pipe 61 and three refrigerant heat transfer pipes 62, 63, 64. An inside of the twisted pipe 61 constitutes a heat medium flow path. Specifically, water flows through the twisted pipe 61. An inside of each of the refrigerant heat transfer pipes 62, 63, 64 constitutes a refrigerant flow path. The refrigerant is divided to flow through the three refrigerant heat transfer pipes 62, 63, 64 in parallel. In Figure 2, for easy distinction among the refrigerant heat transfer pipes 62, 63, 64, the refrigerant heat transfer pipes 62, 64 are hatched for convenience. Specifically, the hatching in Figure 2 does not show a cross section. The twisted pipe 61 has three parallel helical grooves 61a, 61b, 61c in an outer periphery thereof. The refrigerant heat transfer pipes 62, 63, 64 are fitted in the grooves 61a, 61 b, 61c, respectively, and wound into a helical along shapes of the grooves 61a, 61b, 61c. Such a configuration can increase a contact heat transfer area between the twisted pipe 61 and the refrigerant heat transfer pipes 62, 63,64.

[0024] The first condenser 3A, the first condenser 3B, and the second condenser 4 are each constituted by a heat exchanger having substantially the same structure as the heat exchanger 60 described above. Specifically, the first condenser 3A, the first condenser 3B, and the second condenser 4 each include one heat medium flow path and three refrigerant flow paths. In Figure 1, for simplicity, the heat medium flow path in each of the first condenser 3A, the first condenser 3B, and the second condenser 4 is shown by one line.

[0025] As described above, the first condensers 3A, 3B function as one condenser. The first condensers 3A, 3B are constituted by two heat exchangers 60 connected in parallel. Thus, the first condensers 3A, 3B as a whole have two heat medium flow paths and six refrigerant flow paths. A sectional area of the refrigerant flow path in the second condenser 4 is smaller than a sectional area of the refrigerant flow path in the first condensers 3A, 3B. The reason therefor will be described later. When the refrigerant flow path in the condenser is ramified into a plurality of paths, "the sectional area of the refrigerant flow path in the condenser" is a sum of sectional areas of the plurality of refrigerant flow paths. Specifically, the sectional area of the refrigerant flow path in the first condensers 3A, 3B is a sum of sectional areas of six refrigerant flow paths, and the sectional area of the refrigerant flow path in the second condenser 4 is a sum of sectional areas of three refrigerant flow paths. If a sectional area of one refrigerant flow path in the first condensers 3A, 3B is equal to a sectional area of one refrigerant flow path in the second condenser 4, in this embodiment 1, the sectional area of the refrigerant flow path in the second condenser 4 is one-half of the sectional area of the refrigerant flow path in the first condensers 3A, 3B.

[0026] The first condenser and the second condenser in the present invention are not limited to the twisted pipe type heat exchanger as described above, but may be a heat exchanger of a different type such as a plate type heat exchanger. The numbers of the refrigerant flow paths and the heat medium flow paths are not limited to those in the above example.

[0027] Figure 3 is a configuration diagram of a hot water storage type hot water supply system including the refrigeration cycle device 1A of this embodiment 1 and a tank unit 20. As shown in Figure 3, a hot water storage tank 21 and a water pump 22 are provided in the tank unit 20. The refrigeration cycle device 1A and the hot water storage tank 21 are connected by water channels 23, 24. The refrigeration cycle device 1A and the tank unit 20 are connected by electric wiring (not shown). One end of the water channel 23 is connected to the water inlet 91 of the refrigeration cycle device 1A. The other end of the water channel 23 is connected to a lower part of the hot water storage tank 21 in the tank unit 20. A water pump 22 is provided in a middle of the water channel 23 in the tank unit 20. One end of the water channel 24 is connected to the water outlet 92 of the refrigeration cycle device 1A. The other end of the water channel 24 is connected to an upper part of the hot water storage tank 21 in the tank unit 20. Instead of the shown configuration, the water pump 22 may be placed in the refrigeration cycle device 1A.

[0028] A water supply pipe 25 is further connected to the lower part of the hot water storage tank 21 in the tank unit 20. Water supplied from an external water source such as waterworks flows through the water supply pipe 25 into the hot water storage tank 21 and is stored. The water from the water supply pipe 25 flows into the hot water storage tank 21, which is always kept filled with water. A hot water supplying mixing valve 26 is further provided in the tank unit 20. The hot water supplying mixing valve 26 is connected to the upper part of the hot water storage tank 21 by a hot water pipe 27. Also, a water supply branch pipe 28 branching off from the water supply pipe 25 is connected to the hot water supplying mixing valve 26. One end of a hot water supply pipe 29 is further connected to the hot water supplying mixing valve 26. The other end of the hot water supply pipe 29 is connected to a hot water supply terminal such as a tap, a shower, or a bathtub, although not shown.

[0029] In a heat accumulating operation for increasing a heat storage amount of the hot water storage tank 21, the water stored in the hot water storage tank 21 is fed by the water pump 22 through the water channel 23 to the refrigeration cycle device 1A, and heated in the refrigeration cycle device 1A into high temperature hot water. The high temperature hot water generated in the refrigeration cycle device 1A returns through the water channel 24 to the tank unit 20, and flows into the hot water storage tank 21 from the upper part. By the heat accumulating operation, hot water is stored in the hot water storage tank 21 so as to form a temperature stratification with a high temperature upper side and a low temperature lower side.

[0030] When hot water is supplied from the hot water supply pipe 29 to the hot water supply terminal, high temperature hot water in the hot water storage tank 21 is supplied through the hot water pipe 27 to the hot water supplying mixing valve 26, and low temperature water is supplied through the water supply branch pipe 28 to the hot water supplying mixing valve 26. The high temperature hot water and the low temperature water are mixed by the hot water supplying mixing valve 26, and then supplied through the hot water supply pipe 29 to the hot water supply terminal. The hot water supplying mixing valve 26 adjusts a mixing ratio between the high temperature hot water and the low temperature water so as to achieve a hot water supply temperature set by a user.

[0031] A reheating heat exchanger 30 for reheating a bathtub is further provided in the tank unit 20. Pipes for circulating bathtub water to the reheating heat exchanger 30, and pipes for switching connection of the water channels 23, 24 from the hot water storage tank 21 to the reheating heat exchanger 30 are provided in the tank unit 20, although not shown. In a bathtub reheating operation, the pipes are used to circulate the bathtub water and the high temperature hot water generated in the refrigeration cycle device 1A to the reheating heat exchanger 30 and exchange heat therebetween, thereby increasing a temperature of an inside of the bathtub.

[0032] Figure 4 is a flowchart showing a control operation in the refrigeration cycle device 1A of this embodiment 1. In step S1 in Figure 4, the control device 50 compares a feed-water temperature detected by the entry heat medium temperature sensor 13 with a previously set reference temperature α. In this embodiment 1, the reference temperatureα is 50°C. If the feed-water temperature is lower than the reference temperature α in step S1, the control device 50 moves to step S2. In step S2, the refrigeration cycle device 1A performs a low temperature water input operation. On the other hand, if the feed-water temperature is not lower than the reference temperature α in step S1, the control device 50 moves to step S3. In step S3, the refrigeration cycle device 1A performs a high temperature water input operation. The control device 50 controls an operation of the flow path switching valve 11 so that a bypass rate in the high temperature water input operation is larger than a bypass rate in the low temperature water input operation. In this embodiment 1, a bypass percentage in the low temperature water input operation is 0%. Specifically, in step S2, the control device 50 controls the operation of the flow path switching valve 11 so that all of water flowing in from the water inlet 91 flows through the second condenser 4. In this embodiment 1, a bypass percentage in the high temperature water input operation is 100%. Specifically, in step S3, the control device 50 controls the operation of the flow path switching valve 11 so that all of water flowing in from the water inlet 91 flows through the second condenser bypass passage 10 rather than the second condenser 4.

[0033] In order to prevent frequent switching between the low temperature water input operation and the high temperature water input operation when the feed-water temperature is close to the reference temperature α, two reference temperatures may be set to provide hysteresis to switching between the low temperature water input operation and the high temperature water input operation.

[0034] If the low temperature water supplied from the water supply pipe 25 is located on a lower side in the hot water storage tank 21, the feed-water temperature in a heat accumulating operation is similar to the outside air temperature. The reference temperature α is higher than the outside air temperature. Thus, in the heat accumulating operation when the low temperature water supplied from the water supply pipe 25 is located on the lower side in the hot water storage tank 21, the feed-water temperature is lower than the reference temperature α, and thus the refrigeration cycle device 1A performs the low temperature water input operation.

[0035] On the other hand, in a heat accumulating operation for reheating hot water in the hot water storage tank 21 reduced in temperature by thermal dissipation or the like, the feed-water temperature may be higher than the reference temperature α. Also in the bathtub reheating operation, the feed-water temperature may be higher than the reference temperature α. In such cases, the refrigeration cycle device 1A performs the high temperature water input operation.

[0036] Figure 5 shows the low temperature water input operation of the refrigeration cycle device 1A of this embodiment 1. In the low temperature water input operation, the water having flowed in from the water inlet 91 is heated in the second condenser 4 and then divided into two streams to flow through the first condensers 3A, 3B in parallel and further heated.

[0037] The refrigerant flows out of the compressor 2 and is then divided into two streams to flow through the first condensers 3A, 3B in parallel. Immediately before an inlet of a heat transfer portion in the first condenser 3A, the refrigerant is further divided to flow into the three refrigerant flow paths. Similarly, immediately before an inlet of a heat transfer portion in the first condenser 3B, the refrigerant is further divided to flow into the three refrigerant flow paths. In the first condensers 3A, 3B, the refrigerant is partially condensed into a gas-liquid two-phase state. Streams of the refrigerant having passed through the first condensers 3A, 3B merge and then flow to the second condenser 4. Immediately before an inlet of a heat transfer portion in the second condenser 4, the refrigerant is divided to flow into the three refrigerant flow paths. The refrigerant is further condensed in the second condenser 4.

[0038] Figure 6 shows an example of changes in temperature of the refrigerant and the water in the first condensers 3A, 3B and the second condenser 4 in the low temperature water input operation of the refrigeration cycle device 1A of this embodiment 1. In Figure 6, the abscissa represents enthalpy of the refrigerant, and the ordinate represents temperature. In this example, a temperature difference at a pinch point where a temperature difference between the refrigerant and the water is minimum is about 3 K. When a feed-water temperature is 9°C, a condensation temperature of the refrigerant is 62°C (at saturation pressure of 4.11 MPa), and a temperature of a refrigerant gas at inlets of the first condensers 3A, 3B is 126°C, a water temperature at outlets of the first condensers 3A, 3B is 80°C, and a temperature of a refrigerant liquid at an outlet of the second condenser 4 is 12°C. As such, with the refrigeration cycle device 1A of this embodiment 1, hot water of 80°C can be produced at high pressure side pressure of 4.25 MPa or lower that is design pressure for a typical air conditioner. Thus, specifications of the compressor 2 may be common to those of the air conditioner, thereby reducing cost. In the description below, a water temperature at the outlets of the first condensers 3A, 3B is referred to as an "output hot water temperature".

