

Description

BACKGROUND OF THE INVENTION

Field of the Invention

[0001] The present invention relates to an indoor unit of an air conditioner, and specifically relates to an air-conditioning/hot water-supply system capable of simultaneously supplying hot/cool heat necessary for air cooling, air heating and hot water supply, in which a refrigeration cycle for generating hot water for hot water supply is installed and heat exchange is performed between an air-conditioning refrigerant and a hot water-supply refrigerant via a cascade heat exchanger.

Description of the Related Art

[0002] Conventionally, there are air-conditioning/hot water-supply systems capable of simultaneously supplying hot/cool heat necessary for air cooling, air heating and hot water supply.

[0003] In an air-conditioning/hot water-supply system such as above, a refrigeration cycle in which an air-conditioning refrigerant circulates and a refrigeration cycle in which a hot water-supply refrigerant circulates form what is called a dual refrigeration cycle in which these refrigeration cycles are thermally connected in a cascade heat exchanger.

[0004] As a conventional air-conditioning/hot water-supply system, for example, a technique that is a hot water supply device including a hot water-supply refrigerant circuit in which a compressor, a first heat exchanger, an expansion mechanism and a second heat exchanger are connected and a carbon dioxide refrigerant is charged, the first heat exchanger being a heat exchanger for hot water production, the second heat exchanger being a cascade heat exchanger, thereby the hot water supply device being unitized (see, for example, Japanese Patent Laid-Open No. 2004-132647 (Japanese Patent No. 3925383)).

[0005] In a heat generation unit, in order to produce a heating medium having a high temperature of 90°C, it is necessary to bring a temperature Td of a discharged refrigerant from a hot water-supply compressor to no less than 100°C. However, in order to prevent impairing a durability of the compressor because of acceleration of deterioration of, e.g., a refrigerant oil enclosed in the hot water-supply compressor 310 and an insulating coating of a winding of a motor, an upper limit use temperature for the temperature of the discharged refrigerant from the hot water-supply compressor is generally set to 110°C.

[0006] Furthermore, if carbon dioxide that has flown into the hot water-supply compressor turns into a supercritical fluid state, the reliability of the hot water-supply compressor is likely to be impaired, and thus it is necessary that an evaporating pressure Pe in a refrigeration cycle in which the carbon dioxide for hot water supply

circulates be no more than 7.4 MPa, which is a critical pressure of carbon dioxide.

[0007] In general, a condensation temperature Tvc in a refrigeration cycle in which an air-conditioning refrigerant circulates is 30°C to 55°C.

[0008] Here, if the condensation temperature Tvc in the refrigeration cycle in which the air-conditioning refrigerant circulates is 30°C, an evaporation temperature Te in a refrigeration cycle in which carbon dioxide circulates is set as Te = 20°C (corresponding to an evaporating pressure of 5.7 MPa for carbon dioxide) so that a temperature difference of 10 K from Tvc is maintained.

[0009] A suction superheat degree in a hot water-supply compressor is set as 5 K, which does not cause a liquid backflow that impairs a reliability of the compressor and provides high performance of the refrigeration cycle, that is, is set so that a suction temperature in the compressor is 25°C.

[0010] Also, generally, a temperature of a heating medium to be supplied to a hot water-supply heat exchanger is set in a range of 5 to 30°C, and a temperature difference between the carbon dioxide and the hot water-supply heating medium is set as 10 K, and thus the temperature of the carbon dioxide at an exit of a refrigerant flow passage in the hot water-supply heat exchanger is 15 to 40°C.

[0011] As a result, the sucked refrigerant in the hot water-supply compressor has a pressure of 5.7 MPa, a temperature of 25°C and a density of 170 kg/m³, and after compression in an isenthalpic change by hot water-supply compressor, the discharged refrigerant has a pressure of 15.0 MPa and a density of 332 kg/m³ at a temperature Td of 100°C.

[0012] On the other hand, if the condensation temperature Tvc in the refrigeration cycle in which the air-conditioning refrigerant circulates is 55°C, as mentioned above, the operation is controlled so that the evaporating pressure Pe in the refrigeration cycle in which the carbon dioxide circulates is no more than 7.4 MPa. Also, since, generally, the temperature of the heating medium supplied to the hot water-supply heat exchanger is set in a range of 5 to 30°C and the temperature difference between the carbon dioxide and the heating medium for hot water supply is set to 10K, the temperature of the carbon dioxide at the exit of the refrigerant flow passage in the hot water-supply heat exchanger is 15 to 40°C. A high pressure in the hot water supply cycle is approximately 13 MPa, and the carbon dioxide at the exit of the refrigerant flow passage in the hot water-supply heat exchanger is subjected to pressure reduction in an isenthalpic change to a low pressure of 7.4 MPa by a carbon dioxide flow adjustment valve and flows into a cascade heat exchanger, and the temperature of the carbon dioxide here is 12 to 30°C.

[0013] In the cascade heat exchanger, assuming that a logarithmic mean temperature difference that can be obtained from the temperature difference between the carbon dioxide and the air-conditioning refrigerant is 10K,

which is equivalent to that when the condensation temperature T_{vc} in the refrigeration cycle in which the air-conditioning refrigerant circulates is 30°C, the condensation temperature T_{vc} of the air-conditioning refrigerant flowing in the cascade heat exchanger is 55°C and the temperature of the carbon dioxide flowing into the cascade heat exchanger is 12 to 30°C, and thus, the temperature of the carbon dioxide flowing out from the cascade heat exchanger is 52 to 54°C.

[0014] As a result, the sucked refrigerant in the hot water-supply compressor 310 has a pressure of 7.4 MPa, has a temperature of 52 to 54°C and a density of 179 to 183 kg/m³, and the discharged refrigerant from the hot water-supply compressor 310 has a pressure of 12.8 to 13.2 MPa and a density of 266 to 276 kg/m³ at a temperature T_d of 100°C.

[0015] In other words, the density of the sucked refrigerant in the hot water-supply compressor when the condensation temperature T_{vc} in the refrigeration cycle in which the air-conditioning refrigerant circulates is 30°C is low compared to the density of the sucked refrigerant in the hot water-supply compressor when the condensation temperature T_{vc} is 55°C.

[0016] Also, the density of the discharged refrigerant from the hot water-supply compressor when the condensation temperature T_{vc} in the refrigeration cycle in which the air-conditioning refrigerant circulates is 30°C is high compared to the density of the discharged refrigerant from the hot water-supply compressor when the condensation temperature T_{vc} is 55°C.

[0017] In general, volumes on the high pressure-side and the low pressure-side of the refrigeration cycle in which the volume carbon dioxide circulates are fixed, and in consideration of the condensation temperature T_{vc} in the refrigeration cycle in which the air-conditioning refrigerant circulates varying in a range of 30 to 55°C, the temperature of the discharged refrigerant from the hot water-supply compressor may increase beyond 100°C, resulting in a problem of a durability of the compressor being impaired because of, e.g., acceleration of, e.g., deterioration of a motor winding coating in the hot water-supply compressor.

[0018] More specifically, if the high pressure-side circuit internal volume $V_c \gg$ the low pressure-side circuit internal volume V_e in the refrigeration cycle in which the carbon dioxide circulates, an amount of the refrigerant charged in the entire circuit is substantially determined by the product of the high pressure-side circuit internal volume and the density of the discharged refrigerant from the hot water-supply compressor.

[0019] For example, a case where, in a state in which the carbon dioxide is enclosed so that the condensation temperature T_{vc} in the refrigeration cycle in which the air-conditioning refrigerant circulates is 30°C and the temperature of the discharged refrigerant from the hot water-supply compressor is 100°C, the condensation temperature T_{vc} sharply increases to 55°C because of, e.g., reduction in number of operating indoor machines

connected to the refrigeration cycle in which the air-conditioning refrigerant circulates will be considered.

[0020] When the condensation temperature T_{vc} varies from 30°C to 55°C, if an attempt is made to maintain the temperature of the discharged refrigerant at 100°C, as mentioned above, the density of the discharged refrigerant from the hot water-supply compressor 310 becomes relatively low compared to that in the case where the condensation temperature T_{vc} is 30°C, resulting in an excess of refrigerant in the refrigeration cycle in which the carbon dioxide circulates. As a result, the high pressure is increased, and the temperature of the discharged refrigerant from the hot water-supply compressor increases beyond 110°C, which is a design limit, resulting in impairing the durability of the compressor because of acceleration of deterioration of, e.g., the refrigerant oil enclosed in the hot water-supply compressor and an insulating coating of a winding of the motor.

[0021] Conversely, if the high pressure-side circuit internal volume $V_c \ll$ low pressure-side circuit internal volume V_e in the refrigeration cycle in which the carbon dioxide circulates, the amount of refrigerant charged in the entire circuit is substantially determined by the product of the low pressure-side circuit internal volume and the density of the sucked refrigerant in the hot water-supply compressor.

[0022] For example, a case where, in a state in which the carbon dioxide is enclosed in such a manner that the condensation temperature T_{vc} in the refrigeration cycle in which the air-conditioning refrigerant circulates is 55°C and the temperature of the discharged refrigerant from the hot water-supply compressor is 100°C, the condensation temperature T_{vc} sharply decreases to 30°C because of, e.g., reduction in number of operating indoor machines connected to the refrigeration cycle in which the air-conditioning refrigerant circulates.

[0023] When the condensation temperature T_{vc} varies from 55°C to 30°C, if an attempt is made to maintain the temperature of the discharged refrigerant at 100°C, as mentioned above, the density of the sucked refrigerant in the hot water-supply compressor becomes relatively low compared to that in the case where the condensation temperature T_{vc} is 55°C, resulting in an excess of refrigerant in the refrigeration cycle in which the carbon dioxide circulates. As a result, the high pressure is increased, and the temperature of the discharged refrigerant from the hot water-supply compressor increases beyond 110°C, which is a design limit, resulting in impairing the durability of the compressor because of acceleration of deterioration of, e.g., the refrigerant oil enclosed in the hot water-supply compressor and an insulating coating of a winding of the motor.

[0024] As described above, in Japanese Patent Laid-Open No. 2004-132647 (Japanese Patent No. 3925383), the internal volumes on the high pressure-side and the low pressure-side of the refrigeration cycle in which the carbon dioxide circulates are not mentioned, and thus, there is a problem in that, unless a proper volume ratio

Ve/Vc between the low pressure-side circuit internal volume Ve and the high pressure-side circuit internal volume Vc of the refrigeration cycle in which the carbon dioxide circulates is set, a condition in which the discharge temperature Td in the refrigeration cycle in which the carbon dioxide circulates increases is generated within a range of variation of the condensation temperature Tvc in the refrigeration cycle in which the air-conditioning refrigerant circulates, which results in impairing the durability of the compressor.

[0025] The present invention is intended to solve the aforementioned problem, and an object of the present invention is to provide an air-conditioning/hot water-supply system that, even if a condensation temperature Tvc in a refrigeration cycle in which an air-conditioning refrigerant circulates varies, prevents a durability of a hot water-supply compressor from being impaired, enabling enhancement in reliability.

SUMMARY OF THE INVENTION

[0026] In order to achieve the above object, the present invention provides an air-conditioning/hot water-supply system including: a first refrigeration cycle including a hot water-supply compressor that compresses a hot water-supply refrigerant, a hot water-supply heat exchanger that allows heat exchange between the hot water-supply refrigerant and a hot water-supply heating medium, and a cascade heat exchanger that allows heat exchange between the hot water-supply refrigerant and an air-conditioning refrigerant; and a second refrigeration cycle including a first circuit in which the cascade heat exchanger and a second refrigerant flow control device that controls a flow of the air-conditioning refrigerant to be supplied to the cascade heat exchanger are serially connected and at least one second circuit in which an indoor heat exchanger that allows heat exchange between the air-conditioning refrigerant and indoor air and a third refrigerant flow control device that controls a flow of the air-conditioning refrigerant to be supplied to the indoor heat exchanger are serially connected, the second refrigeration cycle connecting a thermal load circuit in which the first circuit and the second circuit are parallelly connected, to an air-conditioning compressor that compresses the air-conditioning refrigerant and an outdoor heat exchanger, wherein a volume ratio between a low pressure-side circuit internal volume Ve and a high pressure-side circuit internal volume Vc in the first refrigeration cycle is set as $Ve/Vc = 0.2$ to 10.1 .

[0027] In the air-conditioning/hot water-supply system according to the present invention, the volume ratio between the low pressure-side circuit internal volume Ve and the high pressure-side circuit internal volume Vc in the first refrigeration cycle is set in a proper range so that the discharge temperature Td from the first refrigeration cycle falls within a range of 100 to 110°C at any condensation temperature Tvc in the second refrigeration cycle where the condensation temperature Tvc varies in a

range of 30°C to 55°C .

[0028] Consequently, irrespective of the condensation temperature Tvc in the second refrigeration cycle, no excessive increase in discharge temperature Td in the first refrigeration cycle occurs and thus, a durability of the compressor is not impaired, enabling enhancement in reliability of the first refrigeration cycle.

