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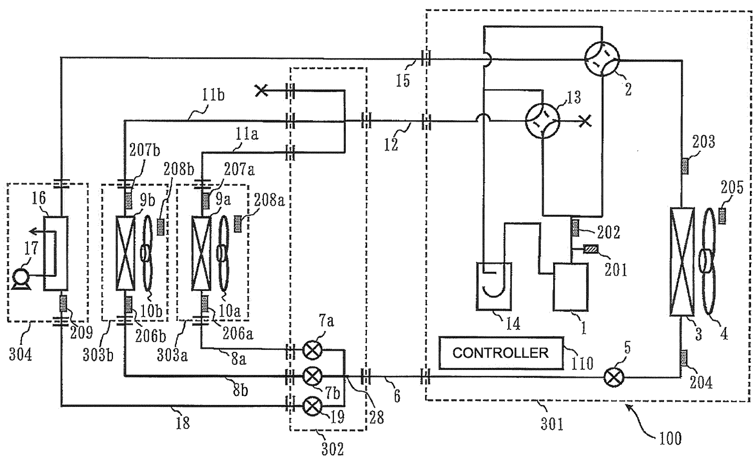
(54) REFRIGERATION CYCLE DEVICE AND REFRIGERATION CYCLE CONTROL METHOD

(57) An excessive increase in high pressure during a high-temperature-water supply is suppressed and a predetermined hot-water-supply capacity within a usage range of a compressor is ensured in a refrigeration cycle apparatus that can perform the hot-water-supply operation of an integrated air-conditioning and hot-water-supply system. An integrated air-conditioning and hot-water-supply system 100 includes a compressor 1, a plate-type water heat exchanger 16, a hot-water-supply pressure-reducing mechanism 19, and an outdoor heat exchanger 3. Moreover, the integrated air-conditioning and hot-water-supply system 100 includes a high-pressure sensor 201 that detects a high pressure in the compressor 1, and a controller 110 that calculates a condensing temperature of the plate-type water heat exchanger 16 based on the high pressure detected by the high-pressure sensor 201. When the calculated condensing temperature is higher than or equal to a preset target condensing-temperature value, the controller 110 performs condensing-temperature control for controlling the operating frequency of the compressor 1 based on a difference between the calculated condensing temperature and the target condensing-temperature value, and performs opening-degree control for controlling the opening degree of the hot-water-supply pressure-reducing mechanism 19 concurrently with the condensing-temperature control based on a difference between a current opening degree of the hot-water-supply pressure-reducing mechanism 19 and a preset target opening-degree value.

FIG. 1



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Description

Technical Field

[0001] The present invention relates to integrated air-conditioning and hot-water-supply systems that can perform an air-conditioning operation (i.e., cooling operation or heating operation) and a hot-water-supply operation at the same time, and more specifically, to an integrated air-conditioning and hot-water-supply system that determines a high-temperature-water supply state when the condensing temperature becomes higher than or equal to a predetermined value during a hot-water supply and that suppresses an excessive increase in high pressure by controlling the condensing temperature of a compressor and the opening degree of a pressure-reducing mechanism so as to achieve a predetermined hot-water-supply capacity within a usage range of the compressor.

Background Art

[0002] In the related art, a hot-water-suppliable heat pump system that is equipped with a refrigerant circuit formed by connecting a hot-water-supply unit (i.e., a water heater) to a heat source unit (i.e., an outdoor unit) by pipes and that can perform the hot-water-supply operation is known. When the hot-water-supply temperature becomes high (e.g., 60 degrees C) in such a hot-water-supply system, the condensing temperature increases, causing an excessive increase in high pressure. This is a problem in that it is difficult to ensure a hot-water-supply capacity. For this reason, there have been efforts to solve this problem (e.g., see Patent Literature 1 and Patent Literature 2).
In a heat-pump bath hot-water-supply device discussed in Patent Literature 1, the valve opening degree of a pressure-reducing device is controlled in accordance with a discharge temperature or a discharge pressure as a target. The operation efficiency is set in accordance with a discharge temperature or a discharge pressure that has a maximum value relative to the valve opening degree of the pressure-reducing device and that corresponds to the maximum operation efficiency as a target control value. By changing the target control value in accordance with a bathtub temperature, a boiling temperature, a water-side inlet temperature, and a compressor frequency, high operation efficiency can be achieved even when the bathtub temperature, the boiling temperature, the water-side inlet temperature, and the compressor frequency change.
[0003] In a heat-pump hot-water-supply device discussed in Patent Literature 2, the discharge pressure is monitored during the hot-water-supply operation, and discharge-pressure control is performed on an expansion valve when the discharge pressure increases, so that the operation can be continuously performed without the discharge pressure exceeding the usage range of the compressor.

Citation List

Patent Literature

[0004]

Patent Literature 1: Japanese Unexamined Patent Application Publication No. 2004-53118
Patent Literature 2: Japanese Unexamined Patent Application Publication No. 2005-98530

Summary of Invention

Technical Problem

[0005] In the heat-pump bath hot-water-supply device discussed in Patent Literature 1, the pressure-reducing device is controlled in accordance with the discharge temperature or the discharge pressure corresponding to the maximum operation efficiency. However, in the case where a high-temperature-water supply is performed and the requested hot-water-supply capacity and the compressor frequency are high, the control is performed based on the operation efficiency regardless of an increase in the discharge pressure of the pressure-reducing device. This causes the discharge pressure to increase, possibly resulting in an excessive increase in the condensing temperature.
[0006] In the heat-pump hot-water-supply device discussed in Patent Literature 2, in the case where the high-temperature-water supply is performed and the requested hot-water-supply capacity and the compressor frequency are high, an increase in high pressure sometimes cannot be suppressed by simply controlling the pressure-reducing device, resulting in an excessive increase in the condensing temperature.
[0007] Furthermore, in an integrated air-conditioning and hot-water-supply system that is equipped with a refrigerant circuit formed by connecting a use-side unit (i.e., an indoor unit) by pipes in addition to a hot-water-supply unit and that

can perform the air-conditioning operation and the hot-water-supply operation at the same time, if there are an air conditioning load and a high-temperature-water-supply request at the same time during the high-temperature-water supply, an operation method that satisfies both of them needs to be established.

[0008] In the present invention, when the condensing temperature becomes higher than or equal to a predetermined value during the hot-water supply, a high-temperature-water supply state is determined, condensing-temperature control is performed on a compressor, and opening-degree control is performed on a pressure-reducing mechanism. Accordingly, an integrated air-conditioning and hot-water-supply system that can suppress an excessive increase in condensing temperature and can ensure a hot-water-supply capacity within a usage range of a compressor during the high-temperature-water supply.

Solution to Problem

[0009] A refrigeration cycle apparatus according to the present invention includes a refrigeration cycle mechanism having a compressor whose operating frequency is controllable, a first radiator, a first pressure-reducing mechanism whose opening degree is controllable, and a first evaporator, and in which a refrigerant sequentially circulates through the compressor, the first radiator, the first pressure-reducing mechanism, and the first evaporator; a high-pressure sensor that detects a high pressure between a discharge side of the compressor and a liquid side of the first pressure-reducing mechanism; and a controller that calculates a condensing temperature of the first radiator based on the high pressure detected by the high-pressure sensor. When the calculated condensing temperature of the first radiator is higher than or equal to a preset target condensing-temperature value, the controller performs condensing-temperature control for controlling the operating frequency of the compressor based on a difference between the calculated condensing temperature and the target condensing-temperature value, and performs opening-degree control for controlling the opening degree of the first pressure-reducing mechanism concurrently with the condensing-temperature control based on a difference between a current opening degree of the first pressure-reducing mechanism and a preset target opening-degree value.

Advantageous Effects of Invention

[0010] The present invention can provide a refrigeration cycle apparatus that can suppress an excessive increase in condensing temperature and can ensure a hot-water-supply capacity within a usage range of a compressor during the high-temperature-water supply.

Brief Description of Drawings

[0011]

[Fig. 1] Fig. 1 illustrates the configuration of an integrated air-conditioning and hot-water-supply system 100 according to Embodiment 1.

[Fig. 2] Fig. 2 schematically illustrates the flow of water from a hot-water-supply unit 304 to a hot-water-supply tank 305 in Embodiment 1.

[Fig. 3] Fig. 3 schematically illustrates a controller 110 in Embodiment 1.

[Fig. 4] Fig. 4 illustrates an operation of four-way valves relative to operation modes in Embodiment 1.

[Fig. 5] Fig. 5 illustrates a method for determining a target evaporating-temperature value from a maximum cooled-room temperature difference in compressor control in Embodiment 1.

[Fig. 6] Fig. 6 illustrates a method for determining a target condensing-temperature value from a maximum heated-room temperature difference in compressor control in Embodiment 1.

[Fig. 7] Fig. 7 illustrates the relationships among a target opening degree, a hot-water-supply capacity, and operation efficiency in Embodiment 1.

[Fig. 8] Fig. 8 illustrates tests performed when performing control for changing a target opening-degree value of a hot-water-supply pressure-reducing mechanism in accordance with a compressor frequency in Embodiment 1.

[Fig. 9] Fig. 9 illustrates the relationship between an outdoor-air temperature and the target opening-degree value in Embodiment 1.

[Fig. 10] Fig. 10 illustrates the relationships among the hot-water-supply capacity, an evaporating capacity, and a compressor input in Embodiment 1.

[Fig. 11] Fig. 11 illustrates the contents of tests performed at a development stage when performing control for changing the target opening-degree value in accordance with the hot-water-supply capacity in Embodiment 1.

[Fig. 12] Fig. 12 is a flowchart illustrating the flow for determining whether a high-temperature-water supply is to be performed or a normal hot-water supply is to be performed in Embodiment 1.

[Fig. 13] Fig. 13 is a flowchart illustrating an operation method during a high-temperature-water supply in a simultaneous heating and hot-water-supply operation in Embodiment 1.

[Fig. 14] Fig. 14 is a flowchart illustrating an operation method during a high-temperature-water supply in a simultaneous cooling and hot-water-supply operation in Embodiment 1.

Description of Embodiments

Embodiment 1

[0012] Embodiment 1 will be described below with reference to Figs. 1 to 14.

Fig. 1 is a refrigerant circuit configuration diagram of an integrated air-conditioning and hot-water-supply system 100 (refrigeration cycle apparatus) according to Embodiment 1. In the following figures including Fig. 1, the dimensional relationships among components may sometimes differ from actual dimensional relationships. Furthermore, when a symbol used in a numerical expression first appears in this specification, the unit of the symbol will be written in parenthesis []. If a symbol is dimensionless (i.e., has no units), the unit will be expressed as [-].

[0013] Fig. 2 schematically illustrates the flow of water from a hot-water-supply unit 304 to a hot-water-supply tank 305 in the integrated air-conditioning and hot-water-supply system 100.

Fig. 3 schematically illustrates various kinds of sensors in the integrated air-conditioning and hot-water-supply system 100, and a measuring unit 101, a calculating unit 102, a control unit 103, and a storage unit 104 in a controller 110. The configuration of the integrated air-conditioning and hot-water-supply system 100 will be described below with reference to Figs. 1 to 3. The integrated air-conditioning and hot-water-supply system 100 is a triple-pipe multisystem integrated air-conditioning and hot-water-supply system that performs a vapor-compression refrigeration cycle operation so as to simultaneously perform a cooling operation or heating operation selected in a use-side unit and a hot-water-supply operation in a hot-water-supply unit. The integrated air-conditioning and hot-water-supply system 100 is an integrated air-conditioning and hot-water-supply system that can ensure a hot-water-supply capacity by suppressing an excessive increase in high pressure during a supply of high-temperature water when the hot-water-supply operation is performed in the hot-water-supply unit. Fig. 1 illustrates the refrigerant circuit configuration, and Fig. 2 illustrates a water circuit configuration from the hot-water-supply unit 304 to the hot-water-supply tank 305.

System Configuration

[0014] The integrated air-conditioning and hot-water-supply system 100 has a heat source unit 301, a branch unit 302, use-side units 303a and 303b, the hot-water-supply unit 304, and the hot-water-supply tank 305. The heat source unit 301 and the branch unit 302 are connected by a liquid extension pipe 6 serving as a refrigerant pipe and a gas extension pipe 12 serving as a refrigerant pipe. The hot-water-supply unit 304 has one end connected to the heat source unit 301 via a hot-water-supply gas extension pipe 15 serving as a refrigerant pipe and another end connected to the branch unit 302 via a hot-water-supply liquid pipe 18 serving as a refrigerant pipe. The use-side units 303a and 303b and the branch unit 302 are connected by indoor gas pipes 11a and 11 b serving as refrigerant pipes and indoor liquid pipes 8a and 8b serving as refrigerant pipes. The hot-water-supply tank 305 and the hot-water-supply unit 304 are connected by an upstream water pipe 20 serving as a water pipe and a downstream water pipe 21 serving as a water pipe.

[0015] Although one heat source unit, two use-side units, one hot-water-supply unit, and one hot-water-supply tank 305 are connected as an example in Embodiment 1, the configuration is not limited to this and the numbers thereof may be more than or fewer than those shown. Furthermore, although a refrigerant used in the integrated air-conditioning and hot-water-supply system 100 is R410A, the refrigerant used in the integrated air-conditioning and hot-water-supply system 100 is not limited to this kind of refrigerant. Other alternatives include, for example, an HFC (hydrofluorocarbon) refrigerant, such as R407C or R404A, an HCFC (hydrochlorofluorocarbon) refrigerant, such as R22 or R134a, and a refrigerant that operates at a critical pressure or higher, such as CO_2.

[0016] As shown in Fig. 1, the integrated air-conditioning and hot-water-supply system 100 includes the controller 110. The controller 110 includes the measuring unit 101, the calculating unit 102, the control unit 103, and the storage unit 104. Control to be described below is entirely performed by the controller 110. Although the controller 110 is disposed in the heat source unit 301 in Fig. 1, this is only an example. The position where the controller 110 is disposed is not limited.

Operation Modes of Heat Source Unit 301

[0017] Operation modes that can be performed by the integrated air-conditioning and hot-water-supply system 100 will be briefly described. In the integrated air-conditioning and hot-water-supply system 100, an operation mode of the heat source unit 301 is determined based on whether there are a hot-water-supply load in the connected hot-water-supply unit 304 and cooling loads or heating loads in the use-side units 303a and 303b. The integrated air-conditioning

and hot-water-supply system 100 is capable of performing the following five operation modes, which includes
a cooling operation mode A,
a heating operation mode B,
a hot-water-supply operation mode C,
a simultaneous heating and hot-water-supply operation mode D, and
a simultaneous cooling and hot-water-supply operation mode E.

[0018]

(1) The cooling operation mode A is an operation mode of the heat source unit 301 when there is no hot-water-supply request signal (also referred to as "hot-water-supply request") and the cooling operation is performed by the use-side units 303a and 303b.
(2) The heating operation mode B is an operation mode of the heat source unit 301 when there is no hot-water-supply request and the heating operation is performed by the use-side units 303a and 303b.
(3) The hot-water-supply operation mode C is an operation mode of the heat source unit 301 when there is no air conditioning load and the hot-water-supply operation is performed by the hot-water-supply unit 304.
(4) The simultaneous heating and hot-water-supply operation mode D is an operation mode of the heat source unit 301 when the heating operation by the use-side units 303a and 303b and the hot-water-supply operation by the hot-water-supply unit 304 are simultaneously performed.
(5) The simultaneous cooling and hot-water-supply operation mode E is an operation mode of the heat source unit 301 when the cooling operation by the use-side units 303a and 303b and the hot-water-supply operation by the hot-water-supply unit 304 are simultaneously performed.

Use-Side Units 303a and 303b

[0019] The use-side units 303a and 303b are connected to the heat source unit 301 via the branch unit 302. The use-side units 303a and 303b are installed in areas (e.g., by being concealed in or suspended from a ceiling indoors or being hung on a wall) where the units can blow conditioned air to an air conditioning target region. The use-side units 303a and 303b are connected to the heat source unit 301 via the branch unit 302, the liquid extension pipe 6, and the gas extension pipe 12, and constitute a part of the refrigerant circuit.

[0020] The use-side units 303a and 303b each include an indoor-side refrigerant circuit that constitutes a part of the refrigerant circuit. These indoor-side refrigerant circuits are constituted by respective indoor heat exchangers 9a and 9b serving as use-side heat exchangers. Furthermore, the use-side units 303a and 303b are respectively provided with indoor air-sending devices 10a and 10b for supplying conditioned air, after having exchanged heat with the refrigerant in the indoor heat exchangers 9a and 9b, to the air conditioning target region, such as an indoor space.

[0021] The indoor heat exchangers 9a and 9b may each be formed of, for example, a cross-fin-type fin-and-tube heat exchanger constituted of a heat transfer pipe and multiple fins. Alternatively, the indoor heat exchangers 9a and 9b may each be formed of a micro-channel heat exchanger, a shell-and-tube heat exchanger, a heat-pipe heat exchanger, or a double-pipe heat exchanger. When the operation mode performed by the use-side units 303a and 303b is the cooling operation mode A, the indoor heat exchangers 9a and 9b function as refrigerant evaporators and cool the air in the air conditioning target region. When the operation mode is the heating operation mode B, the indoor heat exchangers 9a and 9b function as refrigerant condensers (or radiators) and heat the air in the air conditioning target region.

[0022] The indoor air-sending devices 10a and 10b have a function of suctioning indoor air into the use-side units 303a and 303b, making the indoor air exchange heat with the refrigerant at the indoor heat exchangers 9a and 9b, and then supplying the indoor air as conditioned air to the air conditioning target region. Specifically, in the use-side units 303a and 303b, the indoor air taken in by the indoor air-sending devices 10a and 10b and the refrigerant flowing through the indoor heat exchangers 9a and 9b can exchange heat with each other. The indoor air-sending devices 10a and 10b are capable of adjusting the flow rate of conditioned air to be supplied to the indoor heat exchangers 9a and 9b and each include a fan, such as a centrifugal fan or a multi-blade fan, and a motor, such as a DC fan motor, for driving this fan.