[0039] Figure 7 shows a P-h diagram, that is, a Mollier diagram of the low temperature water input operation of the refrigeration cycle device 1A of this embodiment 1. As shown in Figure 7, in the low temperature water input operation, a low pressure refrigerant gas is compressed by the compressor 2 from a point E1 to a point A1 into a high pressure refrigerant gas. The high pressure refrigerant gas is cooled in the first condensers 3A, 3B from the point A1 to a point B1, and starts to condense during that time. The point B1 is a gas-liquid two-phase state. The high pressure refrigerant in the gas-liquid two-phase state is further condensed in the second condenser 4 into a supercooled liquid. Specifically, the high pressure refrigerant is changed from the point B1 to the point C1 in the second condenser 4. The refrigerant in the supercooled liquid state of the point C1 is expanded and reduced in pressure to a point D1 by the expansion valve 5 into a low pressure refrigerant in the gas-liquid two-phase state. The low pressure refrigerant in the gas-liquid two-phase state absorbs heat in the evaporator 6 from the point D1 to a point E1 so as to evaporate.

[0040] In the low temperature water input operation, the refrigerant is supercooled to a temperature close to the feed-water temperature, and thus an enthalpy difference of the refrigerant is increased, thereby increasing COP of the refrigeration cycle device 1A. Figure 8 shows an example of a relationship between the outside air temperature and the feed-water temperature in the low temperature water input operation. The example of the feed-water temperature of 9°C in Figure 5 corresponds to a case of the outside air temperature of 7°C. The feed-water temperature also increases with increasing outside air temperature.

[0041] Figure 9 shows the high temperature water input operation of the refrigeration cycle device 1A of this embodiment 1. In the high temperature water input operation, the water having flowed in from the water inlet 91 passes through the second condenser bypass passage 10 rather than the second condenser 4, and is divided into two streams to flow through the first condensers 3A, 3B in parallel and heated. In the high temperature water input operation, the refrigerant flows along the same path as in the low temperature water input operation. However, heat exchange with water is not performed in the second condenser 4, and thus the refrigerant is not condensed in the second condenser 4.

[0042] Figure 10 shows an example of changes in temperature of the refrigerant and the water in the first condensers 3A, 3B in the high temperature water input operation of the refrigeration cycle device 1A of this embodiment 1. In Figure 10, the abscissa represents enthalpy of the refrigerant, and the ordinate represents temperature. In this example, a temperature difference at a pinch point where a temperature difference between the refrigerant and the water is minimum is about 3 K. When a feed-water temperature is 50°C, a condensation temperature of the refrigerant is 62°C (at saturation pressure of 4.11 MPa), and a temperature of a refrigerant gas at the inlets of the first condensers 3A, 3B is 126°C, a water temperature at the outlets of the first condensers 3A, 3B, that is, an output hot water temperature, is 80°C.

[0043] Figure 11 is a P-h diagram of the high temperature water input operation of the refrigeration cycle device 1A of this embodiment. As shown in Figure 11, in the high temperature water input operation, a low pressure refrigerant gas is compressed by the compressor 2 from a point E2 to a point A2 into a high pressure refrigerant gas. The high pressure refrigerant gas is cooled in the first condensers 3A, 3B from the point A2 to a point B2, and starts to condense during that time. The point B2 is a gas-liquid two-phase state. In the second condenser 4, water does not flow and heat exchange is not performed. Thus, in the second condenser 4, the refrigerant is not reduced in enthalpy but is reduced in pressure due to pressure loss. Specifically, the refrigerant is changed from the point B2 to the point C2 in the second condenser 4. The refrigerant in the gas-liquid two-phase state at the point C2 is expanded to a point D2 and reduced in pressure by the expansion valve 5 into a low pressure refrigerant in the gas-liquid two-phase state. The low pressure refrigerant in the gas-liquid two-phase state absorbs heat in the evaporator 6 from the point D2 to a point E2 so as to evaporate.

[0044] Average refrigerant dryness from the point B2 to the point C2 of the second condenser 4 in the high temperature water input operation is higher than average refrigerant dryness from the point B1 to the point C1 of the second condenser 4 in the low temperature water input operation. Thus, an average refrigerant density in the second condenser 4 in the high temperature water input operation is lower than an average refrigerant density in the second condenser 4 in the low temperature water input operation. Also, average refrigerant dryness from the point D2 to the point E2 of the evaporator 6 in the high temperature water input operation is higher than average refrigerant dryness from the point D1 to the point E1 of the evaporator 6 in the low temperature water input operation. Thus, an average refrigerant density in the evaporator 6 in the high temperature water input operation is lower than an average refrigerant density in the evaporator 6 in the low temperature water input operation. For these reasons, in the high temperature water input operation, an amount of the refrigerant required for the second condenser 4 and the evaporator 6 is smaller than in the low temperature water input operation, thereby producing a surplus of the refrigerant in the refrigerant circuit. In the high temperature water input operation, the redundant refrigerant is stored as a refrigerant liquid in the accumulator 7. As such, in this embodiment 1, the accumulator 7 corresponds to a storage portion configured to store redundant refrigerant. However, in the present invention, a liquid receiver (not shown) provided between the second condenser 4 and the expansion valve 5 may be used as the storage portion, the evaporator 6 may also serve as the storage portion, or the redundant refrigerant may be stored in two or more of the accumulator 7, the liquid receiver, and the evaporator 6.

[0045] Figure 12 shows an example of a relationship between positions and temperatures of the refrigerant and the water in the first condensers 3A, 3B and the second condenser 4 of the refrigeration cycle device 1A of this embodiment 1. In Figure 12, the ordinate represents temperature. In Figure 12, the abscissa represents a distance ratio from a water inlet of the second condenser 4 when a sum of a length of one heat medium flow path in the first condensers 3A, 3B and a length of one heat medium flow path in the second condenser 4 is one. Here, the length of the heat medium flow path is a length of a central axis in a flowing direction of the heat medium flow path. An operation condition in the example in Figure 12 is the same as an operation condition in Figure 6 or 10 described above.

[0046] In the example in Figure 12, Lp1:Lp2 = 0.55:0.45, where Lp1 is the length of one heat medium flow path in the first condensers 3A, 3B and Lp2 is the length of one heat medium flow path in the second condenser 4. In this embodiment 1, there are two heat medium flow paths in the first condensers 3A, 3B and one heat medium flow path in the second condenser 4. Thus, L1:L2 = 1.10:0.45 $\approx$ 2.4:1.0, where L1 is a total length of the heat medium flow paths in the first condensers 3A, 3B and L2 is a total length of the heat medium flow path in the second condenser 4.

[0047] With the ratio between the length of the heat medium flow path in the first condensers 3A, 3B and the length of the heat medium flow path in the second condenser 4 as described above, in the low temperature water input operation at a feed-water temperature of, for example, 9°C, as shown in Figure 12, water can be heated from 9°C to 50°C by the second condenser 4 and then heated from 50°C to 80°C by the first condensers 3A, 3B. In the high temperature water input operation at a feed-water temperature of, for example, 50°C, water can be heated from 50°C to 80°C by the first condensers 3A, 3B.

[0048] With the refrigeration cycle device 1A of this embodiment 1, advantages can be obtained as described below.

(Advantage 1)

[0049] In the low temperature water input operation, the refrigerant is supercooled in the second condenser 4 and the refrigerant temperature at the outlet of the second condenser 4 is reduced to increase an enthalpy difference, thereby increasing COP. The refrigerant in the supercooled liquid state has a low flow speed and a lower heat-transfer coefficient than a gas-liquid two-phase part by its nature. However, in this embodiment 1, the sectional area of the refrigerant flow path in the second condenser 4 is smaller than the sectional area of the refrigerant flow path in the first condensers 3A, 3B, thereby inhibiting a reduction in flow speed of the refrigerant in the supercooled liquid state in the second condenser 4 and thus inhibiting a reduction in heat-transfer coefficient. Thus, in the low temperature water input operation, heat exchange efficiency in the second condenser 4 can be increased to further increase COP. In particular, in this embodiment 1, the number of the refrigerant flow paths in the second condenser 4 is smaller than the number of refrigerant flow paths in the first condensers 3A, 3B, thereby more reliably preventing a reduction in heat-transfer coefficient of the refrigerant in the second condenser 4.

(Advantage 2)

[0050] In the high temperature water input operation, the refrigerant in the second condenser 4 is in the gas-liquid two-phase state or a gas state, thereby increasing a flow speed as compared to a supercooled liquid. Thus, the pressure loss of the refrigerant in the second condenser 4 in the high temperature water input operation is larger than the pressure loss of the refrigerant in the second condenser 4 in the low temperature water input operation. At this time, the sectional area of the refrigerant flow path is smaller and the number of refrigerant flow paths is smaller in the second condenser 4 than in the first condensers 3A, 3B, which is likely to increase the pressure loss of the refrigerant. Thus, in the high temperature water input operation, the refrigerant in the second condenser 4 is reduced in temperature due to the pressure loss. This reduces a temperature difference between the refrigerant and the water, thereby reducing a heat exchange rate at constant pressure. If the pressure loss of the refrigerant further increases in the second condenser 4, a part where the refrigerant temperature is lower than the feed-water temperature is created. In a part where the refrigerant temperature is lower than the water temperature, the refrigerant draws heat from the water to cause loss of heat. This reduces efficiency of the refrigeration cycle device 1A heating the water. However, in this embodiment 1, the water is not passed through the second condenser 4 in the high temperature water input operation, and thus even if the part where the refrigerant temperature is lower than the feed-water temperature is created in the second condenser 4, the refrigerant can be reliably inhibited from drawing heat from the water, thereby inhibiting loss of heat. This can reliably inhibit a reduction in efficiency of the refrigeration cycle device 1A heating the water. Further, the sectional area of the refrigerant flow path is larger and the number of refrigerant flow paths is larger in the first condensers 3A, 3B than in the second condenser 4, thereby causing smaller refrigerant pressure loss. Thus, in the first condensers 3A, 3B, a sufficient heat exchange rate can be ensured without increasing condensation pressure even in the high temperature water input operation at a high feed-water temperature.

[0051] Also in this embodiment 1, R32 is used as the refrigerant to provide an advantage described below.