BRIEF DESCRIPTION OF THE DRAWINGS

[0029]

FIG. 1 is a configuration diagram of a refrigeration cycle in an air-conditioning/hot water-supply system according to Embodiment 1 of the present invention; FIG. 2 is a plan view illustrating an inner structure of a heat generation unit in the present embodiment; FIG. 3 is a front view illustrating an inner structure of a heat generation unit in the present embodiment; FIG. 4 is a p-h diagram for a first refrigeration cycle when a condensation temperature of a second refrigeration cycle is 30°C and 55°C ;

FIG. 5 is a diagram of a relationship between a volume ratio Ve/Vc between a low pressure-side circuit internal volume Ve and a high pressure-side circuit internal volume Vc in a first refrigeration cycle when a temperature of a heating medium supplied to a hot water-supply heat exchanger is 10°C , and a temperature of a discharged refrigerant from a hot water-supply compressor;

FIG. 6 is a diagram of a relationship between a volume ratio Ve/Vc between a low pressure-side circuit internal volume Ve and a high pressure-side circuit internal volume Vc in a first refrigeration cycle when a temperature of a heating medium supplied to a hot water-supply heat exchanger is 5°C , and a temperature of a discharged refrigerant from a hot water-supply compressor;

FIG. 7 is a diagram of a relationship between a volume ratio Ve/Vc between a low pressure-side circuit internal volume Ve and a high pressure-side circuit internal volume Vc in a first refrigeration cycle when a temperature of a heating medium supplied to a hot water-supply heat exchanger is 19°C , and a temperature of a discharged refrigerant from a hot water-supply compressor;

FIG. 8 is a diagram of a relationship between a volume ratio Ve/Vc between a low pressure-side circuit internal volume Ve and a high pressure-side circuit internal volume Vc in a first refrigeration cycle when a temperature of a heating medium supplied to a hot water-supply heat exchanger is 20°C , and a temperature of a discharged refrigerant from a hot water-supply compressor; and

FIG. 9 is a diagram of a relationship between a volume ratio Ve/Vc between a low pressure-side circuit internal volume Ve and a high pressure-side circuit internal volume Vc in a first refrigeration cycle when

a temperature of a heating medium supplied to a hot water-supply heat exchanger is 30°C, and a temperature of a discharged refrigerant from a hot water-supply compressor.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0030] A first aspect of the present invention provides an air-conditioning/hot water-supply system including: a first refrigeration cycle including a hot water-supply compressor that compresses a hot water-supply refrigerant, a hot water-supply heat exchanger that allows heat exchange between the hot water-supply refrigerant and a hot water-supply heating medium, and a cascade heat exchanger that allows heat exchange between the hot water-supply refrigerant and an air-conditioning refrigerant; and a second refrigeration cycle including a first circuit in which the cascade heat exchanger and a second refrigerant flow control device that controls a flow of the air-conditioning refrigerant to be supplied to the cascade heat exchanger are serially connected and at least one second circuit in which an indoor heat exchanger that allows heat exchange between the air-conditioning refrigerant and indoor air and a third refrigerant flow control device that controls a flow of the air-conditioning refrigerant to be supplied to the indoor heat exchanger are serially connected, the second refrigeration cycle connecting a thermal load circuit in which the first circuit and the second circuit are parallelly connected, to an air-conditioning compressor that compresses the air-conditioning refrigerant and an outdoor heat exchanger, wherein a volume ratio between a low pressure-side circuit internal volume V_e and a high pressure-side circuit internal volume V_c in the first refrigeration cycle is set as $V_e/V_c = 0.2$ to 10.1 .

[0031] Consequently, the volume ratio between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle is made to fall within a proper range, whereby a discharge temperature T_d in the first refrigeration cycle falls within a range of 100 to 110°C at any condensation temperature T_{vc} in the second refrigeration cycle where the condensation temperature T_{vc} varies in a range of 30°C to 55°C.

[0032] Therefore, regardless of whether the condensation temperature T_{vc} in the second refrigeration cycle is high or low, no excessive increase in discharge temperature T_d in the first refrigeration cycle occurs, which prevents impairing a durability of the compressor, enabling enhancement in reliability of the second refrigeration cycle.

[0033] An embodiment of the present invention will be described below with reference to the drawings. However, the present invention is not limited by this embodiment.

(Embodiment 1)

[0034] FIG. 1 is a diagram of a cycle configuration of an air-conditioning/hot water-supply system according to an embodiment of the present invention.

[0035] The air-conditioning/hot water-supply system illustrated in FIG. 1 includes an outdoor unit 10, indoor machines 30 and a heat generation unit 40. The present embodiment employs a configuration in which two indoor machines 30 and one heat generation unit 40 are connected to one outdoor unit 10 respectively. However, the refrigeration cycle configuration is not limited to the one illustrated in FIG. 1. For example, two or more outdoor units 10, one or three or more indoor machines 30 and two or more heat generation units 40 can be connected in parallel.

[0036] The outdoor unit 10, the indoor machines 30 and the heat generation unit 40 are connected via a piping in which an air-conditioning refrigerant circulates.

[0037] The outdoor unit 10 and the indoor machines 30 are connected via a gas pipe 25 in which the high-temperature, high-pressure gasified air-conditioning refrigerant flows, a suction pipe 26 in which the low-pressure air-conditioning refrigerant flows and a liquid pipe 27 in which the high-pressure liquefied air-conditioning refrigerant flows. If there are two indoor machines 30 as illustrated in FIG. 1, the indoor machines 30 are parallelly connected to the three pipings. On the other hand, the outdoor unit 10 and the heat generation unit 40, which are parallelly connected to pipes like the indoor machines 30, are, however, connected via the gas pipe 25 and the liquid pipe 27.

[0038] The outdoor unit 10 includes an air-conditioning compressor 11 that compresses an air-conditioning refrigerant. An accumulator 12 that supplies a gas refrigerant to the air-conditioning compressor 11 is connected to the suction side of the air-conditioning compressor 11. An oil separator 13 that separates a refrigerant oil contained in a gaseous air-conditioning refrigerant to be discharged is connected to the discharge side of the air-conditioning compressor 11. The refrigerant oil separated from the air-conditioning refrigerant by the oil separator 13 is returned to the air-conditioning compressor 11 via an oil return pipe 14. Communication with the oil return pipe 14 is controlled by opening/closing of an oil return pipe opening/closing valve 15.

[0039] Also, the outdoor unit 10 includes an outdoor heat exchanger 16, and in the vicinity of the outdoor heat exchanger 16, an outdoor fan 17 that supplies air around the outdoor unit 10 to the outdoor heat exchanger 16 is provided. The outdoor heat exchanger 16 is configured so that heat exchange is performed between air sent by the outdoor fan 17 and the air-conditioning refrigerant, and for the outdoor heat exchanger 16, in general, a fin tube-type or microtube-type heat exchanger is employed.

[0040] The outdoor unit 10 includes an outdoor refrigerant flow adjustment valve 18 that adjusts a flow of the

air-conditioning refrigerant to be supplied to the outdoor heat exchanger 16, an outdoor gas pipe opening/closing valve 19 that controls a flow of the air-conditioning refrigerant in the gas pipe 25, and an outdoor suction pipe opening/closing valve 20 that controls a flow of the air-conditioning refrigerant in the suction pipe 26.

[0041] Each indoor machine 30 includes an indoor heat exchanger 31, an indoor fan 32 that supplies air around the indoor machine 30 to the indoor heat exchanger 31, and an indoor refrigerant flow adjustment valve 33 (third refrigerant flow control device) that adjusts a flow of the air-conditioning refrigerant to be supplied to the indoor heat exchanger 31. The indoor heat exchanger 31 is configured so that heat exchange is performed between air sent by the indoor fan 32 and the air-conditioning refrigerant, and for the indoor heat exchanger 31, in general, a fin tube-type or microtube-type heat exchanger is employed.

[0042] Also, each indoor machine 30 includes an indoor gas pipe opening/closing valve 34 that controls whether or not the air-conditioning refrigerant flows from the gas pipe 25, and an indoor suction pipe opening/closing valve 35 that controls whether or not the air-conditioning refrigerant flows to the suction pipe 26.

[0043] The heat generation unit 40 includes a hot water-supply compressor 41 that compresses a hot water-supply refrigerant, a hot water-supply heat exchanger 42 that allows heat exchange between the hot water-supply refrigerant and a heating medium containing water as a main component, and a hot water-supply refrigerant flow adjustment valve 43 that adjusts a flow of the hot water-supply refrigerant.

[0044] Also, the heat generation unit 40 includes a cascade heat exchanger 44 in which heat exchange is performed between the air-conditioning refrigerant supplied from the gas pipe 25 and the hot water-supply refrigerant, a heat generation unit refrigerant flow adjustment valve 45 (second refrigerant flow control device) that adjusts a flow of the air-conditioning refrigerant to be supplied to the cascade heat exchanger 44, and a heating medium pump 46 that supplies the heating medium to the hot water-supply heat exchanger 42.

[0045] The heating medium boiled up to 70 to 90°C by the hot water-supply heat exchanger 42 is stored in a hot water storage tank (not illustrated). If the heating medium is drinking water, the heating medium is directly used for hot water supply. On the other hand, if the heating medium is not drinking water such as an antifreeze liquid, the heating medium is supplied to, e.g., a radiator installed indoors for use in air heating or the heating medium delivers heat to drinking water in the hot water storage tank and thus is used for hot water supply.

[0046] Here, a first refrigeration cycle 100 is configured by connecting the hot water-supply compressor 41, the hot water-supply heat exchanger 42, the hot water-supply refrigerant flow adjustment valve 43 and the cascade heat exchanger 44 in a loop.

[0047] Also, a second refrigeration cycle 200 includes

a first circuit 201 in which the cascade heat exchanger 44 and the heat generation unit refrigerant flow adjustment valve 45 are serially connected, and at least one second circuit 202 in which the indoor heat exchanger 31 and the indoor refrigerant flow adjustment valve 33 that controls the flow of the air conditioning refrigerant to be supplied to the indoor heat exchanger 31 are serially connected, and is configured by connecting a heat load circuit in which the first circuit 201 and the second circuit 202 are parallelly connected, to the air-conditioning compressor 11 and the outdoor heat exchanger 16.

[0048] Next, an inner structure of the heat generation unit 40 in the present embodiment will be described.

[0049] FIG. 2 is a plan view illustrating the inner structure of the heat generation unit 40 in the present embodiment, and FIG. 3 is a front view illustrating the inner structure of the heat generation unit 40.

[0050] In the heat generation unit 40, a refrigeration cycle formed by the hot water-supply compressor 41, the hot water-supply heat exchanger 42, the hot water-supply refrigerant flow adjustment valve 43 and the cascade heat exchanger 44, the heat generation unit refrigerant flow adjustment valve 45 and the heating medium pump 46 are housed in the casing 50.

[0051] In the present embodiment, for the hot water-supply heat exchanger 42, for example, a double pipe-type heat exchanger is used. A double pipe-type heat exchanger is a heat exchanger formed by inserting at least one pipe (inner pipe) into a tube (outer pipe) having a rough circular cross-sectional shape. If there are a plurality of inner pipes, the inner pipes are helically twisted and inserted into the outer pipe. If a carbon dioxide refrigerant is used for the hot water-supply refrigerant, the carbon dioxide refrigerant is made to flow in the inner pipe of the hot water-supply heat exchanger 42 and the heating medium is made to flow between the outer pipe and the inner pipe.

[0052] For a material of a double pipe-type heat exchanger, a copper pipe having high heat conductivity is often used.

[0053] Also, for the hot water-supply heat exchanger 42, for example, a plate-type heat exchanger or a shell-and-tube-type heat exchanger may be used. Also, for the cascade heat exchanger 44, for example, a plate-type heat exchanger or a shell-and-tube-type heat exchanger is used.

[0054] A heat exchange capability of a double pipe-type heat exchanger is proportional to a length of a double pipe. Therefore, in order to ensure a maximum heat exchange capacity in a limited installation capacity, a double pipe-type heat exchanger is formed by winding a double pipe. When installing a double pipe-type heat exchanger, a double pipe-type is placed so as to be as horizontal as possible, in order to prevent a significant decrease in heat exchange capability due to retention of air in a part of the double pipe-type through which a heating medium passes.

[0055] As illustrated in FIGS. 2 and 3, the hot water-

supply compressor 41 is fixed to a bottom plate member 51 of the casing 50 by a fixing member 67, with an antivibration member 60 such as rubber inserted therebetween.

[0056] Also, the hot water-supply heat exchanger 42 is also fixed to the bottom plate member 51, and the cascade heat exchanger 44 is fixed to an upper surface of the hot water-supply heat exchanger 42.

[0057] Also, as illustrated in FIG. 2, the heating medium pump 46 is installed in such a manner that the heating medium pump 46 is fixed to a side plate member 53 on a back surface side of the casing 50 and a lower end surface of the heating medium pump 46 is placed at a position that is lower than a lower end surface of the cascade heat exchanger 44.

[0058] The hot water-supply heat exchanger 42 and the cascade heat exchanger 44 illustrated in FIGS. 2 and 3 each include a heat insulating material such as foamed polystyrene or a thick felt and a component member further surrounding the heat insulating material. In particular, the hot water-supply heat exchanger 42 is surrounded by a high-strength iron plate to protect a surface of the heat insulating material because deformation of the heat insulating material due to the weight of the cascade heat exchanger 44 installed on an upper part of the hot water-supply heat exchanger 42 is contemplated.

[0059] Here, the cascade heat exchanger 44 does not necessarily need to be in contact with the component member surrounding the hot water-supply heat exchanger 42. In this case, the cascade heat exchanger 44 and the heat insulating material around the cascade heat exchanger 44 are surrounded by the component member having a strength enough to support the weights of the cascade heat exchanger 44 and the heat insulating material and then fixed to a side plate member 52 of the heat generation unit 40.

[0060] Furthermore, as illustrated in FIGS. 2 and 3, in the bottom plate member 51, a drain outlet 62 is provided within a region in which the hot water-supply heat exchanger 42 and the heating medium pump 46 are located on the bottom plate member 51 as viewed vertically. An upper surface of the bottom plate member 51 is properly inclined toward the drain outlet 62 so that water can quickly be discharged from the drain outlet 62 to the outside of the heat generation unit 40.

