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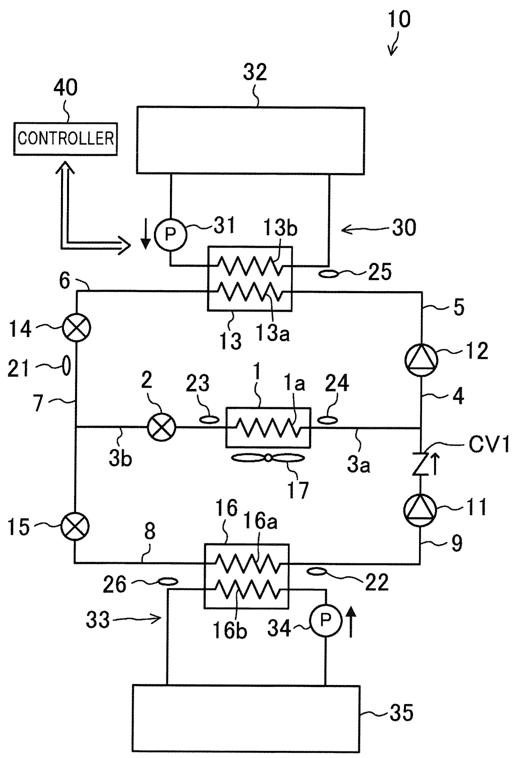
(54) **HEAT PUMP**

(57) The present invention is intended to prevent a reduction in efficiency of a heat pump even if an auxiliary heat exchanger is used to control heat balance of a refrigerant circuit. A refrigerant circuit (10) is provided with an auxiliary heat exchanger (1) which exchanges heat between the refrigerant in the refrigerant circuit (10) and outdoor air. The auxiliary heat exchanger (1) is connected so as to communicate between a connection path (4) between a low-stage compressor (11) and a high-stage compressor (12) and a connection path (7) between a low-stage expansion valve (15) and a high-stage expansion valve (14).

FIG.1



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Description

TECHNICAL FIELD

[0001] The present invention relates to heat pumps, and specifically relates to heat pumps which include a refrigerant circuit capable of providing cooling and heating at the same time.

BACKGROUND ART

[0002] Heat pumps capable of processing a heating load and a cooling load at the same time have been known. These heat pumps include a heat pump having a refrigerant circuit to which a heating heat exchanger that processes the heating load, a cooling heat exchanger that processes the cooling load, and also an auxiliary heat exchanger are connected (see Patent Document 1).

[0003] The auxiliary heat exchanger is for controlling heat balance of the refrigerant circuit to avoid thermal imbalance of the refrigerant circuit depending on conditions of the heating load and the cooling load. In the case where the heating load becomes larger than the cooling load and the refrigerant circuit dissipates excessive heat, the auxiliary heat exchanger serves as an evaporator to increase a heat absorption rate, thereby maintaining the heat balance. That is, a refrigerant condensed by the heating heat exchanger is evaporated by both of the cooling heat exchanger and the auxiliary heat exchanger.

[0004] On the other hand, in the case where the heating load becomes smaller than the cooling load and the refrigerant circuit absorbs excessive heat, the auxiliary heat exchanger serves as a condenser to increase a heat dissipation rate, thereby maintaining the heat balance. That is, the refrigerant evaporated in the cooling heat exchanger is condensed by both of the heating heat exchanger and the auxiliary heat exchanger.

CITATION LIST

[0005] Patent Document 1: Japanese Unexamined Patent Publication No. 2001-349639

SUMMARY OF THE INVENTION

TECHNICAL PROBLEM

[0006] In the auxiliary heat exchanger, outdoor air and the refrigerant are heat exchanged. Thus, the amount of heat exchanged by the auxiliary heat exchanger is not used to process the heating load and the cooling load, and does not contribute to the performance of the heat pump. However, since part of power of a compressor is wastefully used as power for supplying the refrigerant to the auxiliary heat exchanger, efficiency of the heat pump decreases.

[0007] The present invention is thus intended to operate a heat pump which uses an auxiliary heat exchanger to adjust heat balance of a refrigerant circuit, more efficiently than before.

SOLUTION TO THE PROBLEM

[0008] The first aspect of the present invention is a heat pump including: a refrigerant circuit (10) in which a low-stage compression mechanism (11), a high-stage compression mechanism (12), a high-temperature heat exchanger (13), a high-stage expansion mechanism (14), a low-stage expansion mechanism (15), and a low-temperature heat exchanger (16) are sequentially connected together by a refrigerant path, and in which a refrigerant dissipates heat into a high-temperature fluid in the high-temperature heat exchanger (13) and the refrigerant absorbs heat from a low-temperature fluid and evaporates in the low-temperature heat exchanger (16), thereby performing a refrigeration cycle, and an auxiliary heat exchanger (1) which is connected to communicate between the refrigerant path between the low-stage compression mechanism (11) and the high-stage compression mechanism (12) and the refrigerant path between the low-stage expansion mechanism (15) and the high-stage expansion mechanism (14), and which exchanges heat between the refrigerant of the refrigerant circuit (10) and a heat-source fluid.

[0009] In the case of conventional heat pumps, the refrigerant in the refrigerant circuit (10) circulates through single-stage compression. Thus, the only way to make the auxiliary heat exchanger (1) function as a condenser is through communicating the auxiliary heat exchanger (1) with a high-pressure line (a channel in which a high-pressure refrigerant flows) of the refrigerant circuit (10) (see FIG. 13(A), and the only way to make the auxiliary heat exchanger (1) function as an evaporator is through communicating the auxiliary heat exchanger (1) with a low-pressure line (a channel in which a low-pressure refrigerant flows) of the refrigerant circuit (10).

[0010] In the first aspect of the present invention, the refrigerant circuit (10) is formed of a circuit of two-stage compression and two-stage expansion, and the auxiliary heat exchanger (1) is positioned at an intermediate-pressure line (a channel where an intermediate-pressure refrigerant flows) of the refrigerant circuit (10) (see FIG. 13(B)). Thus, when the auxiliary heat exchanger (1) functions as a condenser, only part of the refrigerant compressed in the low-stage compression mechanism (11) needs to be supplied into the auxiliary heat exchanger (1). The compression power required for the refrigerant circuit (10) is therefore reduced.

[0011] Further, when the auxiliary heat exchanger (1) functions as an evaporator, only part of the refrigerant decompressed in the high-stage expansion mechanism needs to be supplied to the auxiliary heat exchanger (1) and evaporated, and thereafter sucked into the high-stage compression mechanism (12). The compression power required for the refrigerant circuit (10) is therefore reduced.

[0012] The second aspect of the present invention is characterized by including, in the first aspect of the present invention, a compression mechanism control section (41) which controls operational capacity of the high-stage compression mechanism (12) according to a heating load of the high-temperature heat exchanger (13) and controls operational capacity of the low-stage compression mechanism (11) according to a cooling load of the low-temperature heat exchanger (16).

[0013] In the second aspect of the present invention, the operational capacity of the high-stage compression mechanism (12) is increased when the heating load becomes large, and the operational capacity of the high-stage compression mechanism (12) is reduced when the heating load becomes small. The operational capacity of the low-stage compression mechanism (11) is increased when the cooling load becomes large, and the operational capacity of the low-stage compression mechanism (11) is reduced when the cooling load becomes small.

[0014] When the heating load is greater than the cooling load, and the operational capacity of the high-stage compression mechanism (12) becomes larger than the operational capacity of the low-stage compression mechanism (11), the refrigerant evaporated in the auxiliary heat exchanger (1) is sucked into the high-stage compression mechanism (12) together with the refrigerant discharged from the low-stage compression mechanism (11).

[0015] On the other hand, when the cooling load is greater than the heating load, and the operational capacity of the low-stage compression mechanism (11) becomes larger than the operational capacity of the high-stage compression mechanism (12), the amount of the refrigerant discharged from the low-stage compression mechanism (11) becomes larger than the amount of the refrigerant suctioned by the high-stage compression mechanism (12). As a result, the refrigerant discharged from the low-stage compression mechanism (11) and is not sucked into the high-stage compression mechanism (12) flows into the auxiliary heat exchanger (1). The auxiliary heat exchanger (1) is made to function as a condenser, and the refrigerant is condensed in the auxiliary heat exchanger (1).

[0016] As described above, it is possible to make the auxiliary heat exchanger (1) function as an evaporator when the heating load is greater than the cooling load, and possible to make the auxiliary heat exchanger (1) function as a condenser when the cooling load is greater than the heating load.

[0017] The third aspect of the present invention is characterized by including in the second aspect of the present invention, a switching mechanism (51, 52) which, in the case where the heating load is greater than the cooling load, and both of the auxiliary heat exchanger (1) and the low-temperature heat exchanger (16) function as evaporators, switches between a low-stage suction state in which the refrigerant flowing out of the auxiliary heat exchanger (1) is guided to a suction side of the low-stage compression mechanism (11) when a pressure difference between an evaporating pressure of the auxiliary heat exchanger (1) and an evaporating pressure of the low-temperature heat exchanger (16) is smaller than a predetermined value, or when the evaporating pressure of the auxiliary heat exchanger (1) is lower than or equal to the evaporating pressure of the low-temperature heat exchanger (16), and a high-stage suction state in which the refrigerant flowing out of the auxiliary heat exchanger (1) is guided to a suction side of the high-stage compression mechanism (12), when the pressure difference is larger than or equal to the predetermined value and the evaporating pressure of the auxiliary heat exchanger (1) is higher than the evaporating pressure of the low-temperature heat exchanger (16).

[0018] The smaller the pressure difference between the evaporating pressure of the auxiliary heat exchanger (1) and the evaporating pressure of the low-temperature heat exchanger (16), the closer the suction pressure and the discharge pressure of the low-stage compression mechanism (11) and the smaller the effects of enhancing operational efficiency of the heat pump by two-stage compression. Further, if the evaporating pressure of the auxiliary heat exchanger (1) is lower than the evaporating pressure of the low-temperature heat exchanger (16), a pressure of the refrigerant suctioned by the low-stage compression mechanism (11) and a pressure of the refrigerant discharged from the low-stage compression mechanism (11) are inverted, which causes a malfunction of the low-stage compression mechanism (11). In practice, the pressure of the refrigerant sucked by the low-stage compression mechanism (11) is lowered to continue operation, which, however, is lower than an optimal evaporating pressure of the low-temperature heat exchanger (16) in this case, and therefore, the operational efficiency of the heat pump may be reduced. The "predetermined value" as used herein is within a range of pressure differences within which it is possible to enhance operational efficiency of the heat pump by two-stage compression.

[0019] In the third aspect of the present invention, the switching mechanism (51, 52) is in the low-stage suction state when the pressure difference between the evaporating pressure of the auxiliary heat exchanger (1) and the evaporating pressure of the low-temperature heat exchanger (16) is smaller than a predetermined value, or when the evaporating pressure of the auxiliary heat exchanger (1) is lower than or equal to the evaporating pressure of the low-temperature heat exchanger (16). Thus, the refrigerant evaporated in the auxiliary heat exchanger (1) is sucked into the low-stage compression mechanism (11).

[0020] On the other hand, the switching mechanism (51, 52) is in the high-stage suction state, because the heat pump can operate with higher efficiency by sucking the refrigerant evaporated in the auxiliary heat exchanger (1) into the high-stage compression mechanism (12) when the pressure difference between the evaporating pressure of the auxiliary heat exchanger (1) and the evaporating pressure of the low-temperature heat exchanger (16) is larger than or equal to the predetermined value and the evaporating pressure of the auxiliary heat exchanger (1) is higher than the evaporating pressure of the low-temperature heat exchanger (16).

[0021] The fourth aspect of the present invention is characterized by including in any one of the first to third aspects of the present invention, an economizer pipe (53) separated from a refrigerant pipe between the high-temperature heat exchanger (13) and the high-stage expansion mechanism (14) and connected to a refrigerant pipe between the low-stage compression mechanism (11) and the high-stage compression mechanism (12), a decompression mechanism (54) which decompresses the refrigerant in the economizer pipe (53), and an economizer heat exchanger (55) which exchanges heat between the refrigerant in the economizer pipe (53) decompressed by the decompression mechanism (54) and a high-pressure refrigerant flowing from the high-temperature heat exchanger (13) to the high-stage expansion mechanism (14).

[0022] In the fourth aspect of the present invention, the degree of subcooling of the refrigerant flowing from the high-temperature heat exchanger (13) to the high-stage expansion mechanism (14) can be greater than in the case where the economizer heat exchanger (55) is not provided. The heat pump can therefore operate with high efficiency.