[0023] The use-side units 303a and 303b are provided with the following various kinds of sensors, which include:

(1) indoor liquid temperature sensors 206a and 206b that are provided at the liquid side of the indoor heat exchangers 9a and 9b and detect the temperature of a liquid refrigerant;
(2) indoor gas temperature sensors 207a and 207b that are provided at the gas side of the indoor heat exchangers 9a and 9b and detect the temperature of a gas refrigerant; and
(3) indoor suction temperature sensors 208a and 208b that are provided at the indoor-air suction side of the use-side units 303a and 303b and detect the temperature of indoor air flowing into the units.

[0024] As shown in Fig. 3, the operation of the indoor air-sending devices 10a and 10b is controlled by the control unit

103 that functions as normal-operation control means that performs a normal operation including the cooling operation mode A and the heating operation mode B of the use-side units 303a and 303b.

Hot-Water-Supply Unit 304

[0025] The hot-water-supply unit 304 is connected to the heat source unit 301 via the branch unit 302. As shown in Fig. 2, the hot-water-supply unit 304 has a function of supplying hot water to the hot-water-supply tank 305 installed, for example, outdoors and boiling the water in the hot-water-supply tank 305 by heating the water. A plate-type water heat exchanger 16 of the hot-water-supply unit 304 includes a connection section 25 (i.e., an inflowing-water-pipe connection section) connected to the downstream water pipe 21 (i.e., an inflowing water pipe), a connection section 26 (i.e., an outflowing water pipe connection section) connected to the upstream water pipe 20 (i.e., an outflowing water pipe), and a water pipe 27 through which water flowing therein from the downstream water pipe 21 flows out toward the upstream water pipe 20. Furthermore, the hot-water-supply unit 304 has one end connected to the heat source unit 301 via the hot-water-supply gas extension pipe 15 and another end connected to the branch unit 302 via the hot-water-supply liquid pipe 18, and constitutes a part of the refrigerant circuit in the integrated air-conditioning and hot-water-supply system 100.
[0026] The hot-water-supply unit 304 includes a hot-water-supply-side refrigerant circuit that constitutes a part of the refrigerant circuit. The hot-water-supply-side refrigerant circuit has the plate-type water heat exchanger 16 serving as a hot-water-supply-side heat exchanger as an elemental function. Furthermore, the hot-water-supply unit 304 is provided with a feed pump 17 for supplying hot water, after having exchanged heat with the refrigerant in the plate-type water heat exchanger 16, to the hot-water-supply tank 305, etc.
[0027] When the hot-water-supply operation mode C is performed by the hot-water-supply unit 304, the plate-type water heat exchanger 16 functions as a refrigerant condenser and heats the water to be supplied by the feed pump 17. The feed pump 17 has a function of supplying the water into the hot-water-supply unit 304, making the water exchange heat at the plate-type water heat exchanger 16 so as to turn the water into hot water, and then supplying the hot water into the hot-water-supply tank 305 so as to make the hot water exchange heat with the water in the hot-water-supply tank 305. Specifically, in the hot-water-supply unit 304, the water supplied by the feed pump 17 and the refrigerant flowing through the plate-type water heat exchanger 16 can exchange heat with each other, and the water supplied by the feed pump 17 and the water in the hot-water-supply tank 305 can exchange heat with each other. Moreover, the flow rate of water to be supplied to the plate-type water heat exchanger 16 can be adjusted.
[0028] The hot-water-supply unit 304 is provided with the following various kinds of sensors, which include:

(1) a hot-water-supply liquid temperature sensor 209 that is provided at the liquid side of the plate-type water heat exchanger 16 and detects the temperature of a liquid refrigerant.

[0029] The operation of the feed pump 17 is controlled by the control unit 103 that functions as normal-operation control means that performs the normal operation including the hot-water-supply operation mode C of the hot-water-supply unit 304 (see Fig. 3).

Hot-Water-Supply Tank 305

[0030] The hot-water-supply tank 305 is installed, for example, outdoors and has a function of storing the hot water boiled by the hot-water-supply unit 304. Furthermore, the hot-water-supply tank 305 has one end connected to the hot-water-supply unit 304 via the upstream water pipe 20 and another end connected to the hot-water-supply unit 304 via the downstream water pipe 21, and constitutes a part of a water circuit in the integrated air-conditioning and hot-water-supply system 100. The hot-water-supply tank 305 is of a full-water type that makes the hot water flow out of the upper portion of the tank when the hot water is consumed by the user and that is supplied with water from the lower portion of the tank in accordance with the consumed amount.
[0031] The water fed by the feed pump 17 in the hot-water-supply unit 304 becomes hot water by being heated by the refrigerant at the plate-type water heat exchanger 16 and then travels through the upstream water pipe 20 so as to flow into the hot-water-supply tank 305. The hot water exchanges heat with the water in the hot-water-supply tank 305 as intermediate water without being mixed with the water in the tank, thereby turning into cold water. Subsequently, the water flows out of the hot-water-supply tank 305 and travels through the downstream water pipe 21 so as to flow into the hot-water-supply unit 304 again. After being fed by the feed pump 17 again, the water turns into hot water at the plate-type water heat exchanger 16. As a result of this process, the water is boiled in the hot-water-supply tank 305.
[0032] The method for heating the water in the hot-water-supply tank 305 is not limited to the intermediate-water-based heat exchange method as in Embodiment 1. As an alternative heating method, the water in the hot-water-supply tank 305 may flow directly into a pipe, turn into hot water by exchanging heat at the plate-type water heat exchanger 16, and then return to the hot-water-supply tank 305.

[0033] The hot-water-supply tank 305 is provided with the following various kinds of sensors, which include:

(1) a hot-water-supply-tank water temperature sensor 210 that is provided on a side surface at the lower portion of the hot-water-supply tank 305 and detects the temperature of the hot water in the tank.

Heat Source Unit 301

[0034] The heat source unit 301 is installed, for example, outdoors and is connected to the use-side units 303a and 303b via the liquid extension pipe 6, the gas extension pipe 12, and the branch unit 302. Moreover, the heat source unit 301 is connected to the hot-water-supply unit 304 via the hot-water-supply gas extension pipe 15, the liquid extension pipe 6, and the branch unit 302, and constitutes a part of the refrigerant circuit in the integrated air-conditioning and hot-water-supply system 100.

[0035] The heat source unit 301 includes an outdoor-side refrigerant circuit that constitutes a part of the refrigerant circuit. As elemental devices, the outdoor-side refrigerant circuit has a compressor 1 that compresses the refrigerant, two four-way valves (i.e., a first four-way valve 2 and a second four-way valve 13) for switching the flowing direction of the refrigerant in accordance with the outdoor operation mode, an outdoor heat exchanger 3 as a heat-source-side heat exchanger, and an accumulator 14 for retaining an excess refrigerant. Furthermore, the heat source unit 301 is constituted of an outdoor air-sending device 4 for supplying air to the outdoor heat exchanger 3 and an outdoor pressure-reducing mechanism 5 as a heat-source-side pressure-reducing mechanism for controlling the distributive flow rate of the refrigerant.

[0036] The compressor 1 suctions the refrigerant and compresses this refrigerant to a high-temperature high-pressure state. The compressor 1 equipped in Embodiment 1 is capable of adjusting the operation capacity and is, for example, a positive-displacement compressor that is driven by a motor (not shown) controlled by an inverter. Although only a single compressor 1 is shown as an example in Embodiment 1, the configuration is not limited to this, and two or more compressors 1 may be connected in parallel to each other in accordance with, for example, the connected number of use-side units 303a and 303b and hot-water-supply units 304. Furthermore, a discharge-side pipe connected to the compressor 1 is bifurcated at an intermediate section of the pipe and has one end connected to the gas extension pipe 12 via the second four-way valve 13 and another end connected to the hot-water-supply gas extension pipe 15 via the first four-way valve 2.

[0037] The first four-way valve 2 and the second four-way valve 13 each function as a flow switching device that switches the flowing direction of the refrigerant in accordance with the operation mode of the heat source unit 301.
Fig. 4 illustrates the operational contents of the four-way valves relative to the operation modes. The terms "solid line" and "dash line" shown in Fig. 4 correspond to solid lines and dashed lines shown in Fig. 1 that denote the switched statuses of the first four-way valve 2 and the second four-way valve 13.

[0038] In the cooling operation mode A, the first four-way valve 2 is switched to the "solid line" state. Specifically, in the cooling operation mode A, in order to make the outdoor heat exchanger 3 function as a condenser for the refrigerant compressed by the compressor 1, the first four-way valve 2 is switched so as to connect the discharge side of the compressor 1 to the gas side of the outdoor heat exchanger 3. In the heating operation mode B, the hot-water-supply operation mode C, the simultaneous heating and hot-water-supply operation mode D, or the simultaneous cooling and hot-water-supply operation mode E, the first four-way valve 2 is switched to the "dash line" state. Specifically, in the heating operation mode B, the hot-water-supply operation mode C, the simultaneous heating and hot-water-supply operation mode D, or the simultaneous cooling and hot-water-supply operation mode E, in order to make the outdoor heat exchanger 3 function as a refrigerant evaporator, the first four-way valve 2 is switched so as to connect the discharge side of the compressor 1 to the gas side of the plate-type water heat exchanger 16 and also to connect the suction side of the compressor 1 to the gas side of the outdoor heat exchanger 3.

[0039] In the cooling operation mode A, the hot-water-supply operation mode C, or the simultaneous cooling and hot-water-supply operation mode E, the second four-way valve 13 switched to the "solid line" state. Specifically, the second four-way valve 13 is switched so as to connect the suction side of the compressor 1 to the gas side of the indoor heat exchangers 9a and 9b, such that the indoor heat exchangers 9a and 9b are made to function as evaporators for the refrigerant compressed by the compressor 1 in the cooling operation mode A or the simultaneous cooling and hot-water-supply operation mode E or such that the refrigerant is prevented from flowing to the use-side units 303a and 303b in the hot-water-supply operation mode C. In the heating operation mode B, the hot-water-supply operation mode C, and the simultaneous heating and hot-water-supply operation mode D, the second four-way valve 13 is switched to the "dash line" state. Specifically, in the heating operation mode B, the hot-water-supply operation mode C, and the simultaneous heating and hot-water-supply operation mode D, in order to make the indoor heat exchangers 9a and 9b function as refrigerant condensers, the second four-way valve 13 is switched so as to connect the discharge side of the compressor 1 to the gas side of the indoor heat exchangers 9a and 9b.

[0040] The outdoor heat exchanger 3 has its gas side connected to the first four-way valve 2 and its liquid side connected to the outdoor pressure-reducing mechanism 5. The outdoor heat exchanger 3 may be formed of, for example, a cross-fin-type fin-and-tube heat exchanger constituted of a heat transfer pipe and multiple fins. Alternatively, the outdoor heat exchanger 3 may be formed of a micro-channel heat exchanger, a shell-and-tube heat exchanger, a heat-pipe heat exchanger, or a double-pipe heat exchanger. In the cooling operation mode A, the outdoor heat exchanger 3 functions as a refrigerant condenser and cools the refrigerant. In the heating operation mode B, the hot-water-supply operation mode C, the simultaneous heating and hot-water-supply operation mode D, and the simultaneous cooling and hot-water-supply operation mode E, the outdoor heat exchanger 3 functions as a refrigerant evaporator and heats the refrigerant.

[0041] The outdoor air-sending device 4 has a function of suctioning outdoor air into the heat source unit 301, making the outdoor air exchange heat at the outdoor heat exchanger 3, and then discharging the air to the outside. Specifically, in the heat source unit 301, the outdoor air taken in by the outdoor air-sending device 4 and the refrigerant flowing through the outdoor heat exchanger 3 can exchange heat with each other. The outdoor air-sending device 4 is capable of adjusting the flow rate of air to be supplied to the outdoor heat exchanger 3 and includes a fan, such as a propeller fan, and a motor, such as a DC fan motor, for driving this fan.

[0042] The accumulator 14 is provided at the suction side of the compressor 1 and has a function of retaining the liquid refrigerant to prevent it from flowing back to the compressor 1 when there is a malfunction in the integrated air-conditioning and hot-water-supply system 100 or during a transient response of an operational state caused by a change in operation control.

[0043] The heat source unit 301 is provided with the following various kinds of sensors, which include:

(1) a high-pressure sensor 201 that is provided at the discharge side of the compressor 1 and detects a high-pressure side high pressure;
(2) a discharge temperature sensor 202 that is provided at the discharge side of the compressor 1 and detects a discharge temperature;
(3) an outdoor gas temperature sensor 203 that is provided at the gas side of the outdoor heat exchanger 3 and detects a gas refrigerant temperature;
(4) an outdoor liquid temperature sensor 204 that is provided at the liquid side of the outdoor heat exchanger 3 and detects a liquid refrigerant temperature; and
(5) an outdoor-air temperature sensor 205 that is provided at the outdoor-air suction side of the heat source unit 301 and detects the temperature of outdoor air flowing into the unit.

[0044] The operation of each of the compressor 1, the first four-way valve 2, the outdoor air-sending device 4, the outdoor pressure-reducing mechanism 5, and the second four-way valve 13 is controlled by the control unit 103 that functions as normal-operation control means that performs the normal operation including the cooling operation mode A, the heating operation mode B, the hot-water-supply operation mode C, the simultaneous heating and hot-water-supply operation mode D, and the simultaneous cooling and hot-water-supply operation mode C.

Branch Unit 302

[0045] The branch unit 302 is installed, for example, indoors, is connected to the heat source unit 301 via the liquid extension pipe 6 and the gas extension pipe 12, is connected to the use-side units 303a and 303b via the indoor liquid pipes 8a and 8b and the indoor gas pipes 11a and 11 b, is connected to the hot-water-supply unit 304 via the hot-water-supply liquid pipe 18, and constitutes a part of the refrigerant circuit in the integrated air-conditioning and hot-water-supply system 100. The branch unit 302 has a function of controlling the flow of the refrigerant in accordance with a requested operation in the use-side units 303a and 303b and the hot-water-supply unit 304.

[0046] The branch unit 302 includes a branch refrigerant circuit that constitutes a part of the refrigerant circuit. As elemental devices, the branch refrigerant circuit has indoor pressure-reducing mechanisms 7a and 7b as use-side pressure-reducing mechanisms for controlling the distributive flow rate of the refrigerant, and a hot-water-supply pressure-reducing mechanism 19 for controlling the distributive flow rate of the refrigerant.

[0047] The indoor pressure-reducing mechanisms 7a and 7b are respectively provided in the indoor liquid pipes 8a and 8b. The hot-water-supply pressure-reducing mechanism 19 is provided in the hot-water-supply liquid pipe 18 in the branch unit 302. Each of the indoor pressure-reducing mechanisms 7a and 7b functions as a pressure-reducing valve and an expansion valve, and reduces the pressure of and expands the refrigerant flowing through the liquid extension pipe 6 in the cooling operation mode A and reduces the pressure of and expands the refrigerant flowing through the hot-water-supply pressure-reducing mechanism 19 in the simultaneous cooling and hot-water-supply operation mode E. In the heating operation mode B and the simultaneous heating and hot-water-supply operation mode D, the indoor pressure-reducing mechanisms 7a and 7b reduce the pressure of and expand the refrigerant flowing through the indoor liquid pipes 8a and 8b. The hot-water-supply pressure-reducing mechanism 19 functions as a pressure-reducing valve and an expansion valve and reduces the pressure of and expands the refrigerant flowing through the hot-water-supply liquid pipe 18 in the hot-water-supply operation mode C and the simultaneous heating and hot-water-supply operation mode D. The indoor pressure-reducing mechanisms 7a and 7b and the hot-water-supply pressure-reducing mechanism 19 may each be of a type whose opening degree is variably controllable, such as precise flow control means using an electronic expansion valve or inexpensive refrigerant flow control means such as a capillary tube.

[0048] As shown in Fig. 3, the operation of the hot-water-supply pressure-reducing mechanism 19 is controlled by the control unit 103 of the controller 110, which functions as normal-operation control means that performs the normal operation including the hot-water-supply operation mode C of the hot-water-supply unit 304. The operation of each of the indoor pressure-reducing mechanisms 7a and 7b is controlled by the control unit 103 functioning as normal-operation control means that performs the normal operation including the cooling operation mode A and the heating operation mode B of the use-side units 303a and 303b.

Controller 110

[0049] As shown in Fig. 3, the values detected by the various kinds of temperature and pressure sensors are input to the measuring unit 101 and are processed by the calculating unit 102. Then, based on the processed result of the calculating unit 102, the control unit 103 controls the compressor 1, the first four-way valve 2, the outdoor air-sending device 4, the outdoor pressure-reducing mechanism 5, the indoor pressure-reducing mechanisms 7a and 7b, the indoor air-sending devices 10 and 10b, the second four-way valve 13, the feed pump 17, and the hot-water-supply pressure-reducing mechanism 19. Specifically, the overall operation of the integrated air-conditioning and hot-water-supply system 100 is controlled by the controller 110 equipped with the measuring unit 101, the calculating unit 102, and the control unit 103. The controller 110 may be constituted of a microcomputer. Calculation expressions to be described in Embodiment 1 below are calculated by the calculating unit 102, and the control unit 103 controls each of the devices, such as the compressor 1, in accordance with the calculation results. The storage unit 104 stores data to be used in the calculating unit 102 and the calculation results.

[0050] Specifically, based on commands, such as an operation mode (e.g., a cooling request signal for requesting the cooling operation of the use-side units 303) received via a remote controller, a hot-water-supply request signal, to be described below, and a preset temperature, and information detected by the various sensors, the control unit 103 performs each operation mode by controlling the following:

the operating frequency of the compressor 1,
the switching of the first four-way valve 2,
the rotation speed (including an on/off operation) of the outdoor air-sending device 4,
the opening degree of the outdoor pressure-reducing mechanism 5,
the opening degrees of the indoor pressure-reducing mechanisms 7a and 7b,
the rotation speeds (including an on/off operation) of the indoor air-sending devices 10a and 10b,
the switching of the second four-way valve 13,
the rotation speed (including an on/off operation) of the feed pump 17, and
the opening degree of the hot-water-supply pressure-reducing mechanism 19.

The measuring unit 101, the calculating unit 102, and the control unit 103 may be integrally provided or may be provided independently of each other. Furthermore, the measuring unit 101, the calculating unit 102, and the control unit 103 may be provided in any one of the units. Moreover, the measuring unit 101, the calculating unit 102, and the control unit 103 may be provided in each of the units.