(Advantage 3)

[0052] Figure 13 shows a comparison between compressor discharge temperatures of an R410A refrigerant and an R32 refrigerant. In the example in Figure 13, compressor suction pressure is 0.81 MPa that is saturation vapor pressure of R32 at 0°C, compressor discharge pressure is 4.25 MPa equal to design pressure of the air conditioner, a degree of superheat of the refrigerant sucked into the compressor 2 is 0 K, and compressor efficiency is assumed to be 100%. Under these conditions, the compressor discharge temperature of R410A is 91°C, while the compressor discharge temperature of R32 is 110°C. The degree of superheat refers to a rise in temperature from an evaporation temperature, that is, a saturation temperature. In the high temperature water input operation, the redundant refrigerant liquid is stored in the accumulator 7 as described above, and thus the degree of superheat of the refrigerant sucked into the compressor 2 is 0 K (or 0 K or less). When the degree of superheat of the refrigerant sucked into the compressor 2 is 0 K, the R410A refrigerant is reduced in compressor discharge temperature to 91°C as described above. Thus, if R410A is used as the refrigerant, it is difficult to increase the output hot water temperature in the high temperature water input operation. However, for the R32 refrigerant, even if the degree of superheat of the refrigerant sucked into the compressor 2 is 0 K, the compressor discharge temperature can be increased to 110°C. Thus, using R32 as the refrigerant can increase the output hot water temperature in the high temperature water input operation to be higher than when using the R410A refrigerant. This can increase a heat storage amount with the same capacity of the hot water storage tank 21. In the refrigeration cycle device 1A of this embodiment 1, when R32 is used as the refrigerant and the design pressure is substantially equal to that of the air conditioner, the output hot water temperature is about 80°C maximum. Thus, the hot water storage temperature in the hot water storage tank 21 is also about 80°C maximum. An output hot water temperature of a heat pump hot water supply device using CO_2 as the refrigerant is about 90°C maximum, and the hot water storage temperature is also about 90°C maximum. Thus, with the same capacity of the hot water storage tank 21, the heat storage amount of the heat pump hot water supply device using the CO_2 refrigerant is larger. However, since the temperature of hot water supplied from the hot water supply pipe 29 to the hot water supply terminal is about 40 to 60°C, there is no problem in the hot water storage temperature of 80°C. In the refrigeration cycle device 1A of this embodiment 1, also for the high temperature water input operation at the feed-water temperature of about 50°C or higher, an efficient operation can be performed with the output hot water temperature of 80°C or higher. Thus, if the hot water storage temperature and the heat storage amount are reduced by thermal dissipation from the hot water storage tank 21 or the like, a heat accumulating operation of the high temperature water input operation by the refrigeration cycle device 1A can be performed to efficiently reheat the hot water reduced in temperature in the hot water storage tank 21. Also, a critical temperature of CO_2 is about 31°C, while a critical temperature of R32 is about 78°C and high. Thus, with the refrigeration cycle device 1A of this embodiment 1, condensation latent heat of the refrigerant can be used even in the high temperature water input operation, thereby allowing an operation with high COP. Also, since too high a hot water storage temperature increases thermal dissipation from the hot water storage tank 21 to outside air, heat loss can be reduced by storing hot water at 80°C rather than at 90°C and again operating the heat accumulating operation when the heat storage amount is reduced. In the present invention, not only when refrigerant containing 100% R32 is used but also when refrigerant mainly containing R32 is used, similar advantages as described above can be obtained. When refrigerant mainly containing R32 is used, refrigerant containing 70 mass % or more, more preferably, 90 mass % or more of R32 may be used.

[0053] Here, a ratio of the number of refrigerant flow paths in the first condenser to the number of refrigerant flow paths in the second condenser is defined as a ratio between the numbers of refrigerant flow paths. As described above, in this embodiment 1, the number of the refrigerant flow paths in the first condensers 3A, 3B is six, and the number of the refrigerant flow paths in the second condenser 4 is three, and thus the ratio between the numbers of refrigerant flow paths is two. Figure 14 shows a relationship between the ratio between the numbers of refrigerant flow paths and a magnitude of refrigerant pressure loss in the first condenser. In Figure 14, the ordinate represents the magnitude of refrigerant pressure loss in the first condenser, which is 100% when the ratio between the numbers of refrigerant flow paths is one. As shown in Figure 14, the pressure loss of the refrigerant in the first condenser decreases with increasing ratio between the numbers of refrigerant flow paths. However, if the ratio between the numbers of refrigerant flow paths exceeds 2.5, further reducing the pressure loss of the refrigerant is less effective. With too high a ratio between the numbers of refrigerant flow paths, the reduction in refrigerant flow speed reduces the heat-transfer coefficient, which may reduce the heat exchange rate. From the above, the ratio between the numbers of refrigerant flow paths is desirably about 1.5 to 2.5, and as in this embodiment 1, the ratio between the numbers of refrigerant flow paths is particularly desirably two. Also, in this embodiment 1, the first condensers 3A, 3B and the second condenser 4 are constituted by heat exchangers having substantially the same structure. Specifically, two heat exchangers having substantially the same structure as the second condenser 4 are connected in parallel to constitute the first condensers 3A, 3B. Thus, the above advantage can be achieved with an easy design.

[0054] In the low temperature water input operation in this embodiment 1, a water bypass percentage is 0% and all water is heated in the second condenser 4, thereby increasing the output hot water temperature. However, in the present invention, the water bypass percentage does not need to be always 0%, but a small amount of water may be passed through the second condenser bypass passage 10 in the low temperature water input operation. In the high temperature water input operation in this embodiment 1, the water bypass percentage is 100% and all water flows through the second condenser bypass passage 10, thereby reliably preventing the water from drawing heat from the refrigerant in the second condenser 4. However, in the present invention, the water bypass percentage does not need to be always 100% in the high temperature water input operation, but a small amount of water may be passed through the second condenser 4.

[0055] In the present invention, a first reference temperature and a second reference temperature higher than the first reference temperature may be set, and the control device 50 may control the operation of the flow path switching valve 11 so that the bypass percentage is 0% when the feed-water temperature is lower than the first reference temperature, the bypass percentage is 100% when the feed-water temperature is higher than the second reference temperature, and the bypass percentage continuously or stepwise increases with increasing feed-water temperature when the feed-water temperature is between the first reference temperature and the second reference temperature. This allows smooth transition between the low temperature water input operation and the high temperature water input operation.

Embodiment 2

[0056] Next, with reference to Figures 15 and 16, embodiment 2 of the present invention will be described. Differences from embodiment 1 described above will be mainly described, and like or corresponding components are denoted by like reference numerals and descriptions thereof will be omitted.

[0057] Figure 15 is a configuration diagram of a refrigeration cycle device according to embodiment 2 of the present invention. As compared to the refrigeration cycle device 1A of embodiment 1, a refrigeration cycle device 1B of this embodiment 2 shown in Figure 15 includes a second condenser bypass passage 16 and a bypass valve 17 rather than the second condenser bypass passage 10 and the flow path switching valve 11. The second condenser bypass passage 16 bypasses a refrigerant flow path 41 in a second condenser 4. One end of the second condenser bypass passage 16 is connected to a refrigerant pipe between refrigerant flow paths 31 in first condensers 3A, 3B and the refrigerant flow path 41 in the second condenser 4. The other end of the second condenser bypass passage 16 is connected to a refrigerant pipe between an expansion valve 5 and an evaporator 6. The bypass valve 17 is provided in a middle of the second condenser bypass passage 16 and opens/closes the second condenser bypass passage 16. The bypass valve 17 also functions as a pressure reducing device for reducing pressure of and expanding a high pressure refrigerant. The bypass valve 17 preferably has a changeable opening. An entry heat medium temperature sensor 13 is provided in a middle of a heat medium path 9 between a water inlet 91 and the second condenser 4.

[0058] In this embodiment 2, out of a total flow of the refrigerant having passed through the first condensers 3A, 3B, a percentage of the refrigerant flowing through the second condenser bypass passage 16 rather than the second condenser 4 is referred to as a "bypass percentage". In this embodiment 2, the expansion valve 5 and the bypass valve 17 correspond to a flow path controlling element that can vary a bypass rate that is a flow rate of the refrigerant flowing through the second condenser bypass passage 16. Also, in this embodiment 2, all of water having flowed in from the water inlet 91 flows through the second condenser 4 both in a low temperature water input operation and in a high temperature water input operation.

[0059] The refrigeration cycle device 1B performs the low temperature water input operation when a feed-water temperature is lower than a reference temperature α, and performs the high temperature water input operation when the feed-water temperature is higher than the reference temperature α. The reference temperature α is 50°C as in embodiment 1. The control device 50 controls operations of the expansion valve 5 and the bypass valve 17 so that a bypass rate in the high temperature water input operation is larger than a bypass rate in the low temperature water input operation. In this embodiment 2, the bypass percentage in the low temperature water input operation is 0%, and the bypass percentage in the high temperature water input operation is 100% for description.

[0060] Figure 15 shows the low temperature water input operation of the refrigeration cycle device 1B of this embodiment 2. When the low temperature water input operation is performed, the control device 50 closes the bypass valve 17 to an opening that prevents the refrigerant from flowing. This causes all of refrigerant having passed through the first condensers 3A, 3B to flow through the second condenser 4 and the expansion valve 5 to the evaporator 6. The low temperature water input operation of the refrigeration cycle device 1B is substantially the same as the low temperature water input operation of the refrigeration cycle device 1A of embodiment 1.

[0061] Figure 16 shows the high temperature water input operation of the refrigeration cycle device 1B of this embodiment 2. As shown in Figure 16, when the high temperature water input operation is performed, the control device 50 opens the bypass valve 17, and closes the expansion valve 5 to an opening that prevents the refrigerant from flowing. Thus, all of refrigerant having passed through the first condensers 3A, 3B flows through the second condenser bypass passage 16 rather than the second condenser 4. The high pressure refrigerant having passed through the first condensers 3A, 3B into the second condenser bypass passage 16 is expanded and reduced in pressure by the bypass valve 17, and flows toward the evaporator 6. In the high temperature water input operation, water flows through the second condenser 4, while the refrigerant does not flow through the second condenser 4, and thus the water is not changed in temperature in the second condenser 4.

[0062] With the refrigeration cycle device 1B of this embodiment 2, similar advantages as in embodiment 1 can be obtained. Specifically, according to this embodiment 2, the refrigerant does not flow through the second condenser 4 in the high temperature water input operation, and thus a part where the refrigerant temperature is lower than the feed-water temperature can be reliably inhibited from being created in the second condenser 4. This can reliably inhibit the refrigerant from drawing heat from water, and thus reliably inhibiting a reduction in efficiency of the refrigeration cycle device 1B heating the water. Also, in the high temperature water input operation, the refrigerant in the gas-liquid two-phase state or a gas state having passed through the first condensers 3A, 3B does not need to flow through the second condenser 4 having a small sectional area of the refrigerant flow path, thereby avoiding a temperature reduction of the refrigerant in the second condenser 4 due to pressure loss.

[0063] Also, the refrigerant does not flow through the second condenser 4 in the high temperature water input operation, thereby further reducing the pressure loss of the refrigerant as compared to in embodiment 1. This can more reliably inhibit an increase in condensation pressure in the first condensers 3A, 3B and more reliably ensure a sufficient heat exchange rate even in the high temperature water input operation.

[0064] In the low temperature water input operation in this embodiment 2, the refrigerant bypass percentage is 0% and the total flow of the refrigerant flows through the second condenser 4, thereby increasing an output hot water temperature. However, in the present invention, the refrigerant bypass percentage does not need to be always 0% in the low temperature water input operation, but a small portion out of the total flow of the refrigerant may be passed through the second condenser bypass passage 16. In the high temperature water input operation in this embodiment 2, the refrigerant bypass percentage is 100% and the total flow of the refrigerant flows through the second condenser bypass passage 16, thereby reliably reducing the pressure loss of the refrigerant. However, in the present invention, the refrigerant bypass percentage does not need to be always 100% in the high temperature water input operation, but a small portion out of the total flow of the refrigerant may be passed through the second condenser 4.

Embodiment 3

[0065] Next, with reference to Figures 17 to 19, embodiment 3 of the present invention will be described. Differences from embodiment 2 described above will be mainly described, and like or corresponding components are denoted by like reference numerals and descriptions thereof will be omitted.

[0066] Figure 17 is a configuration diagram of a refrigeration cycle device according to embodiment 3 of the present invention. As shown in Figure 17, a configuration of a refrigeration cycle device 1C of this embodiment 3 is the same as in embodiment 2, and descriptions thereof will be omitted.

[0067] Figure 18 is a flowchart showing a control operation of the refrigeration cycle device 1C of this embodiment 3. In step S11 in Figure 18, the control device 50 compares a feed-water temperature detected by an entry heat medium temperature sensor 13 with a previously set first reference temperature β. In this embodiment 3, the first reference temperature βis 30°C. If the feed-water temperature is not higher than the first reference temperature β in step S11, the control device 50 moves to step S12. In step S12, the refrigeration cycle device 1C performs a low temperature water input operation. This low temperature water input operation is the same as the low temperature water input operation in embodiment 2 (Figure 15). Specifically, in step S12, the control device 50 opens an expansion valve 5 and closes a bypass valve 17 to an opening that prevents refrigerant from flowing.