[0061] The flow of the heating medium in heating medium pipings 63, 64 and 65 is generated by driving of the heating medium pump 46. The heating medium that has flown into the heat generation unit 40 flows into the heating medium pump 46 through the heating medium piping 63 and is sent out to the heating medium piping 64. Furthermore, the heating medium enters the hot water-supply heat exchanger 42 and is heated by the hot water-supply refrigerant so as to have a high temperature of 70 to 90°C, and is then sent out to the outside of the heat generation unit 40 through the heating medium piping 65.

[0062] For the heating medium pipings 63, 64 and 65, mostly, copper pipes, which have high workability, are

used, but resin materials are also used. On the other hand, for a heating medium suction portion and a heating medium discharge portion of the heating medium pump 46, a resin material is often used. Also, for the hot water-supply heat exchanger 42, which is a double pipe-type heat exchanger, as described above, copper is often used, and a connection port is also made of a copper pipe.

[0063] As described above, in a passage in which the heating medium flows (heating medium piping 63 → heating medium pump 46 → heating medium piping 64 → hot water-supply heat exchanger 42 → heating medium piping 65), there are both resin materials and copper and thus there are parts of connection between different materials. A seal material (not illustrated) is inserted between the heating medium pump connection parts 66, which are then fixed to each other, thereby preventing leakage of the heating medium.

[0064] Here, R410A, R32, R407C, or the like, which are refrigerants generally used for a household air conditioner or a building air conditioner, are used for the air-conditioning refrigerant, and carbon dioxide is used for the hot water-supply refrigerant. A case where carbon dioxide is used as the hot water-supply refrigerant will be described below.

[0065] Here, in order to produce hot water having a high-temperature of 90°C, it is necessary to bring a temperature of a discharged refrigerant from the hot water-supply compressor 41 to no less than 100°C. On the other hand, an excessive increase in temperature of the discharged refrigerant from the hot water-supply compressor 41 may cause deterioration of, e.g., the refrigerant oil enclosed in the hot water-supply compressor 41 and an insulating coating of a winding of a motor.

[0066] Thus, for a temperature of 100°C of the discharged refrigerant of the hot water-supply compressor 41 for producing hot water having a high temperature of 90°C, 110°C, which is 10 K higher than the temperature, is generally set as an upper limit temperature enabling ensuring reliability of the hot water-supply compressor 41, that is, an upper limit use temperature, and thus, in order to ensure reliability of the compressor, it is necessary that the compressor operates at the upper limit use temperature or less.

[0067] Also, if the sucked refrigerant in the hot water-supply compressor 41 turns into a supercritical fluid state, an amount of dissolved refrigerant relative to the refrigerant oil enclosed in the hot water-supply compressor 41 increases and a viscosity of the refrigerant oil thus decreases, which accelerates abrasion of a sliding material, which may impair the reliability of the hot water-supply compressor 41. Thus, it is necessary that an evaporating pressure P_e of carbon dioxide in the first refrigeration cycle 100 be no more than 7.4 MPa, which is a critical pressure of carbon dioxide.

[0068] Also, in general, the temperature of the heating medium to be supplied to the hot water-supply heat exchanger 42 is in a range of 5 to 30°C and a difference in temperature between carbon dioxide and the hot water-

supply heating medium is 10K. Therefore, in the hot water-supply heat exchanger 42, the temperature of carbon dioxide after heat exchange with the heating medium is 15 to 40°C.

[0069] In the below, for property values of the carbon dioxide, values derived from Reference Fluid Thermodynamic and Transport Properties Ver. 9.0 (hereinafter, abbreviated as "Refprop Ver. 9.0") issued by National Institute of Standards and Technology (hereinafter, abbreviated as NIST) are used.

[0070] In the air-conditioning/hot water-supply system according to the present embodiment, a volume ratio between a low pressure-side circuit internal volume V_e and a high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 is set to $V_e/V_c = 0.2$ to 10.1 . Here, the low pressure-side circuit internal volume V_e in the first refrigeration cycle 100 is an internal volume from an exit portion of the hot water-supply refrigerant flow adjustment valve 43 to an entrance portion of the hot water-supply compressor 41, and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 is an internal volume from an exit portion of the hot water-supply compressor 41 to an entrance portion of the hot water-supply refrigerant flow adjustment valve 43.

[0071] Therefore, when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is any temperature in the range of 5 to 30°C, under any condensation temperature condition where a condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 varies in a range of 30°C to 55°C, a temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 in the first refrigeration cycle 100 falls in a range of 100 to 110°C, which is equal to or below the use upper limit discharge temperature of the hot water-supply compressor 41.

[0072] A reason of the above will be described with reference to the p-h diagram in FIG. 4.

[0073] FIG. 4 is a p-h diagram for the first refrigeration cycle 100 when the condensation temperature in the second refrigeration cycle 200 is 30°C and 55°C. In FIG. 4, the abscissa axis represents the specific enthalpy of carbon dioxide, and the ordinate axis represents the pressure of carbon dioxide.

[0074] Reference numerals 500 and 510 denote refrigerant state transition of the first refrigeration cycle 100 when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 are 30°C and 55°C, respectively, and the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 is 100°C.

[0075] Reference numeral 501 denotes a state of the sucked refrigerant in the hot water-supply compressor 41 in the refrigerant state transition 500, reference numeral 502 denotes a state of the discharged refrigerant from the hot water-supply compressor 41, reference numeral 503 denotes a state of the refrigerant at the entrance of the hot water-supply refrigerant flow adjustment

valve 43, and reference numeral 504 denotes a state of the refrigerant at the exit of the hot water-supply refrigerant flow adjustment valve 43.

[0076] Also, reference numeral 511 denotes a state of the sucked refrigerant in the hot water-supply compressor 41 in the refrigerant state transition 510, reference numeral 512 denotes a state of the discharged refrigerant in the hot water-supply compressor 41, reference numeral 513 denotes a state of the refrigerant at the entrance of the hot water-supply refrigerant flow adjustment valve 43, and reference numeral 514 denotes a state of the refrigerant at the exit of the hot water-supply refrigerant flow adjustment valve 43.

[0077] Also, reference numeral 540 denotes an isotherm of carbon dioxide at 100°C, and reference numeral 550 denotes an isotherm of carbon dioxide at 110°C. Also, reference numeral 560 denotes a saturation line of carbon dioxide.

[0078] In this case, the refrigerant state transition 500 is determined through the below process.

[0079] First, the refrigerant state 501 is determined. The evaporation temperature T_e of the carbon dioxide in the first refrigeration cycle 100 is 20°C in consideration of the temperature difference of 10 K being maintained relative to the condensation temperature T_{vc} of the second refrigeration cycle 200 = 30°C. Therefore, the refrigerant pressure in 501 is 5.7 MPa as a result of pressure conversion based on the evaporation temperature $T_e = 20^\circ\text{C}$. On the other hand, the refrigerant temperature in 501 is 25°C, which is a result of addition of a suction superheat degree of 5 K in the hot water-supply compressor 41 to 20°C, which is the evaporation temperature T_e of the carbon dioxide in the first refrigeration cycle 100. Therefore, a refrigerant density in 501 is 170 kg/m³ with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0080] Next, the refrigerant state 502 is determined. Reference numeral 502 denotes a state after adiabatic compression in the hot water-supply compressor 41 from the refrigerant state 501, and the temperature in that state is 100°C. Assuming that the specific enthalpy does not vary in the adiabatic compression step, the refrigerant pressure and the refrigerant density in 502 are 15.0 MPa and 332 kg/m³, respectively, with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0081] Next, the refrigerant state 503 is determined. The refrigerant pressure in 503 does not vary from the refrigerant state 502 and thus is 15.0 MPa. Also, if the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C, the refrigerant temperature in 503 is 20°C in consideration of the temperature difference of 10 K from the carbon dioxide which with the heating medium exchanges heat. In this case, the refrigerant density is 904 kg/m³ with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0082] Next, the refrigerant state 504 is determined. The refrigerant pressure in 504 is 5.7 MPa, which is the same as the pressure in 501. In an expansion step from

503 to 504, assuming that the specific enthalpy does not vary, the refrigerant temperature and the refrigerant density in 504 are 15°C and 838 kg/m³, respectively, with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0083] Likewise, the refrigerant state transition 510 can be determined by a process that is similar to that of the refrigerant state transition 500 described above, and when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C, the refrigerant state 511 is such that the pressure is 7.4 MPa, the temperature is 54°C and the density is 180 kg/m³. Also, the refrigerant state 512 is such that the pressure is 12.8 MPa, the temperature is 100°C and the density is 267 kg/m³, the refrigerant state 513 is such that the pressure is 12.8 MPa, the temperature is 20°C and the density is 886 kg/m³, and the refrigerant state 514 is such that the pressure is 7.4 MPa, the temperature is 17°C and the density is 847 kg/m³.

[0084] A high pressure-side average refrigerant density in the refrigerant state transition 500 is calculated as an average of the refrigerant densities in 502 and 503, that is, is 618 kg/m³, and a low pressure-side average refrigerant density is calculated as an average of the refrigerant densities in 501 and 504, that is, 504 kg/m³.

[0085] Also, a high pressure-side average refrigerant density in the refrigerant state transition 510 is calculated as an average of the refrigerant densities in 512 and 513, that is, 577 kg/m³, and a low pressure-side average refrigerant density is calculated as an average of the refrigerant densities in 511 and 514, that is, 513 kg/m³.

[0086] Here, assuming that a cross-sectional area of each of the low pressure-side and high pressure-side refrigerant passages is constant, a proper refrigerant amount Mref for the circuit of the first refrigeration cycle 100 can be calculated as a sum of the product of a high pressure-side circuit internal volume Vc and a high pressure-side average refrigerant density and the product of a low pressure-side circuit internal volume Ve and a low pressure-side average refrigerant density. In the refrigerant state transition 500, a proper refrigerant amount Mref500 for the circuit of the first refrigeration cycle 100 is 618Vc+504Ve, and in the refrigerant state transition 510, a proper refrigerant amount Mref510 for the circuit of the first refrigeration cycle 100 is 577Vc+513Ve.

[0087] Here, if a volume ratio Ve/Vc between the low pressure-side circuit internal volume Ve and the high pressure-side circuit internal volume Vc, the volume ratio Ve/Vc making Mref500 and Mref510 be equal to each other, is calculated, $(618-577)/(513-504) = 4.6$.

[0088] In other words, if the refrigerant amount Mref500 of the carbon dioxide is enclosed in the circuit of the first refrigeration cycle 100 whose volume ratio Ve/Vc between the low pressure-side circuit internal volume Ve and the high pressure-side circuit internal volume Vc = 4.6, when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C, in each of cases where the condensation temper-

ature Tvc of the air-conditioning refrigerant in the second refrigeration cycle 200 is 30°C and 55°C, the temperature Td of the discharged refrigerant from the hot water-supply compressor 41 is 100°C, which falls within a range of no less than 100°C, which is necessary for production of a heating medium having a high temperature of 90°C, and no more than 110°C, which is the upper limit use temperature for the compressor.

[0089] Where the volume ratio Ve/Vc between the low pressure-side circuit internal volume Ve and the high pressure-side circuit internal volume Vc < 4.6, Mref500 > Mref510, and thus, enclosure of the refrigerant amount Mref500 of the carbon dioxide in the circuit of the first refrigeration cycle 100 causes an excess of the refrigerant in the refrigerant state transition 510, and the high pressure is increased so that the high pressure-side refrigerant density increases, and the temperature of the discharged refrigerant from the hot water-supply compressor 41 becomes higher than 100°C. In this case, a volume ratio Ve/Vc for making the temperature of the discharged refrigerant from the hot water-supply compressor 41 be equal to or below the upper limit use temperature of 110°C in the second refrigeration cycle 200 is calculated.

[0090] In FIG. 4, reference numeral 520 denotes a refrigerant state transition in which when the condensation temperature Tvc in the second refrigeration cycle 200 is 55°C, the temperature Td of the discharged refrigerant from the hot water-supply compressor 41 in the first refrigeration cycle 100 is 110°C.

[0091] Here, reference numeral 521 denotes a state of the sucked refrigerant in the hot water-supply compressor 41 in the refrigerant state transition 520, reference numeral 522 denotes a state of the discharged refrigerant from the hot water-supply compressor 41, reference numeral 523 denotes a state of the refrigerant at the entrance of the hot water-supply refrigerant flow adjustment valve 43, and reference numeral 524 denotes a state of the refrigerant at the exit of the hot water-supply refrigerant flow adjustment valve 43.

[0092] The refrigerant state transition 520 is determined through the below process.

[0093] First, the refrigerant state 521 is determined. The refrigerant state 521, which is equal to the refrigerant state 511, is such that the pressure is 7.4 MPa, the temperature is 54°C and the density is 180 kg/m³.

[0094] Next, the refrigerant state 522 is determined. Reference numeral 522 denotes a state after adiabatic compression in the hot water-supply compressor 41 from the refrigerant state 521, and the temperature in that state is 110°C. Assuming that the specific enthalpy does not vary in the adiabatic compression step, the refrigerant pressure and the refrigerant density in 522 are 14.5 MPa and 288 kg/m³, respectively, with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0095] Next, the refrigerant state 523 is determined. In the refrigerant pressure in 523, the pressure does not vary from the refrigerant state 522 and thus is 14.5 MPa.