[0023] The fifth aspect of the present invention is characterized by including in the first aspect of the present invention, a low-stage bypass path (18) which bypasses the low-stage compression mechanism (11), and a compression mechanism control section (41) which, in the case where the heating load is greater than the cooling load, controls operations of the low-stage compression mechanism (11) and the high-stage compression mechanism (12) by switching at least between a high-stage-only compression operation and a two-stage compression operation, wherein the high-stage-only compression operation is an operation in which operational capacity of the high-stage compression mechanism (12) is controlled according to a heating load of the high-temperature heat exchanger (13), and the low-stage compression mechanism (11) is stopped, when a pressure difference between an evaporating pressure of the auxiliary heat exchanger (1) and an evaporating pressure of the low-temperature heat exchanger (16) is smaller than a predetermined value or when the evaporating pressure of the auxiliary heat exchanger (1) is lower than or equal to the evaporating pressure of the low-temperature heat exchanger (16), and the two-stage compression operation is an operation in which the operational capacity of the high-stage compression mechanism (12) is controlled according to the heating load of the high-temperature heat exchanger (13), and operational capacity of the low-stage compression mechanism (11) is controlled according to a cooling load of the low-temperature heat exchanger (16), when the pressure difference is larger than or equal to the predetermined value and the evaporating pressure of the auxiliary heat exchanger (1) is higher than the evaporating pressure of the low-temperature heat exchanger (16).

[0024] The smaller the pressure difference between the evaporating pressure of the auxiliary heat exchanger (1) and the evaporating pressure of the low-temperature heat exchanger (16), the closer the suction pressure and the discharge pressure of the low-stage compression mechanism (11) and the smaller the effects of enhancing operational efficiency of the heat pump by two-stage compression. Further, if the evaporating pressure of the auxiliary heat exchanger (1) is lower than the evaporating pressure of the low-temperature heat exchanger (16), a pressure of the refrigerant suctioned by the low-stage compression mechanism (11) and a pressure of the refrigerant discharged from the low-stage compression mechanism (11) are inverted, which causes a malfunction of the low-stage compression mechanism (11). In practice, the pressure of the refrigerant sucked by the low-stage compression mechanism (11) is lowered to continue operation, which, however, is lower than an optimal evaporating pressure of the low-temperature heat exchanger (16) in this case, and therefore, the operational efficiency of the heat pump may be reduced. The "predetermined value" as used herein is within a range of pressure differences within which it is possible to enhance operational efficiency of the heat pump by two-stage compression.

[0025] In the fifth aspect of the present invention, the low-stage compression mechanism (11) is stopped and only the high-stage compression mechanism (12) is actuated (i.e., the high-stage-only compression operation) when the pressure difference between the evaporating pressure of the auxiliary heat exchanger (1) and the evaporating pressure of the low-temperature heat exchanger (16) is smaller than a predetermined value, or when the evaporating pressure of the auxiliary heat exchanger (1) is lower than or equal to the evaporating pressure of the low-temperature heat exchanger (16). Since the low-stage compression mechanism (11) is stopped, the refrigerant evaporated in the low-temperature heat exchanger (16) passes through the low-stage bypass path (18) and is sucked into the high-stage compression mechanism (12) together with the refrigerant evaporated in the auxiliary heat exchanger (1).

[0026] The sixth aspect of the present invention is characterized by including, in the first aspect of the present invention, a high-stage bypass path (19) which bypasses the high-stage compression mechanism (12), and a compression mechanism control section (41) which, in the case where the heating load is smaller than the cooling load, controls operations of the low-stage compression mechanism (11) and the high-stage compression mechanism (12) by switching at least between a low-stage-only compression operation and a two-stage compression operation, wherein the low-stage-only compression operation is an operation in which the high-stage compression mechanism (12) is stopped and operational capacity of the low-stage compression mechanism (11) is controlled according to a cooling load of the low-temperature heat exchanger (16) when a pressure difference between a condensing pressure of the auxiliary heat exchanger (1) and a condensing pressure of the high-temperature heat exchanger (13) is smaller than a predetermined value, or when the condensing pressure of the auxiliary heat exchanger (1) is higher than or equal to the condensing pressure of the high-temperature heat exchanger (13), the two-stage compression operation is an operation in which operational capacity of the high-stage compression mechanism (12) is controlled according to a heating load of the high-temperature heat exchanger (13), and the operational capacity of the low-stage compression mechanism (11) is controlled according to the cooling load of the low-temperature heat exchanger (16), when the pressure difference is larger than or equal to the predetermined value and the condensing pressure of the auxiliary heat exchanger (1) is lower than the condensing pressure of the high-temperature heat exchanger (13).

[0027] The smaller the pressure difference between the condensing pressure of the auxiliary heat exchanger (1) and the condensing pressure of the high-temperature heat exchanger (13), the closer the suction pressure and the discharge pressure of the high-stage compression mechanism (12) and the smaller the effects of enhancing operational efficiency of the heat pump by two-stage compression. If the condensing pressure of the auxiliary heat exchanger (1) is higher than the condensing pressure of the high-stage compression mechanism (12), a pressure of the refrigerant sucked by the high-stage compression mechanism (12) and a pressure of the refrigerant discharged from the high-stage compression mechanism (12) are inverted, which causes a malfunction of the high-stage compression mechanism (12). In practice, the pressure of the refrigerant discharged from the high-stage compression mechanism (12) is increased to continue operation, which however, is higher than an optimal condensing pressure of the high-temperature heat exchanger (13), and therefore, the operational efficiency of the heat pump may be reduced. The "predetermined value" as used herein is within a range of pressure differences within which it is possible to enhance operational efficiency of the heat pump by two-stage compression.

[0028] In the sixth aspect of the present invention, the high-stage compression mechanism (12) is stopped and only the low-stage compression mechanism (11) is actuated (i.e., low-stage-only compression operation) when the pressure difference between the condensing pressure of the auxiliary heat exchanger (1) and the condensing pressure of the high-temperature heat exchanger (13) is smaller than a predetermined value, or when the condensing pressure of the auxiliary heat exchanger (1) is higher than or equal to the condensing pressure of the high-temperature heat exchanger (13). Since the high-stage compression mechanism (12) is stopped, the flow of refrigerant discharged from low-stage compression mechanism (11) is separated into both of the auxiliary heat exchanger (1) and the high-stage bypass path (19).

[0029] The seventh aspect of the present invention is characterized by including in any one of the second to sixth aspects of the present invention, a flow rate control mechanism (2) which controls a flow rate of the refrigerant flowing in the auxiliary heat exchanger (1), and a flow rate control mechanism control section (43) which, in the case where the heating load is greater than the cooling load, controls the flow rate control mechanism (2) such that a degree of superheat of the refrigerant having flowed out of the auxiliary heat exchanger (1) is a predetermined value.

[0030] In the seventh aspect of the present invention, the refrigerant flowing into the auxiliary heat exchanger (1) can be reliably evaporated due to the flow rate control mechanism control section (43) in the case where the heating load is greater than the cooling load and the auxiliary heat exchanger (1) functions as an evaporator.

[0031] The eighth aspect of the present invention is characterized by including in any one of the second to the sixth aspects of the present invention, a flow rate control mechanism (2) which controls a flow rate of the refrigerant flowing in the auxiliary heat exchanger (1), and a flow rate control mechanism control section (43) which, in the case where the heating load is smaller than the cooling load, controls the flow rate control mechanism (2) such that a degree of subcooling of the refrigerant having flowed out of the auxiliary heat exchanger (1) is a predetermined value.

[0032] In the eighth aspect of the present invention, the refrigerant flowing into the auxiliary heat exchanger (1) can be reliably condensed due to the flow rate control mechanism control section (43) in the case where the heating load is smaller than the cooling load and the auxiliary heat exchanger (1) functions as a condenser.

[0033] The ninth aspect of the present invention is characterized by including in any one of the second to fourth aspects of the present invention, a high-stage expansion mechanism control section (44) which sets the high-stage expansion mechanism (14) to be fully open in the case where the heating load is greater than the cooling load.

[0034] In the ninth aspect of the present invention, the refrigerant flowing to the auxiliary heat exchanger (1) can be controlled using only the flow rate control mechanism (2) by fully opening the high-stage expansion mechanism (14).

[0035] The tenth aspect of the present invention is characterized by including in any one of the second to eighth aspects of the present invention, a high-stage expansion mechanism control section (44) which, in the case where the heating load is smaller than the cooling load, controls the high-stage expansion mechanism (14) such that an outlet temperature of the refrigerant from the high-stage expansion mechanism (14) is a temperature between an outlet temperature of the refrigerant from the auxiliary heat exchanger (1) and an outlet temperature of the refrigerant from the low-temperature heat exchanger (16).

[0036] In the tenth aspect of the present invention, the pressure of the refrigerant flowing out of the high-stage expansion mechanism (14) can be set to an intermediate pressure of the refrigerant circuit (10) with reliability.

ADVANTAGES OF THE INVENTION

[0037] According to the present invention, the auxiliary heat exchanger (1) is located at the intermediate-pressure line of the refrigerant circuit (10). Thus, the compression power of the refrigerant circuit (10) which is used to supply a refrigerant to the auxiliary heat exchanger (1) can be reduced compared to the case in which the auxiliary heat exchanger (1) is located at the high-pressure line or the low-pressure line. Further, only a necessary amount of refrigerant flows in the auxiliary heat exchanger (1) without control. It is therefore possible to improve operational efficiency of the heat pump, compared to conventional cases.

[0038] According to the second aspect of the present invention, the high-stage compression mechanism (12) is controlled according to the heating load, and the low-stage compression mechanism (11) is controlled according to the cooling load. Therefore, the auxiliary heat exchanger (1) can function as an evaporator when the heating load is greater than the cooling load, and can function as a condenser when the cooling load is greater than the heating load. As a result, without providing a switching valve at the refrigerant circuit (10), it is possible to make the auxiliary heat exchanger (1) function as an evaporator or a condenser according to conditions of the heating load and the cooling load.

[0039] According to the third aspect of the present invention, the switching mechanism (51, 52) switches between the low-stage suction state and the high-stage suction state, based on the evaporating pressures of the auxiliary heat exchanger (1) and the low-temperature heat exchanger (16). Thus, the refrigerant evaporated in the auxiliary heat exchanger (1) can be sucked into the low-stage compression mechanism (11) or the high-stage compression mechanism (12) as needed, and the heat pump can always operate with high efficiency.

[0040] According to the fourth aspect of the present invention, the degree of subcooling of the refrigerant flowing from the high-temperature heat exchanger (13) to the high-stage expansion mechanism (14) can be greater, compared to the case in which the economizer heat exchanger (55) is not provided. The efficiency of the heat pump can therefore be improved.

[0041] According to the fifth aspect of the present invention, the compression mechanism control section (41) switches the operation to the two-stage compression operation or the high-stage-only compression operation, based on the evaporating pressures of the auxiliary heat exchanger (1) and the low-temperature heat exchanger (16). Thus, the heat pump can perform the two-stage compression or the single-stage compression as needed, and the heat pump can always operate with high efficiency.

[0042] According to the sixth aspect of the present invention, the compression mechanism control section (41) switches the operation to the two-stage compression operation or the low-stage-only compression operation, based on the condensing pressures of the auxiliary heat exchanger (1) and the high-temperature heat exchanger (13). The heat pump can perform the two-stage compression or the single-stage compression as needed, and the heat pump can always operate with high efficiency.

[0043] According to the seventh aspect of the present invention, the refrigerant flowing in the auxiliary heat exchanger (1) can be completely evaporated due to the flow rate control mechanism control section (43), and the heat exchange rate of the auxiliary heat exchanger (1) can be ensured. Thus, the heat balance of the refrigerant circuit (10) can be reliably maintained in the state in which the heating load is greater than the cooling load.

[0044] According to the eighth aspect of the present invention, the refrigerant flowing in the auxiliary heat exchanger (1) can be completely condensed due to the flow rate control mechanism control section (43), and the heat exchange rate of the auxiliary heat exchanger (1) can be ensured. Thus, the heat balance of the refrigerant circuit (10) can be reliably maintained in the state in which the heating load is smaller than the cooling load.

[0045] According to the ninth aspect of the present invention, the refrigerant flowing to the auxiliary heat exchanger (1) can be controlled using only the flow rate control mechanism (2), which means that the flow rate control of the refrigerant flowing to the auxiliary heat exchanger (1) can be simplified.

[0046] According to the tenth aspect of the present invention, the pressure of the refrigerant flowing out of the high-stage expansion mechanism (14) can be set to an intermediate pressure of the refrigerant circuit (10) with reliability, and heat can be exchanged between the refrigerant and the heat-source fluid in the auxiliary heat exchanger (1) with reliability.

BRIEF DESCRIPTION OF THE DRAWINGS

[0047]

FIG. 1 shows a refrigerant circuit diagram of a heat pump of the present embodiment.
FIG. 2 shows refrigerant flow in an excessive heating operation of the present embodiment.
FIG. 3 shows refrigerant flow in an excessive cooling operation of the present embodiment.
FIG. 4 shows refrigerant flow in a heating-only operation of the present embodiment.
FIG. 5 shows refrigerant flow in a cooling-only operation of the present embodiment.

FIG. 6 shows a refrigerant circuit diagram of a heat pump of the first variation of the present embodiment.

FIG. 7 shows a refrigerant circuit diagram of a heat pump of the second variation of the present embodiment.

FIG. 8 shows a refrigerant circuit diagram of a heat pump of the third variation of the present embodiment.