[0068] If the feed-water temperature is higher than the first reference temperature β in step S11, the control device 50 moves to step S13. In step S13, the control device 50 compares the feed-water temperature with a previously set second reference temperature γ. In this embodiment 3, the second reference temperature γ is 50°C. If the feed-water temperature is not lower than the second reference temperature γ in step S 13, the control device 50 moves to step S14. In step S14, the refrigeration cycle device 1C performs a high temperature water input operation. This high temperature water input operation is the same as the high temperature water input operation in embodiment 2 (Figure 16). Specifically, in step S14, the control device 50 opens the bypass valve 17 and closes the expansion valve 5 to an opening that prevents refrigerant from flowing.

[0069] If the feed-water temperature is lower than the second reference temperature γ in step S13, that is, if the feed-water temperature is between the first reference temperature β and the second reference temperature γ, the control device 50 moves to step S15. In step S15, the refrigeration cycle device 1C performs a middle temperature water input operation.

[0070] Figure 17 shows the middle temperature water input operation of the refrigeration cycle device 1C of this embodiment 3. In the middle temperature water input operation, the control device 50 controls the openings of the expansion valve 5 and the bypass valve 17 so that refrigerant having passed through first condensers 3A, 3B is divided to flow through a second condenser 4 and a second condenser bypass passage 16.

[0071] Figure 19 shows a relationship between the feed-water temperature and a bypass percentage in the middle temperature water input operation of the refrigeration cycle device 1C of this embodiment 3. As shown in Figure 19, in the middle temperature water input operation, the control device 50 controls the openings of the expansion valve 5 and the bypass valve 17 so that the bypass percentage continuously increases with increasing feed-water temperature.

[0072] Here, the following expression is satisfied;

$$Rb = Grb/(Grc + Grb) \times 100$$

where, Rb [%] is a bypass percentage, Grc is a flow rate of the refrigerant flowing through the second condenser 4, and Grb is a flow rate of the refrigerant flowing through the second condenser bypass passage 16.

[0073] In Figure 12, the first reference temperature β is desirably approximately a water temperature at a position where dryness of the refrigerant in the second condenser 4 is zero, that is, a water temperature at a position where the refrigerant is between a gas-liquid two-phase zone and a supercooled zone. In the example in Figure 12, the water temperature at the position where dryness of the refrigerant is zero is about 30°C. Thus, in this embodiment 3, the first reference temperature β is 30°C.

[0074] When a total flow of the refrigerant flows through the second condenser 4, at constant pressure, an average flow speed of the refrigerant in the second condenser 4 increases and refrigerant pressure loss in the second condenser 4 increases with increasing feed-water temperature. In this embodiment 3, when the feed-water temperature is between the first reference temperature β (30°C) and the second reference temperature γ (50°C), the middle temperature water input operation for causing a portion of the refrigerant to flow through the second condenser bypass passage 16 can be performed to reduce the flow rate of the refrigerant flowing through the second condenser 4 and reduce pressure loss. Thus, according to this embodiment 3, when the feed-water temperature is between 30°C and 50°C, refrigerant pressure loss can be advantageously more reduced than in the embodiment 2.

[0075] In the middle temperature water input operation, the following expression is satisfied;

$$\Delta h = \Delta h1 + Grc/(Grc + Grb) \cdot \Delta h2$$

where Δh1 is a refrigerant enthalpy difference in the first condensers 3A, 3B, Δh2 is a refrigerant enthalpy difference in the second condenser 4, and Δh is an overall refrigerant enthalpy difference in the first condensers 3A, 3B and the second condenser 4.

[0076] In this embodiment 3, when the feed-water temperature is between the first reference temperature β and the second reference temperature γ, the overall refrigerant enthalpy difference of the first condensers 3A, 3B and the second condenser 4 is Δh calculated by the above expression. On the other hand, if the total flow of the refrigerant flows through the second condenser bypass passage 16 when the feed-water temperature is not lower than the first reference temperature β, the overall refrigerant enthalpy difference of the first condensers 3A, 3B and the second condenser 4 is Δh1. As such, according to this embodiment 3, the refrigerant enthalpy difference can be more increased than in a case where the total flow of the refrigerant flows through the second condenser bypass passage 16 when the feed-water temperature is not lower than the first reference temperature β, thereby further increasing COP.

[0077] Further, according to this embodiment 3, the middle temperature water input operation is performed between the low temperature water input operation and the high temperature water input operation, thereby allowing smooth transition between the operations. In this embodiment 3, the openings of the expansion valve 5 and the bypass valve 17 are controlled so that the bypass percentage continuously increases with increasing feed-water temperature in the middle temperature water input operation, but in the present invention, the openings of the expansion valve 5 and the bypass valve 17 may be controlled so that the bypass percentage increases in a stepwise fashion with increasing feed-water temperature in the middle temperature water input operation.

Reference Signs List

[0078]

1A, 1B, 1C	refrigeration cycle device
2	compressor
3A, 3B	first condenser
4	second condenser
5	expansion valve
6	evaporator
7	evaporator
9	heat medium path
10	second condenser bypass passage
11	flow path switching valve
12	blower
13	entry heat medium temperature sensor
16	second condenser bypass passage
17	bypass valve
20	tank unit
21	hot water storage tank
22	water pump
23, 24	water channel
25	water supply pipe
26	hot water supplying mixing valve
27	hot water pipe
28	water supply branch pipe
29	hot water supply pipe
30	reheating heat exchanger
31	refrigerant flow path
32	heat medium flow path
41	refrigerant flow path
42	heat medium flow path
50	control device
50a	processor
50b	memory
60	heat exchanger
61	twisted pipe
61a, 61b, 61c	groove
62, 63, 64	refrigerant heat transfer pipe
91	water inlet
92	water outlet

Claims

1. A refrigeration cycle device comprising:

   a compressor configured to compress refrigerant;
   a first condenser including a refrigerant flow path and a heat medium flow path, the first condenser being configured to condense the refrigerant compressed by the compressor;
   a second condenser including a refrigerant flow path having a small sectional area as compared with the refrigerant flow path in the first condenser, and a heat medium flow path, the second condenser being configured to further condense the refrigerant having passed through the first condenser;
   an evaporator configured to evaporate the refrigerant;
   a heat medium path configured to allow a liquid heat medium subjected to heat exchange with the refrigerant to pass through the second condenser and the first condenser in this order;
   a second condenser bypass passage configured to bypass the refrigerant flow path or the heat medium flow path in the second condenser;
   a flow path controlling element capable of varying a bypass rate that is a flow rate of the refrigerant or the heat medium flowing through the second condenser bypass passage; and
   control means for controlling an operation of the flow path controlling element so that the bypass rate in a case where an entry heat medium temperature that is a temperature of the heat medium before heat exchange with the refrigerant is higher than a reference temperature is larger than the bypass rate in a case where the entry heat medium temperature is lower than the reference temperature.

2. The refrigeration cycle device according to claim 1, wherein the second condenser bypass passage is configured to bypass the heat medium flow path in the second condenser.

3. The refrigeration cycle device according to claim 1, wherein the second condenser bypass passage is configured to bypass the refrigerant flow path in the second condenser.

4. The refrigeration cycle device according to any one of claims 1 to 3, wherein out of a total flow of the refrigerant or the heat medium, a percentage of the refrigerant or the heat medium flowing through the second condenser bypass passage is a bypass percentage, and
   the control means is configured to cause the bypass percentage to be 0% when the entry heat medium

temperature is lower than the reference temperature.

5. The refrigeration cycle device according to any one of claims 1 to 4, wherein out of a total flow of the refrigerant or the heat medium, a percentage of the refrigerant or the heat medium flowing through the second condenser bypass passage is a bypass percentage, and
the control means is configured to cause the bypass percentage to be 100% when the entry heat medium temperature is higher than the reference temperature.

6. The refrigeration cycle device according to any one of claims 1 to 3, wherein out of a total flow of the refrigerant or the heat medium, a percentage of the refrigerant or the heat medium flowing through the second condenser bypass passage is a bypass percentage, and
the control means is configured to:

cause the bypass percentage to be 0% when the entry heat medium temperature is lower than a first reference temperature;
cause the bypass percentage to be 100% when the entry heat medium temperature is higher than a second reference temperature higher than the first reference temperature;
and
control an operation of the flow path controlling element, when the entry heat medium temperature is between the first reference temperature and the second reference temperature, so that the bypass percentage continuously or stepwise increases with increasing the entry heat medium temperature.

7. The refrigeration cycle device according to any one of claims 1 to 6, wherein the refrigerant flow path in the first condenser is ramified into a plurality of paths, and
a ratio of the number of the refrigerant flow paths in the first condenser to the number of the refrigerant flow path(s) in the second condenser is 1.5 to 2.5.

8. The refrigeration cycle device according to any one of claims 1 to 7, wherein the refrigerant is R32, or the refrigerant mainly contains R32.

9. The refrigeration cycle device according to any one of claims 1 to 8, wherein there is a surplus of the refrigerant in a refrigerant circuit when the entry heat medium temperature is higher than the reference temperature, and
the refrigeration cycle device further comprises a storage portion configured to store redundant refrigerant in the refrigerant circuit.