Operation Modes

[0051] The integrated air-conditioning and hot-water-supply system 100 controls each of the devices equipped in the heat source unit 301, the branch unit 302, the use-side units 303a and 303b, and the hot-water-supply unit 304 in accordance with requested air conditioning loads of the use-side units 303a and 303b and a requested hot-water-supply load of the hot-water-supply unit 304. With this control, the integrated air-conditioning and hot-water-supply system 100 performs the cooling operation mode A, the heating operation mode B, the hot-water-supply operation mode C, the simultaneous heating and hot-water-supply operation mode D, or the simultaneous cooling and hot-water-supply operation mode E.

[0052] The simultaneous cooling and hot-water-supply operation mode E further includes a "hot-water-supply priority mode" in which the operating frequency of the compressor 1 is controlled in accordance with a hot-water-supply request signal from the hot-water-supply unit 304 and a "cooling priority mode" in which the operating frequency of the compressor 1 is controlled in accordance with cooling loads of the use-side units 303a and 303b. The hot-water-supply request signal is output from the hot-water-supply unit 304 when the temperature of the water stored in the hot-water-supply tank 305 is lower than a preset hot-water-supply temperature. When the hot-water-supply request signal is output, the control unit 103 estimates a cooling load and a heating load from a temperature difference (i.e., an indoor temperature difference) between an indoor suction temperature and a preset indoor temperature and performs control based on an assumption that the larger the indoor temperature difference, the larger the cooling load and the heating load.

Operation

[0053] Specific refrigerant flowing methods and normal control methods in the cooling operation mode A, the heating operation mode B, the hot-water-supply operation mode C, the simultaneous heating and hot-water-supply operation mode D, and the simultaneous cooling and hot-water-supply operation mode E performed by the integrated air-conditioning and hot-water-supply system 100 will now be described. The operation of each four-way valve in each of the operation modes is as shown in Fig. 4. For each of the hot-water-supply operation mode C, the simultaneous heating and hot-water-supply operation mode D, and the simultaneous cooling and hot-water-supply operation mode E, a control method for high-temperature-water supply will be described in addition to a normal control method.

Cooling Operation Mode A

[0054] In the cooling operation mode, the hot-water-supply pressure-reducing mechanism 19 is completely closed. In the cooling operation mode A, the first four-way valve 2 is in the solid-line state, meaning that the discharge side of the compressor 1 is connected to the gas side of the outdoor heat exchanger 3. Furthermore, the second four-way valve 13 is in the solid-line state, meaning that the suction side of the compressor 1 is connected to the indoor heat exchangers 9a and 9b via the gas extension pipe 12.

[0055] While the refrigerant circuit is in this state, the compressor 1, the outdoor air-sending device 4, and the indoor pressure-reducing mechanisms 7a and 7b are activated. This causes a low-pressure gas refrigerant to be suctioned into and compressed by the compressor 1, thereby becoming a high-temperature high-pressure gas refrigerant. Subsequently, the high-temperature high-pressure gas refrigerant travels through the first four-way valve 2 and flows into the outdoor heat exchanger 3 where the gas refrigerant condenses by exchanging heat with outdoor air supplied by the outdoor air-sending device 4, thereby becoming a high-pressure liquid refrigerant. After flowing out of the outdoor heat exchanger 3, the high-pressure liquid refrigerant flows to the outdoor pressure-reducing mechanism 5 where the high-pressure liquid refrigerant is reduced in pressure. Subsequently, the liquid refrigerant travels through the liquid extension pipe 6 and flows into the branch unit 302. At this time, the outdoor pressure-reducing mechanism 5 is controlled to a maximum opening degree. The refrigerant flowing into the branch unit 302 is reduced in pressure by the indoor pressure-reducing mechanisms 7a and 7b so as to become a low-pressure two-phase gas-liquid refrigerant. Subsequently, the refrigerant flows out of the branch unit 302 and travels through the indoor liquid pipes 8a and 8b so as to flow into the use-side units 303a and 303b.

[0056] The refrigerant flowing into the use-side units 303a and 303b flows into the indoor heat exchangers 9a and 9b where the refrigerant evaporates by exchanging heat with indoor air supplied by the indoor air-sending devices 10a and 10b, thereby becoming a low-pressure gas refrigerant. In this case, each of the indoor pressure-reducing mechanisms 7a and 7b is controlled such that a temperature difference (i.e., a cooled-room temperature difference) obtained by subtracting a preset temperature from an indoor suction temperature detected by the indoor suction temperature sensor 208a or 208b in corresponding use-side unit 303a or 303b is eliminated. Therefore, the flow rate of refrigerant flowing through the indoor heat exchangers 9a and 9b corresponds to the cooling load requested in the air-conditioned space where the use-side units 303a and 303b are installed.

[0057] The refrigerant flowing out of the indoor heat exchangers 9a and 9b flows out of the use-side units 303a and 303b and then travels through the indoor gas pipes 11a and 11b and the branch unit 302 so as to flow into the gas extension pipe 12. The refrigerant then travels through the second four-way valve 13 and passes through the accumulator 14 so as to be suctioned into the compressor 1 again.

[0058] The operating frequency of the compressor 1 is controlled by the control unit 103 such that the evaporating temperature is made equal to a predetermined value. The predetermined evaporating-temperature value is the temperature detected by the indoor liquid temperature sensor 206a or 206b. The predetermined evaporating-temperature value is determined from a temperature difference (i.e., a cooled-room temperature difference), which is obtained by subtracting a preset temperature from an indoor suction temperature detected by the indoor suction temperature sensor 208a or 208b, in the use-side unit 303a or 303b that has the maximum temperature difference in the use-side units 303a and 303b. Fig. 5 illustrates a method for determining a target-evaporating-temperature value from a maximum cooled-room temperature difference in compressor control. Specifically, as shown in Fig. 5, a target-evaporating-temperature value in a corresponding range is set on the basis of a maximum cooled-room temperature difference ΔTje [-]. Target-evaporating-temperature values A1 to A4 in respective maximum cooled-room temperature difference ranges are determined from tests, etc. Furthermore, the quantity of air from the outdoor air-sending device 4 is controlled by the control unit 103 such that the condensing temperature is made equal to a predetermined value in accordance with the outdoor-air temperature detected by the outdoor-air temperature sensor 205. The condensing temperature in this case is a saturation temperature calculated based on the pressure detected by the high-pressure sensor 201.

Heating Operation Mode B

[0059] In the heating operation mode, the hot-water-supply pressure-reducing mechanism 19 (i.e., a first pressure-reducing mechanism) is completely closed. Therefore, the refrigerant does not flow to the first four-way valve 2 and the hot-water-supply unit 304. In the heating operation mode B, the first four-way valve 2 is in the dash-line state, meaning that the discharge side of the compressor 1 is connected to the gas side of the plate-type water heat exchanger 16 (i.e., a first radiator) and the suction side of the compressor 1 is connected to the gas side of the outdoor heat exchanger 3 (i.e., a first evaporator). The second four-way valve 13 is in the dash-line state, meaning that the discharge side of the compressor 1 is connected to the gas side of the indoor heat exchangers 9a and 9b.

[0060] While the refrigerant circuit is in this state, the compressor 1, the outdoor air-sending device 4, the indoor air-sending devices 10a and 10b, and the feed pump 17 are activated. This causes a low-pressure gas refrigerant to be suctioned into and compressed by the compressor 1, thereby becoming a high-temperature high-pressure gas refrigerant. Subsequently, the high-temperature high-pressure gas refrigerant flows through the second four-way valve 13.

[0061] The refrigerant flowing into the second four-way valve 13 flows out of the heat source unit 301 and travels through the gas extension pipe 12 so as to flow into the branch unit 302. Subsequently, the refrigerant travels through the indoor gas pipes 11a and 11 b so as to flow into the use-side units 303a and 303b. The refrigerant flowing into the use-side units 303a and 303b flows into the indoor heat exchangers 9a and 9b where the refrigerant condenses by exchanging heat with indoor air supplied by the indoor air-sending devices 10a and 10b so as to become a high-pressure liquid refrigerant, which then flows out of the indoor heat exchangers 9a and 9b. The refrigerant having heated the indoor air at the indoor heat exchangers 9a and 9b flows out of the use-side units 303a and 303b and travels through the indoor liquid pipes 8a and 8b so as to flow into the branch unit 302. The refrigerant is then reduced in pressure by the indoor pressure-reducing mechanisms 7a and 7b, thereby becoming a low-pressure, two-phase gas-liquid or liquid-phase refrigerant. Subsequently, the refrigerant flows out of the branch unit 302.

[0062] Each of the indoor pressure-reducing mechanisms 7a and 7b is controlled such that a temperature difference (i.e., a heated-room temperature difference) obtained by subtracting a preset indoor temperature from an indoor suction temperature detected by the indoor suction temperature sensor 208a or 208b in corresponding use-side unit 303a or 303b is eliminated. Therefore, the flow rate of refrigerant flowing through the indoor heat exchangers 9a and 9b corresponds to the heating load requested in the air-conditioned space where the use-side units 303a and 303b are installed.

[0063] The refrigerant flowing out of the branch unit 302 travels through the liquid extension pipe 6, flows into the heat source unit 301, passes through the outdoor pressure-reducing mechanism 5, and then flows into the outdoor heat exchanger 3. The opening degree of the outdoor pressure-reducing mechanism 5 is controlled so that it is in a completely open state. The refrigerant flowing into the outdoor heat exchanger 3 evaporates by exchanging heat with outdoor air supplied by the outdoor air-sending device 4, thereby becoming a low-pressure gas refrigerant. This refrigerant flows out of the outdoor heat exchanger 3, travels through the first four-way valve 2, passes through the accumulator 14, and is then suctioned into the compressor 1 again.

[0064] The operating frequency of the compressor 1 is controlled by the control unit 103 such that the condensing temperature is made equal to a target value. The method for determining the condensing temperature is the same as that in the cooling operation. The target condensing-temperature value is determined from a temperature difference (i.e., a heated-room temperature difference), which is obtained by subtracting a preset indoor temperature from an indoor suction temperature detected by the indoor suction temperature sensor 208a or 208b, in the use-side unit 303a or 303b that has the maximum heated-room temperature difference in the use-side units 303a and 303b.

Fig. 6 illustrates a method for determining a target-condensing-temperature value from a maximum heated-room temperature difference in compressor control. Specifically, as shown in Fig. 6, a target-condensing-temperature value in a corresponding range is set on the basis of a maximum heated-room temperature difference ΔTjc [-]. Target-condensing-temperature values B1 to B4 in respective maximum heated-room temperature difference ranges are determined from tests, etc. Furthermore, the quantity of air from the outdoor air-sending device 4 is controlled by the control unit 103 such that the evaporating temperature is made equal to a predetermined value in accordance with the outdoor-air temperature detected by the outdoor-air temperature sensor 205. The evaporating temperature in this case is determined based on the temperature detected by the outdoor liquid temperature sensor 204.

Hot-Water-Supply Operation Mode C

[0065] In the hot-water-supply operation mode C, the first four-way valve 2 is in the dash-line state, meaning that the discharge side of the compressor 1 is connected to the gas side of the plate-type water heat exchanger 16 and the suction side of the compressor 1 is connected to the gas side of the outdoor heat exchanger 3. The second four-way valve 13 is in the solid-line state, meaning that the suction side of the compressor 1 is connected to the indoor heat exchangers 9a and 9b via the gas extension pipe 12.

[0066] While the refrigerant circuit is in this state, the compressor 1, the outdoor air-sending device 4, the indoor air-sending devices 10a and 10b, and the feed pump 17 are activated. This causes a low-pressure gas refrigerant to be suctioned into and compressed by the compressor 1, thereby becoming a high-temperature high-pressure gas refrigerant. Subsequently, the high-temperature high-pressure gas refrigerant flows through the first four-way valve 2.

[0067] The refrigerant flowing into the first four-way valve 2 flows out of the heat source unit 301 and travels through the hot-water-supply gas extension pipe 15 so as to flow into the hot-water-supply unit 304. The refrigerant flowing into the hot-water-supply unit 304 flows into the plate-type water heat exchanger 16 where the refrigerant condenses by exchanging heat with water supplied by the feed pump 17 so as to become a high-pressure liquid refrigerant, which then flows out of the plate-type water heat exchanger 16. The refrigerant having heated the water at the plate-type water heat exchanger 16 flows out of the hot-water-supply unit 304, travels through the hot-water-supply liquid pipe 18, flows into the branch unit 302, and is then reduced in pressure by the hot-water-supply pressure-reducing mechanism 19, thereby becoming a low-pressure two-phase gas-liquid refrigerant. Subsequently, the refrigerant flows out of the branch unit 302 and flows into the heat source unit 301 via the liquid extension pipe 6.

[0068] In the hot-water-supply operation mode, the opening degree of the hot-water-supply pressure-reducing mechanism 19 is controlled by the control unit 103 such that the degree of subcooling at the liquid side of the plate-type water heat exchanger 16 is made equal to a predetermined value. The degree of subcooling at the liquid side of the plate-type water heat exchanger 16 is determined by calculating a saturation temperature (i.e., a calculated condensing temperature) from a pressure (i.e., a high pressure) detected by the high-pressure sensor 201 (i.e., a high-pressure sensor) and then subtracting a temperature detected by the hot-water-supply liquid temperature sensor 209 therefrom. The hot-water-supply pressure-reducing mechanism 19 controls the flow rate of refrigerant flowing through the plate-type water heat exchanger 16 so that the degree of subcooling of the refrigerant at the liquid side of the plate-type water heat exchanger 16 is made equal to the predetermined value. Therefore, the high-pressure liquid refrigerant condensed by the plate-type water heat exchanger 16 turns into a state with a predetermined degree of subcooling. Accordingly, the flow rate of refrigerant flowing through the plate-type water heat exchanger 16 corresponds to a hot-water-supply request according to the usage condition of hot water in a facility where the hot-water-supply unit 304 is installed.

[0069] The refrigerant flowing out of the branch unit 302 travels through the liquid extension pipe 6, flows into the heat source unit 301, passes through the outdoor pressure-reducing mechanism 5, and then flows into the outdoor heat exchanger 3. The opening degree of the outdoor pressure-reducing mechanism 5 is controlled so that it is in a completely open state. The refrigerant flowing into the outdoor heat exchanger 3 evaporates by exchanging heat with outdoor air supplied by the outdoor air-sending device 4, thereby becoming a low-pressure gas refrigerant. This refrigerant flows out of the outdoor heat exchanger 3, travels through the first four-way valve 2, passes through the accumulator 14, and is then suctioned into the compressor 1 again.

[0070] The operating frequency of the compressor 1 is controlled to a high value by the control unit 103. Specifically, in the case of the hot-water-supply operation, the controller 110 ensures a high hot-water-supply capacity so as to increase the water temperature in the hot-water-supply tank 305 to a preset hot-water-supply temperature as quickly as possible in response to a hot-water-supply request signal detected by the hot-water-supply-tank water temperature sensor 210. Furthermore, the quantity of air from the outdoor air-sending device 4 is controlled by the control unit 103 such that the evaporating temperature is made equal to a predetermined value in accordance with the outdoor-air temperature detected by the outdoor-air temperature sensor 205. The evaporating temperature in this case is the temperature detected by the outdoor liquid temperature sensor 204.

[0071] If the hot-water-supply temperature is high (e.g., 60 degrees C), the inlet water temperature (i.e., the temperature of water flowing into the connection section 25) of the plate-type water heat exchanger 16 also becomes high, causing the condensing temperature to increase. In this case, if the operating frequency of the compressor 1 is controlled to a high value, the high pressure increases to a value outside an appropriate operating range of the compressor 1. Therefore, if a condensing temperature calculated from a detected value of the high-pressure sensor 201 reaches an upper limit value (e.g., 60 degrees C), condensing-temperature control shown in expressions (1) and (2) is performed on the compressor 1 so as to prevent the condensing temperature from increasing.

[0072] [Math. 1]

$$Fm = F + \Delta F \qquad (1)$$

[0073] [Math. 2]

$$\Delta F = (CTm - CT) \times k_{CT,comp} \qquad (2)$$

[0074] In this case, Fm denotes a target operating frequency [Hz] of the compressor 1, F denotes a current operating frequency [Hz] of the compressor 1, ΔF denotes a change [Hz] in the operating frequency of the compressor 1, CTm denotes a target condensing-temperature value [degrees C], CT denotes a calculated condensing temperature [degrees C], and $k_{CT,comp}$ denotes gain compensation [-] for a change in the operating frequency of the compressor.
The target condensing-temperature value CTm is, for example, a maximum condensing-temperature value (e.g., 60 degrees C) allowable in an appropriate usage range of the compressor 1. The condensing temperature CT is a saturation temperature calculated from the pressure detected by the high-pressure sensor 201. The gain compensation $k_{CT,comp}$ for a change in the operating frequency of the compressor is set to a value based on tests or simulation such that the condensing temperature CT does not increase from the target condensing-temperature value CTm and that the frequency does not decrease rapidly. Although the high-pressure sensor 201 is provided between the compressor 1 and the first four-way valve 2 in Embodiment 1, the configuration is not limited to this. The high-pressure sensor 201 may be provided at any position between the liquid side of the hot-water-supply pressure-reducing mechanism 19 and the discharge side of the compressor 1, which is located at the high-pressure side of the refrigeration cycle. If the high-pressure sensor 201 is disposed between the first four-way valve 2 and the liquid side of the hot-water-supply pressure-reducing mechanism 19, an additional pressure sensor for determining the condensing temperature in the heating operation mode B is disposed between the compressor 1 and the second four-way valve 13.

[0075] If the calculated condensing temperature CT reaches the target condensing-temperature value CTm during the high-temperature-water supply, CT becomes higher than CTm. In that case, the operating frequency of the compressor 1 is decreased in accordance with expressions (1) and (2), whereby the condensing temperature CT can be prevented from being higher than the target condensing-temperature value CTm. When the operating frequency of the compressor 1 decreases, the hot-water-supply capacity decreases. In order to adjust the amount of decrease in the hot-water-supply capacity, pressure-reducing-mechanism opening-degree control is performed so that a predetermined hot-water-supply capacity can be ensured. In Embodiment 1, the opening degree of the hot-water-supply pressure-reducing mechanism 19 is controlled. Specifically, the opening degree of the hot-water-supply pressure-reducing mechanism 19 is controlled in accordance with expressions (3) and (4) so that the predetermined hot-water-supply capacity can be ensured.