FIG. 9 shows refrigerant flow in a high-stage-only compression operation of the third variation.

FIG. 10 shows refrigerant flow in a low-stage-only compression operation of the third variation.

FIG. 11 shows a configuration of a controller.

FIG. 12 shows a refrigerant circuit diagram of a heat pump of the fourth variation of the present embodiment.

FIG. 13 schematically shows a relationship between the refrigeration cycle and respective heat exchangers on a P-h diagram. FIG. 13(A) shows a case in which the auxiliary heat exchanger is placed at a high-pressure line. FIG. 13(B) shows a case in which the auxiliary heat exchanger is placed at an intermediate-pressure line.

DESCRIPTION OF EMBODIMENTS

[0048] Embodiments of the present invention will be described in detail below based on the drawings.

[0049] A heat pump of the present embodiment is for industrial use. The heat pump is capable of providing cold thermal energy and hot thermal energy simultaneously. The heat pump is provided with a refrigerant circuit (10) and a controller (40).

-Refrigerant Circuit-

[0050] The refrigerant circuit (10) performs a refrigeration cycle in two-stage compression and two-stage expansion. The refrigerant circuit (10) is provided with a low-stage compressor (a low-stage compression mechanism) (11), a high-stage compressor (a high-stage compression mechanism) (12), a heating heat exchanger (a high-temperature heat exchanger) (13), a high-stage expansion valve (a high-stage expansion mechanism) (14), a low-stage expansion valve (a low-stage expansion mechanism) (15), a cooling heat exchanger (a low-temperature heat exchanger) (16), a flow rate control valve (a flow rate control mechanism) (2), and an auxiliary heat exchanger (1).

[0051] Both of the low-stage compressor (11) and the high-stage compressor (12) are hermetic compressors. A low-stage side inverter (not shown) is connected to the low-stage compressor (11), and a high-stage side inverter (not shown) is connected to the high-stage compressor (12). These inverters enable the compressors (11, 12) to have variable rotational speed. Further, a discharge port of the low-stage compressor (11) and a suction port of the high-stage compressor (12) are connected to each other by a connection pipe (4) on the compressor side. A check valve (CV1) is attached to the connection pipe (4) at a location near the low-stage compressor (11). The check valve (CV1) allows the refrigerant to flow in a direction from the low-stage compressor (11) to the high-stage compressor (12) and prevents the refrigerant to flow in the adverse direction.

[0052] The heating heat exchanger (13) has a refrigerant channel (13a) and a water channel (13b). The refrigerant channel (13a) has an inlet port connected to the discharge port of the high-stage compressor (12) by a first refrigerant pipe (5), and an outlet port connected to an inlet port of the high-stage expansion valve (14) by a second refrigerant pipe (6). On the other hand, the water channel (13b) of the heating heat exchanger (13) communicates with a hot water channel (30). A hot water pump (31) and a hot water tank (32) are connected to the hot water channel (30). The heating heat exchanger (13) is configured such that when a high-pressure refrigerant discharged from the high-stage compressor (12) passes through the refrigerant channel (13a) and water having flowed out of the hot water pump (31) passes through the water channel (13b), the heating heat exchanger (13) exchanges heat between the high-pressure refrigerant and the water.

[0053] The high-stage expansion valve (14) and the low-stage expansion valve (15) are electronic expansion valves whose degree of opening can be adjusted. An outlet port of the high-stage expansion valve (14) and an inlet port of the low-stage expansion valve (15) are connected to each other by a connection pipe (7) on the expansion valve side.

[0054] The cooling heat exchanger (16) has a refrigerant channel (16a) and a water channel (16b). The refrigerant channel (16a) has an inlet port connected to an outlet port of the low-stage expansion valve (15) by a third refrigerant pipe (8), and an outlet port connected to a suction port of the low-stage compressor (11) by a fourth refrigerant pipe (9). On the other hand, the water channel (16b) of the cooling heat exchanger (16) communicates with a cold water channel (33). A cold water pump (34) and a cold water tank (35) are connected to the cold water channel (33). The cooling heat exchanger (16) is configured such that when a low-pressure refrigerant having flowed of from the low-stage expansion valve (15) passes through the refrigerant channel (16a) and water having flowed out of the cold water pump (34) passes through the water channel (16b), the cooling heat exchanger (16) exchanges heat between the low-pressure refrigerant and the water.

[0055] As described above, the refrigerant circuit (10) has a closed circuit in which the low-stage compressor (11), the high-stage compressor (12), the heating heat exchanger (13), the high-stage expansion valve (14), the low-stage expansion valve (15), and the cooling heat exchanger (16) are sequentially connected together. The auxiliary heat exchanger (1) and the flow rate control valve (2) are connected to the closed circuit.

<Auxiliary Heat Exchanger>

[0056] The auxiliary heat exchanger (1) is intended to maintain heat balance of a refrigeration cycle of the refrigerant circuit (10).

[0057] The auxiliary heat exchanger (1) is, for example, a cross-fin type fin-and-tube heat exchanger, and includes a refrigerant path (1a) and an air passageway (not shown). A branch pipe (3a) branched from the connection pipe (4) on the compressor side is connected to one end of the refrigerant path (1a) of the auxiliary heat exchanger (1), and a branch pipe (3b) branched from the connection pipe (7) on the expansion valve side is connected to the other end. The flow rate control valve (2) is provided at the branch pipe (3b).

[0058] Further, an air blowing fan (17) is provided near the auxiliary heat exchanger (1). The auxiliary heat exchanger (1) is configured such that when a refrigerant discharged from the low-stage compressor (11) or a refrigerant having flowed out of the high-stage expansion valve (14) passes through the refrigerant path (1a) and outdoor air blown by the air blowing fan (17) passes through the air passageway, the auxiliary heat exchanger (1) exchanges heat between the refrigerant and the outdoor air.

-Controller-

[0059] The controller (40) is configured to control operation of the heat pump. As illustrated in FIG. 11, the controller (40) includes a compressor control section (a compression mechanism control section) (41), a load determination section (42), a flow rate control valve control section (a flow rate control mechanism control section) (43), a high-stage expansion valve control section (a high-stage expansion mechanism control section) (44), and a low-stage expansion valve control section (a low-stage expansion mechanism control section) (45). Further, a plurality of temperature sensors (21-26) are electrically connected to the controller (40).

[0060] Specifically, the plurality of temperature sensors (21-26) are a high-stage expansion valve temperature sensor (21) which detects an outlet temperature of the refrigerant from the high-stage expansion valve (14), a cooling heat exchanger temperature sensor (22) which detects an outlet temperature of the refrigerant from the cooling heat exchanger (16), first and second auxiliary heat exchanger temperature sensors (23, 24) which detect temperatures of the refrigerant before and after passing through the auxiliary heat exchanger (1), a hot water temperature sensor (25) which detects an outlet temperature of hot water from the heating heat exchanger (13), and a cold water temperature sensor (26) which detects a outlet temperature of cold water from the cooling heat exchanger (16).

<Compressor control section>

[0061] Detection values of the hot water temperature sensor (25) and the cold water temperature sensor (26), a hot water setting value of the outlet temperature of the hot water from the heating heat exchanger (13), and a cold water setting value of the outlet temperature of the cold water from the cooling heat exchanger (16) are input in the compressor control section (41).

[0062] In the case where the detection value of the hot water temperature sensor (25) is lower than the hot water setting value, the compressor control section (41) outputs a signal for increasing the rotational speed of the high-stage compressor (12) to the high-stage side inverter. In the case where the detection value of the hot water temperature sensor (25) is higher than the hot water setting value, the compressor control section (41) outputs a signal for reducing the rotational speed of the high-stage compressor (12) to the high-stage side inverter.

[0063] Further, in the case where the detection value of the cold water temperature sensor (26) is higher than the cold water setting value, the compressor control section (41) outputs a signal for increasing the rotational speed of the low-stage compressor (11) to the low-stage side inverter. In the case where the detection value of the cold water temperature sensor (26) is lower than the cold water setting value, the compressor control section (41) outputs a signal for reducing the rotational speed of the low-stage compressor (11) to the low-stage side inverter.

[0064] As described above, the compressor control section (41) controls an operational capacity of the high-stage compressor (12) according to the heating load, and an operational capacity of the low-stage compressor (11) according to the cooling load.

<Load Determination Section>

[0065] Frequency instruction values of the low-stage side inverter and the high-stage side inverter are input in the load determination section (42). The load determination section (42) detects a cooling load based on the frequency instruction value of the low-stage side inverter, and detects a heating load based on the frequency instruction value of the high-stage side inverter. In the case where the frequency instruction value of the high-stage side inverter is larger than the frequency instruction value of the low-stage side inverter, the load determination section (42) determines that the heating load is greater than the cooling load, and outputs a signal that indicates excessive heating. In the case where the frequency instruction value of the high-stage side inverter is smaller than the frequency instruction value of the low-stage side inverter, the load determination section (42) determines that the heating load is smaller than the cooling load, and outputs a signal that indicates excessive cooling.

<Flow Rate Control Valve Control Section>

[0066] Detection values of the first and second auxiliary heat exchanger temperature sensors (23, 24) and a signal determined by the load determination section (42) are input to the flow rate control valve control section (43). A detection value of an auxiliary heat exchanger internal temperature sensor (not shown) which detects a temperature of the refrigerant flowing in the auxiliary heat exchanger (1) is input to the flow rate control valve control section (43).

[0067] When the flow rate control valve control section (43) receives the signal that indicates excessive heating from the load determination section (42), the flow rate control valve control section (43) determines that the value detected by the auxiliary heat exchanger internal temperature sensor is an evaporation temperature in the auxiliary heat exchanger (1), and based on this evaporation temperature, calculates a degree of superheat at the outlet of the auxiliary heat exchanger (1) from the value detected by the second auxiliary heat exchanger temperature sensor (24). A signal for controlling a degree of opening is appropriately outputs from the flow rate control valve control section (43) to the flow rate control valve (2), and the degree of opening of the flow rate control valve (2) is controlled so that the degree of superheat at the outlet will be a predetermined value (e.g., 3°C).

[0068] On the other hand, when the flow rate control valve control section (43) receives the signal that indicates excessive cooling from the load determination section (42), the flow rate control valve control section (43) determines that the value detected by the auxiliary heat exchanger internal temperature sensor is a condensation temperature of the auxiliary heat exchanger (1), and based on this condensation temperature, calculates a degree of subcooling at the outlet of the auxiliary heat exchanger (1) from the value detected by the first auxiliary heat exchanger temperature sensor (23). A signal for controlling a degree of opening is appropriately outputs from the flow rate control valve control section (43) to the flow rate control valve (2), and the degree of opening of the flow rate control valve (2) is controlled so that the degree of subcooling will be a predetermined value (e.g., 2°C).

<High-Stage Expansion Valve Control Section>

[0069] A detection value of the high-stage expansion valve temperature sensor (21), a detection value of the cooling heat exchanger temperature sensor (22), a detection value of the second auxiliary heat exchanger temperature sensor (24), and a signal determined by the load determination section (42) are input to the high-stage expansion valve control section (44).

[0070] When the high-stage expansion valve control section (44) receives the signal that indicates excessive heating from the load determination section (42), the signal for controlling a degree of opening is output from the high-stage expansion valve control section (44) to the high-stage expansion valve (14), and the high-stage expansion valve (14) is fully opened.

[0071] On the other hand, when the high-stage expansion valve control section (44) receives the signal that indicates excessive cooling from the load determination section (42), the signal for controlling a degree of opening is appropriately output from the high-stage expansion valve control section (44) to the high-stage expansion valve (14), and the degree of opening of the high-stage expansion valve (14) is controlled such that the outlet temperature of the refrigerant from the high-stage expansion valve (14) (the value detected by the high-stage expansion valve temperature sensor (21)) will be a temperature between the outlet temperature of the refrigerant from the auxiliary heat exchanger (1) (the value detected by the second auxiliary heat exchanger temperature sensor (24)) and the outlet temperature of the refrigerant from the low-temperature heat exchanger (16) (the value detected by the cooling heat exchanger temperature sensor (22)).

<Low-Stage Expansion Valve Control Section>

[0072] A detection value of the cooling heat exchanger temperature sensor (22) is input to the low-stage expansion valve control section (45). Further, a detection value of a cooling heat exchanger internal temperature sensor (not shown) which detects a temperature of the refrigerant flowing in the cooling heat exchanger (16) is input to the low-stage expansion valve control section (45).

[0073] The low-stage expansion valve control section (45) determines that the value detected by the cooling heat exchanger internal temperature sensor is an evaporation temperature in the cooling heat exchanger (16), and based on this evaporation temperature, calculates a degree of superheat at the outlet of the cooling heat exchanger (16) from the value detected by the cooling heat exchanger temperature sensor (22). A signal for controlling a degree of opening is appropriately output from the low-stage expansion valve control section (45) to the low-stage expansion valve (15), and the degree of opening of the low-stage expansion valve (15) is controlled so that the degree of superheat at the outlet will be a predetermined value (e.g., 3°C).