FIG. 1

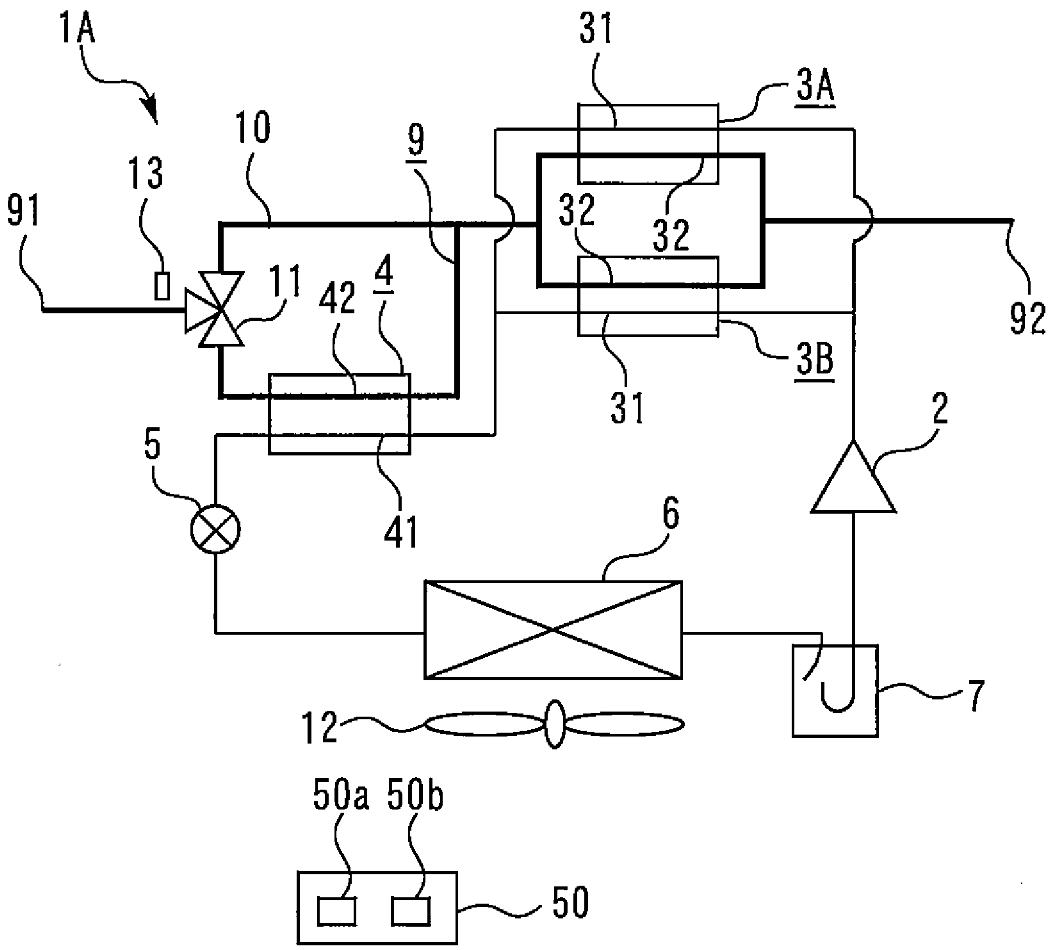


FIG. 2

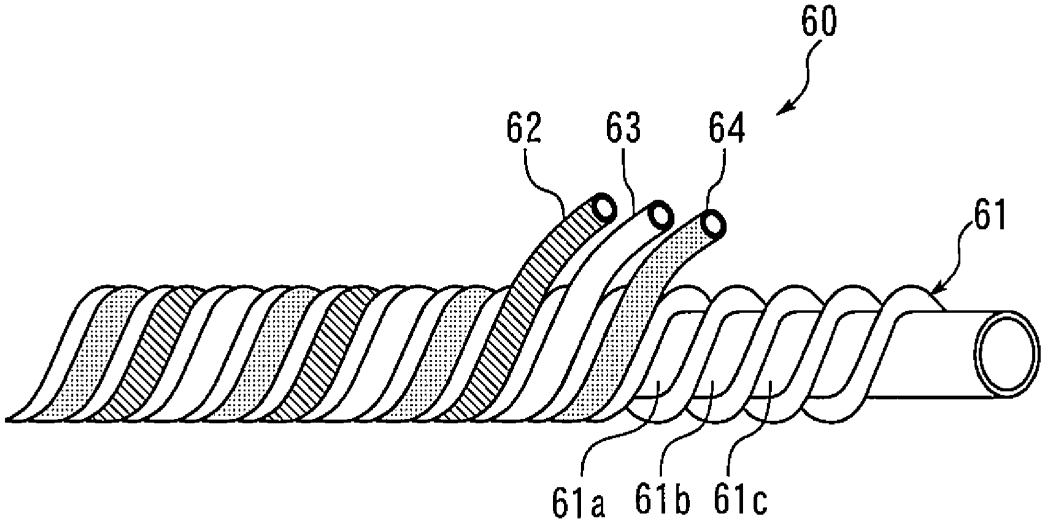


FIG. 3

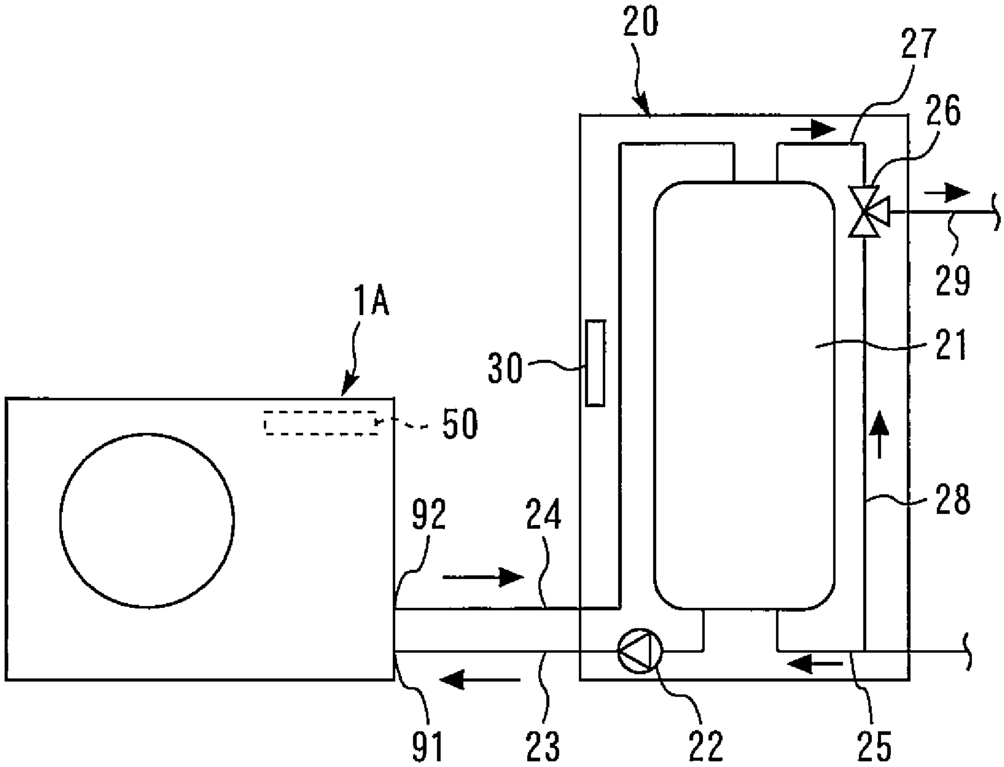


FIG. 4

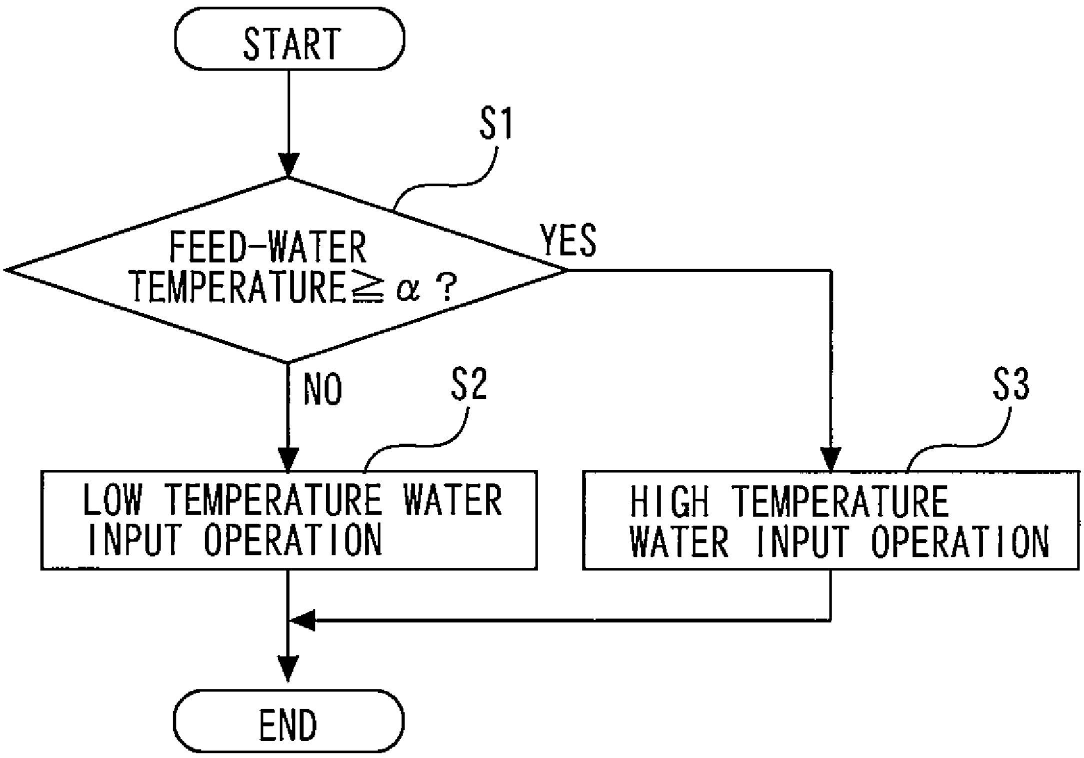

START
S1
FEED-WATER
TEMPERATURE≧α?
YES
NO
S2
S3
LOW TEMPERATURE WATER
INPUT OPERATION
HIGH TEMPERATURE
WATER INPUT OPERATION
END

FIG. 5

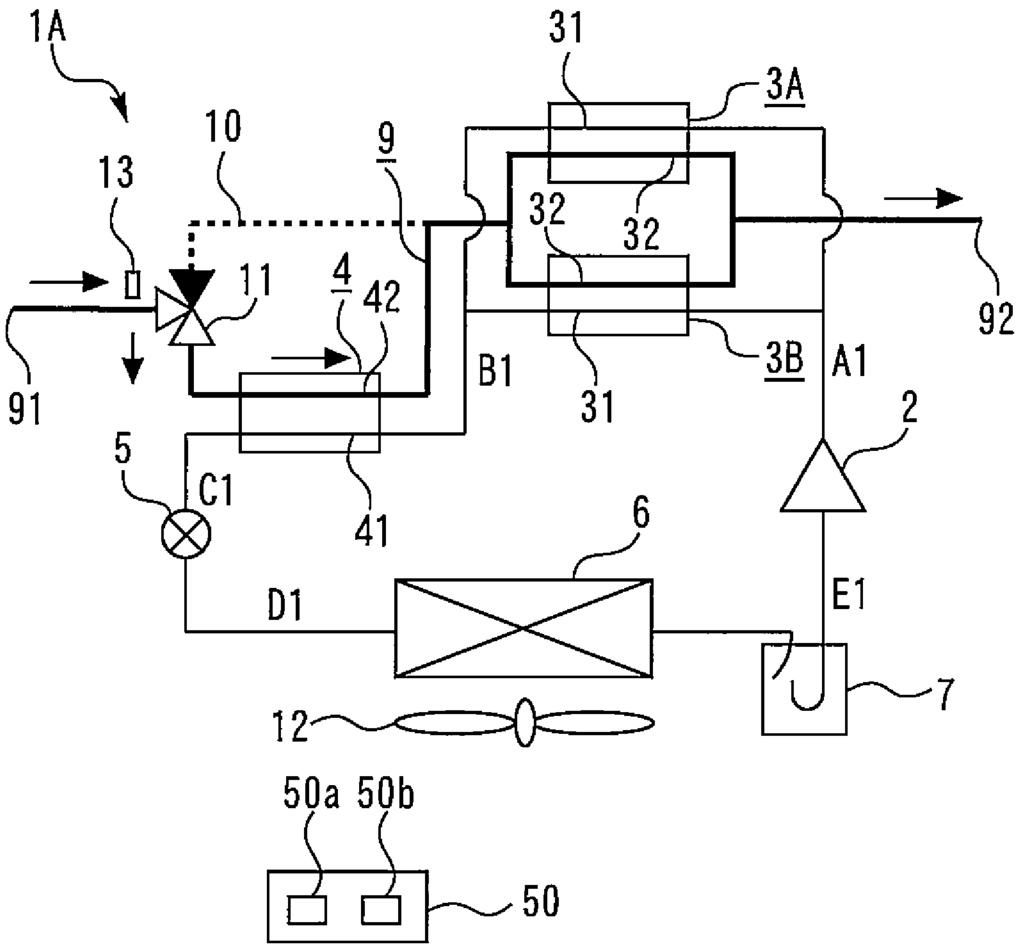

1A
10
9
13
11
4
42
91
31
3A
32
32
32
31
3B
92
A1
B1
2
5
C1
41
6
D1
E1
7
12
50a
50b
50