[0076] [Math. 3]

$$S_j = S_{j-1} + \Delta S_j \qquad (3)$$

[0077] [Math. 4]

$$\Delta S_j = (S_j m - S_{j-1}) \qquad (4)$$

[0078] In this case, S_j denotes an opening degree [pulse] of the pressure-reducing mechanism after changing the opening degree thereof, S_{j-1} denotes a current opening degree [pulse] of the pressure-reducing mechanism, ΔS_j denotes a change [pulse] in the opening degree of the pressure-reducing mechanism, and $S_j m$ denotes a target opening degree [pulse] of the pressure-reducing mechanism (sometimes referred to as "target pressure-reducing-mechanism opening-degree value").

[0079] The target opening degree $S_j m$ [pulse] of the pressure-reducing mechanism can be determined at the development stage in the following manner.
Fig. 7 illustrates the relationship between the hot-water-supply capacity and the operation efficiency. Fig. 7(a) illustrates the hot-water-supply capacity of the plate-type water heat exchanger 16 relative to the opening degree of the hot-water-supply pressure-reducing mechanism 19. The abscissa axis denotes the opening degree of the hot-water-supply pressure-reducing mechanism 19, whereas the ordinate axis denotes a target hot-water-supply capacity value of the plate-type water heat exchanger 16. Fig. 7(b) illustrates the operation efficiency (COP) relative to the opening degree of the hot-water-supply pressure-reducing mechanism 19. The abscissa axis denotes the opening degree of the hot-water-supply pressure-reducing mechanism 19, whereas the ordinate axis denotes the operation efficiency. When the inlet water temperature during high-temperature-water supply increases and condensing-temperature control is to be performed on the compressor 1, the hot-water-supply capacity of the plate-type water heat exchanger 16 and the operation efficiency (COP) change as shown in Figs. 7(a) and 7(b) relative to the opening degree of the hot-water-supply pressure-

reducing mechanism 19. Because the operating frequency of the compressor 1 becomes higher as the opening degree of the hot-water-supply pressure-reducing mechanism 19 increases, the hot-water-supply capacity increases. In contrast, the operation efficiency decreases. The target opening degree S_jm of the pressure-reducing mechanism can be set based on Fig. 7 as an opening degree that achieves a minimum-required hot-water-supply capacity to be ensured. Specifically, the target opening-degree value is set in correspondence with a target value for the hot-water-supply capacity (i.e., heat-radiation capacity) of the plate-type water heat exchanger 16 (i.e., first radiator). The target opening degree S_jm of the pressure-reducing mechanism is determined based on tests or simulation at the development stage. Furthermore, as the hot-water-supply temperature becomes higher and the inlet water temperature increases (i.e., as CT increases when CT>CTm), the operating frequency of the compressor 1 is decreased by performing the condensing-temperature control (i.e., expressions (1) and (2)) on the compressor 1, causing the hot-water-supply capacity to decrease. Therefore, the target opening degree is determined when the inlet water temperature is at the maximum. The inlet water temperature is estimated such that, for example, when the maximum value of the hot-water-supply temperature is 60 degrees C and the hot-water-supply capacity is a rated hot-water-supply capacity, the amount of flowing water causes the temperature difference between the inlet water temperature and the outlet water temperature of the plate-type water heat exchanger 16 to be 5 degrees C. In this case, since the hot-water-supply temperature is 60 degrees C, the outlet water temperature is 60 degrees C and the inlet water temperature is 55 degrees C. In other words, the maximum inlet water temperature is 55 degrees C. Because the hot-water-supply capacity increases as the inlet water temperature decreases, the minimum-required hot-water-supply capacity (i.e., the heat-radiation capacity of the plate-type water heat exchanger 16) can be ensured by determining the target opening degree when the inlet water temperature is at the maximum. Furthermore, it is obvious from Fig. 7 that, by lowering the target hot-water-supply capacity and lowering the target opening degree S_jm, the operation efficiency can be increased.
The target hot-water-supply capacity value of the plate-type water heat exchanger 16 may be set in correspondence with an upper limit value in design for the inlet water temperature of the water flowing into the water pipe of the plate-type water heat exchanger 16 from the downstream water pipe 21.

[0080] When an operation is actually performed with the target opening degree S_jm of the pressure-reducing mechanism described above, the condensing-temperature control is performed on the compressor 1, and the operation is performed with the target opening degree S_jm of the pressure-reducing mechanism as a fixed value regardless of the operating frequency of the compressor 1. Therefore, the minimum-required hot-water-supply capacity can be ensured when the inlet water temperature is 55 degrees C, and the operating frequency of the compressor is increased when the inlet water temperature is low at 54 degrees C or 53 degrees C. Because the hot-water-supply capacity increases in proportion to the operating frequency of the compressor, the hot-water-supply capacity is excessive when the inlet water temperature is low, leading to reduced operating efficiency even though the time required for completing the hot-water-supply operation can be shortened. If the minimum-required hot-water-supply capacity can be ensured, it is desirable that the hot-water-supply operation be performed at the highest possible operation efficiency. Therefore, when the inlet water temperature is low at 54 degrees C or 53 degrees C, the opening degree of the hot-water-supply pressure-reducing mechanism 19 may be reduced to suppress an excessive hot-water-supply capacity, so that the minimum-required hot-water-supply capacity can be ensured. Reducing the opening degree of the hot-water-supply pressure-reducing mechanism 19 causes a pressure difference in the hot-water-supply pressure-reducing mechanism 19 to increase and the condensing temperature to increase, resulting in a lower operating frequency of the compressor 1.

[0081] Fig. 8 illustrates tests performed when performing control for changing the target opening-degree value of the hot-water-supply pressure-reducing mechanism in accordance with the frequency of the compressor. The contents of the tests are shown in Fig. 8 for explaining how the control is performed in detail. For determining the target opening degree of the pressure-reducing mechanism at the development stage mentioned above, the tests are performed when the inlet water temperature is at the maximum at 55 degrees C and also when the inlet water temperature is 54 degrees C and 53 degrees C, and target opening degrees S_jm of the pressure-reducing mechanism that achieve the minimum-required hot-water-supply capacity to be ensured when the condensing-temperature control is performed on the compressor 1 (the target condensing temperature is set to, for example, 60 degrees C) are determined. In this case, a compressor frequency F is also recorded, and a function f(F) of the target opening degree S_jm of the pressure-reducing mechanism relative to the compressor frequency F is created from a point obtained from each test. The function of the target opening degree S_jm of the pressure-reducing mechanism can be obtained with higher accuracy by increasing the number of tested inlet-water-temperature points. Furthermore, because the operating frequency of the compressor 1 becomes higher and the refrigerant flow rate increases as the inlet water temperature decreases, the target opening degree S_jm of the pressure-reducing mechanism also increases. In the actual operation, when the condensing-temperature control is performed on the compressor 1, the target opening degree S_jm of the pressure-reducing mechanism is determined from the function f(F) shown in expression (5), which is created at the development stage. The controller 110 stores the following expression (5) in the storage unit 104 as frequency/opening-degree correspondence information.

[0082] [Math. 5]

$$S_j m = f(F) \qquad (5)$$

[0083] By performing the operation in this manner, high operating efficiency can be achieved while the minimum-required hot-water-supply capacity is ensured.

[0084] Fig. 9 illustrates the relationship between the outdoor-air temperature and the target opening-degree value. As shown in Fig. 9, since the pressure at the low-pressure side increases and the pressure at the high-pressure side increases as the outdoor-air temperature increases, the operating frequency of the compressor 1 decreases, causing the target opening-degree value $S_j m$ for ensuring the hot-water-supply capacity to increase. By changing the target opening-degree value $S_j m$ in accordance with the outdoor-air temperature, a constant hot-water-supply capacity can also be ensured relative to a change in the outdoor-air temperature, such as an increase in the outdoor-air temperature. The controller 110 stores the relationship between the outdoor-air temperature and the target opening-degree value shown in Fig. 9 in the storage unit 104 as outdoor-air-temperature/opening-degree correspondence information. When the condensing-temperature control and the opening-degree control are concurrently performed, the control unit 103 of the controller 110 refers to the outdoor-air-temperature/opening-degree correspondence information so as to identify a target opening-degree value corresponding to an outdoor-air temperature detected by the outdoor-air temperature sensor 205 from the outdoor-air-temperature/opening-degree correspondence information, and uses the identified target opening-degree value as a target opening-degree value in the opening-degree control.

[0085] For determining the target opening degree $S_j m$ of the pressure-reducing mechanism, a target opening degree is determined from tests performed at the development stage so as to achieve a constant hot-water-supply capacity. However, because there are individual differences among pressure-reducing mechanisms in actuality, a constant hot-water-supply capacity is sometimes not achieved even if the same pressure-reducing mechanism is used. The following configuration can be used to solve this problem. By determining the hot-water-supply capacity directly from the operational state of the actual system in operation and setting a target opening degree of the pressure-reducing mechanism that can at least ensure a "target constant hot-water-supply capacity" by using the determined hot-water-supply capacity, variations in hot-water-supply capacity caused by individual differences among pressure-reducing mechanisms or degradation over time can be prevented, thereby preventing an unexpected decrease in hot-water-supply capacity.

[0086] Fig. 10 illustrates the relationships among a hot-water-supply capacity Qc, an evaporating capacity Qe, and a compressor input W. The following description relates to a specific method. A sum of the evaporating capacity of the outdoor heat exchanger 3 and the input of the compressor 1 is equal to the hot-water-supply capacity of the plate-type water heat exchanger 16. Therefore, the hot-water-supply capacity is determined by determining the evaporating capacity of the outdoor heat exchanger 3 and the input of the compressor 1 (i.e., compressing work done on the refrigerant by the compressor 1). A table showing the evaporating capacity relative to a temperature difference between the outdoor-air temperature and the evaporating temperature is created based on tests, and the evaporating capacity of the outdoor heat exchanger 3 is determined by using the table.

Fig. 11 illustrates the contents of tests performed at the development stage when performing control for changing the target opening-degree value in accordance with the hot-water-supply capacity. The contents of the tests are shown in Fig. 11. The condensing-temperature control is performed on the compressor 1, and the opening degree of the hot-water-supply pressure-reducing mechanism 19 that can ensure the hot-water-supply capacity at a maximum inlet water temperature of 55 degrees C is determined. The "difference between the outdoor-air temperature and the evaporating temperature" and the "evaporating capacity" of the outdoor heat exchanger 3 at that point are recorded. In Embodiment 1, the evaporating temperature is based on a detected value of the outdoor liquid temperature sensor 204. Subsequently, in a state where the opening degree is slightly changed (to, for example, about 50 pulses) from the previously-determined opening degree of the hot-water-supply pressure-reducing mechanism 19, the "difference between the outdoor-air temperature and the evaporating temperature" and the evaporating capacity of the outdoor heat exchanger 3 at that point are recorded. The blanks in Fig. 11 are filled in this manner. By completing the table in Fig. 11 and applying it to the actual operation, the evaporating capacity can be calculated from the outdoor-air temperature and the evaporating temperature. If a difference between the outdoor-air temperature and the evaporating temperature that is not determined in the tests is detected in the actual operation, the values on the table are linearly-interpolated so as to determine the evaporating capacity. Specifically, the relationship between the "difference between the outdoor-air temperature and the evaporating temperature" and the evaporating capacity obtained in Fig. 11 is input to the controller 110. The controller 110 interpolates the results of the relationship (three sets thereof in Fig. 11) between the "difference between the outdoor-air temperature and the evaporating temperature" and the evaporating capacity and calculates a function of the evaporating capacity and the "difference between the outdoor-air temperature and the evaporating temperature".

[0087] The input W [kW] of the compressor 1 can be calculated from the operating frequency F [Hz] of the compressor 1, the condensing temperature CT [degrees C], and an evaporating temperature ET [degrees C] by using the following expression (6). The degree of superheat at the inlet of the compressor is simply set to zero.

[0088] [Math. 6]

$$W = f(F, CT, ET) \qquad (6)$$

[0089] The operating frequency F of the compressor 1 is obtained as operation information. The condensing temperature CT is obtained as a saturation pressure detected by the high-pressure sensor 201. The evaporating temperature ET is determined in a manner similar to how the evaporating capacity is calculated. Accordingly, since the evaporating capacity Qe [kW] and the input W [kW] of the compressor 1 can be determined, the hot-water-supply capacity Qc [kW] can be determined from expression (7).

[0090] [Math. 7]

$$Qc = Qe + W \qquad (7)$$

[0091] The target opening degree $S_j m$ of the pressure-reducing mechanism can be determined from the determined hot-water-supply capacity Qc and a target minimum-required hot-water-supply capacity value Qcm [kW].

[0092] [Math. 8]

$$Sj,m = (Qcm - Qc) \times k_{Qc,Sjm} \qquad (8)$$

[0093] In this case, $k_{Qc,Sjm}$ denotes gain compensation [-] for a change in the target opening degree of the pressure-reducing mechanism and is determined from tests or simulation. By determining the hot-water-supply capacity from the evaporating capacity and the input of the compressor 1 in this manner, the target opening degree $S_j m$ of the pressure-reducing mechanism is determined. Accordingly, variations in hot-water-supply capacity caused by individual differences among pressure-reducing mechanisms can be suppressed, so that the minimum-required hot-water-supply capacity can be ensured during the high-temperature-water supply in any actual system. Because the target opening degree $S_j m$ of the pressure-reducing mechanism is calculated by determining the hot-water-supply capacity by using the outdoor-air temperature in this method, the outdoor-air temperature compensation shown in Fig. 9 is not necessary.

[0094] This will be described in detail below. The controller 110 receives data of two or more sets of a temperature difference between the outdoor-air temperature around the outdoor heat exchanger 3 and the evaporating temperature of the outdoor heat exchanger 3 and the evaporating capacity of the outdoor heat exchanger 3 corresponding to this temperature difference. Based on the input data, the controller 110 determines a functional relationship between the temperature difference and the evaporating capacity by interpolation and refers to the determined functional relationship so as to identify, from the functional relationship, the evaporating capacity that corresponds to the temperature difference between the outdoor-air temperature detected by the outdoor-air temperature sensor 205 and the evaporating temperature detected by the outdoor liquid temperature sensor 204. Then, the controller 110 calculates a compressor input W, which indicates the compressing work done on the refrigerant by the compressor, from the operating frequency of the compressor 1, the calculated condensing temperature, and the evaporating temperature detected by the outdoor liquid temperature sensor 204 (expression (6)). Furthermore, the controller 110 calculates the hot-water-supply capacity Qc of the plate-type water heat exchanger 16 from the identified evaporating capacity Qe and the calculated compressor input W (expression (7)). The controller 110 determines a target opening-degree value in accordance with a difference between the calculated hot-water-supply capacity Qc and a preliminarily-stored target hot-water-supply-capacity value Qcm, and uses the determined target opening-degree value as a target opening-degree value in the opening-degree control (expression (8)).

[0095] Fig. 12 is a flowchart illustrating the flow for determining whether the high-temperature-water supply is to be performed or the hot-water-supply (i.e., normal hot-water supply) other than the high-temperature-water-supply is to be performed. First, in step S11, the controller 110 determines whether the condensing temperature has increased to a value higher than a predetermined value CTm. The predetermined value CTm for the condensing temperature is, for example, a maximum value (e.g., 60 degrees C) of an appropriate usage range of the compressor 1. If the condensing temperature CT has increased to a value higher than the predetermined value, the process proceeds to step S12 where the high-temperature-water-supply operation is performed by performing the condensing-temperature control shown in expressions (1) and (2) on the compressor 1 and performing the opening-degree control shown in expressions (3) and (4) on the hot-water-supply pressure-reducing mechanism 19. If the condensing temperature CT is lower than the predetermined value, the process proceeds to step S13 where the normal hot-water-supply operation is performed by performing normal control on the compressor 1 and the hot-water-supply pressure-reducing mechanism 19. This reliably allows for switching to high-temperature-water supply control in response to an increase in the condensing temperature CT, thereby suppressing an increase in the condensing temperature.

[0096] With the above process, the hot-water-supply operation is performed in response to a hot-water-supply request, and condensing-temperature control is performed on the compressor and opening-degree control is performed on the pressure-reducing mechanism during the high-temperature-water supply in which the condensing temperature becomes higher than the predetermined value CTm, thereby suppressing an excessive increase in high pressure and achieving a predetermined hot-water-supply capacity.

[0097] Although the hot-water-supply pressure-reducing mechanism 19 is used as a pressure-reducing mechanism whose opening degree is controlled during the high-temperature-water supply in which the condensing temperature CT becomes higher than or equal to the predetermined value CTm in Embodiment 1, this is merely an example. The controlled subject is not limited to the hot-water-supply pressure-reducing mechanism 19, and the opening-degree control may alternatively be performed on the outdoor pressure-reducing mechanism 5. In this case, similar to how the opening degree of the outdoor pressure-reducing mechanism 5 is controlled so that it is in a completely open state when the hot-water-supply pressure-reducing mechanism 19 is used as a pressure-reducing mechanism whose opening degree is controlled, the opening degree of the hot-water-supply pressure-reducing mechanism 19 is controlled so that it is in a completely open state.

[0098] Furthermore, although the integrated air-conditioning and hot-water-supply system 100 is described as an example in Embodiment 1, the high-temperature-water-supply control according to the technology developed in the present invention can also be applied to the hot-water-supply operation in a hot-water-supply system in which the heat source unit 301 and the hot-water-supply unit 304 are connected by a refrigerant communication pipe, specifically, a hot-water-supply system that does not have an air-conditioning function but is only capable of performing the hot-water-supply operation.

[0099] Furthermore, although an R410A refrigerant whose operating pressure becomes lower than or equal to the critical pressure is used as the refrigerant in Embodiment 1, the refrigerant is not limited to an R410A refrigerant and may alternatively be, for example, a refrigerant, such as a CO_2 refrigerant, whose operating pressure becomes higher than or equal to the critical pressure (i.e., a refrigerant whose pressure at the high-pressure side, such as the pressure at the discharge side of the compressor, becomes higher than or equal to the critical pressure). In this case, when the pressure (high pressure) detected by the high-pressure sensor 201 of the controller becomes higher than or equal to a predetermined high pressure (e.g., 14.5 MPaG when a CO_2 refrigerant is used), high-pressure control shown in expressions (9) and (10) is performed on the compressor 1 so as to prevent the high pressure from increasing.