Also, in consideration of the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 being 10°C and the temperature difference between the carbon dioxide and the hot water supply heating medium being 10 K, the refrigerant temperature in 523 is 20°C. In this case, the refrigerant density is 900 kg/m³ with reference to the refrigerant property values in Refprop Ver. 9.0 in NIST.

[0096] Next, the refrigerant state 524 is determined. The refrigerant pressure in 524 is 7.4 MPa, which is equal to the pressure in the refrigerant state 521. Assuming that in an expansion step from 523 to 524, the specific enthalpy does not vary, the refrigerant temperature and the refrigerant density in 524 are 16°C and 851 kg/m³, respectively, with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0097] A high pressure-side average refrigerant density in the refrigerant state transition 520 is calculated as an average of the refrigerant densities in 522 and 523, that is, 594 kg/m³, and a low pressure-side average refrigerant density is calculated as an average of the refrigerant densities in 521 and 524, that is, 516 kg/m³. Therefore, a proper refrigerant amount Mref520 for the circuit of the first refrigeration cycle 100 = 594V_c+516V_e.

[0098] Here, upon calculation of a volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c, the volume ratio V_e/V making Mref500 and Mref520 be equal to each other, (618-594)/(516-504) = 2.1.

[0099] In other words, if the refrigerant amount Mref500 of the carbon dioxide is enclosed in the circuit of the first refrigeration cycle 100 whose volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c = 2.1, when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C, in respective cases where the condensation temperatures T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 are 30°C and 55°C, the respective temperatures T_d of the discharged refrigerant from the hot water-supply compressor 41 are 100°C and 110°C, which fall within a range of no less than 100°C, which is necessary for production of a heating medium having a high temperature of 90°C, and no more than 110°C, which is the upper limit use temperature for the compressor.

[0100] Where the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c > 4.6, Mref500 < Mref510, and thus, enclosure of the refrigerant amount Mref510 of the carbon dioxide in the circuit of the first refrigeration cycle 100 causes an excess of the refrigerant in the refrigerant state transition 500, and thus the high pressure is increased so that the high pressure-side refrigerant density increases, resulting in a refrigerant state transition in which the temperature of the discharged refrigerant from the hot water-supply compressor 41 becomes higher than 100°C. In this case, a volume

ratio V_e/V_c for making the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 in the first refrigeration cycle 100 be equal to or below the upper limit use temperature of 110°C is calculated.

[0101] In FIG. 4, reference numeral 530 denotes a refrigerant state transition in which when the condensation temperature T_{vc} in the second refrigeration cycle 200 is 30°C, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 in the first refrigeration cycle 100 is 110°C.

[0102] Here, reference numeral 531 denotes a state of the sucked refrigerant in the hot water-supply compressor 41 in the refrigerant state transition 530, reference numeral 532 denotes a state of the discharged refrigerant from the hot water-supply compressor 41, reference numeral 533 denotes a state of the refrigerant at the entrance of the hot water-supply refrigerant flow adjustment valve 43, and reference numeral 534 denotes a state of the refrigerant at the exit of the hot water-supply refrigerant flow adjustment valve 43.

[0103] The refrigerant state transition 530 is determined through the below process.

[0104] First, the refrigerant state 531 is determined. The refrigerant state 531, which is equal to the refrigerant state 501, is such that the pressure is 5.7 MPa, the temperature is 25°C and the density is 171 kg/m³.

[0105] Next, the refrigerant state 532 is determined. Reference numeral 532 denotes a state after adiabatic compression in the hot water-supply compressor 41 from the refrigerant state 531, and the temperature in that state is 110°C. Assuming that the specific enthalpy does not vary in the adiabatic compression step, the refrigerant pressure and the refrigerant density in 532 are 17.0 MPa and 357 kg/m³, respectively, with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0106] Next, the refrigerant state 533 is determined. In the refrigerant pressure 533, the pressure does not vary from the refrigerant state 532 and thus is 17.0 MPa. Also, in consideration of the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 being 10°C and the temperature difference between the carbon dioxide and the hot water supply heating medium being 10 K, the refrigerant temperature in 533 is 20°C. In this case, the refrigerant density is 886 kg/m³ with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0107] Next, the refrigerant state 534 is determined. The refrigerant pressure in 534 is 5.7 MPa, which is equal to the pressure in the refrigerant state in 531. Assuming that in an expansion step from 533 to 534, the specific enthalpy does not vary, the refrigerant temperature and the refrigerant density in 534 are 14°C and 843 kg/m³, respectively, with reference to the refrigerant property values in Refprop Ver. 9.0 by NIST.

[0108] A high pressure-side average refrigerant density in the refrigerant state transition 530 is calculated as an average of the refrigerant densities in 532 and 533,

that is, 638 kg/m^3 , a low pressure-side average refrigerant density is calculated as an average of the refrigerant densities in 531 and 534, that is, 506 kg/m^3 . Therefore, a proper refrigerant amount Mref530 for the inside of the circuit of the first refrigeration cycle 100 is $638V_c + 506V_e$.

[0109] Here, upon calculation of a volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c , the volume ratio V_e/V_c making Mref500 and Mref530 be equal to each other, $(638-577)/(513-506) = 8.7$.

[0110] In other words, if the refrigerant amount Mref510 of the carbon dioxide is enclosed in the circuit of the first refrigeration cycle 100 whose volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume $V_c = 8.7$, when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C , in respective cases where the condensation temperatures T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 are 30°C and 55°C , the respective temperatures T_d of the discharged refrigerant from the hot water-supply compressor 41 are 110°C and 100°C , which fall within a range of no less than 100°C , which is necessary for production of a heating medium having a temperature of 90°C and no more than 110°C , which is the upper limit use temperature for the compressor.

[0111] Likewise, a volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c , the volume ratio V_e/V_c making a proper refrigerant amount Mref530 for the circuit of the first refrigeration cycle 100 in the refrigerant state transition 530, which is $638V_c + 506V_e$, and a proper refrigerant amount Mref520 for the circuit of the first refrigeration cycle 100 in the refrigerant state transition 520, which is $594V_c + 516V_e$, be equal to each other. In this case, V_e/V_c is $(638-594)/(516-506) = 4.7$.

[0112] In other words, if the refrigerant amount Mref520 of the carbon dioxide is enclosed in the circuit of the first refrigeration cycle 100 whose volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume $V_c = 4.7$, when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C , in respective cases where the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 30°C and 55°C , the respective temperatures T_d of the discharged refrigerant from the hot water-supply compressor 41 are both 110°C , which falls within a range of no less than 100°C , which is necessary for production of a heating medium having a high temperature of 90°C , and no more than 110°C , which is the upper limit use temperature for the compressor.

[0113] The above will be described with reference to FIG. 5.

[0114] FIG. 5 is a diagram of a relationship between the volume ratio V_e/V_c between the low pressure-side

circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 and the temperature of the discharged refrigerant from the hot water-supply compressor 41 when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C .

[0115] In FIG. 5, the abscissa axis represents the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c , and the ordinate axis represents the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 30°C .

[0116] Reference numeral 614 denotes a line indicating variation of the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 30°C , relative to V_e/V_c where, when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C , the amount of enclosed refrigerant in the first refrigeration cycle 100 is set as the refrigerant amount Mref 520, which is a proper enclosure amount in the refrigerant state transition 520 in the first refrigeration cycle 100 in which when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 55°C , the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 is 110°C .

[0117] The right side of the line 614 indicates that when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 55°C , the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 is no more than 110°C .

[0118] Also, reference numeral 615 denotes a line indicating variation of the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 30°C , relative to V_e/V_c where, when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 10°C , the amount of enclosed refrigerant in the first refrigeration cycle 100 is set as the refrigerant amount Mref510, which is a proper enclosure amount in the refrigerant state transition 510 in the first refrigeration cycle 100 in which when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 55°C , the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 is 100°C .

[0119] The left side of the line 615 indicates that when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 55°C , the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 is no less than 100°C .

[0120] Therefore, the area sandwiched between the lines 614 and 615 is an area in which when the conden-

sation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 55°C, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 is no less than 100°C and no more than 110°C.

[0121] Also, an area sandwiched between lines 616 and 617 is an area in which when the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 is 30°C, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 is no less than 100°C and no more than 110°C.

[0122] According to the above, in FIG. 5, in the area surrounded by the lines 614, 615, 616 and 617, that is, a range in which the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c is 2.1 to 8.7, even if the condensation temperature T_{vc} of the air-conditioning refrigerant in the second refrigeration cycle 200 varies from 30°C to 55°C, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 in the first refrigeration cycle 100 falls within the range of 100°C to 110°C, and thus, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 is equal to or below the upper limit use temperature.

[0123] FIG. 6 is a diagram of a relationship between the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 and the temperature of the discharged refrigerant from the hot water-supply compressor 41 when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 5°C.

[0124] When the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 5°C, as illustrated in FIG. 6, as a result of the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 being set as $V_e/V_c = 0.6$ to 4.0, under any condensation temperature condition where the condensation temperature T_{vc} in the second refrigeration cycle 200 varies from 30°C to 55°C, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 falls within the range of 100 to 110°C, which is equal to or lower than the use upper limit discharge temperature for the hot water-supply compressor 41.

[0125] FIG. 7 is a diagram of a relationship between the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 and the temperature of the discharged refrigerant from the hot water-supply compressor 41 when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 19°C.

[0126] When the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is

19°C, as illustrated in FIG. 7, as a result of the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 being set as $V_e/V_c = 0.9$ to 10.1, under any condensation temperature condition where the condensation temperature T_{vc} in the second refrigeration cycle 200 varies from 30°C to 55°C, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 falls within the range of 100 to 110°C, which is equal to or lower than the use upper limit discharge temperature for the hot water-supply compressor 41.

[0127] FIG. 8 is a diagram of a relationship between the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 and the temperature of the discharged refrigerant from the hot water-supply compressor 41 when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 20°C.

[0128] When the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 20°C, as illustrated in FIG. 8, as a result of the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 being set as $V_e/V_c = 0.6$ to 3.9, under any condensation temperature condition where the condensation temperature T_{vc} in the second refrigeration cycle 200 varies from 30°C to 55°C, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 falls within the range of 100 to 110°C, which is equal to or lower than the use upper limit discharge temperature for the hot water-supply compressor 41.

[0129] FIG. 9 is a diagram of a relationship between the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 and the temperature of the discharged refrigerant from the hot water-supply compressor 41 when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 30°C.

[0130] When the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is 30°C, as illustrated in FIG. 9, as a result of the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 being set as $V_e/V_c = 0.2$ to 0.9, under any condensation temperature condition where the condensation temperature T_{vc} in the second refrigeration cycle 200 varies from 30°C to 55°C, the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 falls within the range of 100 to 110°C, which is equal to or lower than the use upper limit discharge temperature for the hot water-supply compressor 41.

[0131] According to the above, as a result of the volume ratio V_e/V_c between the low pressure-side circuit internal

volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 being set as $V_e/V_c = 0.2$ to 10.1 , when the temperature of the heating medium supplied to the hot water-supply heat exchanger 42 is any temperature of 5 to 30°C , at any condensation temperature T_{vc} in the second refrigeration cycle 200 where the condensation temperature T_{vc} varies from 30°C to 55°C , the temperature T_d of the discharged refrigerant from the hot water-supply compressor 41 falls within the range of 100 to 110°C , which is equal to or below the upper limit use temperature for the hot water-supply compressor 41.

[0132] Next, operation of the outdoor unit 10, the indoor machines 30 and the heat generation unit 40 will be described with reference to the refrigeration cycle diagram in FIG. 1.

[0133] In operation for air cooling only, in the outdoor unit 10, the outdoor gas pipe opening/closing valve 19 is set to be opened, and the outdoor suction pipe opening/closing valve 20 is set to be closed, in each of the indoor machines 30, the indoor gas pipe opening/closing valve 34 is set to be closed and the indoor suction pipe opening/closing valve 35 is set to be opened, and in the heat generation unit 40, the heat generation unit refrigerant flow adjustment valve 45 is set to be fully closed.

[0134] The high-temperature, high-pressure air-conditioning refrigerant compressed in the air-conditioning compressor 11 enters the outdoor heat exchanger 16 via the outdoor gas pipe opening/closing valve 19 and is cooled by air around the outdoor unit 10 and turns into a liquid state. In the liquid-state air-conditioning refrigerant flows into the liquid pipe 27 via the outdoor refrigerant flow adjustment valve 18 in a fully-opened state and reaches the respective indoor machines 30.

[0135] The air-conditioning refrigerant that has reached each indoor machine 30 is subjected to pressure reduction by the indoor refrigerant flow adjustment valve 33 and thereby turns into a low-temperature, low-pressure gas-liquid two-phase state, and then flows into the indoor heat exchanger 31 and takes heat from indoor air to cool the air. Through this step, the air-conditioning refrigerant evaporates and enters the suction pipe 26 via the indoor suction pipe opening/closing valve 35 and returns to the outdoor unit 10. The air-conditioning refrigerant that has returned to the outdoor unit 10 returns to the air-conditioning compressor 11 via the accumulator 12.

[0136] Also, in operation for air heating only, in the outdoor unit 10, the outdoor gas pipe opening/closing valve 19 is set to be closed and the outdoor suction pipe opening/closing valve 20 is set to be opened, in each of the indoor machines 30, the indoor gas pipe opening/closing valve 34 is set to be opened and the indoor suction pipe opening/closing valve 35 is set to be closed, and in the heat generation unit 40, the heat generation unit refrigerant flow adjustment valve 45 is set to be fully closed.