FIG. 6

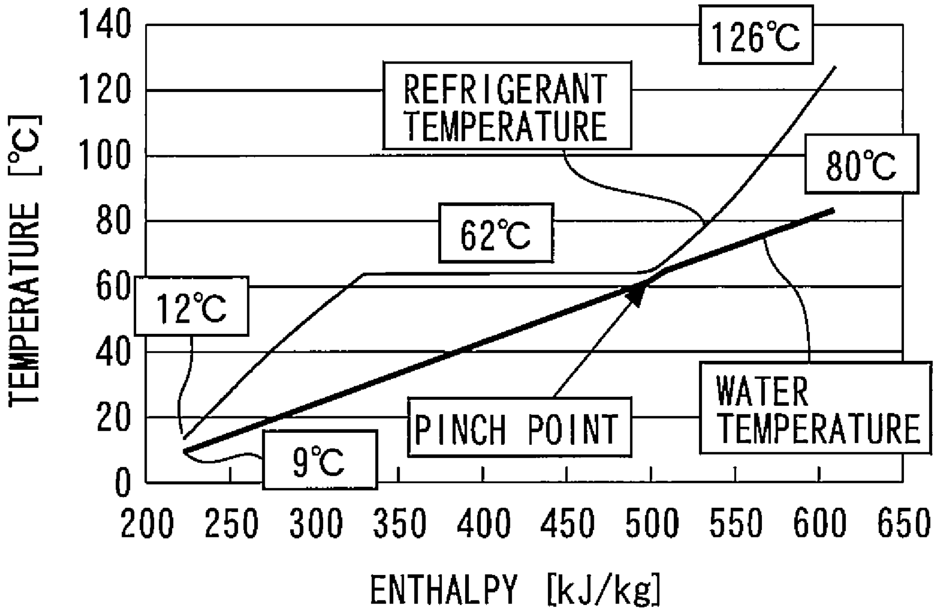

140
120
100
80
60
40
20
0
TEMPERATURE [°C]
126°C
REFRIGERANT TEMPERATURE
80°C
62°C
12°C
PINCH POINT
WATER TEMPERATURE
9°C
200 250 300 350 400 450 500 550 600 650
ENTHALPY [kJ/kg]

FIG. 7

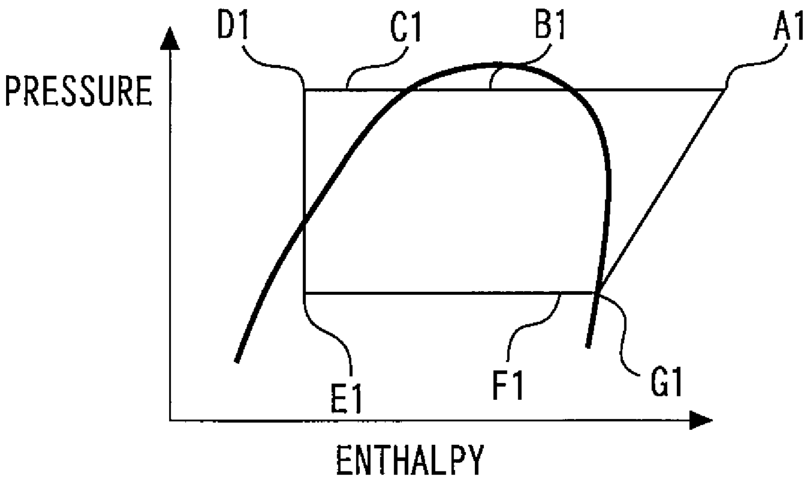

D1
C1
B1
A1
PRESSURE
E1
F1
G1
ENTHALPY

FIG. 8

RELATIONSHIP BETWEEN OUTSIDE AIR TEMPERATURE AND FEED-WATER TEMPERATURE IN LOW TEMPERATURE WATER INPUT OPERATION

OUTSIDE AIR TEMPERATURE	FEED-WATER TEMPERATURE
2℃	5℃
7℃	9℃
16℃	17℃
25℃	24℃

FIG. 9

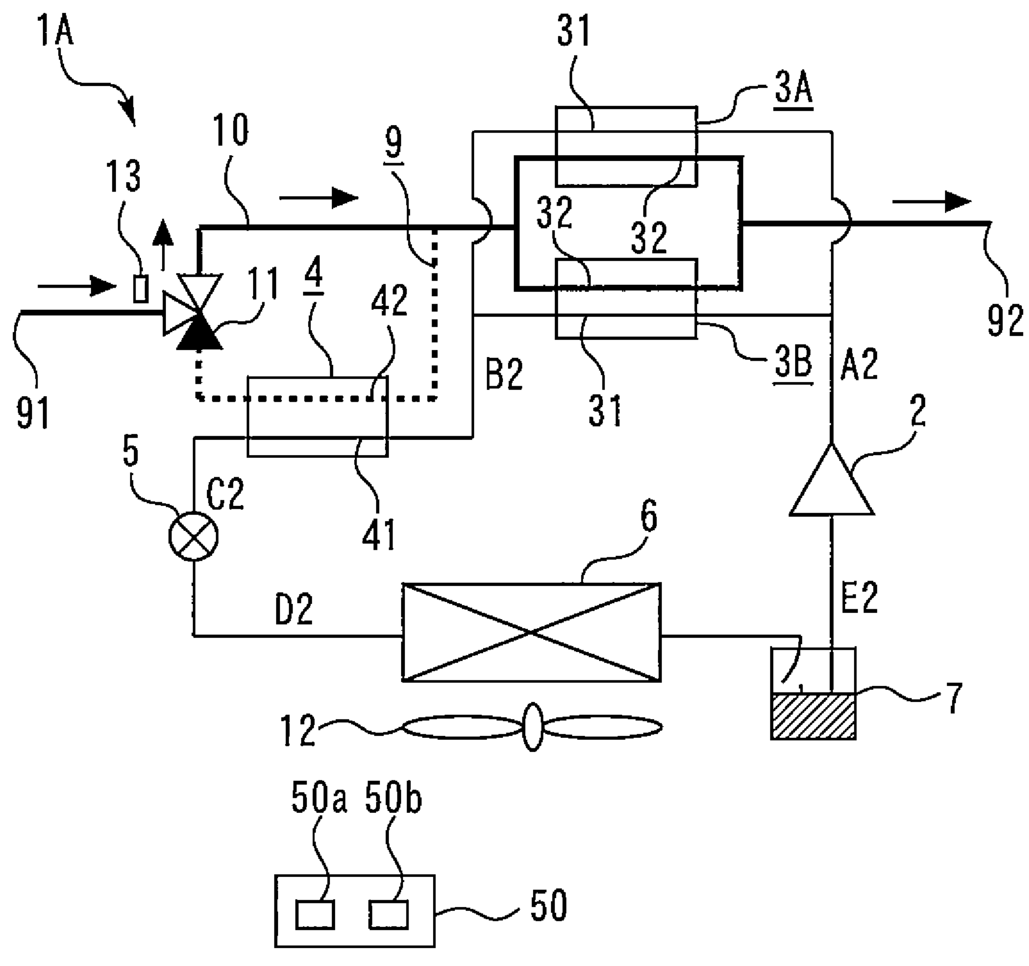

1A
13
10
9
31
3A
32
32
11
4
42
91
B2
31
3B
A2
92
2
5
C2
41
6
D2
E2
7
12
50a
50b
50

FIG. 10

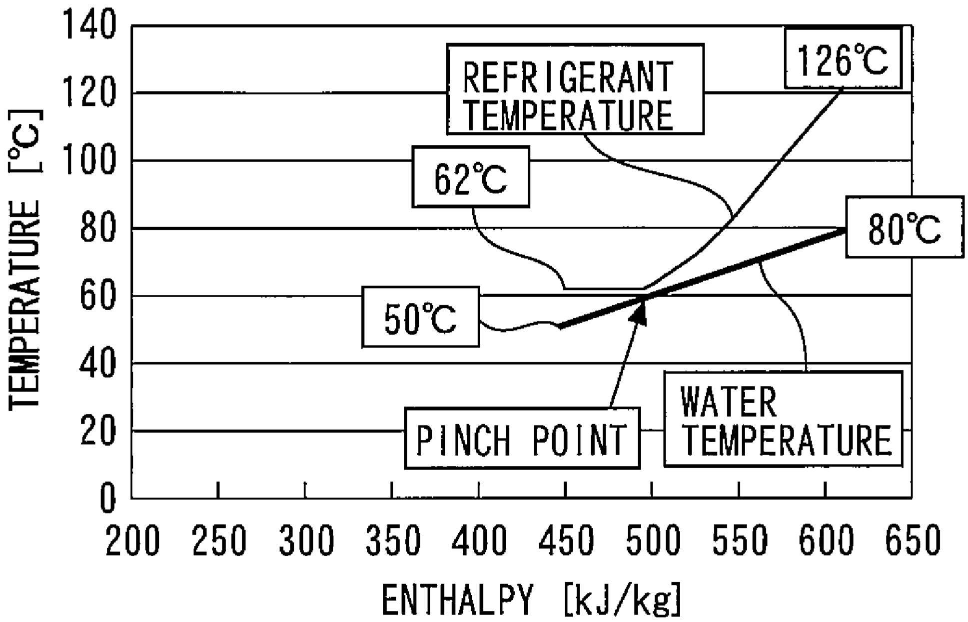

140
120
100
80
60
40
20
0
TEMPERATURE [°C]
REFRIGERANT TEMPERATURE
126℃
62℃
80℃
50℃
PINCH POINT
WATER TEMPERATURE
200
250
300
350
400
450
500
550
600
650
ENTHALPY [kJ/kg]