-Operation of Heat Pump-

[0074] Next, operation of the heat pump will be described. The heat pump can perform an excessive heating operation or an excessive cooling operation without using a switching valve, etc., according to conditions of the heating load and the cooling load. The excessive heating operation and the excessive cooling operation will be described first, and a heating-only operation and a cooling-only operation will be described thereafter.

<Excessive Heating Operation>

[0075] The excessive heating operation shown in FIG. 2 is an operation performed when a heating load of the heat pump is greater than a cooling load. In the present embodiment, the excessive heating operation under the following conditions will be described: the outdoor air temperature is 15°C; the hot water setting value and the cold water setting value determined by the compressor control section (41) are 65°C and 7°C, respectively; and a necessary heating capability and a necessary cooling capability of the heat pump are 90% and 60%, respectively.

[0076] In this excessive heating operation, the compressor control section (41) of the controller (40) controls the rotational speed of the high-stage compressor (12) such that the outlet temperature of hot water from the heating heat exchanger (13) will be the hot water setting value, i.e., 65°C, and controls the rotational speed of the low-stage compressor (11) such that the outlet temperature of cold water from the cooling heat exchanger (16) will be the cold water setting value, i.e., 7°C.

[0077] Further, the high-stage expansion valve control section (44) sets the high-stage expansion valve (14) to be fully open. The flow rate control valve control section (43) controls a degree of opening of the flow rate control valve (2) such that a degree of superheat at the outlet of the auxiliary heat exchanger (1) will be 3°C. The low-stage expansion valve control section (45) controls a degree of opening of the low-stage expansion valve (15) such that a degree of superheat at the outlet of the cooling heat exchanger (16) will be 3°C.

[0078] After actuation of the low-stage compressor (11) and the high-stage compressor (12), the rotational speed of the high-stage compressor (12) exceeds the rotational speed of the low-stage compressor (11) because the heating load is greater than the cooling load, and a refrigerant suction amount of the high-stage compressor (12) becomes larger than a refrigerant discharge amount of the low-stage compressor (11).

[0079] Thus, the refrigerant evaporated in the auxiliary heat exchanger (1) is suctioned into the high-stage compressor (12) together with the refrigerant discharged from the low-stage compressor (11). That is, the refrigerant flows through the auxiliary heat exchanger (1) from the expansion valve side to the compressor side (from the left side to the right side of the auxiliary heat exchanger (1) in FIG. 2).

[0080] The refrigerant discharged from the high-stage compressor (12) dissipates heat into the water in the hot water channel (30) of the heating heat exchanger (13), and is condensed. At this moment, the condensation temperature in the heating heat exchanger (13) is around 70°C, and the water in the hot water channel (30) is heated to 65°C due to the heat dissipation of the refrigerant in the heating heat exchanger (13). The refrigerant condensed in the heating heat exchanger (13) is separated into two flows after passing through the high-stage expansion valve (14) fully opened by the high-stage expansion valve control section (44).

[0081] The refrigerant in one of the separated flows of the refrigerant is decompressed by the low-stage expansion valve (15), and thereafter absorbs heat from the water in the cold water channel (33) in the cooling heat exchanger (16) and evaporates. The evaporation temperature in the cooling heat exchanger (16) is around 0°C, and the water in the cold water channel (33) is cooled to 7°C due to the heat absorption of the refrigerant in the cooling heat exchanger (16). The refrigerant evaporated in the cooling heat exchanger (16) is sucked into the low-stage compressor (11) and compressed, and thereafter discharged toward a suction side of the high-stage compressor (12).

[0082] The refrigerant in the other one of the separated flows of the refrigerant is decompressed by the flow rate control valve (2), and thereafter absorbs heat from the outdoor air in the auxiliary heat exchanger (1) and evaporates. The evaporation temperature at this time is around 10°C. The refrigerant evaporated in the auxiliary heat exchanger (1) is merged with the refrigerant discharged from the low-stage compressor (11), sucked into the high-stage compressor (12) and compressed, and then discharged into the heating heat exchanger (13) again.

[0083] As described above, in the case where the heating load is greater than the cooling load, the direction of the refrigerant flowing in the auxiliary heat exchanger (1) is from the expansion valve side to the compressor side, and the auxiliary heat exchanger (1) functions as an evaporator. Accordingly, the refrigerant circuit (10) can perform a refrigeration cycle while maintaining the heat balance.

<Excessive Cooling Operation>

[0084] The excessive cooling operation shown in FIG. 3 is an operation performed when a heating load of the heat pump is smaller than the cooling load. In the present embodiment, the excessive cooling operation under the following conditions will be described: the outdoor air temperature is 15°C; the hot water setting value and the cold water setting value determined by the compressor control section (41) are 65°C and 7°C, respectively; and a necessary heating capability and a necessary cooling capability of the heat pump are 40% and 80%, respectively.

[0085] In this excessive cooling operation, the compressor control section (41) of the controller (40) controls the rotational speed of the high-stage compressor (12) such that the outlet temperature of hot water from the heating heat exchanger (13) will be the hot water setting value, i.e., 65°C, and controls the rotational speed of the low-stage compressor (11) such that the outlet temperature of cold water from the cooling heat exchanger (16) will be the cold water setting value, i.e., 7°C.

[0086] The high-stage expansion valve control section

(44) controls the degree of opening of the high-stage expansion valve (14) such that the outlet temperature of the refrigerant from the high-stage expansion valve (14) will be a temperature between the outlet temperature of the refrigerant from the auxiliary heat exchanger (1) and the outlet temperature of the refrigerant from the cooling heat exchanger (16). The flow rate control valve control section (43) controls the degree of opening of the flow rate control valve (2) such that the degree of subcooling at the outlet of the auxiliary heat exchanger (1) will be 2°C. Further, the low-stage expansion valve control section (45) controls the degree of opening of the low-stage expansion valve (15) such that the degree of superheat at the outlet of the cooling heat exchanger (16) will be 3°C.

[0087] After actuation of the low-stage compressor (11) and the high-stage compressor (12), the rotational speed of the high-stage compressor (12) becomes lower than rotational speed of the low-stage compressor (11) because the heating load is smaller than the cooling load, and a refrigerant suction amount of the high-stage compressor (12) becomes smaller than a refrigerant discharge amount of the low-stage compressor (11).

[0088] In the above situation, not all the refrigerant discharged from the low-stage compressor (11) can be sucked into the high-stage compressor (12), and part of the refrigerant discharged from the low-stage compressor (11) flows into the auxiliary heat exchanger (1). That is, the refrigerant flows through the auxiliary heat exchanger (1) from the compressor side to the expansion valve side (from the right side to the left side of the auxiliary heat exchanger (1) in FIG. 3).

[0089] The refrigerant separated into a flow from the low-stage compressor (11) to the high-stage compressor (12) is compressed by the high-stage compressor (12) and is discharged to the heating heat exchanger (13) thereafter. The refrigerant discharged from the high-stage compressor (12) dissipates heat into the water in the hot water channel (30) in the heating heat exchanger (13), and is condensed. The condensation temperature at this time is around 70°C, and the water in the hot water channel (30) is heated to 65°C due to the heat dissipation of the refrigerant in the heating heat exchanger (13). The refrigerant condensed in the heating heat exchanger (13) is decompressed by the high-stage expansion valve (14).

[0090] On the other hand, the refrigerant separated into a flow from the low-stage compressor (11) to the auxiliary heat exchanger (1) is compressed by the auxiliary heat exchanger (1) and then flows into the flow rate control valve (2). The condensation temperature of the auxiliary heat exchanger (1) at this time is around 20°C. The refrigerant having flowed into the flow rate control valve (2) is decompressed by the flow rate control valve (2) and thereafter merged with the refrigerant having flowed out of the high-stage expansion valve (14) to flow into the low-stage expansion valve (15).

[0091] The refrigerant having flowed into the low-stage expansion valve (15) is decompressed and thereafter absorbs heat from the water in the cold water channel (33) in the cooling heat exchanger (16) and evaporates. The evaporation temperature of the cooling heat exchanger (16) at this time is around 0°C, and the water in the cold water channel (33) is cooled to 7°C due to the heat absorption of the refrigerant in the cooling heat exchanger (16). The refrigerant evaporated in the cooling heat exchanger (16) is sucked into the low-stage compressor (11) and compressed, and thereafter discharged again to the auxiliary heat exchanger (1) and the high-stage compressor (12).

[0092] As described above, in the case where the heating load is smaller than the cooling load, the direction of the refrigerant flowing in the auxiliary heat exchanger (1) is from the compressor side to the expansion valve side, and the auxiliary heat exchanger (1) functions as a condenser. Accordingly, the refrigerant circuit (10) can perform a refrigeration cycle while maintaining the heat balance.

<Heating-Only Operation>

[0093] The heating-only operation as shown in FIG. 4 is an operation performed when there is not the cooling load and there is the heating load. In the heating-only operation, the high-stage compressor (12) is actuated and the low-stage compressor (11) is stopped. Further, the high-stage expansion valve (14) is in a fully open state, and the low-stage expansion valve (15) is in a fully closed state.

[0094] The refrigerant discharged from the high-stage compressor (12) dissipates heat into the water in the hot water channel (30) in the heating heat exchanger (13) and is condensed. At this moment, the water in the hot water channel (30) is heated by the heat dissipation of the refrigerant in the heating heat exchanger (13). The refrigerant condensed by the heating heat exchanger (13) passes through the high-stage expansion valve (14) in the fully open state and thereafter flows into the flow rate control valve (2).

[0095] The refrigerant having flowed into the flow rate control valve (2) is decompressed by the flow rate control valve (2) to be a low-pressure refrigerant, and thereafter absorbs heat from outdoor air in the auxiliary heat exchanger (1) and evaporates. The refrigerant evaporated in the auxiliary heat exchanger (1) is sucked into the high-stage compressor (12) and compressed, and thereafter discharged to the heating heat exchanger (13) again. The heating heat exchanger (13) functions as a condenser and the auxiliary heat exchanger (1) functions as an evaporator, and the heating load is processed in the heating heat exchanger (13).

<Cooling-Only Operation>

[0096] The cooling-only operation shown in FIG. 5 is an operation performed when there is the cooling load but not the heating load. In the cooling-only operation, the high-stage compressor (12) is stopped and the low-stage compressor (11) is actuated. Further, the high-stage expansion valve (14) is in the fully closed state, and the low-stage expansion valve (15) is in the fully open state.

[0097] The refrigerant discharged from the low-stage compressor (11) dissipates heat into the outdoor air in the auxiliary heat exchanger (1) and is condensed, and is thereafter decompressed by the flow rate control valve (2) to be a low-pressure refrigerant. The low-pressure refrigerant passes through the low-stage expansion valve (15) in the fully open state and thereafter absorbs heat from the water in the cold water channel (33) in the cooling heat exchanger (16) and evaporates. At this moment, the water in the cold water channel (33) is cooled due to the heat absorption of the refrigerant in the cooling heat exchanger (16). The refrigerant evaporated in the cooling heat exchanger (16) is sucked into the low-stage compressor (11) and compressed, and is thereafter discharged again to the auxiliary heat exchanger (1). The auxiliary heat exchanger (1) functions as a condenser and the cooling heat exchanger (16) functions as an evaporator, and the cooling load is processed in the cooling heat exchanger (16).

-Advantages of Embodiment-

[0098] In the present embodiment, the auxiliary heat exchanger (1) is located at the intermediate-pressure line of the refrigerant circuit (10), thereby making it possible to reduce compression power of the refrigerant circuit (10) used to supply refrigerant to the auxiliary heat exchanger (1), compared to the case where the auxiliary heat exchanger (1) is located at the high-pressure line or the low-pressure line. It is therefore possible to avoid a reduction in efficiency of the heat pump. Further, only a necessary amount of refrigerant flows in the auxiliary heat exchanger (1) without control. It is therefore possible to improve operational efficiency of the heat pump, compared to conventional cases.

[0099] In the present embodiment, the high-stage compressor (12) is controlled according to the heating load, and the low-stage compressor (11) is controlled according to the cooling load. Therefore, the auxiliary heat exchanger (1) can function as an evaporator when the heating load is greater than the cooling load, and can function as a condenser when the cooling load is greater than the heating load. As a result, without providing a switching valve at the refrigerant circuit (10), it is possible to make the auxiliary heat exchanger (1) function as an evaporator or a condenser according to conditions of the heating load and the cooling load.

[0100] In the present embodiment, the refrigerant flowing in the auxiliary heat exchanger (1) can be completely evaporated due to the flow rate control valve control section (43), and the heat exchange rate of the auxiliary heat exchanger (1) can be ensured. Thus, the heat balance of the refrigerant circuit (10) can be reliably maintained in the state in which the heating load is greater than the cooling load.

[0101] In the present embodiment, the refrigerant flowing in the auxiliary heat exchanger (1) can be reliably condensed due to the flow rate control valve control section (43), and the heat exchange rate of the auxiliary heat exchanger (1) can be ensured. Thus, the heat balance of the refrigerant circuit (10) can be reliably maintained in the state in which the heating load is smaller than the cooling load.