[0137] The high-temperature, high-pressure air-conditioning refrigerant compressed in the air-conditioning

compressor 11 flows into the gas pipe 25 and reaches the respective indoor machines 30. The air-conditioning refrigerant that has reached each indoor machine 30 flows into the indoor heat exchanger 31 via the indoor gas pipe opening/closing valve 34 and releases heat to indoor air to heat the air. Through this step, the air-conditioning refrigerant is condensed and thereby liquefied, flows into the liquid pipe 27 via the indoor refrigerant flow adjustment valve 33 in a fully-opened state and returns to the outdoor unit 10.

[0138] The air-conditioning refrigerant that has returned to the outdoor unit 10 is subjected to pressure reduction by the outdoor refrigerant flow adjustment valve 18, turns into a low-temperature, low-pressure gas-liquid two-phase state and then enters the outdoor heat exchanger 16, and is heated by air around the outdoor unit 10 and thereby evaporates. The evaporated and gasified air-conditioning refrigerant returns to the air-conditioning compressor 11 via the outdoor suction pipe opening/closing valve 20 and the accumulator 12.

[0139] In operation for hot water supply only, in the outdoor unit 10, the outdoor gas pipe opening/closing valve 19 is set to be closed and the outdoor suction pipe opening/closing valve 20 is set to be opened, in each of the indoor machines 30, the indoor gas pipe opening/closing valve 34 and the indoor suction pipe opening/closing valve 35 are both set to be closed, and in the heat generation unit 40, the heat generation unit refrigerant flow adjustment valve 45 is opened.

[0140] The high-temperature, high-pressure air-conditioning refrigerant compressed in the air-conditioning compressor 11 flows into the gas pipe 25 and reaches the heat generation unit 40. In the heat generation unit 40, the hot water-supply compressor 41 operates, whereby the hot water-supply refrigerant circulates through the hot water-supply compressor 41, the hot water-supply heat exchanger 42, the hot water-supply refrigerant flow adjustment valve 43 and the cascade heat exchanger 44 in this order.

[0141] The air-conditioning refrigerant that has reached the heat generation unit 40 heats the carbon dioxide in the cascade heat exchanger 44 and the air-conditioning refrigerant itself is thereby cooled and liquefied, and then flows into the liquid pipe 27 via the heat generation unit refrigerant flow adjustment valve 45 and returns to the outdoor unit 10.

[0142] The air-conditioning refrigerant that has returned to the outdoor unit 10 is subjected to pressure reduction by the outdoor refrigerant flow adjustment valve 18 and thereby turns into a low-temperature, low-pressure gas-liquid two-phase state, and then enters the outdoor heat exchanger 16 and is heated by air around the outdoor unit 10 and thereby evaporated. The evaporated and gasified air-conditioning refrigerant returns to the air-conditioning compressor 11 via the outdoor suction pipe opening/closing valve 20 and the accumulator 12.

[0143] Meanwhile, the carbon dioxide heated by the

air-conditioning refrigerant in the cascade heat exchanger 44 is gasified and enters the hot water-supply compressor 41. The hot water-supply refrigerant compressed so as to have a high temperature and a high pressure by the hot water-supply compressor 41 enters the hot water-supply heat exchanger 42 and heats the heating medium up to 70 to 90°C. Through this step, the hot water-supply refrigerant is cooled and liquefied, and is subjected to pressure reduction by the hot water-supply refrigerant flow adjustment valve 43 and then returns to the cascade heat exchanger 44 again.

[0144] In operation for both air cooling and air heating, if a cooling load and a heating load are substantially equal to each other, in the outdoor unit 10, the outdoor gas pipe opening/closing valve 19 and the outdoor suction pipe opening/closing valve 20 are both set to be closed. In an indoor machine 30 that performs air cooling, the indoor gas pipe opening/closing valve 34 is set to be closed and the indoor suction pipe opening/closing valve 35 is set to be opened, in an indoor machine 30 that performs the air heating, the indoor gas pipe opening/closing valve 34 is set to be opened and the indoor suction pipe opening/closing valve 35 is set to be closed. Also, in the heat generation unit 40, the heat generation unit refrigerant flow adjustment valve 45 is set to be fully closed.

[0145] The high-temperature, high-pressure air-conditioning refrigerant compressed by the air-conditioning compressor 11 flows into the gas pipe 25 and reaches the indoor machine 30 that performs air heating. The air-conditioning refrigerant that has reached the indoor machine 30 that performs air heating flows into the indoor heat exchanger 31 via the indoor gas pipe opening/closing valve 34, and releases heat to indoor air to heat the air. Through this step, the air-conditioning refrigerant is condensed and thereby liquefied and flows into the liquid pipe 27 via the fully-opened indoor refrigerant flow adjustment valve 33.

[0146] The liquid-state air-conditioning refrigerant that has flown into the liquid pipe 27 reaches the indoor machine 30 that performs air cooling. The air-conditioning refrigerant that has reached the indoor machine 30 that performs air cooling is subjected to pressure reduction by the indoor refrigerant flow adjustment valve 33 and thereby turns into a low-temperature, low-pressure gas-liquid two-phase state, and then flows into the indoor heat exchanger 31 and takes heat from indoor air to cool the air. Through this step, the air-conditioning refrigerant evaporates, enters the suction pipe 26 via the indoor suction pipe opening/closing valve 35, and returns to the outdoor unit 10. The air-conditioning refrigerant that has returned to the outdoor unit 10 returns to the air-conditioning compressor 11 via the accumulator 12.

[0147] If the cooling load is larger than the heating load, the liquid refrigerant supplied to the indoor machine 30 that performs air cooling from the indoor machine 30 that performs air heating is insufficient, and thus, the insufficiency is compensated by the liquid refrigerant produced in the outdoor heat exchanger 16 in the outdoor unit 10.

In other words, with the outdoor suction pipe opening/closing valve 20 kept closed, the outdoor gas pipe opening/closing valve 19 is opened, and a part of the refrigerant discharged by the air-conditioning compressor 11 is supplied to the outdoor heat exchanger 16 and is liquefied, and is supplied to the indoor machine 30 that performs air cooling, via the outdoor refrigerant flow adjustment valve 18 and the liquid pipe 27.

[0148] Conversely, if the heating load is larger than the cooling load, the liquid refrigerant supplied from the indoor machine 30 that performs air heating cannot be fully evaporated in the indoor machine 30 that performs air cooling, and thus, a part of the liquid refrigerant is evaporated in the outdoor heat exchanger 16 in the outdoor unit 10. In other words, with the outdoor gas pipe opening/closing valve 19 kept closed, the outdoor suction pipe opening/closing valve 20 is opened to return the liquid refrigerant flowing out from the indoor machine 30 that performs air heating, to the outdoor unit 10 via the liquid pipe 27.

[0149] The liquid refrigerant that has returned the outdoor unit 10 is subjected to pressure reduction by the outdoor refrigerant flow adjustment valve 18, and then evaporates in the outdoor heat exchanger 16. The gasified air-conditioning refrigerant returns to the accumulator 12 and the air-conditioning compressor 11 via the outdoor suction pipe opening/closing valve 20.

[0150] In operation for both air cooling and hot water supply, if the cooling load and a hot water supply load are substantially equal to each other, in the outdoor unit 10, the outdoor gas pipe opening/closing valve 19 and the outdoor suction pipe opening/closing valve 20 are both set to be closed. In an indoor machine 30 that performs cooling, the indoor gas pipe opening/closing valve 34 is set to be closed and the indoor suction pipe opening/closing valve 35 is set to be opened, and in the heat generation unit 40, the heat generation unit refrigerant flow adjustment valve 45 is opened.

[0151] The high-temperature, high-pressure air-conditioning refrigerant compressed by the air-conditioning compressor 11 flows into the gas pipe 25 and reaches the heat generation unit 40. Meanwhile, in the heat generation unit 40, the hot water-supply compressor 41 operates, whereby the hot water-supply refrigerant circulates through the hot water-supply compressor 41, the hot water-supply heat exchanger 42, the hot water-supply refrigerant flow adjustment valve 43 and the cascade heat exchanger 44 in this order.

[0152] The air-conditioning refrigerant that has reached the heat generation unit 40 heats the carbon dioxide in the cascade heat exchanger 44 and the air-conditioning refrigerant itself is thereby cooled and liquefied, and then flows into the liquid pipe 27 via the heat generation unit refrigerant flow adjustment valve 45.

[0153] The liquid-state air-conditioning refrigerant that has flown into the liquid pipe 27 reaches the indoor machine 30 that performs air cooling. The air-conditioning refrigerant that has reached the indoor machine 30 that

performs air cooling is subjected to pressure reduction by the indoor refrigerant flow adjustment valve 33 and thereby turns into a low-temperature, low-pressure gas-liquid two-phase state, and then enters the indoor heat exchanger 31, and takes heat from indoor air to cool the air. Through this step, the air-conditioning refrigerant evaporates and enters the suction pipe 26 via the indoor suction pipe opening/closing valve 35, and returns to the outdoor unit 10. The air-conditioning refrigerant that has returned to the outdoor unit 10 returns to the air-conditioning compressor 11 via the accumulator 12.

[0154] Meanwhile, the hot water-supply refrigerant heated by the air-conditioning refrigerant in the cascade heat exchanger 44 is gasified and enters the hot water-supply compressor 41. The hot water-supply refrigerant compressed so as to have a high temperature and a high pressure by the hot water-supply compressor 41 enters the hot water-supply heat exchanger 42 and heats the heating medium up to 70 to 90°C. Through this step, the hot water-supply refrigerant is cooled and thereby liquefied, is subjected to pressure reduction by the hot water-supply refrigerant flow adjustment valve 43 and then returns to the cascade heat exchanger 44 again.

[0155] If the cooling load is larger than the hot water supply load, the liquid refrigerant supplied to the indoor machine 30 that performs air cooling from the heat generation unit 40 is insufficient, and thus, the insufficient is compensated by the liquid refrigerant produced in the outdoor heat exchanger 16 in the outdoor unit 10. In other words, with the outdoor suction pipe opening/closing valve 20 kept closed, the outdoor gas pipe opening/closing valve 19 is opened, whereby a part of the refrigerant discharged by the air-conditioning compressor 11 is supplied to the outdoor heat exchanger 16 and is liquefied, and is supplied to the indoor machine 30 that performs air cooling, via the outdoor refrigerant flow adjustment valve 18 and the liquid pipe 27.

[0156] On the other hand, if the hot water supply load is larger than the cooling load, the liquid refrigerant supplied to the heat generation unit 40 cannot be fully evaporated in the indoor machine 30 that performs air cooling, and thus, a part of the liquid refrigerant is evaporated in the outdoor heat exchanger 16 in the outdoor unit 10. In other words, with the outdoor gas pipe opening/closing valve 19 kept closed, the outdoor suction pipe opening/closing valve 20 is opened, and the part of the liquid refrigerant flown out from the indoor machine 30 that performs air heating is returned to the outdoor unit 10 via the liquid pipe 27.

[0157] The liquid refrigerant that has returned to the outdoor unit 10 is subjected to pressure reduction by the outdoor refrigerant flow adjustment valve 18 and then evaporates in the outdoor heat exchanger 16. The gasified air-conditioning refrigerant returns to the accumulator 12 and the air-conditioning compressor 11 via the outdoor suction pipe opening/closing valve 20.

[0158] In operation for both air heating and hot water supply, in the outdoor unit 10, the outdoor gas pipe open-

ing/closing valve 19 is set to be closed and the outdoor suction pipe opening/closing valve 20 is set to be opened, in each of the indoor machines 30, the indoor gas pipe opening/closing valve 34 is set to be opened and the indoor suction pipe opening/closing valve 35 is set to be closed, and in the heat generation unit 40, the heat generation unit refrigerant flow adjustment valve 45 is opened.

[0159] The high-temperature, high-pressure air-conditioning refrigerant compressed by the air-conditioning compressor 11 flows into the gas pipe 25 and reaches the respective indoor machines 30 and the heat generation unit 40. The air-conditioning refrigerant that has reached each indoor machine 30 flows into the indoor heat exchanger 31 via the indoor gas pipe opening/closing valve 34, and releases heat to indoor air to heat the air. Through this step, the air-conditioning refrigerant is condensed and thereby liquefied and flows into the liquid pipe 27 via the fully-opened indoor refrigerant flow adjustment valve 33.

[0160] The air-conditioning refrigerant that has reached the heat generation unit 40 heats the hot water-supply refrigerant in the cascade heat exchanger 44 and the air-conditioning refrigerant itself is thereby cooled and liquefied, and then flows into the liquid pipe 27 via the heat generation unit refrigerant flow adjustment valve 45. The liquid refrigerant merges with the liquid refrigerant that has flown out from the indoor machines 30 that perform air heating and returns to the outdoor unit 10. The liquid refrigerant that has returned to the outdoor unit is subjected to pressure reduction by the outdoor refrigerant flow adjustment valve 18 and then is evaporated in the outdoor heat exchanger 16. The gasified air-conditioning refrigerant returns to the accumulator 12 and the air-conditioning compressor 11 via the outdoor suction pipe opening/closing valve 20.

[0161] Meanwhile, the carbon dioxide heated by the air-conditioning refrigerant in the cascade heat exchanger 44 is gasified and enters the hot water-supply compressor 41. The hot water-supply refrigerant compressed so as to have a high temperature and a high pressure by the hot water-supply compressor 41 enters the hot water-supply heat exchanger 42 and heats the heating medium up to 70 to 90°C. Through this step, the hot water-supply refrigerant is cooled and thereby liquefied, is subjected to pressure reduction by the hot water-supply refrigerant flow adjustment valve 43 and then returns to the cascade heat exchanger 44 again.