FIG. 11

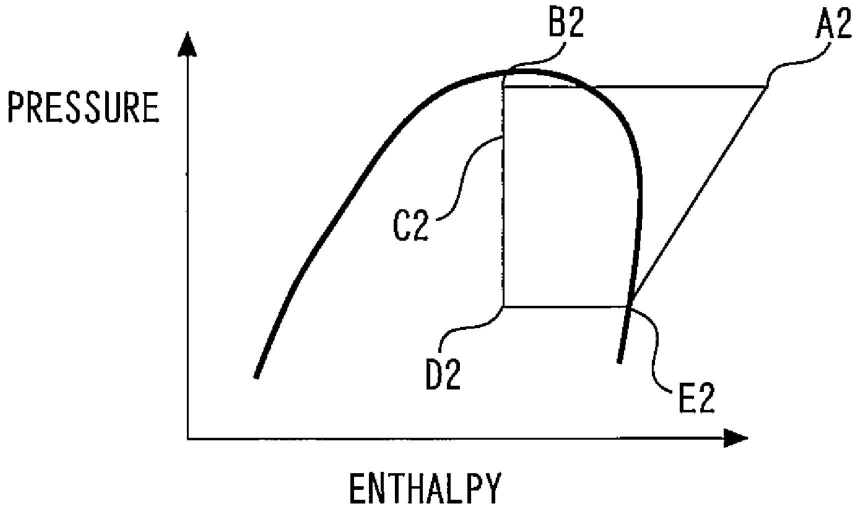

PRESSURE
B2
A2
C2
D2
E2
ENTHALPY

FIG. 12

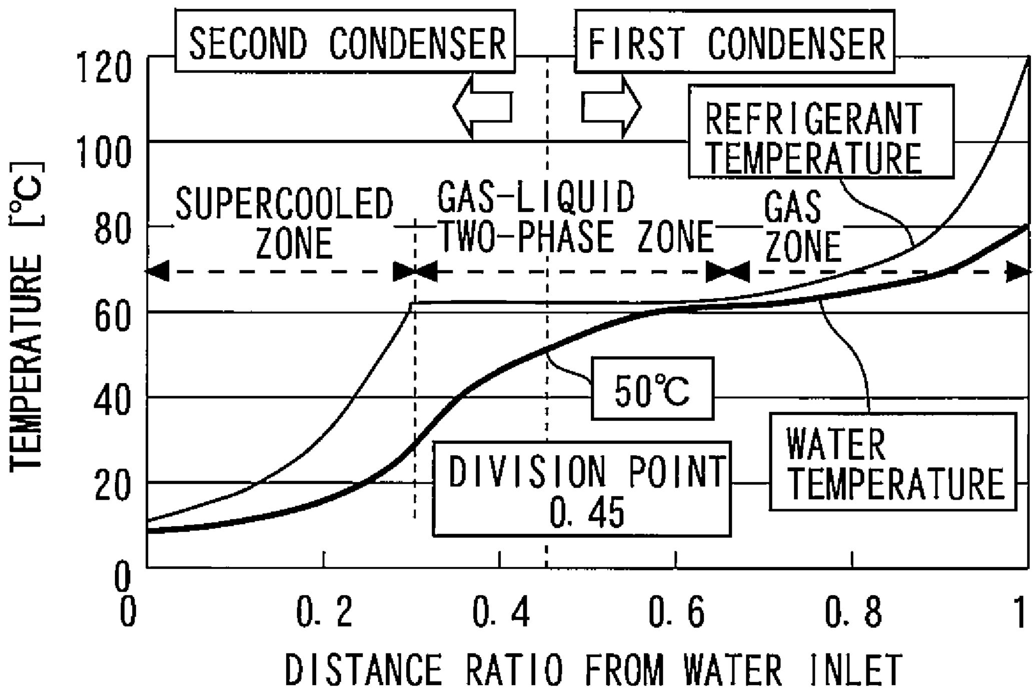


FIG. 13

COMPARISON BETWEEN COMPRESSOR DISCHARGE TEMPERATURES OF R410A AND R32

REFRIGERANT	R410A	R32
SUCTION PRESSURE [MPa]	0.81	
DEGREE OF SUPERHEAT [K]	0	
DISCHARGE PRESSURE [MPa]	4.25	
COMPRESSOR EFFICIENCY [%]	100	
DISCHARGE TEMPERATURE [℃]	91	110

FIG. 14

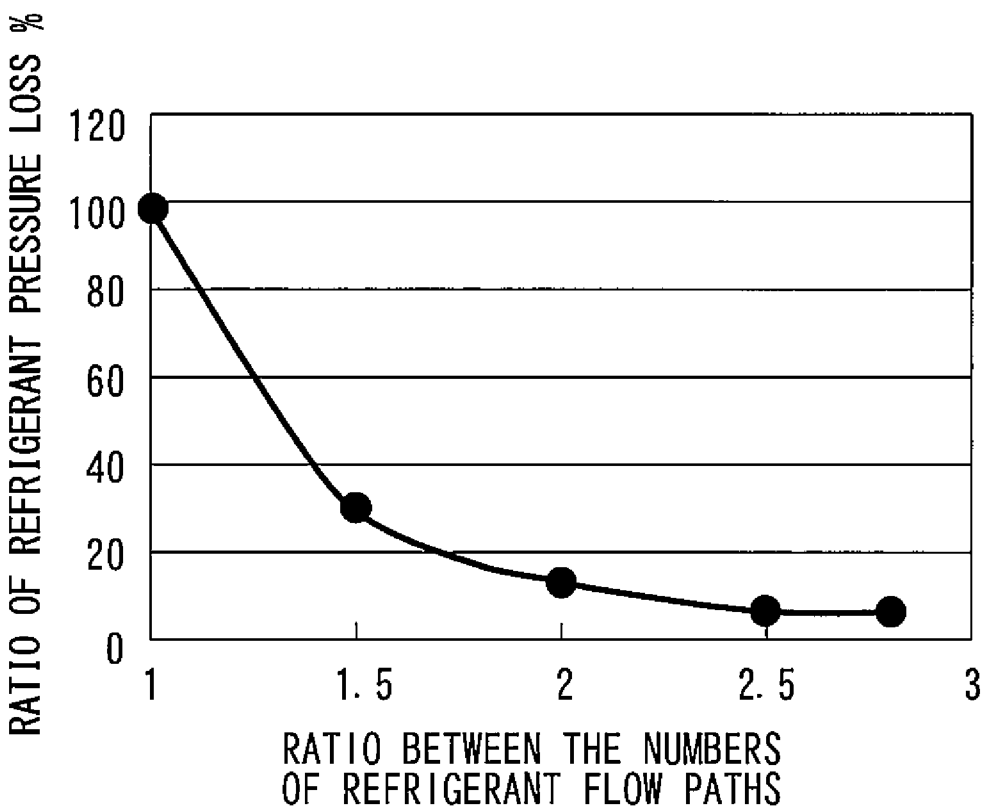

RATIO OF REFRIGERANT PRESSURE LOSS %
120
100
80
60
40
20
0
1
1.5
2
2.5
3
RATIO BETWEEN THE NUMBERS
OF REFRIGERANT FLOW PATHS