[0100] [Math. 9]

$$Fm = F + \Delta F \qquad (9)$$

[0101] [Math. 10]

$$\Delta F = (Pm_{high} - P_{high}) \times k_{P,comp} \qquad (10)$$

[0102] In this case, Fm denotes a target operating frequency [Hz] of the compressor 1, F denotes a current operating frequency [Hz] of the compressor 1, ΔF denotes a change [Hz] in the operating frequency of the compressor 1, Pm_{high} denotes a target high-pressure value [MPaG], P_{high} denotes a calculated condensing temperature [MPaG], and $k_{P,comp}$ denotes gain compensation [-] for a change in the operating frequency of the compressor.

The target high-pressure value Pm_{high} is, for example, a maximum high-pressure value (e.g., 14.5 MPaG when a CO_2 refrigerant is used) allowable in the appropriate usage range of the compressor 1. Furthermore, in order to adjust the amount of decrease in hot-water-supply capacity, the opening degree of the hot-water-supply pressure-reducing mechanism 19 is controlled based on expressions (3) and (4) so that a predetermined hot-water-supply capacity can be ensured. By performing the control in this manner, the technology according to the present invention can be applied to a refrigerant that operates at the critical pressure or higher, similar to a refrigerant that operates at the critical pressure or lower, such as an R410A refrigerant, thereby suppressing an excessive increase in high pressure during the high-temperature-water supply so as to achieve the predetermined hot-water-supply capacity.

Simultaneous Heating and Hot-Water-Supply Operation Mode D

[0103] In the simultaneous heating and hot-water-supply operation mode D (i.e., concurrent heat-radiation operation), the first four-way valve 2 is in the "dash-line" state in Fig. 4. This means that the discharge side of the compressor 1 is connected to the gas side of the plate-type water heat exchanger 16, and the suction side of the compressor 1 is connected to the gas side of the outdoor heat exchanger 3. The second four-way valve 13 is in the "dash-line" state. This means that the discharge side of the compressor 1 is connected to the gas side of the indoor heat exchangers 9a and 9b. Although both the first four-way valve 2 and the second four-way valve 13 are in the "dash-line" state, as in the "heating operation mode", the hot-water-supply pressure-reducing mechanism 19 is open in the simultaneous heating and hot-water-supply operation mode D, unlike in the "heating operation mode" in which the hot-water-supply pressure-reducing mechanism 19 is closed.

[0104] While the refrigerant circuit is in this state, the compressor 1, the outdoor air-sending device 4, the indoor air-sending devices 10a and 10b, and the feed pump 17 are activated. This causes a low-pressure gas refrigerant to be suctioned into and compressed by the compressor 1, thereby becoming a high-temperature high-pressure gas refrigerant. Subsequently, the high-temperature high-pressure gas refrigerant is distributed so as to flow through the first four-way valve 2 and the second four-way valve 13.

[0105] The refrigerant flowing into the first four-way valve 2 flows out of the heat source unit 301 and travels through the hot-water-supply gas extension pipe 15 so as to flow into the hot-water-supply unit 304. The refrigerant flowing into the hot-water-supply unit 304 flows into the plate-type water heat exchanger 16 where the refrigerant condenses by exchanging heat with water supplied by the feed pump 17 so as to become a high-pressure liquid refrigerant, which then flows out of the plate-type water heat exchanger 16. The refrigerant having heated the water at the plate-type water heat exchanger 16 flows out of the hot-water-supply unit 304, travels through the hot-water-supply liquid pipe 18, flows into the branch unit 302, and is then reduced in pressure by the hot-water-supply pressure-reducing mechanism 19, thereby becoming a low-pressure two-phase gas-liquid refrigerant. Subsequently, the refrigerant merges with the refrigerant flowing from the indoor pressure-reducing mechanisms 7a and 7b and flows out of the branch unit 302. A flow path branching from the discharge side of the compressor 1 and extending through the second four-way valve 13, the indoor heat exchangers 9a and 9b, and the indoor pressure-reducing mechanisms 7a and 7b serves as a branch flow path relative to a flow path for the hot-water-supply operation.

[0106] The opening degree of the hot-water-supply pressure-reducing mechanism 19 is controlled by the control unit 103 such that the degree of subcooling at the liquid side of the plate-type water heat exchanger 16 is made equal to a predetermined value. The degree of subcooling at the liquid side of the plate-type water heat exchanger 16 is similar to that in the hot-water-supply operation. The hot-water-supply pressure-reducing mechanism 19 controls the flow rate of refrigerant flowing through the plate-type water heat exchanger 16 so that the degree of subcooling of the refrigerant at the liquid side of the plate-type water heat exchanger 16 is made equal to the predetermined value. Therefore, the high-pressure liquid refrigerant condensed by the plate-type water heat exchanger 16 turns into a state with a predetermined degree of subcooling. Accordingly, the flow rate of refrigerant flowing through the plate-type water heat exchanger 16 corresponds to a hot-water-supply request according to the usage condition of hot water in the facility where the hot-water-supply unit 304 is installed.

[0107] On the other hand, the refrigerant flowing into the second four-way valve 13 flows out of the heat source unit 301 and travels through the gas extension pipe 12 so as to flow to the branch unit 302. Subsequently, the refrigerant travels through the indoor gas pipes 11a and 11b so as to flow into the use-side units 303a and 303b. The refrigerant flowing into the use-side units 303a and 303b flows into the indoor heat exchangers 9a and 9b where the refrigerant condenses by exchanging heat with indoor air supplied by the indoor air-sending devices 10a and 10b so as to become a high-pressure liquid refrigerant, which then flows out of the indoor heat exchangers 9a and 9b. The refrigerant having heated the indoor air at the indoor heat exchangers 9a and 9b flows out of the use-side units 303a and 303b and travels through the indoor liquid pipes 8a and 8b so as to flow into the branch unit 302. The refrigerant is then reduced in pressure by the indoor pressure-reducing mechanisms 7a and 7b, thereby becoming a low-pressure, two-phase gas-liquid or liquid-phase refrigerant. Subsequently, the refrigerant flowing out of the indoor pressure-reducing mechanisms 7a and 7b merges with the refrigerant flowing from the hot-water-supply pressure-reducing mechanism 19 and flows out of the branch unit 302.

[0108] Each of the indoor pressure-reducing mechanisms 7a and 7b is controlled such that a temperature difference (i.e., a heated-room temperature difference) obtained by subtracting a preset indoor temperature from an indoor suction temperature detected by the indoor suction temperature sensor 208a or 208b (i.e., an indoor temperature sensor) in corresponding use-side unit 303a or 303b is eliminated. Therefore, the flow rate of refrigerant flowing through the indoor heat exchangers 9a and 9b corresponds to the heating load requested in the air-conditioned space where the use-side units 303a and 303b are installed.

[0109] The refrigerant flowing out of the branch unit 302 travels through the liquid extension pipe 6, flows into the heat source unit 301, passes through the outdoor pressure-reducing mechanism 5, and then flows into the outdoor heat exchanger 3. The opening degree of the outdoor pressure-reducing mechanism 5 is controlled so that it is in a completely open state. The refrigerant flowing into the outdoor heat exchanger 3 evaporates by exchanging heat with outdoor air supplied by the outdoor air-sending device 4, thereby becoming a low-pressure gas refrigerant. This refrigerant flows out of the outdoor heat exchanger 3, travels through the first four-way valve 2, passes through the accumulator 14, and is then suctioned into the compressor 1 again.

[0110] Since there is a hot-water-supply request signal detected by the hot-water-supply-tank water temperature sensor 210, the operating frequency of the compressor 1 is controlled to a high value by the control unit 103 so that a high hot-water-supply capacity can be ensured. The quantity of air from the outdoor air-sending device 4 is controlled by the control unit 103 such that the evaporating temperature is made equal to a predetermined value in accordance with the outdoor-air temperature detected by the outdoor-air temperature sensor 205. The evaporating temperature in this case is the temperature detected by the outdoor liquid temperature sensor 204.

[0111] If the hot-water-supply temperature is a high temperature (e.g., 60 degrees C), the inlet water temperature of the plate-type water heat exchanger 16 also becomes high, causing the condensing temperature to increase. Unlike the case of the hot-water-supply operation, the heating operation is performed in the use-side units 303a and 303b in the simultaneous heating and hot-water-supply operation mode D. Therefore, even if the condensing-temperature control shown in expressions (1) and (2) is performed on the compressor 1 and the opening-degree control shown in expressions (3) and (4) is performed on the hot-water-supply pressure-reducing mechanism 19, the hot-water-supply capacity sometimes cannot be ensured, and the opening degree of the hot-water-supply pressure-reducing mechanism 19 is controlled regardless of the state in the heated room. Consequently, a heating capacity cannot be ensured in the use-side units 303a and 303b, possibly resulting in a non-heated state. Therefore, in the case of the simultaneous heating and hot-water-supply operation, the simultaneous heating and hot-water-supply operation is stopped if the condensing temperature CT increases to a value higher than a predetermined value. Then, the controller 110 performs a switching process for alternately switching between the heating operation and the hot-water-supply operation so that the heating operation and the hot-water-supply operation are performed.

[0112] Fig. 13 is a flowchart illustrating the flow of an operation method during the high-temperature-water supply in the simultaneous heating and hot-water-supply operation. Specifically, the operation is performed in accordance with the flowchart shown in Fig. 13. First, in step S21, it is determined whether or not the condensing temperature has increased to a value higher than a predetermined value. Similar to the case of the hot-water-supply temperature, the predetermined value for the condensing temperature CT is, for example, a maximum condensing-temperature value (e.g., 60 degrees C) allowable in the appropriate usage range of the compressor 1. If the condensing temperature CT is lower than or equal to the predetermined value, normal control is continuously performed in the simultaneous heating and hot-water-supply operation in step S22. If the condensing temperature is above the predetermined value, the mode is changed to a heating operation mode in step S23. In this case, the use-side units 303a and 303b are set in a heating thermostat OFF state, and the following control is performed for the purpose of changing to a hot-water-supply operation mode. In the heating operation, the indoor pressure-reducing mechanisms 7a and 7b are normally controlled so that a "heated-room temperature difference", which is equal to "indoor suction temperature (detected by indoor suction temperature sensor) - preset indoor temperature", is eliminated. The indoor pressure-reducing mechanisms 7a and 7b are controlled so that the "heated-room temperature difference" is a positive value, such as +1 degree C (i.e., a predetermined positive value) (S23). Moreover, the operating frequency of the compressor 1 is controlled so that the condensing temperature CT is made equal to the target value CTm. Normally, the target value CTm for the condensing temperature is determined from a "heated-room temperature difference" in the use-side unit 303a or 303b that has the maximum heated-room temperature difference. However, the target value CTm for the condensing temperature is determined from a "heated-room temperature difference of -1 degree C" in the use-side unit 303a or 303b that has the maximum heated-room temperature difference of -1 degree C. By performing the control in this manner, the "heated-room temperature difference" (i.e., indoor suction temperature - preset indoor temperature) can be made equal to +1 degree C.

[0113] Subsequently, in step S24, it is determined whether the heated-room temperature difference is greater than or equal to +1 degree C. If the heated-room temperature difference is smaller than +1 degree C, the process returns to step S23. If the heated-room temperature difference is greater than or equal to +1 degree C, the process proceeds to step S25 where the use-side units 303a and 303b are set in a thermostat OFF state and the hot-water-supply unit 304 is set in a thermostat ON state, thereby commencing the hot-water-supply operation mode C. Specifically, the simultaneous heating and hot-water-supply operation mode D is changed to the hot-water-supply operation mode C. In other words, the first four-way valve 2 and the second four-way valve 13 are set to the hot-water-supply operation mode C in Fig. 4. This state is a high-temperature-water-supply state since the condensing temperature CT is higher than or equal to the predetermined value, and the controller 110 performs the condensing-temperature control on the compressor 1 and the opening-degree control on the hot-water-supply pressure-reducing mechanism 19. In step S26, the controller 110 determines whether the heated-room temperature difference (i.e., indoor suction temperature - preset indoor temperature) is greater than or equal to 0 degrees C. If the heated-room temperature difference is smaller than 0 degrees C, the process returns to step S23 where the controller 110 performs the heating operation mode B. If the heated-room temperature difference is greater than or equal to 0 degrees C, the process proceeds to step S27 where the controller 110 determines whether or not there is a hot-water-supply request (i.e., whether the hot-water supply is completed). If there is a hot-water-supply request, the process returns to step S25 where the controller 110 continues to perform the hot-water-supply operation mode C. If there is no hot-water-supply request, the process proceeds to step S28 where the controller 110 stops the hot-water-supply unit 304 and sets the use-side units 303a and 303b to a heating thermostat ON state so as to commence the normal heating operation.

[0114] By performing the above procedure, a constant heating capacity and a constant hot-water-supply capacity can be ensured when there is a heating load and a hot-water-supply request at the same time and when the inlet water temperature is high in the high-temperature-water supply.

Simultaneous Cooling and Hot-Water-Supply Operation Mode E

[0115] In the simultaneous cooling and hot-water-supply operation mode E (i.e., concurrent heat-absorption and heat-radiation operation), the use-side units 303a and 303b perform the cooling operation, and the hot-water-supply unit 304 performs the hot-water-supply operation. As shown in Fig. 4, in the simultaneous cooling and hot-water-supply operation mode E, the first four-way valve 2 is in the dash-line state, and the second four-way valve 13 is in the solid-line state. This means that the discharge side of the compressor 1 is connected to the plate-type water heat exchanger 16 via the hot-water-supply gas extension pipe 15, and the suction side of the compressor 1 is connected to the gas side of the outdoor heat exchanger 3. The refrigerant flowing out of the plate-type water heat exchanger 16 travels through the hot-water-supply pressure-reducing mechanism 19 and subsequently diverges therefrom so as to flow into the indoor pressure-reducing mechanisms 7a and 7b and into the liquid extension pipe 6.

[0116] While the refrigerant circuit is in this state, the compressor 1, the outdoor air-sending device 4, the indoor air-sending devices 10a and 10b, and the feed pump 17 are activated, so that a low-pressure gas refrigerant is suctioned into and compressed by the compressor 1, thereby becoming a high-temperature high-pressure gas refrigerant. Subsequently, the high-temperature high-pressure gas refrigerant flows into the first four-way valve 2.

[0117] The refrigerant flowing into the first four-way valve 2 flows out of the heat source unit 301 and travels through the hot-water-supply gas extension pipe 15 so as to flow into the hot-water-supply unit 304. The refrigerant flowing into the hot-water-supply unit 304 flows into the plate-type water heat exchanger 16 where the refrigerant condenses by exchanging heat with water supplied by the feed pump 17 so as to become a high-pressure liquid refrigerant, which then flows out of the plate-type water heat exchanger 16. The refrigerant having heated the water at the plate-type water heat exchanger 16 flows out of the hot-water-supply unit 304 and travels through the hot-water-supply liquid pipe 18, so as to flow into the branch unit 302.

[0118] The refrigerant flowing into the branch unit 302 is reduced in pressure by the hot-water-supply pressure-reducing mechanism 19, thereby becoming an intermediate-pressure, two-phase gas-liquid or liquid-phase refrigerant. In this case, the hot-water-supply pressure-reducing mechanism 19 is controlled to a maximum opening degree. Subsequently, the refrigerant is distributed so as to flow into the liquid extension pipe 6 and into the indoor pressure-reducing mechanisms 7a and 7b. As shown in Fig. 1, the refrigeration traveling toward the indoor units diverges at a branch section 28. Furthermore, in Fig. 1, flow paths constituted by the indoor pressure-reducing mechanisms 7a and 7b (i.e., second pressure-reducing mechanisms), the indoor heat exchangers 9a and 9b (i.e., second evaporators), and the second four-way valve 13 constitute heat-absorption branch flow paths.

[0119] The refrigerant flowing into the indoor pressure-reducing mechanisms 7a and 7b is reduced in pressure into a low-pressure two-phase gas-liquid state and travels through the indoor liquid pipes 8a and 8b so as to flow into the use-side units 303a and 303b. The refrigerant flowing into the use-side units 303a and 303b flows into the indoor heat exchangers 9a and 9b where the refrigerant evaporates by exchanging heat with indoor air supplied by the indoor air-sending devices 10a and 10b, thereby becoming a low-pressure gas refrigerant.

[0120] In this case, each of the indoor pressure-reducing mechanisms 7a and 7b is controlled such that a temperature difference (i.e., a cooled-room temperature difference) obtained by subtracting a preset temperature from an indoor suction temperature detected by the indoor suction temperature sensor 208a or 208b in corresponding use-side unit 303a or 303b is eliminated. Therefore, the flow rate of refrigerant flowing through the indoor heat exchangers 9a and 9b corresponds to the cooling load requested in the air-conditioned space where the use-side units 303a and 303b are installed.

[0121] The refrigerant flowing out of the indoor heat exchangers 9a and 9b flows out of the use-side units 303a and 303b and then travels through the indoor gas pipes 11a and 11b, the branch unit 302, and the gas extension pipe 12 so as to flow into the heat source unit 301. The refrigerant flowing into the heat source unit 301 passes through the second four-way valve 13 and then merges with the refrigerant having passed through the outdoor heat exchanger 3.

[0122] On the other hand, the refrigerant flowing into the liquid extension pipe 6 flows into the heat source unit 301 and is reduced in pressure by the outdoor pressure-reducing mechanism 5, thereby becoming a low-pressure two-phase gas-liquid refrigerant. Subsequently, the refrigerant flows into the outdoor heat exchanger 3 where the refrigerant evaporates by exchanging heat with outdoor air supplied by the outdoor air-sending device 4. Then, the refrigerant travels through the first four-way valve 2 and merges with the refrigerant having passed through the indoor heat exchangers 9a and 9b. Subsequently, the refrigerant passes through the accumulator 14 and is suctioned into the compressor 1 again.