-First Variation of Embodiment-

[0102] The first variation of the embodiment, shown in FIG. 6, differs from the above embodiment particularly in that switching mechanisms (51, 52) that change refrigerant flow of the refrigerant circuit (10) and a switching mechanism controlling section (not shown) that controls the switching mechanisms (51, 52) are provided. In the following description, descriptions of the same elements as in the above embodiment are omitted, and only differences will be described.

[0103] A refrigerant circuit (10) of the first variation is provided with an auxiliary pipe (50) which connects the branch pipe (3a) and the fourth refrigerant pipe (9). A first on-off valve (51) is provided at the auxiliary pipe (50). A second on-off valve (52) is provided at the branch pipe (3a) closer to the connection pipe (4) on the compressor side. The on-off valves (51, 52) form the switching mechanisms (51, 52) described above.

[0104] The first state of the switching mechanisms (51, 52) is that the first on-off valve (51) is closed and the second on-off valve (52) is open. The second state of the switching mechanisms (51, 52) is that the first on-off valve (51) is open and the second on-off valve (52) is closed.

[0105] A heat pump of the first variation is configured to be able to perform a second excessive heating operation using the first and second on-off valves (51, 52) in addition to the four operations (the excessive heating operation, the excessive cooling operation, the heating-only operation, and the cooling-only operation) described above. In the present embodiment, the above-described four operations can be performed when the switching mechanisms (51, 52) are in the first state, and the second excessive heating operation can be performed when the switching mechanisms (51, 52) are in the second state. The second excessive heating operation is an operation performed when a heating load of the heat pump is greater than a cooling load.

[0106] In the case where the heating load is greater than the cooling load, and both of the auxiliary heat exchanger (1) and the low-temperature heat exchanger (16) function as evaporators, the smaller a pressure difference between an evaporating pressure of the auxiliary heat exchanger (1) and an evaporating pressure of the cooling heat exchanger (16), the closer a suction pressure and a discharge pressure of the low-stage compressor (11) and the smaller the effects of enhancing operational efficiency of the heat pump by two-stage compression. Further, if the evaporating pressure of the auxiliary heat exchanger (1) is lower than the evaporating pressure of the low-temperature heat exchanger (16), a pressure of the refrigerant suctioned by the low-stage compressor (11) and a pressure of the refrigerant discharged from the low-stage compressor (11) are inverted, which results in malfunction of the low-stage compressor (11). In practice, the pressure of the refrigerant sucked by the low-stage compressor (11) is lowered to continue operation, which, however, is lower than an optimal evaporating pressure of the cooling heat exchanger (16) in this case, and therefore, the operational efficiency of the heat pump may be reduced.

[0107] In view of this, the first on-off valve (51) is opened and the second on-off valve (52) is closed (a low-stage suction state of the switching mechanisms (51, 52)) when the pressure difference between the evaporating pressure of the auxiliary heat exchanger (1) and the evaporating pressure of the cooling heat exchanger (16) is smaller than a predetermined value, or when the evaporating pressure of the auxiliary heat exchanger (1) is lower than or equal to the evaporating pressure of the cooling heat exchanger (16). As a result, the refrigerant flows from the auxiliary heat exchanger (1) to a suction side of the low-stage compressor (11).

[0108] The switching mechanism controlling section estimates the evaporating pressure of the auxiliary heat exchanger (1) from the outdoor air temperature, and estimates the evaporating pressure of the cooling heat exchanger (16) from the outlet temperature of cold water from the cooling heat exchanger (16). The switching mechanism controlling section changes the operation to the low-stage suction state, when a temperature difference between the outdoor air temperature and the outlet temperature of cold water from the cooling heat exchanger (16) is smaller than the predetermined value and the outdoor air temperature is lower than and equal to the outlet temperature of cold water from the cooling heat exchanger (16). The "predetermined value" as used herein is within a range of temperature differences that are converted from pressure differences within which it is possible to enhance operational efficiency of the heat pump by two-stage compression.

[0109] The refrigerant circuit (10) in the state in which the first on-off valve (51) is closed and the second on-off valve (52) is opened (a high-stage suction state of the switching mechanisms (51, 52)) is approximately the same as the refrigerant circuit (10) in the above embodiment. Explanation thereof is therefore omitted.

[0110] As described above, the switching mechanism controlling section switches between the low-stage suction state and the high-stage suction state, based on the outdoor air temperature and the outlet temperature of cold water from the cooling heat exchanger (16). Thus, the refrigerant evaporated in the auxiliary heat exchanger (1) can be sucked into the low-stage compressor (11) or the high-stage compressor (12) as needed, and the heat pump can always operate with high efficiency.

-Second Variation of Embodiment-

[0111] The second variation of the embodiment, shown in FIG. 7, differs from the above embodiment particularly in that an economizer heat exchanger (55) is provided. In the following description, descriptions of the same elements as in the above embodiment are omitted, and only differences will be described.

[0112] A refrigerant circuit (10) of the second variation is provided with an economizer pipe (53) which communicates the second refrigerant pipe (6) and the connection pipe (4) on the compressor side. The economizer heat exchanger (55) has a high-temperature channel and a low-temperature channel, and is placed such that the high-temperature channel communicates with the second refrigerant pipe (6) and the low-temperature channel communicates with the economizer pipe (53). Further, a decompression valve (54) is provided at a location between the second refrigerant pipe (6) of the economizer pipe (53) and the economizer heat exchanger (55).

[0113] Part of the refrigerant having flowed out of the heating heat exchanger (13) is separated and decompressed by the decompression valve (54), and thereafter flows into the low-temperature channel of the economizer heat exchanger (55). The rest of the refrigerant having flowed out of the heating heat exchanger (13) flows into the high-temperature channel of the economizer heat exchanger (55).

[0114] In the economizer heat exchanger (55), the refrigerant in the high-temperature channel and the refrigerant in the low-temperature channel are heat-exchanged, thereby cooling the refrigerant in the high-temperature channel. Thus, it is possible to increase a degree of subcooling of the refrigerant flowing from the high-temperature heat exchanger (13) to the high-stage expansion valve (14), and possible to improve efficiency of the heat pump, compared to the case where the economizer heat exchanger (55) is not provided.

-Third Variation of Embodiment-

[0115] The third variation of the embodiment, shown in FIG. 8 to FIG. 10, differs from the above embodiment in that the refrigerant circuit (10) is configured such that the refrigerant therein can bypass the low-stage compressor (11) or the high-stage compressor (12). In the following description, descriptions of the same elements as in the above embodiment are omitted, and only differences will be described.

[0116] A refrigerant circuit (10) of the third variation is provided with a low-stage bypass pipe (a low-stage bypass path) (18) which bypasses the low-stage compressor (11) and a high-stage bypass pipe (a high-stage bypass path) (19) which bypasses the high-stage compressor (12). Further, the bypass pipes (18, 19) are provided with check valves (CV4, CV3), respectively. The check valves (CV3, CV4) are provided in an orientation which allows the refrigerant to flow in a direction from a suction side to a discharge side of each of the compressors (11, 12) and prevents the refrigerant to flow in the adverse direction.

[0117] Unlike the above embodiment, the compressor control section (41) of the controller (40) is configured to control operations of the low-stage compressor (11) and the high-stage compressor (12) by properly switching among two-stage compression operation, high-stage-only compression operation, and low-stage-only compression operation. The two-stage compression operation is the same as the excessive heating operation and the excessive cooling operation in the above embodiment, and therefore, the explanation thereof is omitted.

<High-Stage-Only Compression Operation>

[0118] In the case where the heating load is greater than the cooling load, and both of the auxiliary heat exchanger (1) and the low-temperature heat exchanger (16) function as evaporators, the smaller a pressure difference between an evaporating pressure of the auxiliary heat exchanger (1) and an evaporating pressure of the cooling heat exchanger (16), the closer a suction pressure and a discharge pressure of the low-stage compressor (11) and the smaller the effects of enhancing operational efficiency of the heat pump by the two-stage compression. Further, if the evaporating pressure of the auxiliary heat exchanger (1) is lower than the evaporating pressure of the low-temperature heat exchanger (16), a pressure of the refrigerant suctioned by the low-stage compressor (11) and a pressure of the refrigerant discharged from the low-stage compressor (11) are inverted, which results in malfunction of the low-stage compressor (11).

[0119] In practice, the pressure of the refrigerant sucked by the low-stage compressor (11) is lowered to continue operation, which, however, is lower than an optimal evaporating pressure of the cooling heat exchanger (16) in this case, and therefore, the operational efficiency of the heat pump may be reduced.

[0120] In view of this, the compressor control section (41) switches the operation from the two-stage compression operation to the high-stage-only compression operation when the pressure difference between the evaporating pressure of the auxiliary heat exchanger (1) and the evaporating pressure of the cooling heat exchanger (16) is smaller than a predetermined value, or when the evaporating pressure of the auxiliary heat exchanger (1) is lower than or equal to the evaporating pressure of the cooling heat exchanger (16). In the high-stage-only compression operation, the low-stage compressor (11) is stopped and only the high-stage compressor (12) is actuated. Since the low-stage compressor (11) is stopped, the refrigerant evaporated in the cooling heat exchanger (16) passes through the low-stage bypass pipe (18) and is sucked into the high-stage compressor (12) together with the refrigerant evaporated in the auxiliary heat exchanger (1).

[0121] In the third variation of the present embodiment, the evaporating pressure of the auxiliary heat exchanger (1) is estimated from the outdoor air temperature, and the evaporating pressure of the cooling heat exchanger (16) is estimated from the outlet temperature of cold water. Thus, the compressor control section (41) changes the operation to the high-stage-only compression operation when a temperature difference between the outdoor air temperature and the outlet temperature of cold water is smaller than the predetermined value, or when the outdoor air temperature is lower than or equal to the outlet temperature of cold water. The "predetermined value" as used herein is within a range of temperature differences that are converted from pressure differences within which it is possible to enhance operational efficiency of the heat pump by two-stage compression.

[0122] In the high-stage-only compression operation, the outlet temperature of cold water from the cooling heat exchanger (16) cannot be controlled by controlling the rotational speed of the low-stage compressor (11) because the low-stage compressor (11) needs to be stopped. In the high-stage-only compression operation, the outlet temperature of cold water from the cooling heat exchanger (16) is controlled by controlling a degree of opening of the low-stage expansion valve (15). The outlet temperature of hot water from the heating heat exchanger (13) is controlled by controlling the rotational speed of the high-stage compressor (12) as in the above-described embodiment.

[0123] As described above, the heat pump can perform two-stage compression or single-stage compression as needed, and the heat pump can always operate with high efficiency.

<Low-Stage-Only Compression Operation>

[0124] In the case where the heating load is smaller than the cooling load, and both of the auxiliary heat exchanger (1) and the heating heat exchanger (13) function as condensers, the smaller a pressure difference between a condensing pressure of the auxiliary heat exchanger (1) and a condensing pressure of the heating heat exchanger (13), the closer a suction pressure and a discharge pressure of the high-stage compressor (12) and the smaller the effects of enhancing operational efficiency of the heat pump by two-stage compression. Further, if the condensing pressure of the auxiliary heat exchanger (1) is higher than the condensing pressure of the heating heat exchanger (13), a pressure of the refrigerant sucked by the high-stage compressor (12) and a pressure of the refrigerant discharged from the high-stage compressor (12) are inverted, which results in malfunction of the high-stage compressor (12).

[0125] In practice, the pressure of the refrigerant discharged from the high-stage compressor (12) is increased to continue operation, which, however, is higher than an optimal condensing pressure of the heating heat exchanger (13) in this case, and therefore, the operational efficiency of the heat pump may be reduced.

[0126] In view of this, the compressor control section (41) switches the operation from the two-stage compression operation to the low-stage-only compression operation when the pressure difference between the condensing pressure of the auxiliary heat exchanger (1) and the condensing pressure of the heating heat exchanger (13) is smaller than a predetermined value, or when the condensing pressure of the auxiliary heat exchanger (1) is higher than or equal to the condensing pressure of the heating heat exchanger (13). In the low-stage-only compression operation, the high-stage compressor (12) is stopped and only the low-stage compressor (11) is actuated. Since the high-stage compressor (12) is stopped, the flow of refrigerant discharged from the low-stage compressor (11) is separated into both of the auxiliary heat exchanger (1) and the high-stage bypass pipe (19).

[0127] In the present embodiment, the condensing pressure of the heating heat exchanger (13) is estimated from the outlet temperature of hot water from the heating heat exchanger (13). Thus, the compressor control section (41) changes the operation to the low-stage-only compression operation when a temperature difference between the outdoor air temperature and the outlet temperature of hot water is smaller than a predetermined value, or the outdoor air temperature is higher than or equal to the outlet temperature of hot water. The "predetermined value" as used herein is within a range of temperature differences that are converted from pressure differences within which it is possible to enhance operational efficiency of the heat pump by two-stage compression.