[0162] In operation for all air cooling, air heating and hot water supply, if the cooling load and a sum of the heating load and the hot water supply load are substantially equal to each other, in the outdoor unit 10, the outdoor gas pipe opening/closing valve 19 and the outdoor suction pipe opening/closing valve 20 are both set to be closed. In an indoor machine 30 that perform air cooling, the indoor gas pipe opening/closing valve 34 is set to be closed and the indoor suction pipe opening/closing valve 35 is set to be opened, and in an indoor machine 30 that

performs air heating, the indoor gas pipe opening/closing valve 34 is set to be opened and the indoor suction pipe opening/closing valve 35 is set to be closed. Also, in the heat generation unit 40, the heat generation unit refrigerant flow adjustment valve 45 is opened.

[0163] The high-temperature, high-pressure air-conditioning refrigerant compressed by the air-conditioning compressor 11 flows into the gas pipe 25 and reaches the indoor machine 30 that performs air heating and the heat generation unit 40. Meanwhile, in the heat generation unit 40, the hot water-supply compressor 41 operates, whereby the hot water-supply refrigerant circulates through the hot water-supply compressor 41, the hot water-supply heat exchanger 42, the hot water-supply refrigerant flow adjustment valve 43 and the cascade heat exchanger 44 in this order.

[0164] The air-conditioning refrigerant that has reached the indoor machine 30 that performs air heating flows into the indoor heat exchanger 31 via the indoor gas pipe opening/closing valve 34 and releases heat to indoor air to heat the air. Through this step, the air-conditioning refrigerant is condensed and thereby liquefied and flows into the liquid pipe 27 via the fully-opened indoor refrigerant flow adjustment valve 33.

[0165] The air-conditioning refrigerant that has reached the heat generation unit 40 heats the carbon dioxide in the cascade heat exchanger 44 and the air-conditioning refrigerant itself is thereby cooled and liquefied and then flows into the liquid pipe 27 via the heat generation unit refrigerant flow adjustment valve 45.

[0166] The liquefied air-conditioning refrigerant flowing into the liquid pipe 27 from the indoor machine 30 that performs air heating and the liquefied air-conditioning refrigerant flowing into the liquid pipe 27 from the heat generation unit 40 merge and reach the indoor machine 30 that performs air cooling. The air-conditioning refrigerant that has reached the indoor machine 30 that performs air cooling is subjected to pressure reduction by the indoor refrigerant flow adjustment valve 33 and thereby turns into a low-temperature, low-pressure gas-liquid two-phase state, and then flows into the indoor heat exchanger 31 and takes heat from indoor air to cool the air. Through this step, the air-conditioning refrigerant evaporates, and enters the suction pipe 26 via the indoor suction pipe opening/closing valve 35 and returns to the outdoor unit 10. The air-conditioning refrigerant that has returned to the outdoor unit 10 returns to the air-conditioning compressor 11 via the accumulator 12.

[0167] Meanwhile, the carbon dioxide heated by the air-conditioning refrigerant in the cascade heat exchanger 44 is gasified and enters the hot water-supply compressor 41. The hot water-supply refrigerant compressed so as to have a high temperature and a high pressure by the hot water-supply compressor 41 enters the hot water-supply heat exchanger 42 and heats the heating medium up to 70 to 90°C. Through this step, the hot water-supply refrigerant is cooled and thereby liquefied, and is subjected to pressure reduction by the hot water-supply re-

frigerant flow adjustment valve 43 and then returns to the cascade heat exchanger 44 again.

[0168] Here, if the cooling load is larger than the sum of the heating load and hot water supply load, the liquid refrigerant supplied to the indoor machine 30 from the indoor machine 30 that performs air heating and the heat generation unit 40 is insufficient, and thus, the insufficiency is compensated by the liquid refrigerant produced in the outdoor heat exchanger 16 in the outdoor unit 10. In other words, with the outdoor suction pipe opening/closing valve 20 kept closed, the outdoor gas pipe opening/closing valve 19 is opened, and a part of the refrigerant discharged by the air-conditioning compressor 11 is supplied to the outdoor heat exchanger 16 and is liquefied, and is supplied to the indoor machine 30 that performs air cooling via the outdoor refrigerant flow adjustment valve 18 and the liquid pipe 27.

[0169] On the other hand, if the sum of the heating load and the hot water supply load is larger than the cooling load, the liquid refrigerant supplied from the indoor machine 30 that performs air heating and the heat generation unit 40 cannot be fully evaporated in the indoor machine 30 that performs air cooling, and thus, a part of the liquid refrigerant is evaporated in the outdoor heat exchanger 16 in the outdoor unit 10. In other words, with the outdoor gas pipe opening/closing valve 19 kept closed, the outdoor suction pipe opening/closing valve 20 is opened, and the part of the liquid refrigerant flown out from the indoor machine 30 that performs air heating and the heat generation unit 40 returns to the outdoor unit 10 via the liquid pipe 27.

[0170] The liquid refrigerant that has returned to the outdoor unit 10 is subjected to pressure reduction by the outdoor refrigerant flow adjustment valve 18 and then evaporates in the outdoor heat exchanger 16. The gasified air-conditioning refrigerant returns to the accumulator 12 and the air-conditioning compressor 11 via the outdoor suction pipe opening/closing valve 20.

[0171] Next, operation of the heating medium in the heat generation unit 40 will be described with reference to FIGS. 2 and 3.

[0172] In operation for hot water supply only, operation for both air cooling and hot water supply, operation for air heating and hot water supply and operation for all air cooling, air heating and hot water supply, the hot water-supply compressor 41 and the heating medium pump 46 operate. During the heating medium pump operating, the heating medium flows into the heat generation unit 40 from the outside of the heat generation unit 40 such as a water supply, and enters the heating medium pump 46 through the heating medium piping 63.

[0173] The heating medium that has flown into the heating medium pump 46 flows into the heating medium piping 64 from a discharge port and enters the hot water-supply heat exchanger 42. The heating medium exchanges heat with the high-temperature carbon dioxide discharged by the hot water-supply compressor 41 in the hot water-supply heat exchanger 42, which is a double

pipe-type heat exchanger, and is thereby heated up to 70 to 90°C, and then sent to the outside of the heat generation unit 40 through the heating medium piping 65.

[0174] As described above, in a passage in which the heating medium flows (heating medium piping 63 → heating medium pump 46 → heating medium piping 64 → hot water-supply heat exchanger 42 → heating medium piping 65), there are both resin materials and copper and thus there are parts of connection between different materials.

[0175] In the present embodiment, the hot water-supply heat exchanger 42 is installed so as to be not in contact with the bottom plate member 51 to which the hot water-supply compressor 41 fixes and be fixed to the side plate member 52, and thus, vibration during operation of the hot water-supply compressor 41 is not transmitted directly to the hot water-supply heat exchanger 42 through the bottom plate member 51.

[0176] As described above, in the present embodiment, as a result of the volume ratio V_e/V_c between the low pressure-side circuit internal volume V_e and the high pressure-side circuit internal volume V_c in the first refrigeration cycle 100 being set as $V_e/V_c = 0.2$ to 10.1 , under any condensation temperature condition where the condensation temperature in the second refrigeration cycle 200 varies from 30°C to 55°C, the discharge temperature T_d in the hot water-supply compressor 41 in the first refrigeration cycle 100 falls within a range of 100 to 110°C, which is equal to or below the use upper limit discharge temperature for the hot water-supply compressor 41.

[0177] As a result, irrespective of the condensation temperature T_{vc} in the refrigeration cycle in which the air-conditioning refrigerant circulates, operation is performed in such a manner that the discharge temperature T_d in the hot water-supply compressor 41 in the first refrigeration cycle 100 falls within a range of no more than the use upper limit discharge temperature for the hot water-supply compressor 41, which prevents the durability of the hot water-supply compressor 41 from being impaired, enabling enhancement in reliability of the first refrigeration cycle 100.

[0178] Although an embodiment of the present invention has been described with reference to the drawings, the present invention is not limited to the above-described embodiment, and changes are possible without departing from the spirit of the invention.

Industrial Applicability

[0179] The present invention can favorably be used as one providing a highly-reliable refrigeration cycle in which carbon dioxide that does not impair a durability of a hot water-supply compressor 41 circulates, in an air-conditioning/hot water-supply system that can simultaneously supply hot and cold heat necessary for air cooling, air heating and hot water supply.

Reference Signs List

[0180]

- | | |
|----|--|
| 5 | 10 outdoor unit |
| | 11 air-conditioning compressor |
| | 16 outdoor heat exchanger |
| | 30 indoor machine |
| | 31 indoor heat exchanger |
| 10 | 40 heat generation unit |
| | 41 hot water-supply compressor |
| | 42 hot water-supply heat exchanger |
| | 43 hot water-supply refrigerant flow adjustment valve |
| 15 | 44 cascade heat exchanger |
| | 45 heat generation unit refrigerant flow adjustment valve |
| | 46 heating medium pump |
| | 100 first refrigeration cycle |
| 20 | 200 second refrigeration cycle |
| | 201 first circuit |
| | 202 second circuit |
| | 500 refrigerant state transition in first refrigeration cycle when condensation temperature T_{vc} of second refrigeration cycle is 30°C and temperature T_d of discharged refrigerant from hot water-supply compressor is 100°C |
| 25 | 510 refrigerant state transition in first refrigeration cycle when condensation temperature T_{vc} of second refrigeration cycle is 55°C and temperature T_d of discharged refrigerant from hot water-supply compressor is 100°C |
| 30 | 520 refrigerant state transition in first refrigeration cycle when condensation temperature T_{vc} of second refrigeration cycle is 55°C and temperature of discharged refrigerant from hot water-supply compressor is 110°C |
| 35 | 530 refrigerant state transition in first refrigeration cycle when condensation temperature T_{vc} of second refrigeration cycle is 30°C and temperature of discharged refrigerant from hot water-supply compressor is 110°C |
| 40 | 540 isotherm of carbon dioxide at 100°C |
| | 550 isotherm of carbon dioxide at 110°C |
| 45 | 560 saturation line of carbon dioxide |

Claims

- | | |
|----|---|
| 50 | 1. An air-conditioning/hot water-supply system comprising: |
| | a first refrigeration cycle (100) including a hot water-supply compressor (41) that compresses a hot water-supply refrigerant, a hot water-supply heat exchanger (42) that allows heat exchange between the hot water-supply refrigerant and a hot water-supply heating medium, and |
| 55 | |

a cascade heat exchanger (44) that allows heat exchange between the hot water-supply refrigerant and an air-conditioning refrigerant; and a second refrigeration cycle (200) including a first circuit (201) in which the cascade heat exchanger and a second refrigerant flow control device (45) that controls a flow of the air-conditioning refrigerant to be supplied to the cascade heat exchanger are serially connected and at least one second circuit (202) in which an indoor heat exchanger (31) that allows heat exchange between the air-conditioning refrigerant and indoor air and a third refrigerant flow control device (33) that controls a flow of the air-conditioning refrigerant to be supplied to the indoor heat exchanger are serially connected, the second refrigeration cycle connecting a thermal load circuit in which the first circuit and the second circuit are parallelly connected, to an air-conditioning compressor (11) that compresses the air-conditioning refrigerant and an outdoor heat exchanger (16), **characterized in that** a volume ratio between a low pressure-side circuit internal volume V_e and a high pressure-side circuit internal volume V_c in the first refrigeration cycle is set as $V_e/V_c = 0.2$ to 10.1 .