[0123] In the case where the simultaneous cooling and hot-water-supply operation mode E is in the hot-water-supply priority mode, the water temperature in the hot-water-supply tank 305 is increased to a preset hot-water-supply temperature as quickly as possible in response to a hot-water-supply request of the hot-water-supply unit 304. Thus, in order to ensure a high hot-water-supply capacity, the control unit 103 controls the compressor 1 so as to increase the operating frequency thereof. Therefore, heat absorption is necessary in the outdoor heat exchanger 3 for achieving an equal cooling capacity for the cooling loads of the use-side units 303a and 303b. The opening degree of the outdoor pressure-reducing mechanism 5 is controlled by the control unit 103 such that the degree of superheat at the gas side of the outdoor heat exchanger 3 is made equal to a predetermined value. The degree of superheat at the gas side of the outdoor heat exchanger 3 is determined by subtracting the temperature detected by the outdoor liquid temperature sensor 204 from the temperature detected by the outdoor gas temperature sensor 203. The quantity of air from the outdoor air-sending device 4 is controlled such that the evaporating temperature is made equal to a predetermined value. The evaporating temperature is the temperature detected by the indoor liquid temperature sensor 206a or 206b. The predetermined evaporating-temperature value is determined from a temperature difference (i.e., a cooled-room temperature difference), which is obtained by subtracting a preset temperature from an indoor suction temperature detected by the indoor suction temperature sensor 208a or 208b, in the use-side unit 303a or 303b that has the maximum heated-room temperature difference in the use-side units 303a and 303b.

[0124] In the case where the simultaneous cooling and hot-water-supply operation mode E is the cooling priority mode, the operating frequency of the compressor 1 is controlled by the control unit 103 such that the evaporating temperature is made equal to a predetermined value in accordance with the cooling loads of the use-side units 303a and 303b. The predetermined evaporating-temperature value is determined from a temperature difference (i.e., a cooled-room temperature difference), which is obtained by subtracting a preset temperature from an indoor suction temperature detected by the indoor suction temperature sensor 208a or 208b, in the use-side unit 303a or 303b that has the maximum heated-room temperature difference in the use-side units 303a and 303b. Because the operating frequency of the compressor 1 is set in accordance with the cooling loads of the use-side units 303a and 303b, there is no need to perform heat absorption in the outdoor heat exchanger 3. Therefore, the outdoor pressure-reducing mechanism 5 is controlled to a small opening degree by the control unit 103, and the outdoor air-sending device 4 is stopped by the control unit 103.

[0125] In the simultaneous cooling and hot-water-supply operation mode E, the operation is normally performed in the cooling priority mode and is performed in accordance with the cooling load so as to achieve a favorable level of comfort inside. However, if the cooling load is small, the operating frequency of the compressor 1 becomes low. If this causes the hot-water-supply capacity to be low for a long time, the time that it takes to complete the hot-water-supply operation increases, causing a shortage of hot water. In order to prevent such a shortage of hot water, if a hot-water-supply request is detected continuously for a certain period of time (e.g., two consecutive hours), the simultaneous cooling and hot-water-supply operation mode E is performed in the hot-water-supply priority mode so as to prevent the shortage of hot water.

[0126] If the hot-water-supply temperature is a high temperature (e.g., 60 degrees C), the inlet water temperature of the plate-type water heat exchanger 16 also becomes high, causing the condensing temperature CT to increase. Unlike the case of the hot-water-supply operation, the cooling operation is performed in the use-side units 303a and 303b in the simultaneous cooling and the hot-water-supply operation. Therefore, when the condensing-temperature control shown in expressions (1) and (2) is performed on the compressor 1 and the opening-degree control shown in expressions (3) and (4) is performed on the hot-water-supply pressure-reducing mechanism 19, the operating frequency of the compressor 1 becomes low in the condensing-temperature control. Thus, a cooling capacity cannot be ensured in the use-side units 303a and 303b, sometimes resulting in a "non-cooled" state. Therefore, if the condensing temperature increases to a value higher than a predetermined value during the simultaneous cooling and hot-water-supply operation, the simultaneous operation is stopped, and the cooling operation and the hot-water-supply operation are performed by alternately switching between the cooling operation and the hot-water-supply operation, as in the simultaneous heating and hot-water-supply operation.

[0127] Fig. 14 is a flowchart illustrating the flow of an operation method during the high-temperature-water supply in the simultaneous cooling and hot-water-supply operation mode. Specifically, the operation is performed in accordance with the flowchart shown in Fig. 14. First, in step S31, it is determined whether or not the condensing temperature has increased to a value higher than a predetermined value. Similar to the case of the hot-water-supply temperature, the predetermined value for the condensing temperature is, for example, a maximum condensing-temperature value (e.g., 60 degrees C) allowable in the appropriate usage range of the compressor 1. If the condensing temperature is lower than or equal to the predetermined value, normal control is continuously performed in the simultaneous cooling and hot-water-supply operation in step S32. If the condensing temperature is higher than or equal to the predetermined value, the process proceeds to the cooling operation mode A in step S33. In this case, the use-side units 303a and 303b are set in a cooling thermostat OFF state, and the following control is performed for the purpose of changing to the hot-water-supply operation mode C. In the cooling operation, the indoor pressure-reducing mechanisms 7a and 7b are normally controlled so that a "cooled-room temperature difference", which is equal to "indoor suction temperature (detected by indoor suction temperature sensor) - preset indoor temperature", is eliminated. The indoor pressure-reducing mechanisms 7a and 7b are controlled so that the cooled-room temperature difference is a negative value, such as -1 degree C (i.e., a predetermined negative value). Moreover, the operating frequency of the compressor 1 is controlled so that the evaporating temperature is made equal to a target value. Normally, the target value for the evaporating temperature is determined from a cooled-room temperature difference in the use-side unit 303a or 303b that has the maximum cooled-room temperature difference in the use-side units 303a and 303b. However, the target evaporating-temperature value for the operating frequency of the compressor 1 is determined from a cooled-room temperature difference of +1 degree C in the use-side unit 303a or 303b that has the maximum heated-room temperature difference of +1 degree C. By performing the control in this manner, the cooled-room temperature difference can be made equal to -1 degree C.

[0128] Subsequently, in step S34, it is determined whether the cooled-room temperature difference is smaller than or equal to -1 degree C. If the cooled-room temperature difference is not smaller than or equal to -1 degree C, the process returns to step S33. If the cooled-room temperature difference is smaller than or equal to -1 degree C, the process proceeds to step S35 where the use-side units 303a and 303b are set in a cooling thermostat OFF state and the mode is changed to the hot-water-supply operation mode C. In the hot-water-supply operation mode C, this state is a high-temperature-water-supply state since the condensing temperature is higher than or equal to the predetermined value, and the condensing-temperature control is performed on the compressor 1, and the opening-degree control is performed on the hot-water-supply pressure-reducing mechanism 19. In step S36, it is determined whether the cooled-room temperature difference is smaller than or equal to 0 degrees C. If the cooled-room temperature difference is greater than or equal to 0 degrees C, the process returns to step S33 where the mode is set to the cooling operation mode A. If the cooled-room temperature difference is smaller than or equal to 0 degrees C, the process proceeds to step S37 where it is determined whether or not there is a hot-water-supply request (i.e., whether the hot-water supply is completed). If there is a hot-water-supply request, the process returns to step S35 where the hot-water-supply operation mode C is continuously performed. If there is no hot-water-supply request, the process proceeds to step S38 where the hot-water-supply unit 304 is stopped and the use-side units 303a and 303b are set to a cooling thermostat ON state, thereby commencing the normal cooling operation.

[0129] By performing the above procedure, a constant hot-water-supply capacity can be ensured and the cooling operation can be performed when there is a cooling load and a hot-water-supply request at the same time and when the inlet water temperature is high in the high-temperature-water supply.

[0130] With the integrated air-conditioning and hot-water-supply system 100 according to Embodiment 1, an excessive increase in condensing temperature during the high-temperature-water supply can be suppressed, and a hot-water-supply capacity can be ensured within a usage range of the compressor.

[0131] Although the integrated air-conditioning and hot-water-supply system 100 (refrigeration cycle apparatus) is described in Embodiment 1, the operation of the integrated air-conditioning and hot-water-supply system 100 can be construed as a refrigeration cycle control method.

Reference Signs List

[0132] 1: compressor, 2: first four-way valve, 3: outdoor heat exchanger, 4: outdoor air-sending device, 5: outdoor pressure-reducing mechanism, 6: liquid extension pipe, 7a, 7b: indoor pressure-reducing mechanism, 8a, 8b : indoor liquid pipe, 9a, 9b: indoor heat exchanger, 10a, 10b: indoor air-sending device, 11a, 11b: indoor gas pipe, 12: gas extension pipe, 13: second four-way valve, 14: accumulator, 15: hot-water-supply gas extension pipe, 16: plate-type water heat exchanger, 17: feed pump, 18: hot-water-supply liquid pipe, 19: hot-water-supply pressure-reducing mechanism, 20: upstream water pipe, 21: downstream water pipe, 100: integrated air-conditioning and hot-water-supply system, 110: controller, 101: measuring unit, 102: calculating unit, 103: control unit, 104: storage unit, 201: high-pressure sensor, 202: discharge temperature sensor, 203: outdoor gas temperature sensor, 204: outdoor liquid temperature sensor, 205: outdoor-air temperature sensor, 206a, 206b: indoor liquid temperature sensor, 207a, 207b: indoor gas temperature sensor, 208a, 208b: indoor suction temperature sensor, 209: hot-water-supply liquid temperature sensor, 210: hot-water-supply-tank water temperature sensor, 301: heat source unit, 302: branch unit, 303a, 303b: use-side unit, 304: hot-water-supply unit, 305: hot-water-supply tank

Claims

1. A refrigeration cycle apparatus comprising:

a refrigeration cycle mechanism having a compressor whose operating frequency is controllable, a first radiator, a first pressure-reducing mechanism whose opening degree is controllable, and a first evaporator, wherein a refrigerant sequentially circulates through the compressor, the first radiator, the first pressure-reducing mechanism, and the first evaporator;
a high-pressure sensor that detects a high pressure between a discharge side of the compressor and a liquid side of the first pressure-reducing mechanism; and
a controller that calculates a condensing temperature of the first radiator based on the high pressure detected by the high-pressure sensor, wherein
when the calculated condensing temperature of the first radiator is higher than or equal to a preset target condensing-temperature value, the controller performs condensing-temperature control for controlling the operating frequency of the compressor based on a difference between the calculated condensing temperature and the target condensing-temperature value, and performs opening-degree control for controlling the opening degree of the first pressure-reducing mechanism concurrently with the condensing-temperature control based on a difference between a current opening degree of the first pressure-reducing mechanism and a preset target opening-degree value.

2. The refrigeration cycle apparatus of claim 1, wherein the target opening-degree value of the first pressure-reducing mechanism is set in correspondence with a target heat-radiation capacity value of the first radiator.

3. The refrigeration cycle apparatus of claim 2,
wherein the first radiator includes
an inflowing-water-pipe connection section connected to an inflowing water pipe into which water flows,
an outflowing-water-pipe connection section connected to an outflowing water pipe from which the water flows, and
a water pipe through which the water flowing in from the inflowing water pipe passes and flows out to the outflowing water pipe,
wherein the first radiator heats the water passing through the water pipe by radiating heat to the water, and
wherein the target heat-radiation-capacity value of the first radiator is set in correspondence with an upper limit value in design for an inlet water temperature of the water flowing into the water pipe from the inflowing water pipe.

4. The refrigeration cycle apparatus of claim 1, wherein the controller includes a storage unit that stores frequency/opening-degree correspondence information in which the operating frequency of the compressor and the target opening-degree value of the first pressure-reducing mechanism are stored in correspondence with each other, and wherein when the controller concurrently performs the condensing-temperature control and the opening-degree control, the controller refers to the frequency/opening-degree correspondence information so as to identify the target opening-degree value corresponding to a current operating frequency of the compressor from the frequency/opening-degree correspondence information, and uses the identified target opening-degree value as the target opening-degree value in the opening-degree control.

5. The refrigeration cycle apparatus of claim 1,
wherein the first evaporator is disposed outdoors,
wherein the refrigeration cycle apparatus further comprises an outdoor-air temperature sensor that detects an outdoor-air temperature around the first evaporator, and
wherein the controller includes a storage unit that stores outdoor-air-temperature/opening-degree correspondence information in which the outdoor-air temperature and the target opening-degree value are stored in correspondence with each other, and wherein when the controller concurrently performs the condensing-temperature control and the opening-degree control, the controller refers to the outdoor-air-temperature/opening-degree correspondence information so as to identify the target opening-degree value corresponding to the outdoor-air temperature detected by the outdoor-air temperature sensor from the outdoor-air-temperature/opening-degree correspondence information, and uses the identified target opening-degree value as the target opening-degree value in the opening-degree control.

6. The refrigeration cycle apparatus of claim 1,
wherein the first evaporator is disposed outdoors,
wherein the refrigeration cycle apparatus further comprises:

an outdoor-air temperature sensor that detects an outdoor-air temperature around the first evaporator; and
an evaporating temperature sensor that detects an evaporating temperature of the refrigerant in the first evaporator, and

wherein the controller receives data of two or more sets of a temperature difference between the outdoor-air temperature around the first evaporator and the evaporating temperature of the first evaporator and an evaporating capacity of the first evaporator corresponding to the temperature difference, determines a functional relationship between the temperature difference and the evaporating capacity based on the received data, refers to the determined functional relationship so as to identify the evaporating capacity corresponding to the temperature difference between the outdoor-air temperature detected by the outdoor-air temperature sensor and the evaporating temperature detected by the evaporating temperature sensor from the functional relationship, calculates a compressor input, which indicates compressing work done on the refrigerant by the compressor, from the operating frequency of the compressor, the calculated condensing temperature, and the evaporating temperature detected by the evaporating temperature sensor, calculates a heat-radiation capacity of the first radiator from the identified evaporating capacity and the calculated compressor input, determines the target opening-degree value in accordance with a difference between the calculated heat-radiation capacity and a preliminarily-stored target heat-radiation-capacity value, and uses the determined target opening-degree value as the target opening-degree value in the opening-degree control.

7. The refrigeration cycle apparatus of claim 1, further comprising a branch flow path branching from the discharge side of the compressor and having a second radiator and a second pressure-reducing mechanism, the branch flow path being connected to the second radiator and the second pressure-reducing mechanism sequentially from the discharge side of the compressor and merging with an intermediate section between the first pressure-reducing mechanism and the first evaporator,
wherein the controller performs a concurrent heat-radiation operation in which the refrigerant discharged from the compressor is circulated by being made to flow into the first radiator and the second radiator, and wherein when the calculated condensing temperature becomes higher than or equal to the target condensing-temperature value during the concurrent heat-radiation operation, the controller performs a switching process for alternately switching between a process for making the discharged refrigerant flow into the first radiator and a process for making the discharged refrigerant flow into the second radiator.

8. The refrigeration cycle apparatus of claim 7,
wherein the second radiator exchanges heat with indoor air,
wherein the refrigeration cycle apparatus further comprises an indoor temperature sensor that detects an indoor temperature, and
wherein the controller performs the switching process based on a temperature difference obtained by subtracting a preliminarily-stored preset indoor temperate from the indoor temperature detected by the indoor temperature sensor.

9. The refrigeration cycle apparatus of claim 8, wherein when the discharged refrigerant is made to flow only into the second radiator due to the switching process, the controller controls the operating frequency of the compressor and the opening degree of the first pressure-reducing mechanism so that the temperature difference is greater than a predetermined positive value, and wherein when the temperature difference becomes greater than the predetermined positive value, the controller performs the switching process so as to make the discharged refrigerant flow only into the first radiator.

10. The refrigeration cycle apparatus of claim 1, further comprising a heat-absorption branch flow path that branches from a branch section between the first pressure-reducing mechanism and the first evaporator and merges with a suction side of the compressor, the heat-absorption branch flow path having a second evaporator and a pressure-reducing mechanism for the second evaporator, the heat-absorption branch flow path being connected to the pressure-reducing mechanism for the second evaporator and to the second evaporator sequentially from the branch section toward the discharge side and merging with the suction side of the compressor,
wherein the controller performs a concurrent heat-absorption and heat-radiation operation in which a heat-radiation operation of the first radiator and a heat-absorption operation of the second evaporator are concurrently performed, the heat-radiation operation being operation in which the refrigerant discharged from the compressor is suctioned into the compressor from the suction side thereof via the first radiator, the first pressure-reducing mechanism, the branch section, and the first evaporator, the heat-absorption operation being operation in which the discharged refrigerant is suctioned into the compressor from the suction side thereof via the first radiator, the first pressure-reducing mechanism, the branch section, the pressure-reducing mechanism for the second evaporator, and the second evaporator, and wherein when the calculated condensing temperature becomes higher than or equal to the target condensing-temperature value during the concurrent heat-reception and heat-radiation operation, the controller performs a switching process for alternately switching between the heat-radiation operation and the heat-

absorption operation.