[0128] In the low-stage-only compression operation, the outlet temperature of hot water from the heating heat exchanger (13) cannot be controlled by controlling the rotational speed of the high-stage compressor (12) because the high-stage compressor (12) needs to be stopped. In the low-stage-only compression operation, the outlet temperature of hot water is controlled by controlling a degree of opening of the high-stage expansion valve (14). The outlet temperature of cold water from the cooling heat exchanger (16) is controlled by controlling the rotational speed of the low-stage compressor (11) as in the above-described embodiment.

[0129] As described above, the heat pump can perform two-stage compression or single-stage compression as needed, and the heat pump can always operate with high efficiency.

-Fourth Variation of Embodiment-

[0130] The fourth variation of the embodiment, shown in FIG. 12, differs from the above embodiment in that it is possible to change the operation to any one of the excessive heating operation, the excessive cooling operation, the heating-only operation, the cooling-only operation, the second excessive heating operation, the high-stage-only compression operation, and the low-stage-only compression operation which are described above. In the following description, descriptions of the same elements as in the above embodiment are omitted, and only differences will be described.

[0131] The refrigerant circuit (10) of the fourth variation is the refrigerant circuit (10) of the first variation (see FIG. 6) except that a high-stage bypass pipe (a high-stage bypass path) (19) which bypasses the high-stage compressor (12) is provided in the refrigerant circuit (10). The high-stage bypass pipe (19) is provided with a check valve (CV3). The check valve (CV3) allows the refrigerant to flow in a direction from a suction side to a discharge side of the high-stage compressor (12) and prevents the refrigerant to flow in the adverse direction.

[0132] In the case of the high-stage-only compression operation, the low-stage compressor (11) is stopped and the first and second on-off valves (51, 52) are fully open. Thus, the refrigerant evaporated in the cooling heat exchanger (16) passes through the auxiliary pipe (50) and thereafter merges with the refrigerant evaporated in the auxiliary heat exchanger (1), and the merged refrigerant is sucked into the high-stage compressor (12). The other operations except the high-stage-only compression operation are the same as those described above. Explanation thereof is therefore omitted.

[0133] As described above, heat pump operations are switched as needed, and the heat pump can always operate with high efficiency.

«Other Embodiments»

[0134] The above embodiment may have the following configurations.

[0135] In the present embodiment, the flow rate control valve control section (43) controls a degree of superheat or a degree of subcooling of the refrigerant circuit (10) based on the signal determined by the load determination section (42), but the configuration is not limited to this configuration. For example, the branch pipe (3a) branched from the connection pipe (4) on the compressor side or the branch pipe (3b) branched from the connection pipe (7) on the expansion valve side may be provided with a detecting section which detects a direction of the refrigerant flow, and the degree of superheat or the degree of subcooling of the refrigerant circuit (10) may be controlled based on a detection signal from the detecting section.

[0136] Specifically, when the detecting section detects that the refrigerant flows from the expansion valve side to the compressor side of the auxiliary heat exchanger (1), the flow rate control valve control section (43) controls the degree of superheat. When the detecting section detects that the refrigerant flows from the compressor side to the expansion valve side of the auxiliary heat exchanger (1), the flow rate control valve control section (43) controls the degree of subcooling. It is therefore possible to control the flow rate control valve control section (43) with reliability.

[0137] In the present embodiment, an evaporating pressure/ a condensing pressure of the auxiliary heat exchanger (1) is estimated from the outdoor air temperature; an evaporating pressure of the cooling heat exchanger (16) is estimated from the outlet temperature of the cold water; and a condensing pressure of the heating heat exchanger (13) is estimated from the outlet temperature of hot water. However, the configuration is not limited to this configuration. For example, these pressures may be directly detected by a pressure sensor.

[0138] Further, the temperature of the refrigerant passing through the heat exchangers (1, 13, 16) may be detected by a temperature sensor, and the pressures may be estimated from the detected value. In this case, as well, similar advantages as in the present invention can be obtained.

[0139] The foregoing embodiments are merely preferred examples in nature, and are not intended to limit the scope, applications, and use of the invention.

INDUSTRIAL APPLICABILITY

[0140] As described above, the present invention relates to heat pumps, and is specifically useful for heat pumps which include a refrigerant circuit capable of providing cooling and heating at the same time.

DESCRIPTION OF REFERENCE CHARACTERS

[0141]

1	auxiliary heat exchanger
2	flow rate control valve (flow rate control mechanism)
10	refrigerant circuit
11	low-stage compressor (low-stage compression mechanism)
12	high-stage compressor (high-stage compression mechanism)
13	heating heat exchanger (high-temperature heat exchanger)
14	high-stage expansion valve (high-stage expansion mechanism)
15	low-stage expansion valve (low-stage expansion mechanism)
16	cooling heat exchanger (low-temperature heat exchanger)
17	air blowing fan
21	high-stage expansion valve temperature sensor
22	cooling heat exchanger temperature sensor
23	first auxiliary heat exchanger temperature sensor
24	second auxiliary heat exchanger temperature sensor
25	hot water temperature sensor
26	cold water temperature sensor
30	hot water channel
31	hot water pump
32	hot water tank
33	cold water channel
34	cold water pump
35	cold water tank
40	controller
41	compressor control section (compression mechanism control section)
42	load determination section
43	flow rate control valve control section (flow rate control mechanism control section)
44	high-stage expansion valve control section (high-stage expansion mechanism control section)
45	low-stage expansion valve control section (low-stage expansion mechanism control section)

Claims

1. A heat pump comprising:

a refrigerant circuit (10) in which a low-stage compression mechanism (11), a high-stage compression mechanism (12), a high-temperature heat exchanger (13), a high-stage expansion mechanism (14), a low-stage expansion mechanism (15), and a low-temperature heat exchanger (16) are sequentially connected together by a refrigerant path, and in which a refrigerant dissipates heat into a high-temperature fluid in the high-temperature heat exchanger (13) and the refrigerant absorbs heat from a low-temperature fluid and evaporates in the low-temperature heat exchanger (16), thereby performing a refrigeration cycle, and
an auxiliary heat exchanger (1) which is connected to communicate between the refrigerant path between the low-stage compression mechanism (11) and the high-stage compression mechanism (12) and the refrigerant path between the low-stage expansion mechanism (15) and the high-stage expansion mechanism (14), and which exchanges heat between the refrigerant of the refrigerant circuit (10) and a heat-source fluid.

2. The heat pump of claim 1, further comprising:

a compression mechanism control section (41) which controls operational capacity of the high-stage compression mechanism (12) according to a heating load of the high-temperature heat exchanger (13) and controls operational capacity of the low-stage compression mechanism (11) according to a cooling load of the low-temperature heat exchanger (16).

3. The heat pump of claim 2, further comprising:

a switching mechanism (51, 52) which, in the case where the heating load is greater than the cooling load, and both of the auxiliary heat exchanger (1) and the low-temperature heat exchanger (16) function as evaporators, switches between a low-stage suction state in which the refrigerant flowing out of the auxiliary heat exchanger (1) is guided to a suction side of the low-stage compression mechanism (11) when a pressure difference between an evaporating pressure of the auxiliary heat exchanger (1) and an evaporating pressure of the low-temperature heat exchanger (16) is smaller than a predetermined value, or when the evaporating pressure of the auxiliary heat exchanger (1) is lower than or equal to the evaporating pressure of the low-temperature heat exchanger (16), and a high-stage suction state in which the refrigerant flowing out of the auxiliary heat exchanger (1) is guided to a suction side of the high-stage compression mechanism (12), when the pressure difference is larger than or equal to the predetermined value and the evaporating pressure of the auxiliary heat exchanger (1) is higher than the evaporating pressure of the low-temperature heat exchanger (16).

4. The heat pump of any one of claims 1-3, further comprising:

an economizer pipe (53) separated from a refrigerant pipe between the high-temperature heat exchanger (13) and the high-stage expansion mechanism (14) and connected to a refrigerant pipe between the low-stage compression mechanism (11) and the high-stage compression mechanism (12),
a decompression mechanism (54) which decompresses the refrigerant in the economizer pipe (53), and
an economizer heat exchanger (55) which exchanges heat between the refrigerant in the economizer pipe (53) decompressed by the decompression mechanism (54) and a high-pressure refrigerant flowing from the high-temperature heat exchanger (13) to the high-stage expansion mechanism (14).

5. The heat pump of claim 1, further comprising:

a low-stage bypass path (18) which bypasses the low-stage compression mechanism (11), and
a compression mechanism control section (41) which, in the case where the heating load is greater than the cooling load, controls operations of the low-stage compression mechanism (11) and the high-stage compression mechanism (12) by switching at least between a high-stage-only compression operation and a two-stage compression operation, wherein
the high-stage-only compression operation is an operation in which operational capacity of the high-stage compression mechanism (12) is controlled according to a heating load of the high-temperature heat exchanger (13), and the low-stage compression mechanism (11) is stopped, when a pressure difference between an evaporating pressure of the auxiliary heat exchanger (1) and an evaporating pressure of the low-temperature heat exchanger (16) is smaller than a predetermined value or when the evaporating pressure of the auxiliary heat exchanger (1) is lower than or equal to the evaporating pressure of the low-temperature heat exchanger (16), and
the two-stage compression operation is an operation in which the operational capacity of the high-stage compression mechanism (12) is controlled according to the heating load of the high-temperature heat exchanger (13), and operational capacity of the low-stage compression mechanism (11) is controlled according to a cooling load of the low-temperature heat exchanger (16), when the pressure difference is larger than or equal to the predetermined value and the evaporating pressure of the auxiliary heat exchanger (1) is higher than the evaporating pressure of the low-temperature heat exchanger (16).

6. The heat pump of claim 1, further comprising:

a high-stage bypass path (19) which bypasses the high-stage compression mechanism (12), and
a compression mechanism control section (41) which, in the case where the heating load is smaller than the cooling load, controls operations of the low-stage compression mechanism (11) and the high-stage compression mechanism (12) by switching at least between a low-stage-only compression operation and a two-stage compression operation, wherein
the low-stage-only compression operation is an operation in which the high-stage compression mechanism (12) is stopped and operational capacity of the low-stage compression mechanism (11) is controlled according to a cooling load of the low-temperature heat exchanger (16) when a pressure difference between a condensing pressure of the auxiliary heat exchanger (1) and a condensing pressure of the high-temperature heat exchanger (13) is smaller than a predetermined value, or when the condensing pressure of the auxiliary heat exchanger (1) is higher than or equal to the condensing pressure of the high-temperature heat exchanger (13),

the two-stage compression operation is an operation in which operational capacity of the high-stage compression mechanism (12) is controlled according to a heating load of the high-temperature heat exchanger (13), and the operational capacity of the low-stage compression mechanism (11) is controlled according to the cooling load of the low-temperature heat exchanger (16), when the pressure difference is larger than or equal to the predetermined value and the condensing pressure of the auxiliary heat exchanger (1) is lower than the condensing pressure of the high-temperature heat exchanger (13).

7. The heat pump of any one of claims 2-6, further comprising:

a flow rate control mechanism (2) which controls a flow rate of the refrigerant flowing in the auxiliary heat exchanger (1), and
a flow rate control mechanism control section (43) which, in the case where the heating load is greater than the cooling load, controls the flow rate control mechanism (2) such that a degree of superheat of the refrigerant having flowed out of the auxiliary heat exchanger (1) is a predetermined value.

8. The heat pump of any one of claims 2-6, further comprising:

a flow rate control mechanism (2) which controls a flow rate of the refrigerant flowing in the auxiliary heat exchanger (1), and
a flow rate control mechanism control section (43) which, in the case where the heating load is smaller than the cooling load, controls the flow rate control mechanism (2) such that a degree of subcooling of the refrigerant having flowed out of the auxiliary heat exchanger (1) is a predetermined value.

9. The heat pump of any one of claims 2-4, further comprising:

a high-stage expansion mechanism control section (44) which sets the high-stage expansion mechanism (14) to be fully open in the case where the heating load is greater than the cooling load.

10. The heat pump of any one of claims 2-4, further comprising:

a high-stage expansion mechanism control section (44) which, in the case where the heating load is smaller than the cooling load, controls the high-stage expansion mechanism (14) such that an outlet temperature of the refrigerant from the high-stage expansion mechanism (14) is a temperature between an outlet temperature of the refrigerant from the auxiliary heat exchanger (1) and an outlet temperature of the refrigerant from the low-temperature heat exchanger (16).