FIG. 15

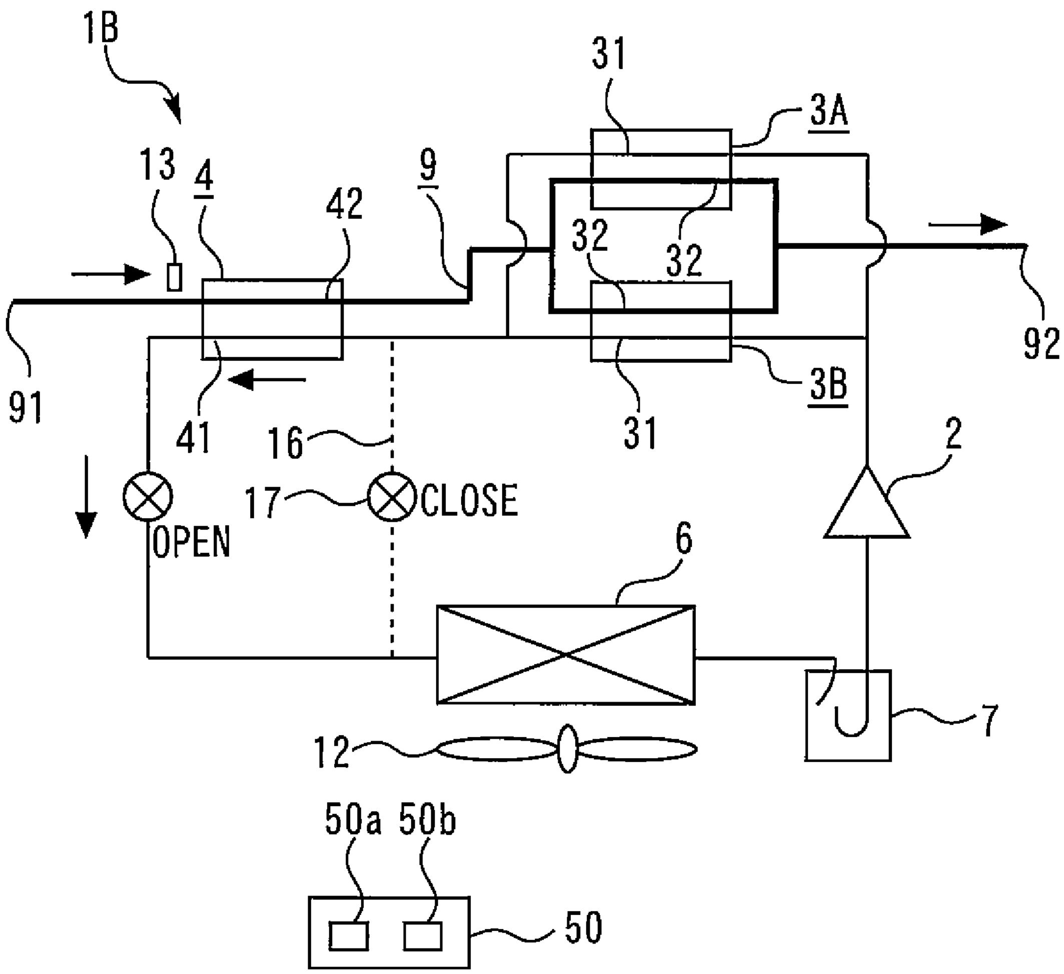

1B
13
4
42
9
31
3A
32
32
32
92
91
41
16
3B
31
2
17
CLOSE
OPEN
6
7
12
50a
50b
50

FIG. 16

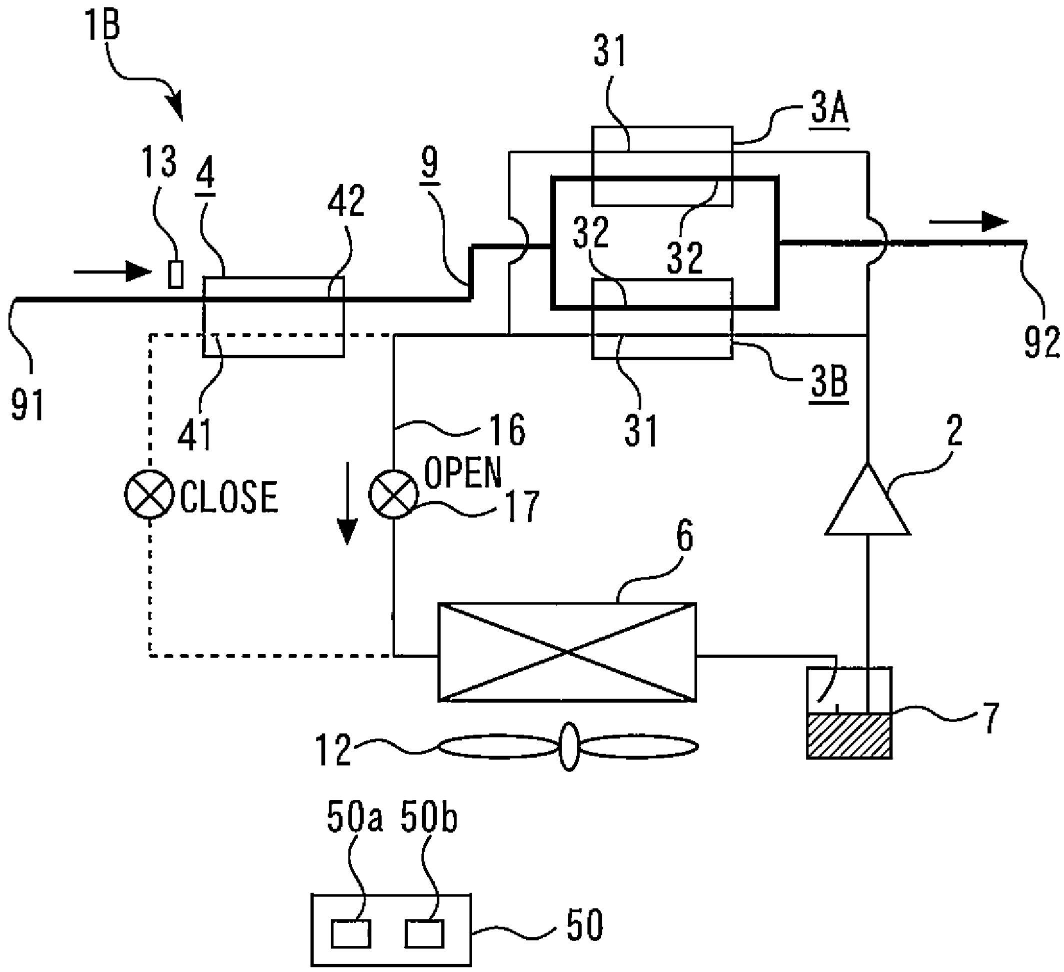

1B
13
4
42
9
31
3A
32
32
92
91
41
16
31
3B
2
CLOSE
OPEN
17
6
7
12
50a
50b
50

FIG. 17

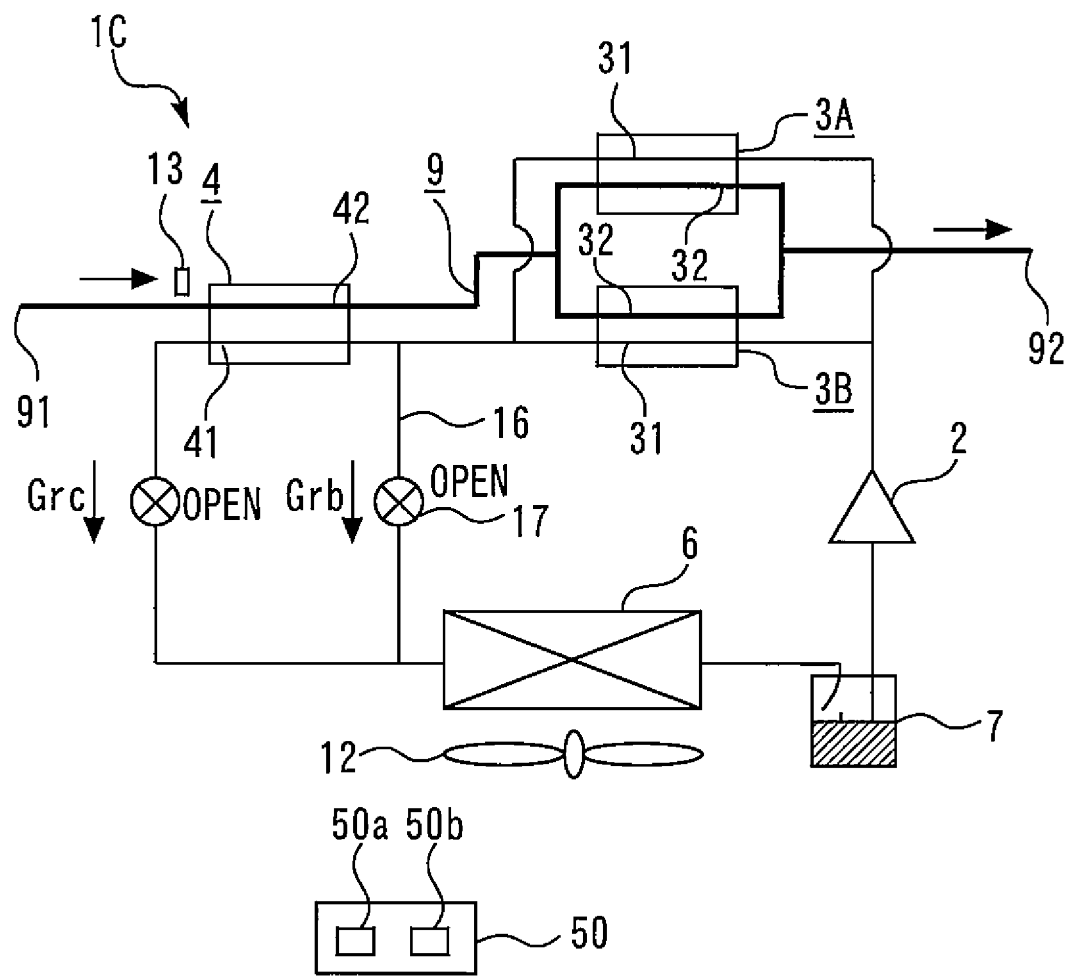

1C
31
3A
13
4
9
42
32
32
32
92
91
41
16
31
3B
2
Grc
OPEN
Grb
OPEN
17
6
7
12
50a
50b
50

FIG. 18

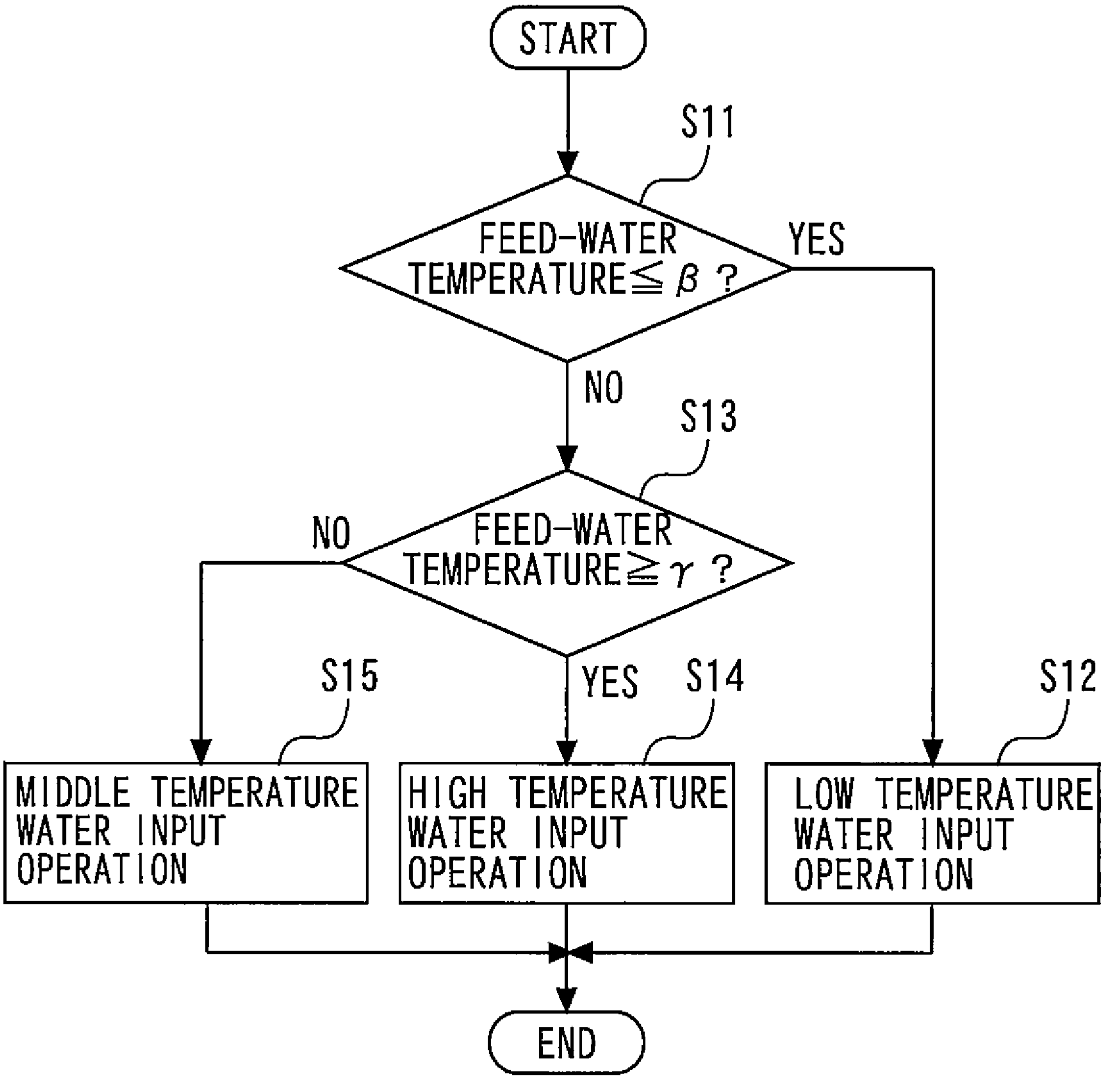
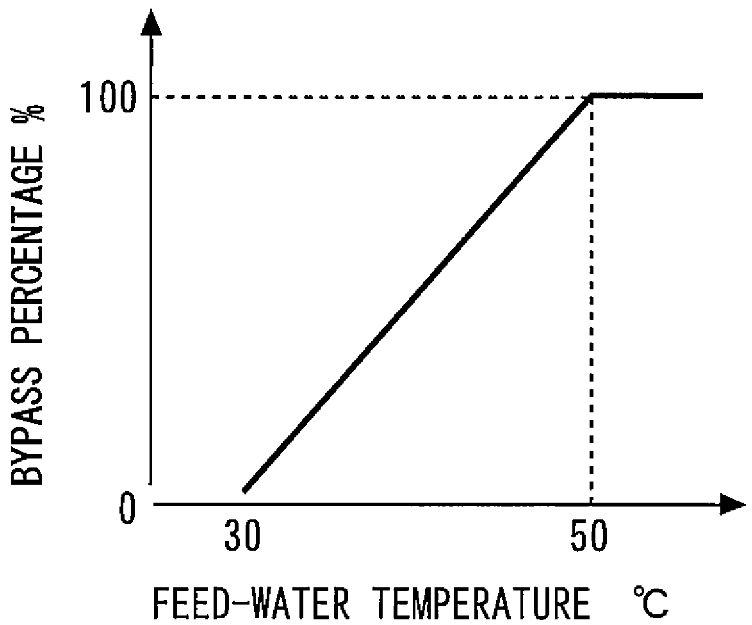

START
S11
FEED-WATER TEMPERATURE≦β ?
YES
NO
S13
FEED-WATER TEMPERATURE≧γ ?
NO
YES
S15
S14
S12
MIDDLE TEMPERATURE WATER INPUT OPERATION
HIGH TEMPERATURE WATER INPUT OPERATION
LOW TEMPERATURE WATER INPUT OPERATION
END

FIG. 19



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2013/078216

A. CLASSIFICATION OF SUBJECT MATTER
F25B1/00(2006.01)i, *F25B6/04*(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, F25B6/04

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 2003-262397 A (Osaka Gas Co., Ltd.), 19 September 2003 (19.09.2003), paragraphs [0025] to [0043]; fig. 1 to 4 (Family: none)	1-5,7-9 6
Y A	JP 2011-137617 A (Hitachi Appliances, Inc.), 14 July 2011 (14.07.2011), paragraphs [0005] to [0054]; fig. 1 to 4 (Family: none)	1-5,7-9 6
Y A	JP 2009-222246 A (Mitsubishi Electric Corp.), 01 October 2009 (01.10.2009), paragraphs [0009] to [0075]; fig. 1 to 16 (Family: none)	1-5,7-9 6

☒ 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	Date of mailing of the international search report
14 January, 2014 (14.01.14)	21 January, 2014 (21.01.14)

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

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2013/078216

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 2006-78048 A (Matsushita Electric Industrial Co., Ltd.), 23 March 2006 (23.03.2006), paragraphs [0040] to [0051]; fig. 6 to 8 (Family: none)	1-9

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2002089958 A **[0005]**
- JP 2002310498 A **[0005]**
- JP 2007232285 A **[0005]**
- JP 2013044441 A **[0005]**
- JP 2009222246 A **[0005]**
- JP 2010014374 A **[0005]**
- JP 2001082818 A **[0005]**