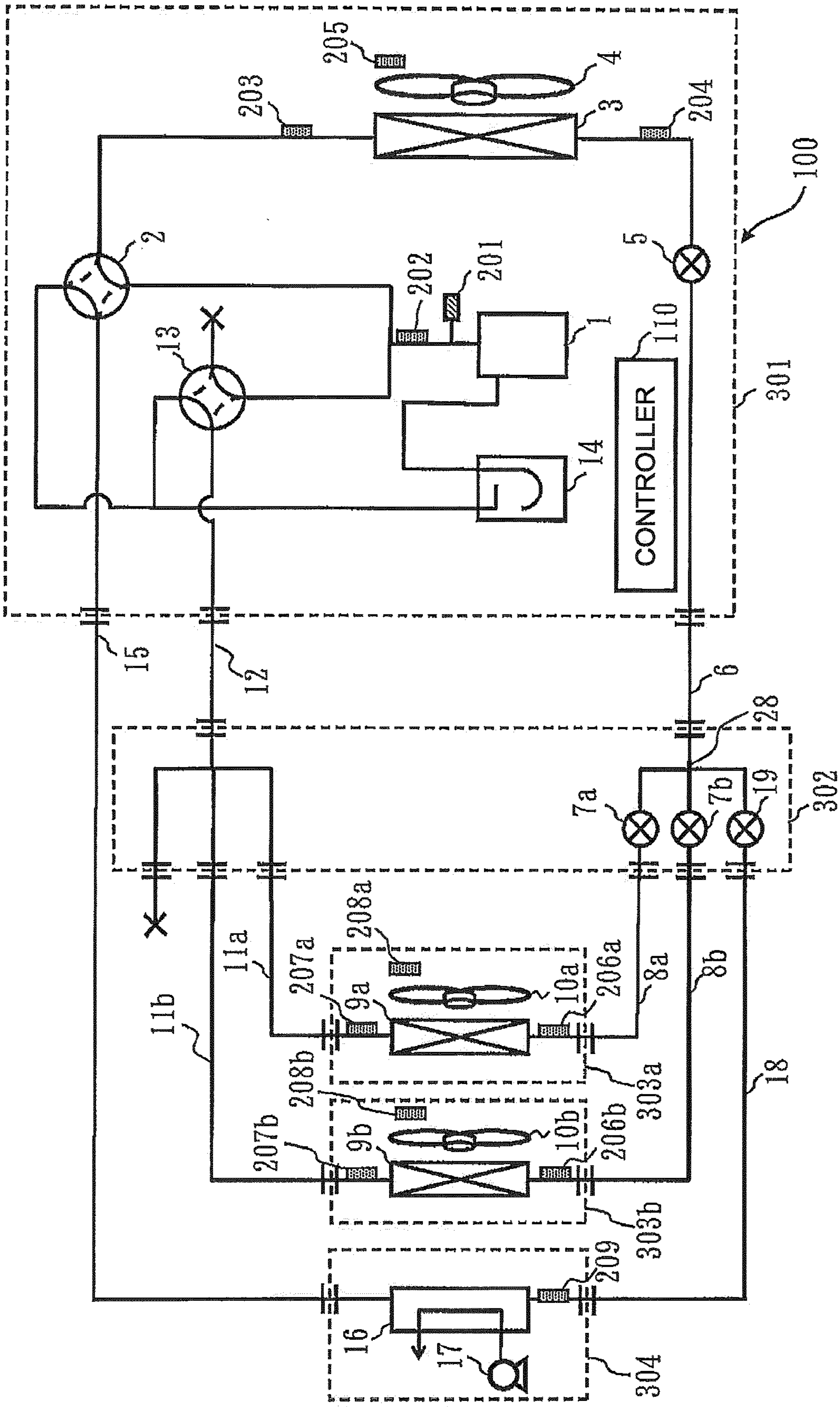
11. The refrigeration cycle apparatus of claim 10,
wherein the second evaporator exchanges heat with indoor air,
wherein the refrigeration cycle apparatus further comprises an indoor temperature sensor that detects an indoor temperature, and
wherein the controller performs the switching process based on a temperature difference obtained by subtracting a preliminarily-stored preset indoor temperature from the indoor temperature detected by the indoor temperature sensor.

12. The refrigeration cycle apparatus of claim 11, wherein when only the heat-absorption operation is performed due to the switching process, the controller controls the operating frequency of the compressor and the opening degree of the first pressure-reducing mechanism so that the temperature difference is smaller than a predetermined negative value, and wherein when the temperature difference becomes smaller than the predetermined negative value, the controller performs the switching process so that only the heat-radiation operation is performed.

13. The refrigeration cycle apparatus of claim 1,
wherein the refrigeration cycle apparatus uses a refrigerant that operates at a critical pressure or higher, and
wherein when the high pressure detected by the high-pressure sensor is higher than or equal to a preset target high-pressure value, the controller performs high-pressure control for controlling the operating frequency of the compressor based on a difference between the high pressure and the target high-pressure value, and performs opening-degree control for controlling the opening degree of the first pressure-reducing mechanism concurrently with the high-pressure control based on the difference between the current opening degree of the first pressure-reducing mechanism and the preset target opening-degree value.

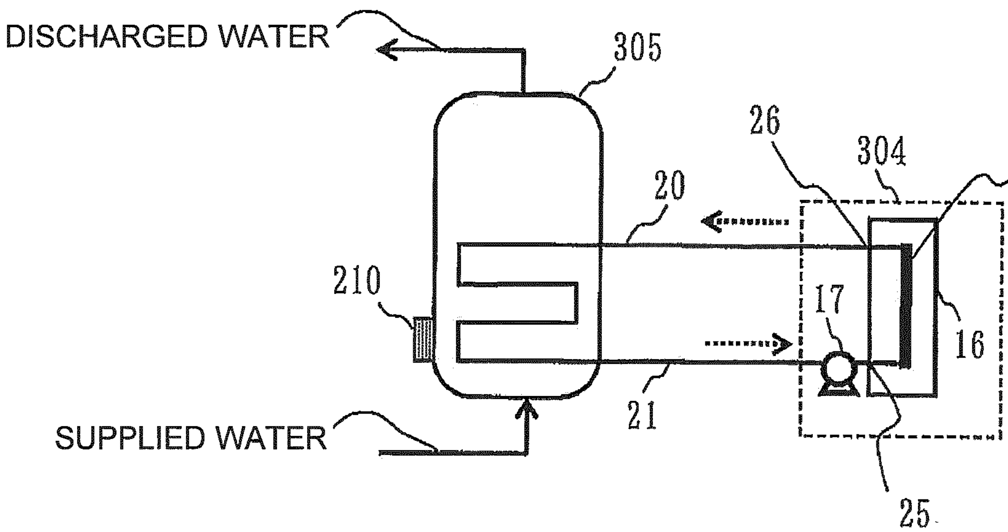
14. A refrigeration cycle control method for performing an operation on a refrigeration cycle apparatus, the refrigeration cycle apparatus including a refrigeration cycle mechanism having a compressor whose operating frequency is controllable, a first radiator, a first pressure-reducing mechanism whose opening degree is controllable, and a first evaporator, wherein a refrigerant sequentially circulates through the compressor, the first radiator, the first pressure-reducing mechanism, and the first evaporator; and a high-pressure sensor that detects a high pressure between a discharge side of the compressor and a liquid side of the first pressure-reducing mechanism, the method comprising:

calculating a condensing temperature of the first radiator based on the high pressure detected by the high-pressure sensor; and
performing condensing-temperature control for controlling the operating frequency of the compressor based on a difference between the calculated condensing temperature and a preset target condensing-temperature value and performing opening-degree control for controlling the opening degree of the first pressure-reducing mechanism concurrently with the condensing-temperature control based on a difference between a current opening degree of the first pressure-reducing mechanism and a preset target opening-degree value when the calculated condensing temperature of the first radiator is higher than or equal to the target condensing-temperature value.


100
301
302
303a
303b
304
1
2
3
4
5
6
7a
7b
8a
8b
9a
9b
10a
10b
11a
11b
12
13
14
15
16
17
18
19
28
110
CONTROLLER
201
202
203
204
205
206a
206b
207a
207b
208a
208b
209

F I G. 1

F I G. 2


DISCHARGED WATER
305
26
27
304
20
210
17
16
21
SUPPLIED WATER
25

F I G. 3

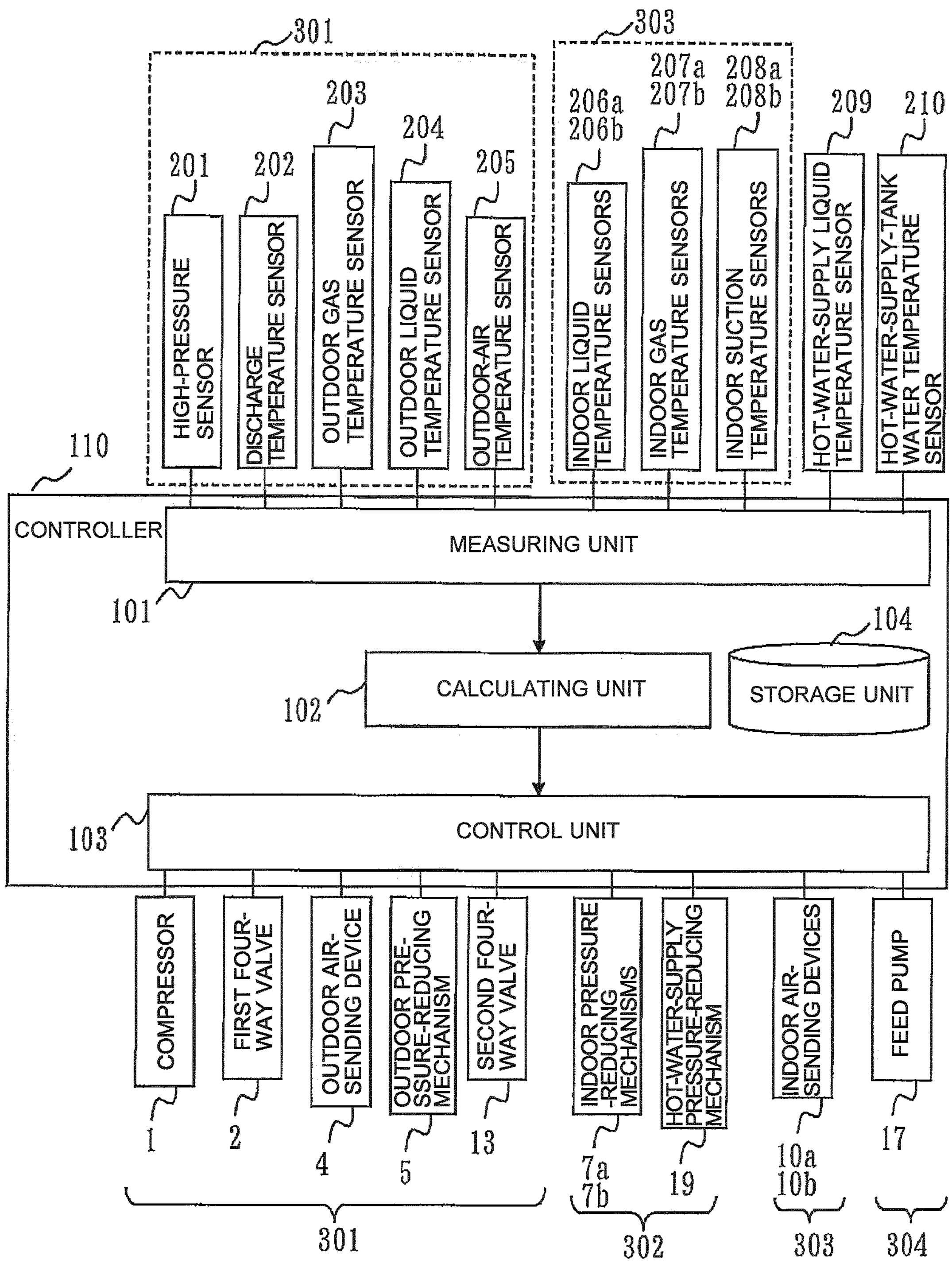

301
303
201
HIGH-PRESSURE SENSOR
202
DISCHARGE TEMPERATURE SENSOR
203
OUTDOOR GAS TEMPERATURE SENSOR
204
OUTDOOR LIQUID TEMPERATURE SENSOR
205
OUTDOOR-AIR TEMPERATURE SENSOR
206a
206b
INDOOR LIQUID TEMPERATURE SENSORS
207a
207b
INDOOR GAS TEMPERATURE SENSORS
208a
208b
INDOOR SUCTION TEMPERATURE SENSORS
209
HOT-WATER-SUPPLY LIQUID TEMPERATURE SENSOR
210
HOT-WATER-SUPPLY-TANK WATER TEMPERATURE SENSOR
110
CONTROLLER
MEASURING UNIT
101
CALCULATING UNIT
102
STORAGE UNIT
104
CONTROL UNIT
103
COMPRESSOR
1
FIRST FOUR-WAY VALVE
2
OUTDOOR AIR-SENDING DEVICE
4
OUTDOOR PRE-SSURE-REDUCING MECHANISM
5
SECOND FOUR-WAY VALVE
13
INDOOR PRESSURE -REDUCING MECHANISMS
7a
7b
HOT-WATER-SUPPLY PRESSURE-REDUCING MECHANISM
19
INDOOR AIR-SENDING DEVICES
10a
10b
FEED PUMP
17
301
302
303
304

FIG. 4

	COOLING OPERATION	HEATING OPERATION	HOT-WATER -SUPPLY OPERATION	SIMULTANEOUS HEATING AND HOT-WATER-SUPPLY OPERATION	SIMULTANEOUS COOLING AND HOT-WATER-SUPPLY OPERATION
FIRST FOUR-WAY VALVE 2	SOLID LINE	DASH LINE	DASH LINE	DASH LINE	DASH LINE
SECOND FOUR-WAY VALVE 13	SOLID LINE	DASH LINE	SOLID LINE	DASH LINE	SOLID LINE

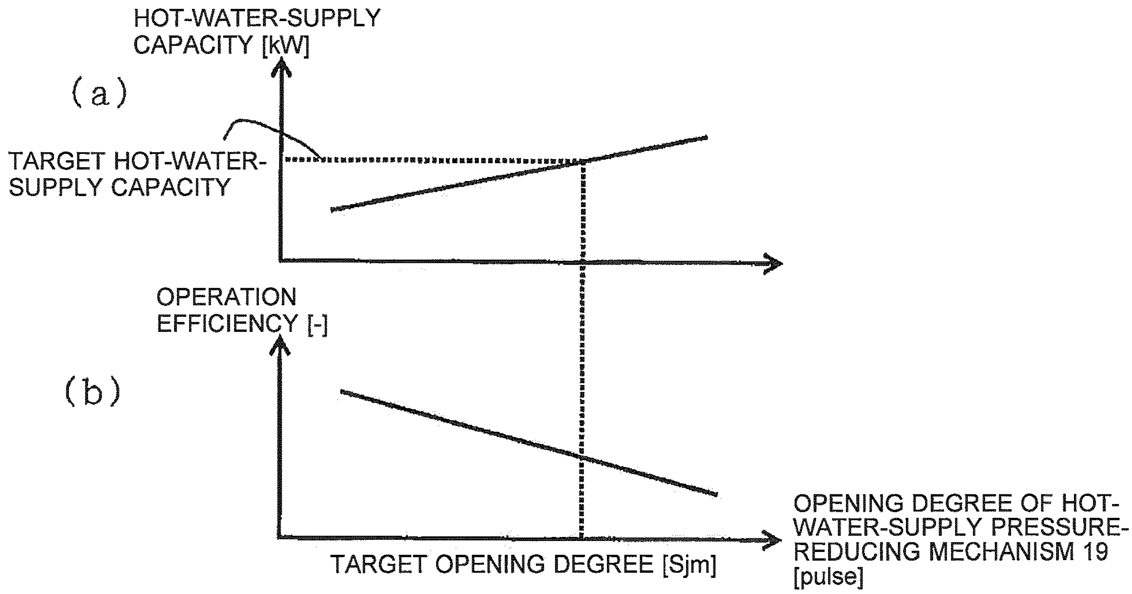
FIG. 5

MAXIMUM COOLED-ROOM TEMPERATURE DIFFERENCE [maxΔTje]	TARGET EVAPORATING-TEMPERATURE VALUE [°C]
$2 \leqq \max\Delta Tje$	A1
$1 \leqq \max\Delta Tje < 2$	A2
$-1 \leqq \max\Delta Tje < 1$	A3
$\max\Delta Tje < -1$	A4

FIG. 6

MAXIMUM HEATED-ROOM TEMPERATURE DIFFERENCE [maxΔTjc]	TARGET CONDENSING-TEMPERATURE VALUE [°C]
$1 \leqq \max\Delta Tjc$	B1
$-1 \leqq \max\Delta Tjc < 1$	B2
$-2 \leqq \max\Delta Tjc < -1$	B3
$\max\Delta Tjc < -2$	B4

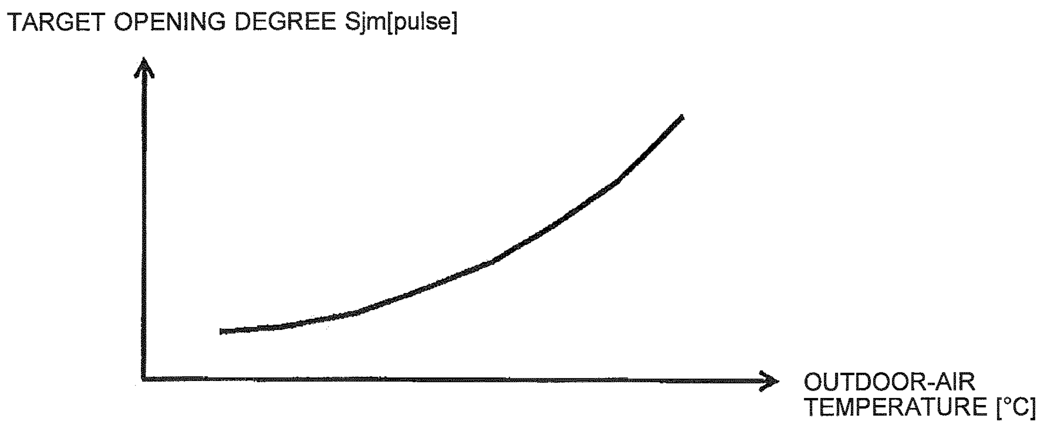
FIG. 7



F I G. 8

TEST NO.		1	2	3
TEST CONDITION	INLET WATER TEMPERATURE OF LATE-TYPE WATER HEAT EXCHANGER 16			
TEST RESULT	OPERATING FREQUENCY OF COMPRESSOR 1			
	OPENING DEGREE OF HOT-WATER-SUPPLY PRESSURE-REDUCING MECHANISM 19			