FIG.1

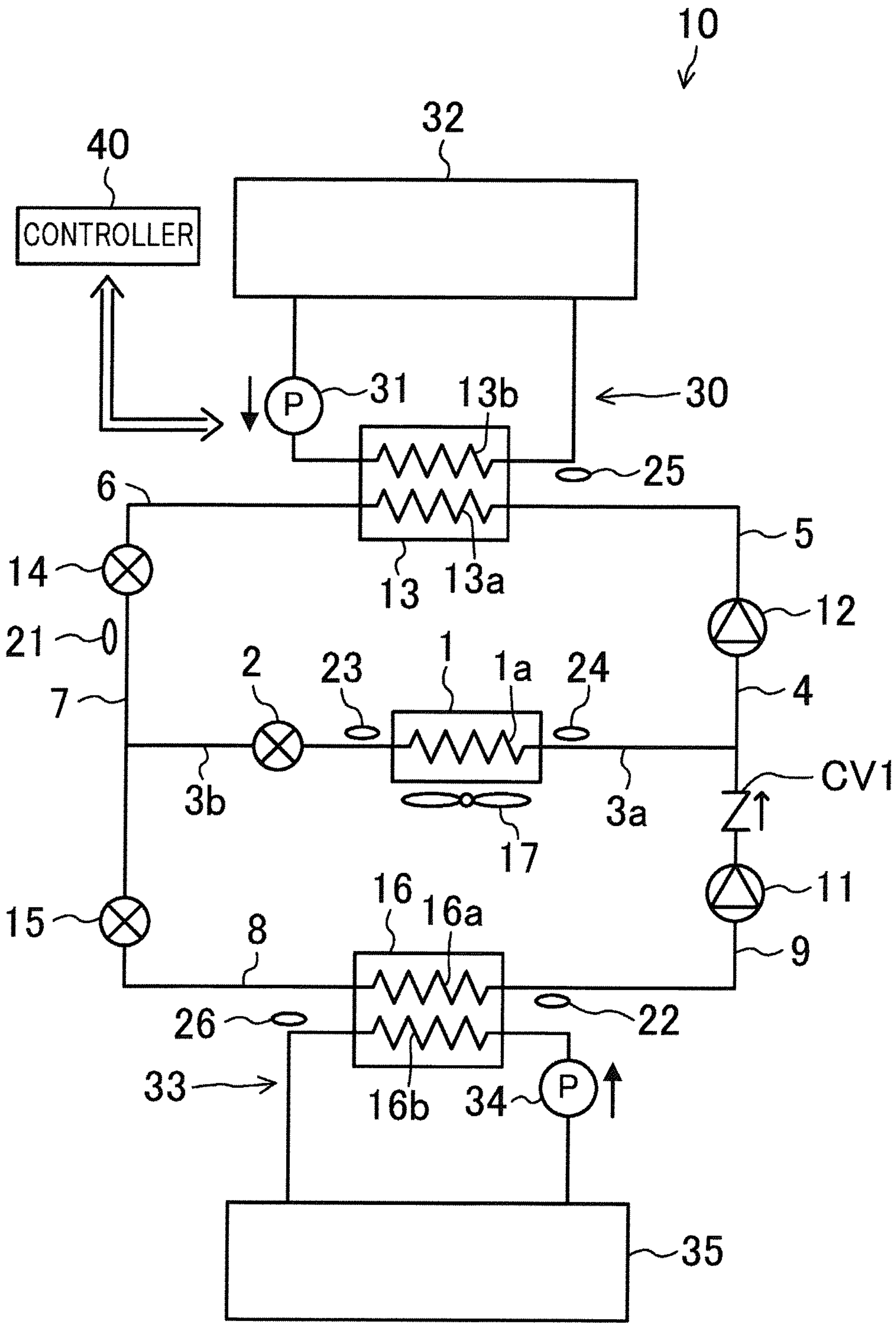
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FIG.2

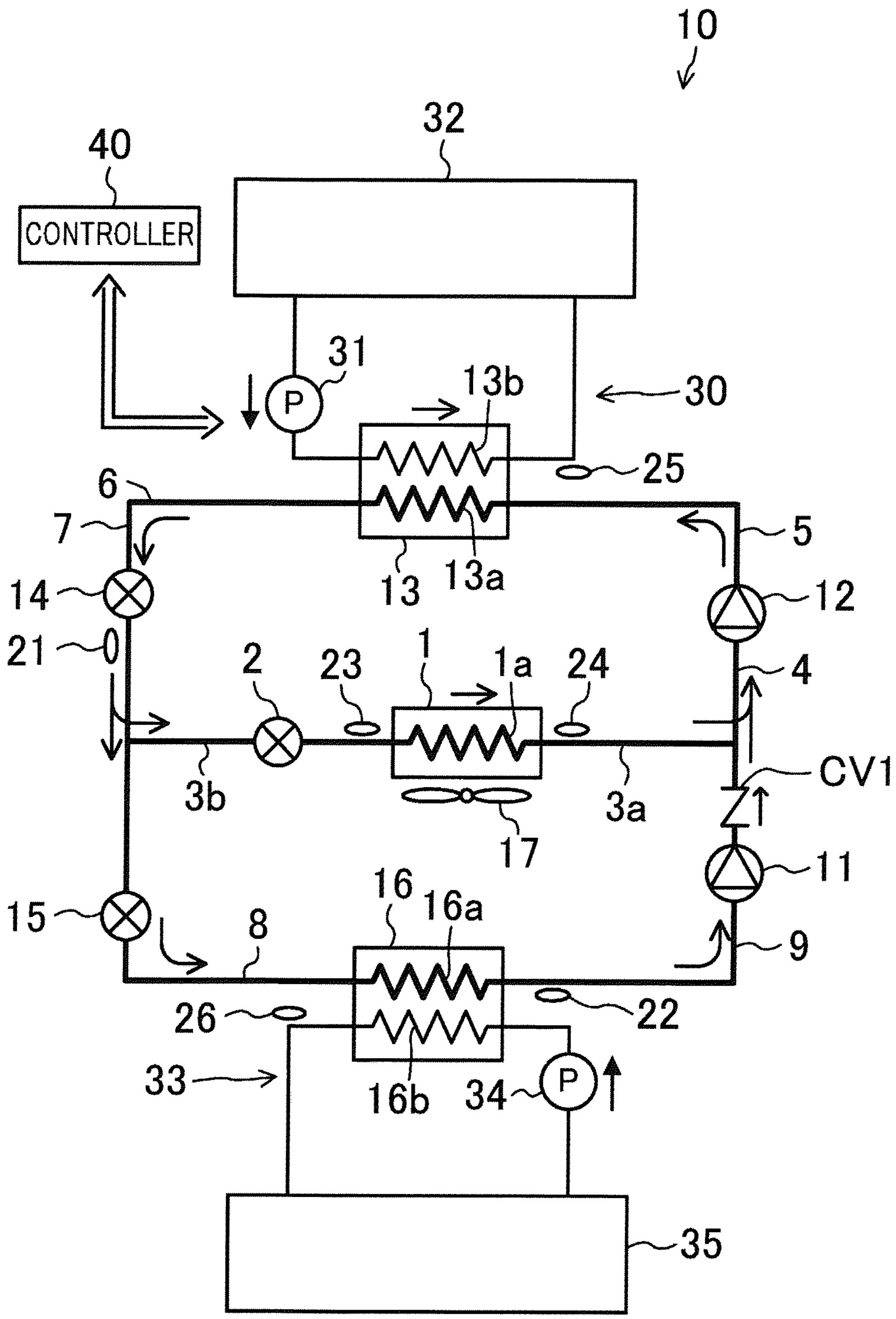
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FIG.3

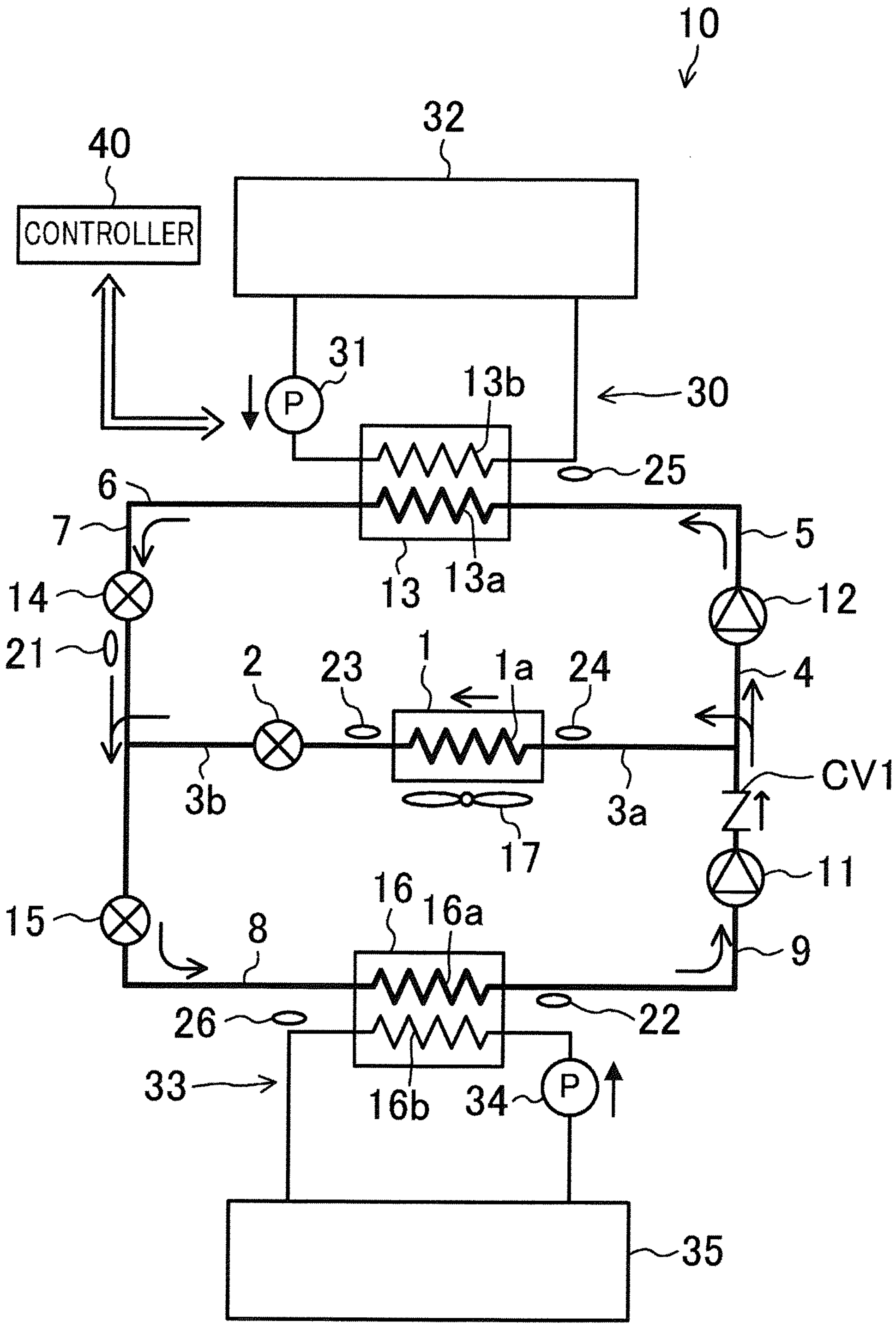
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FIG.4

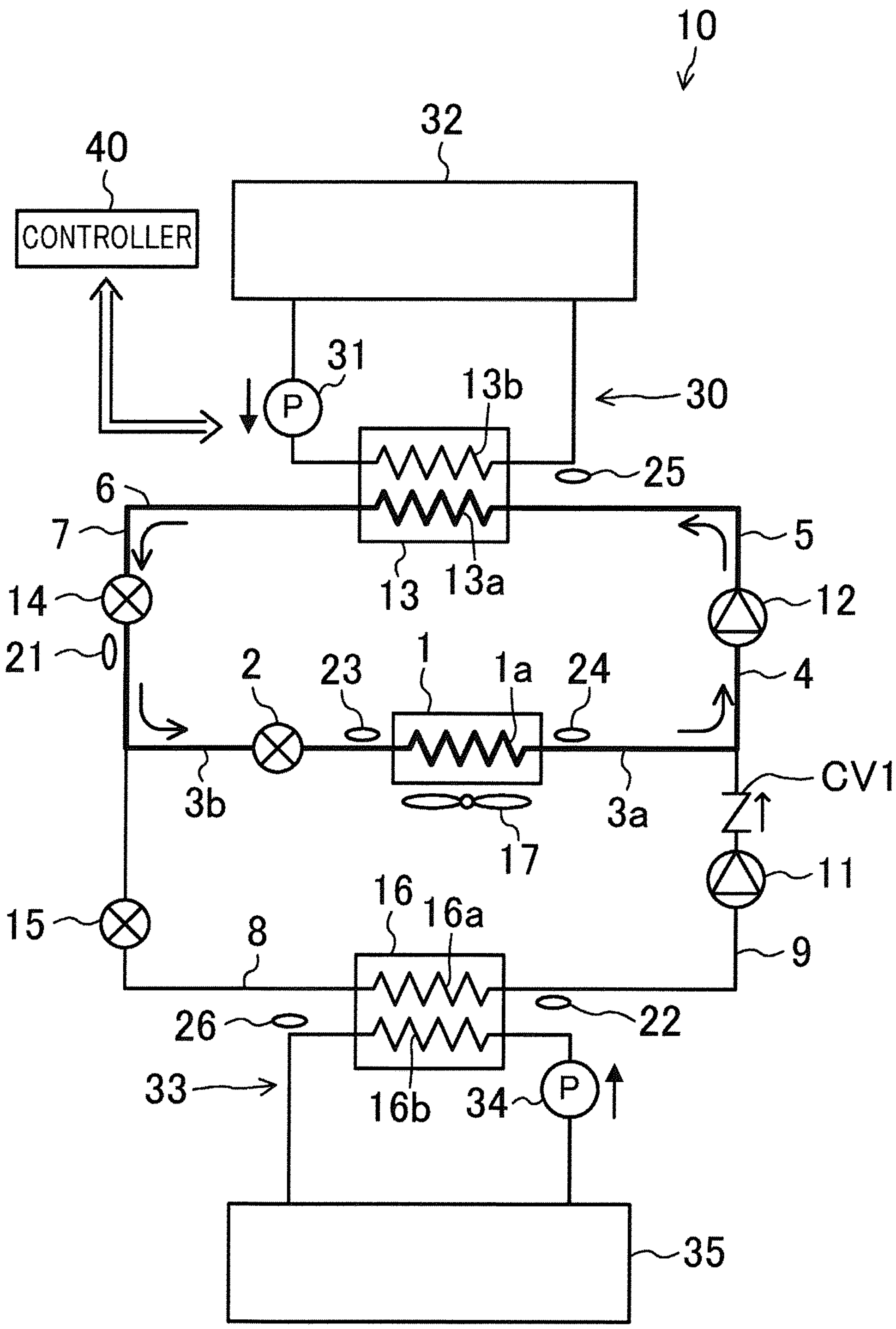
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FIG.5