30

35

40

45

50

55

FIG. 1

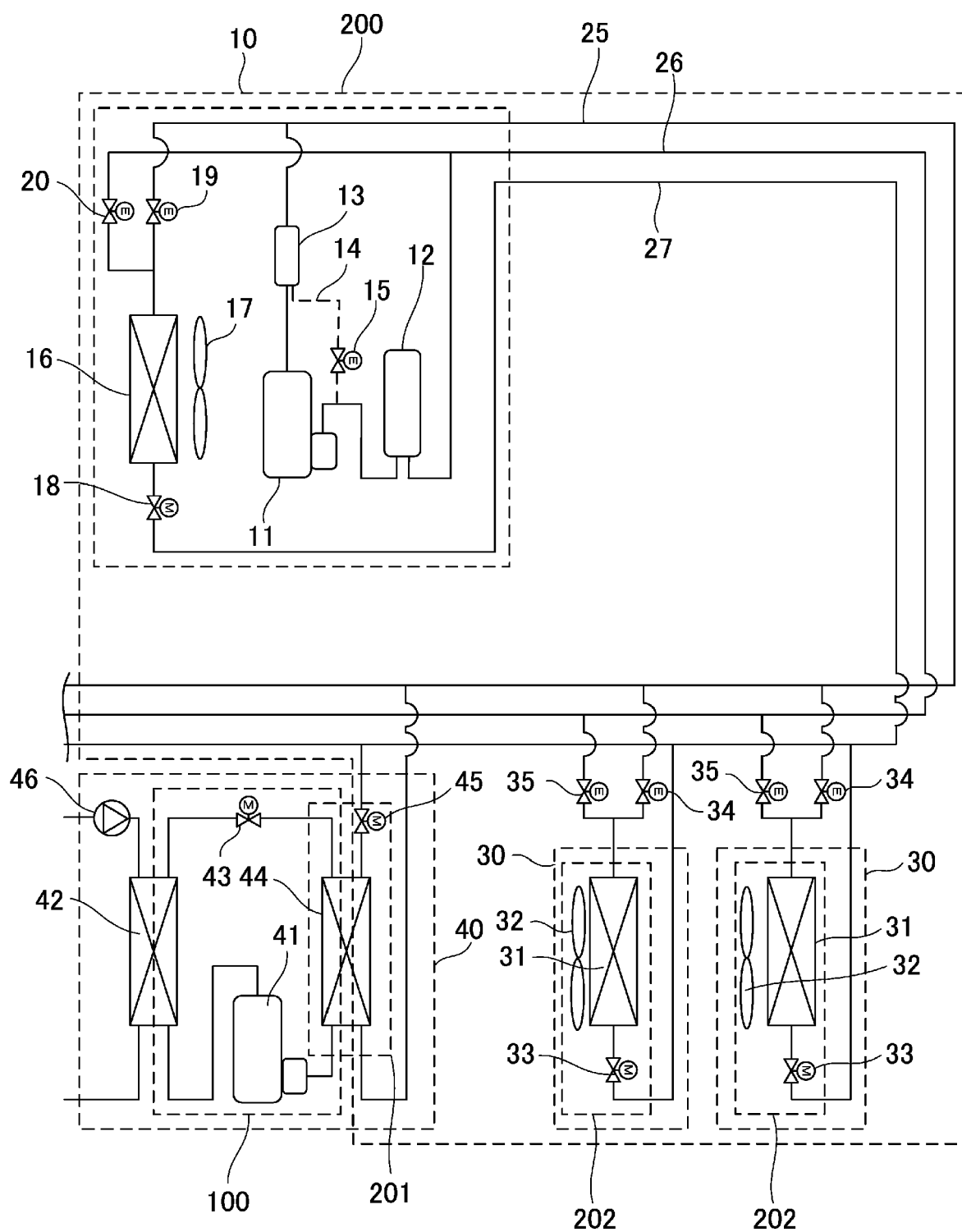


FIG.2

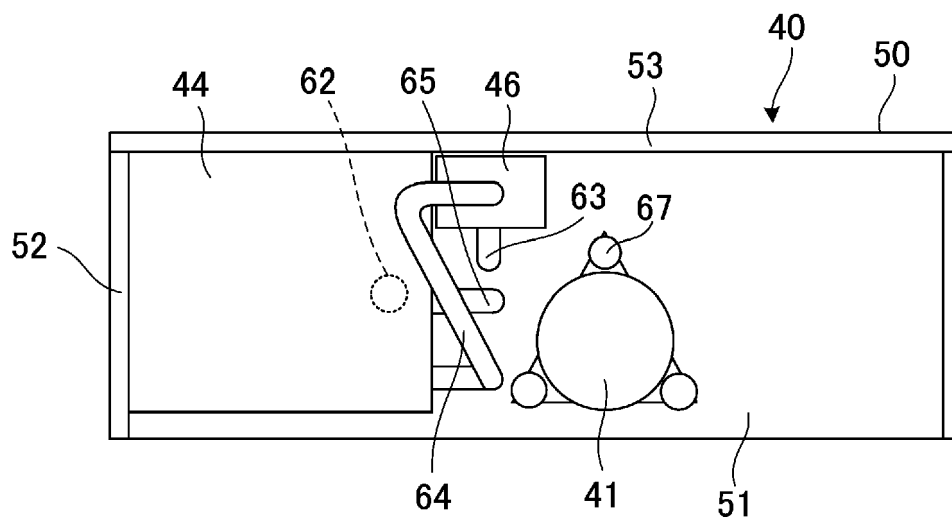


FIG.3

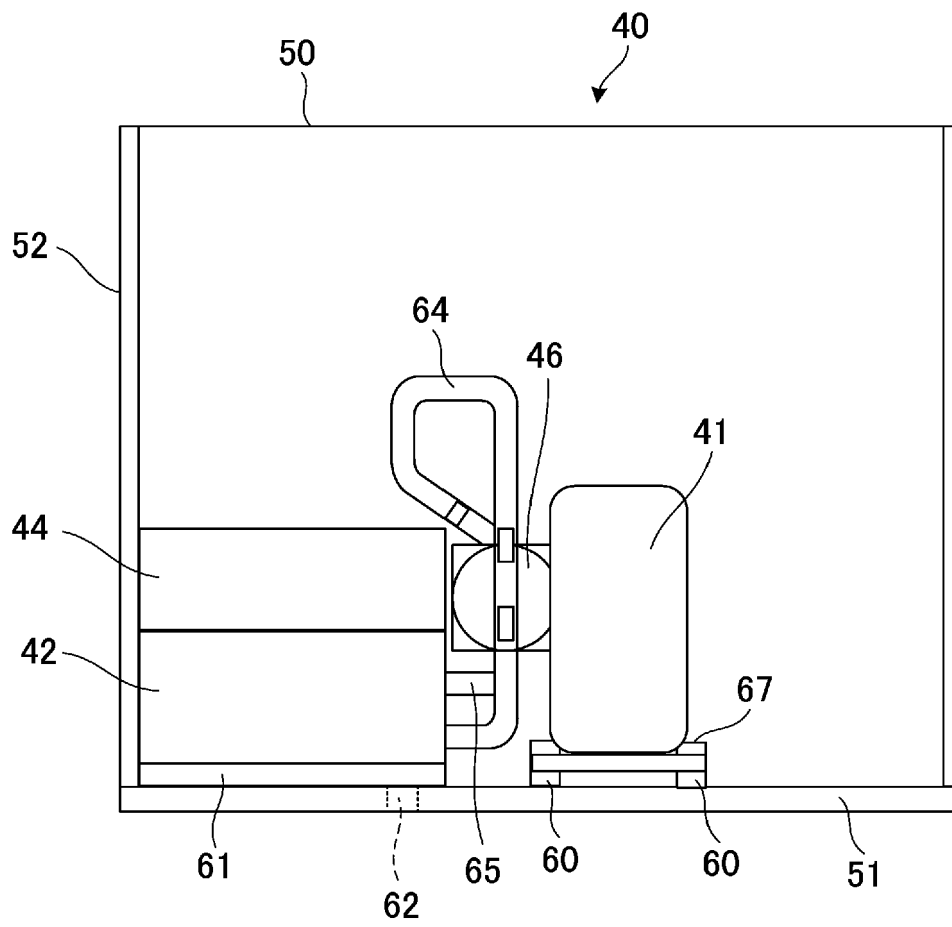


FIG.4

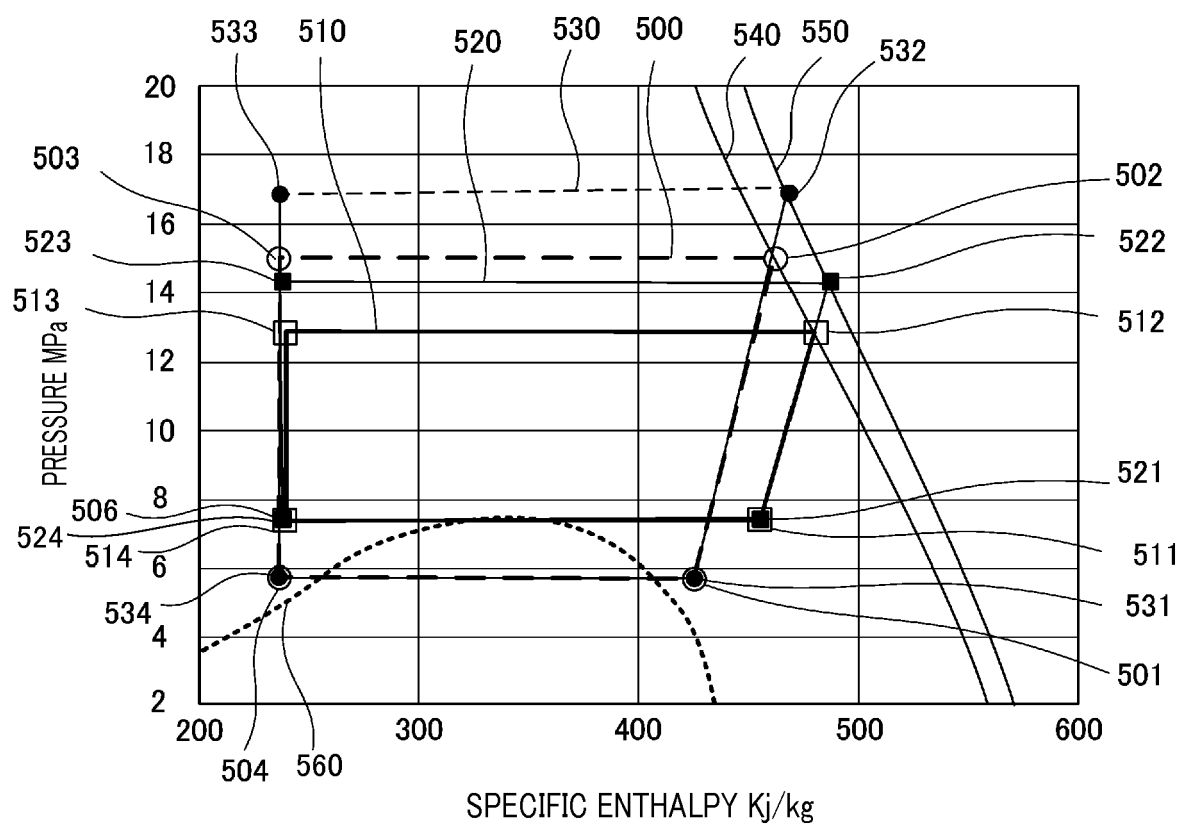


FIG. 5

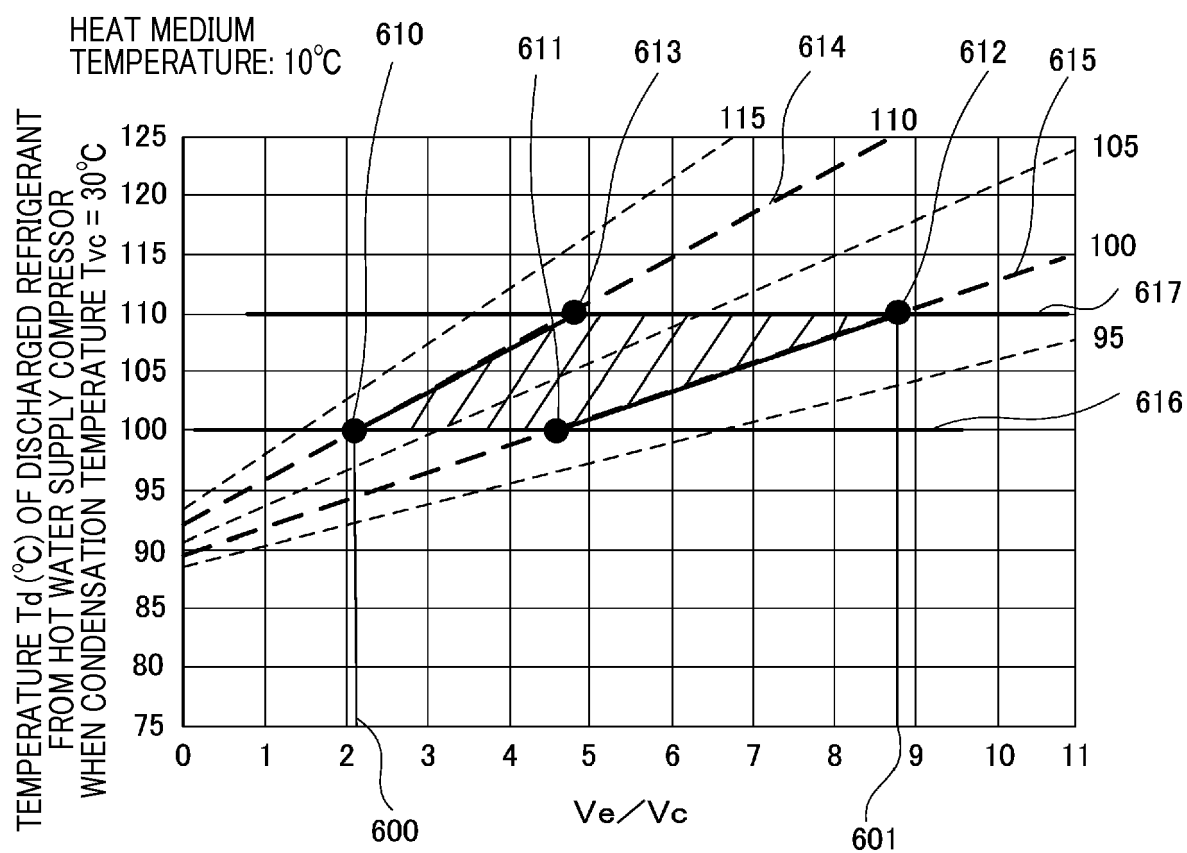


FIG.6

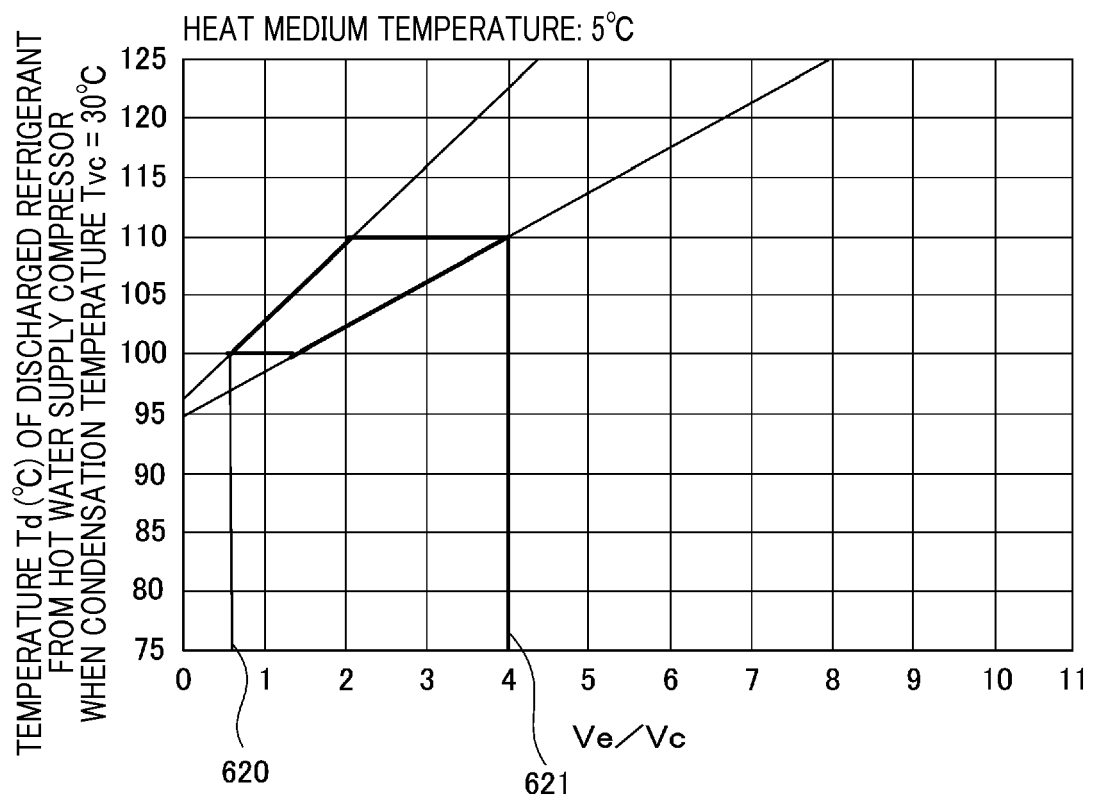


FIG.7

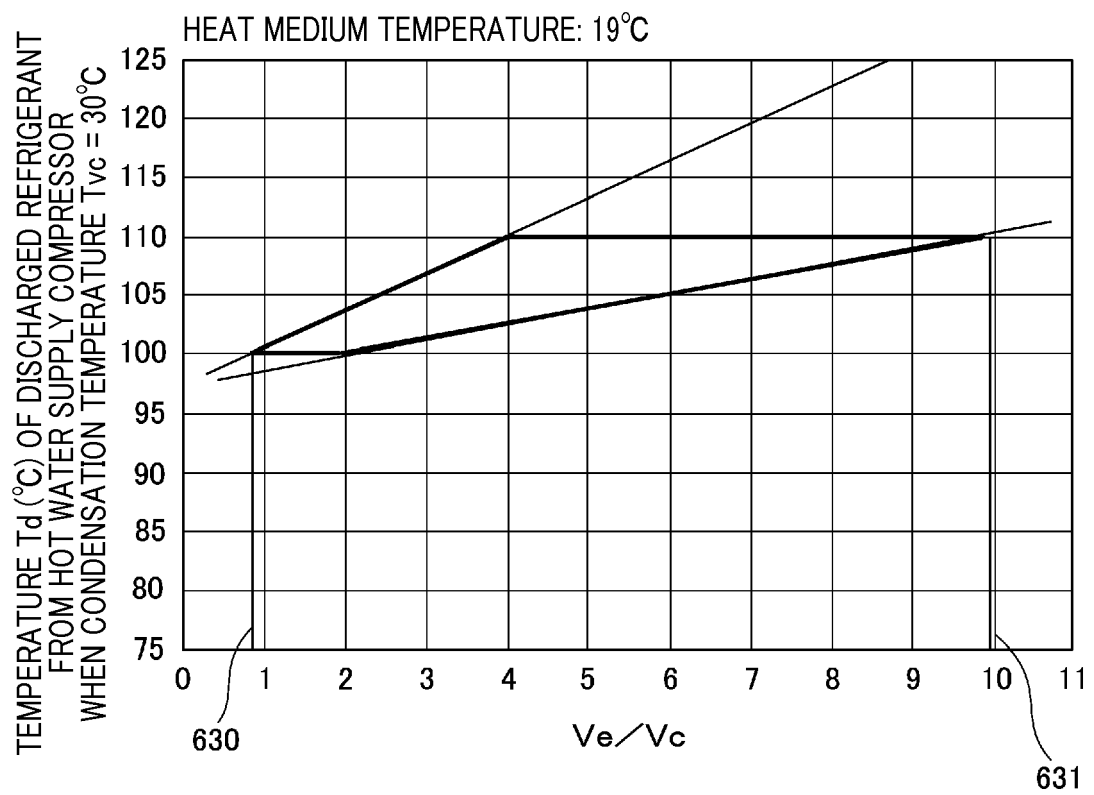


FIG.8

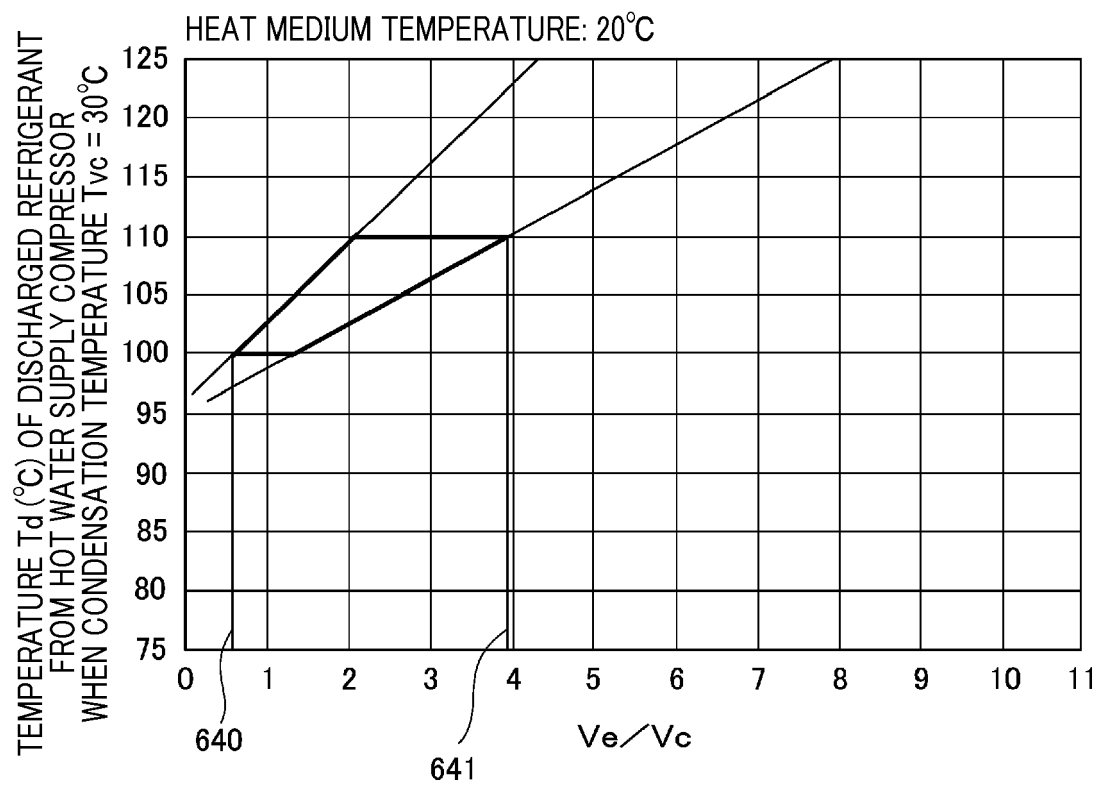
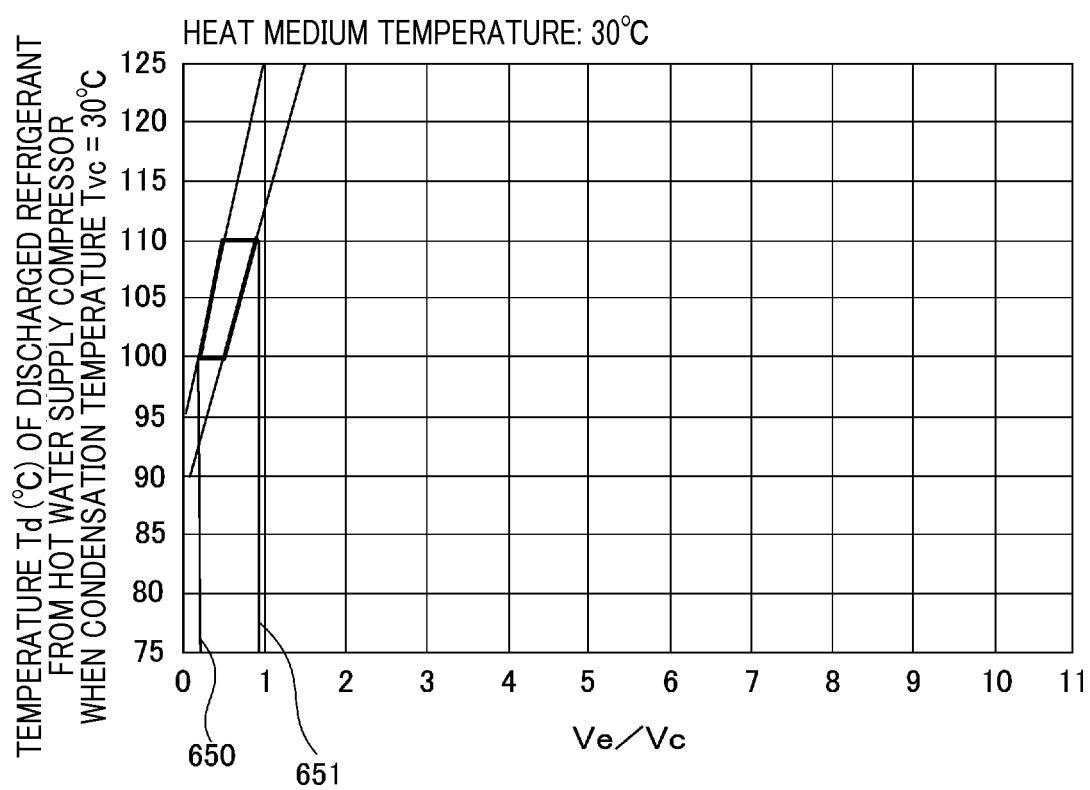


FIG.9





EUROPEAN SEARCH REPORT

Application Number
EP 17 15 7736

5

10

15

20

25

30

35

40

45

50

55

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
Y	US 2011/314848 A1 (TANAKA KOSUKE [JP] ET AL) 29 December 2011 (2011-12-29) * the whole document *	1	INV. F25B49/02 F25B7/00 F25B25/00
Y,D	JP 2004 132647 A (DAIKIN IND LTD) 30 April 2004 (2004-04-30) * abstract; figures 1-3 *	1	
Y	US 2010/300135 A1 (OTAKE MASAHISA [JP] ET AL) 2 December 2010 (2010-12-02) * the whole document *	1	
Y	DE 10 2014 214656 A1 (KONVEKTA AG [DE]) 28 January 2016 (2016-01-28) * paragraph [0027] - paragraph [0052]; figure 1 *	1	
Y	US 2005/178151 A1 (MATSUMOTO KENZO [JP] ET AL) 18 August 2005 (2005-08-18) * paragraph [0098] - paragraph [0100]; figures 4-6 *	1	
Y	JP 2005 226927 A (SANYO ELECTRIC CO) 25 August 2005 (2005-08-25) * paragraph [0052]; figure 2 *	1	TECHNICAL FIELDS SEARCHED (IPC) F25B
The present search report has been drawn up for all claims			