F I G. 9



F I G. 1 0

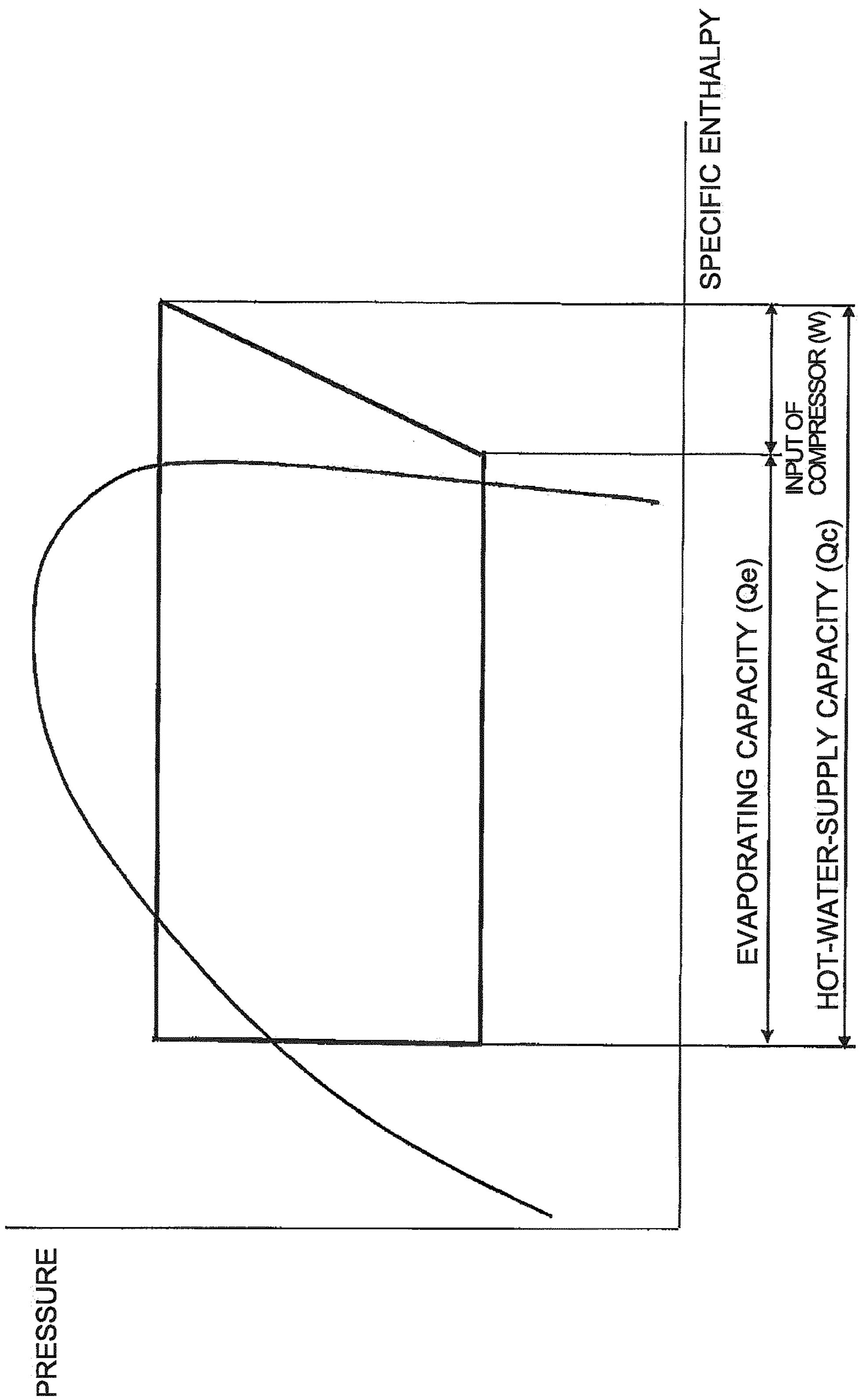

PRESSURE
SPECIFIC ENTHALPY
INPUT OF COMPRESSOR (W)
EVAPORATING CAPACITY (Qe)
HOT-WATER-SUPPLY CAPACITY (Qc)

FIG. 11

TEST NO.		1	2	3
TEST CONDITION	OPERATING FREQUENCY OF COMPRESSOR 1			
	OPENING DEGREE OF HOT-WATER-SUPPLY PRESSURE-REDUCING MECHANISM 19			
TEST RESULT	DIFFERENCE BETWEEN OUTDOOR-AIR TEMPERATURE AND EVAPORATING TEMPERATURE			
	EVAPORATING CAPACITY			

FIG. 12

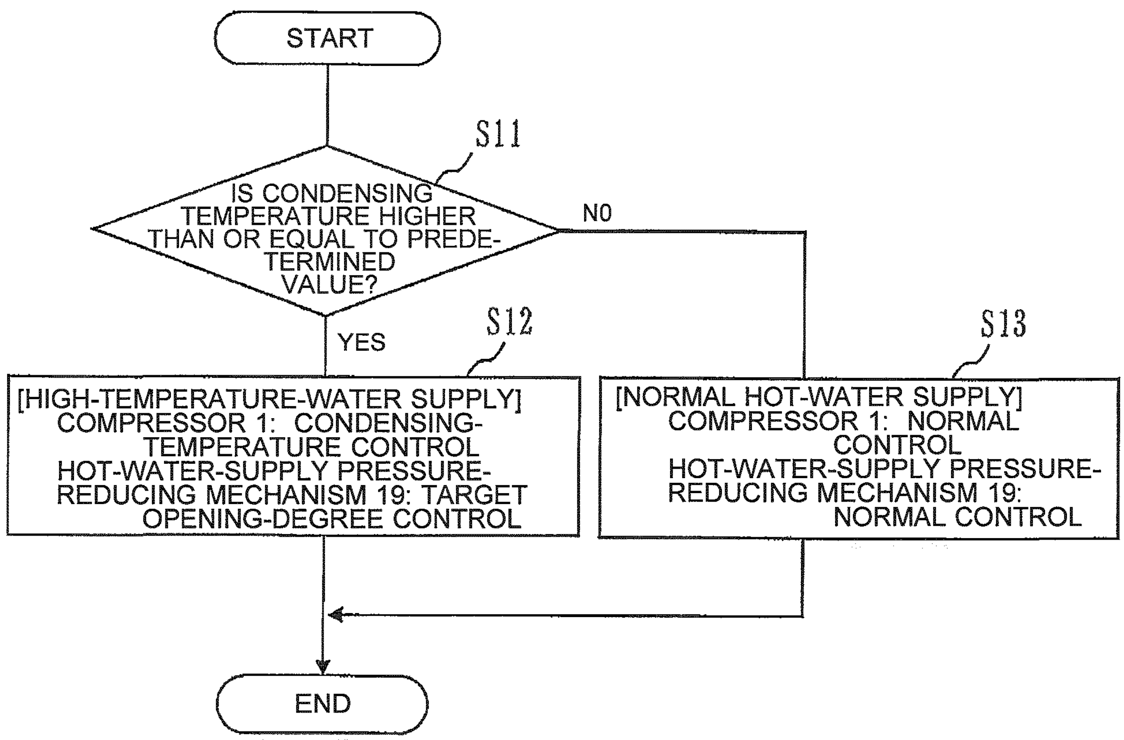
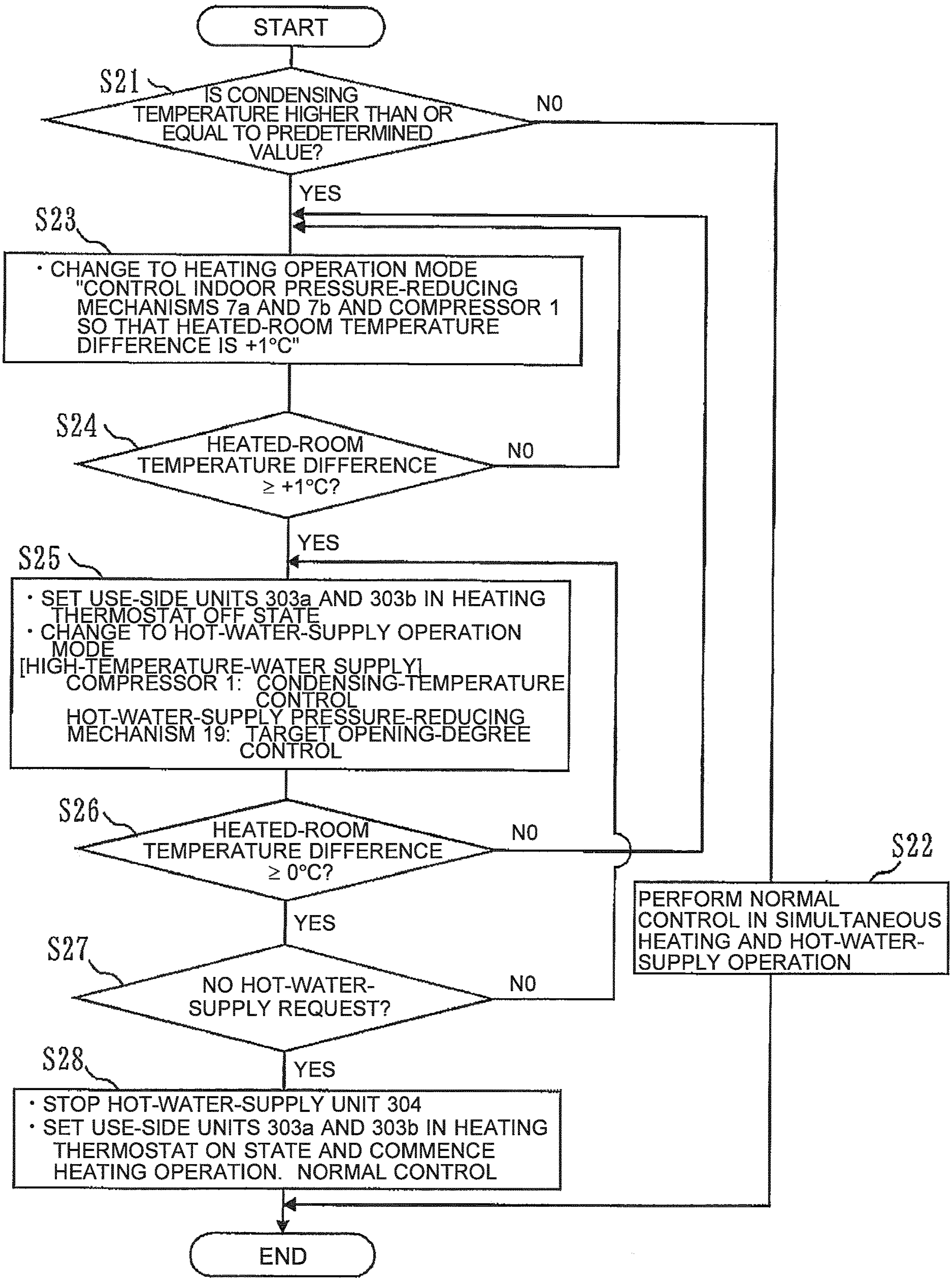
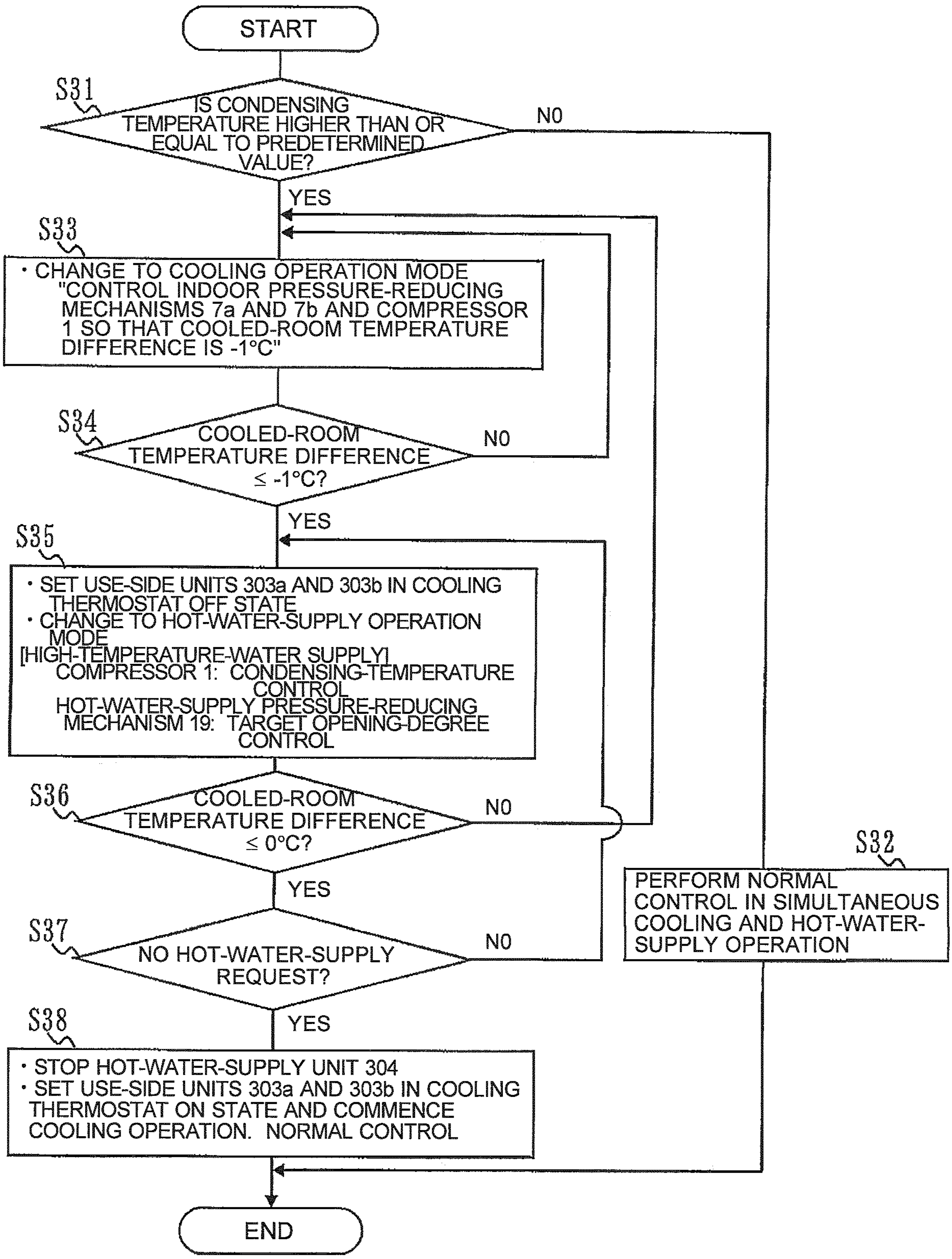


FIG. 13


START
S21
IS CONDENSING TEMPERATURE HIGHER THAN OR EQUAL TO PREDETERMINED VALUE?
NO
YES
S23
・CHANGE TO HEATING OPERATION MODE "CONTROL INDOOR PRESSURE-REDUCING MECHANISMS 7a AND 7b AND COMPRESSOR 1 SO THAT HEATED-ROOM TEMPERATURE DIFFERENCE IS +1°C"
S24
HEATED-ROOM TEMPERATURE DIFFERENCE ≥ +1°C?
NO
YES
S25
・SET USE-SIDE UNITS 303a AND 303b IN HEATING THERMOSTAT OFF STATE
・CHANGE TO HOT-WATER-SUPPLY OPERATION MODE
[HIGH-TEMPERATURE-WATER SUPPLY]
COMPRESSOR 1: CONDENSING-TEMPERATURE CONTROL
HOT-WATER-SUPPLY PRESSURE-REDUCING MECHANISM 19: TARGET OPENING-DEGREE CONTROL
S26
HEATED-ROOM TEMPERATURE DIFFERENCE ≥ 0°C?
NO
YES
S22
PERFORM NORMAL CONTROL IN SIMULTANEOUS HEATING AND HOT-WATER-SUPPLY OPERATION
S27
NO HOT-WATER-SUPPLY REQUEST?
NO
YES
S28
・STOP HOT-WATER-SUPPLY UNIT 304
・SET USE-SIDE UNITS 303a AND 303b IN HEATING THERMOSTAT ON STATE AND COMMENCE HEATING OPERATION. NORMAL CONTROL
END

F I G. 1 4


START
S31
IS CONDENSING TEMPERATURE HIGHER THAN OR EQUAL TO PREDETERMINED VALUE?
NO
YES
S33
・CHANGE TO COOLING OPERATION MODE "CONTROL INDOOR PRESSURE-REDUCING MECHANISMS 7a AND 7b AND COMPRESSOR 1 SO THAT COOLED-ROOM TEMPERATURE DIFFERENCE IS -1°C"
S34
COOLED-ROOM TEMPERATURE DIFFERENCE ≤ -1°C?
NO
YES
S35
・SET USE-SIDE UNITS 303a AND 303b IN COOLING THERMOSTAT OFF STATE
・CHANGE TO HOT-WATER-SUPPLY OPERATION MODE
[HIGH-TEMPERATURE-WATER SUPPLY]
COMPRESSOR 1: CONDENSING-TEMPERATURE CONTROL
HOT-WATER-SUPPLY PRESSURE-REDUCING MECHANISM 19: TARGET OPENING-DEGREE CONTROL
S36
COOLED-ROOM TEMPERATURE DIFFERENCE ≤ 0°C?
NO
YES
S37
NO HOT-WATER-SUPPLY REQUEST?
NO
YES
S38
・STOP HOT-WATER-SUPPLY UNIT 304
・SET USE-SIDE UNITS 303a AND 303b IN COOLING THERMOSTAT ON STATE AND COMMENCE COOLING OPERATION. NORMAL CONTROL
S32
PERFORM NORMAL CONTROL IN SIMULTANEOUS COOLING AND HOT-WATER-SUPPLY OPERATION
END

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2011/055374

A. CLASSIFICATION OF SUBJECT MATTER
F25B1/00(2006.01)i, *F25B6/02*(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, F25B6/02

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	1996-2011
Kokai Jitsuyo Shinan Koho	1971-2011	Toroku Jitsuyo Shinan Koho	1994-2011

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.
X Y	JP 2009-186121 A (Mitsubishi Electric Corp.), 20 August 2009 (20.08.2009), paragraphs [0026] to [0029], [0040]; fig. 1 & US 2009/0199581 A1 & EP 2088390 A2	1-3,13,14 4,5
Y	JP 2010-196985 A (Mitsubishi Heavy Industries, Ltd.), 09 September 2010 (09.09.2010), claim 1; paragraphs [0024] to [0030]; fig. 1 to 4 & EP 2224180 A1	4
Y	JP 2003-90631 A (Sanyo Electric Co., Ltd.), 28 March 2003 (28.03.2003), claim 1; fig. 1 & KR 10-2003-0023531 A & CN 1405519 A	5

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

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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
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"&" document member of the same patent family

Date of the actual completion of the international search 10 May, 2011 (10.05.11)	Date of mailing of the international search report 24 May, 2011 (24.05.11)
Name and mailing address of the ISA/ Japanese Patent Office Facsimile No.	Authorized officer Telephone No.

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2011/055374

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 2009-236392 A (Sanyo Electric Co., Ltd.), 15 October 2009 (15.10.2009), entire text; all drawings & US 2009/0241577 A & EP 2105686 A2 & KR 10-2009-0102709 A	7-12

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2004053118 A **[0004]**
- JP 2005098530 A **[0004]**