Place of search Munich		Date of completion of the search 18 July 2017	Examiner Lucic, Anita
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

EPO FORM 1503 03.82 (P04C01)

**ANNEX TO THE EUROPEAN SEARCH REPORT
ON EUROPEAN PATENT APPLICATION NO.**

EP 17 15 7736

5

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

18-07-2017

10

15

20

25

30

35

40

45

50

55

Patent document cited in search report		Publication date		Patent family member(s)	Publication date
US 2011314848	A1	29-12-2011	CN	102365510 A	29-02-2012
			EP	2416093 A1	08-02-2012
			JP	5042262 B2	03-10-2012
			JP	2010236817 A	21-10-2010
			US	2011314848 A1	29-12-2011
			WO	2010113372 A1	07-10-2010

JP 2004132647	A	30-04-2004	JP	3925383 B2	06-06-2007
			JP	2004132647 A	30-04-2004

US 2010300135	A1	02-12-2010	CN	101900455 A	01-12-2010
			EP	2261570 A2	15-12-2010
			JP	5585003 B2	10-09-2014
			JP	2010276230 A	09-12-2010
			US	2010300135 A1	02-12-2010

DE 102014214656	A1	28-01-2016	CN	105299971 A	03-02-2016
			DE	102014214656 A1	28-01-2016
			JP	2016035377 A	17-03-2016

US 2005178151	A1	18-08-2005	BR	PI0500371 A	27-09-2005
			CN	1654902 A	17-08-2005
			EP	1564507 A2	17-08-2005
			KR	20060041722 A	12-05-2006
			MX	PA05001650 A	16-08-2005
			TW	I324242 B	01-05-2010
			US	2005178151 A1	18-08-2005

JP 2005226927	A	25-08-2005	NONE		

REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- JP 2004132647 A [0004] [0024]
- JP 3925383 B [0004] [0024]