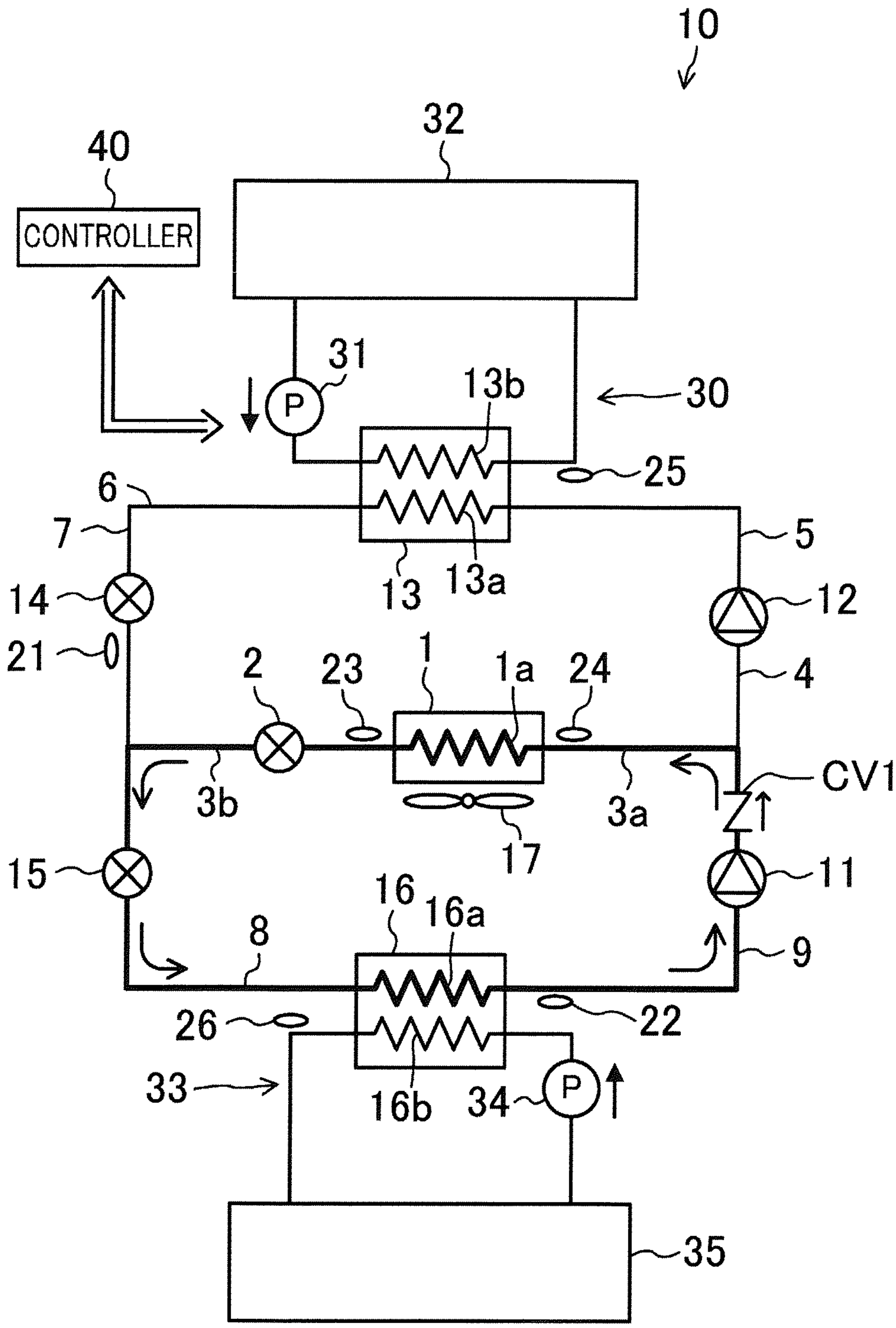
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FIG.6

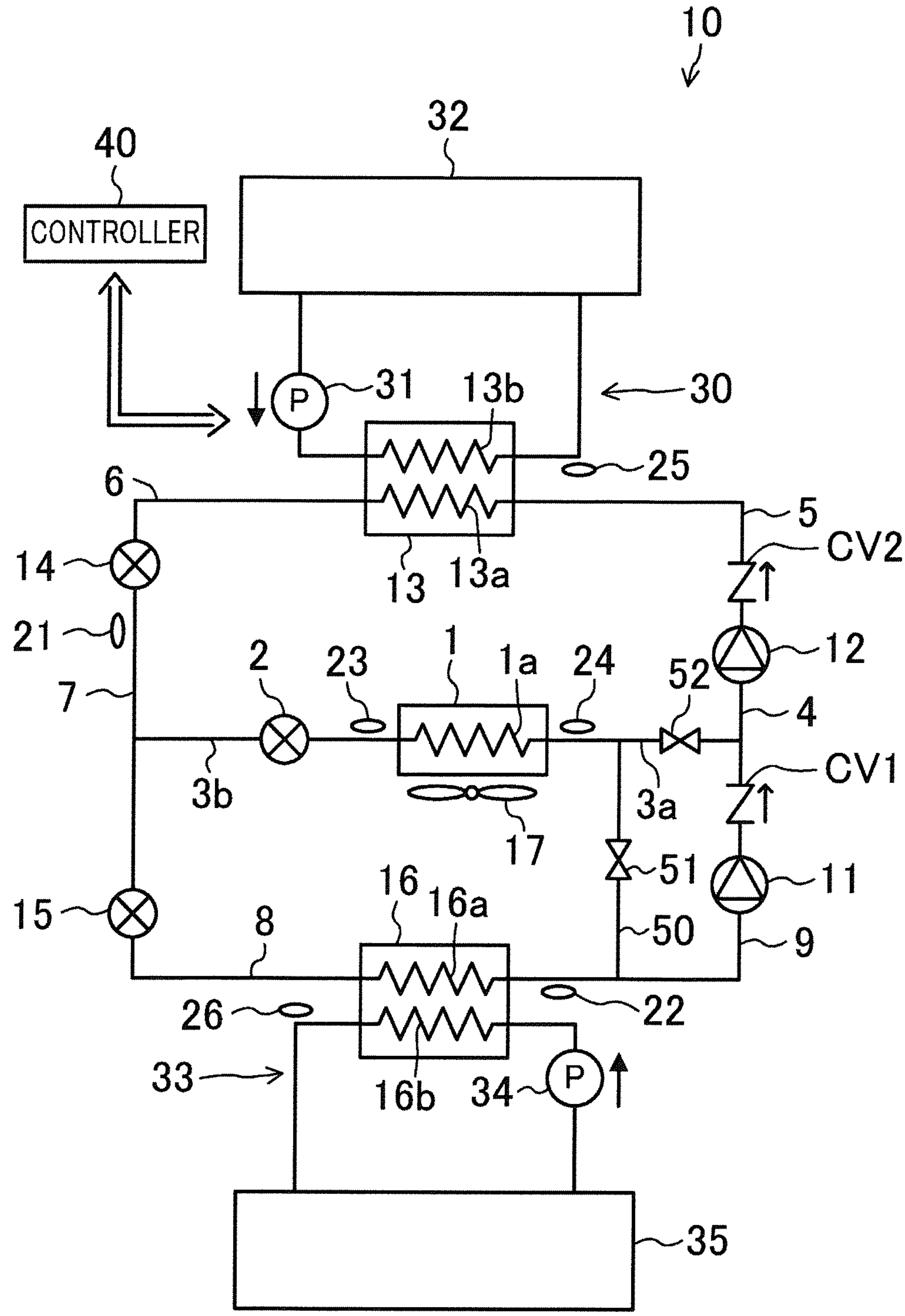
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FIG.7

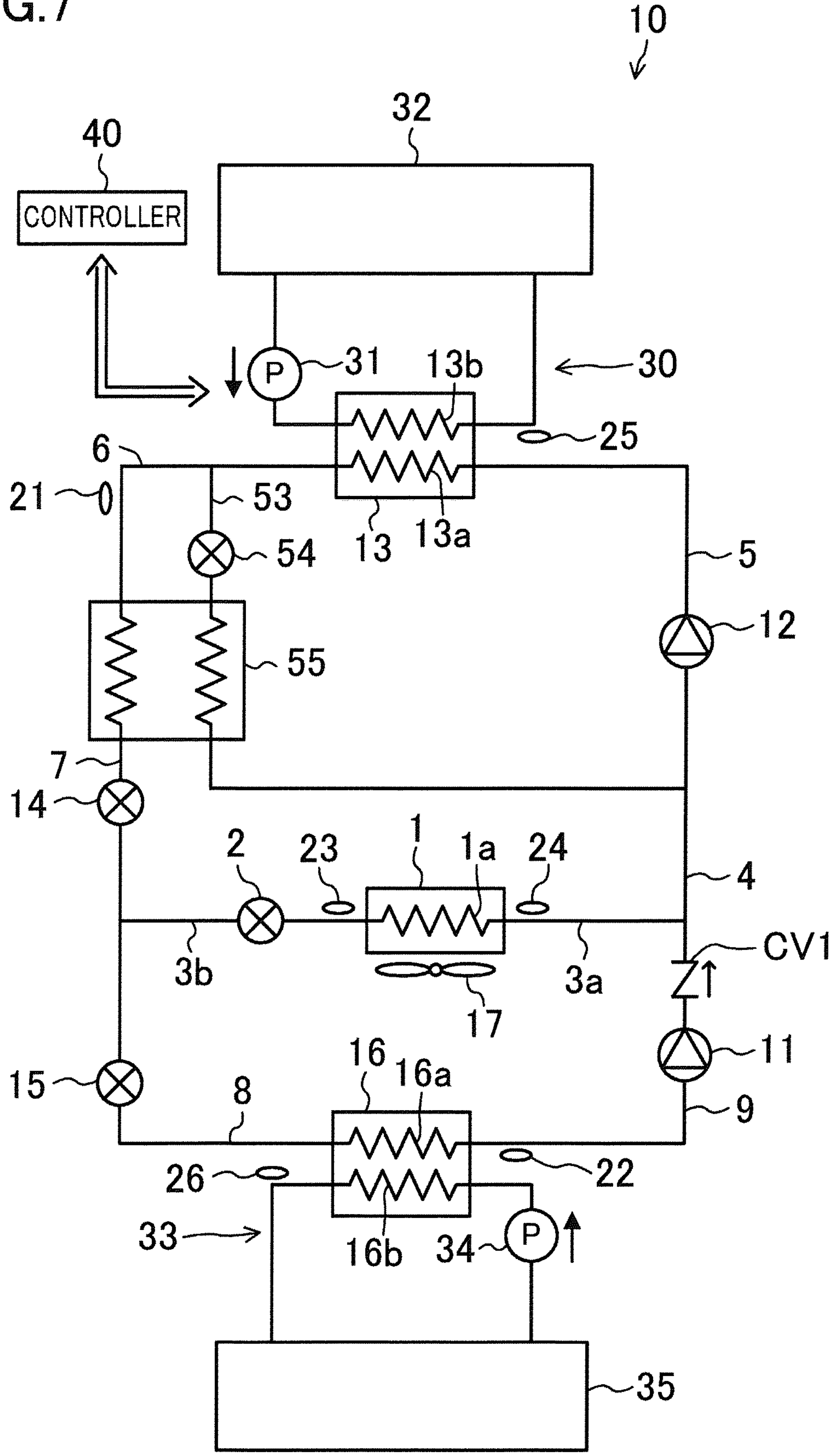
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FIG.8

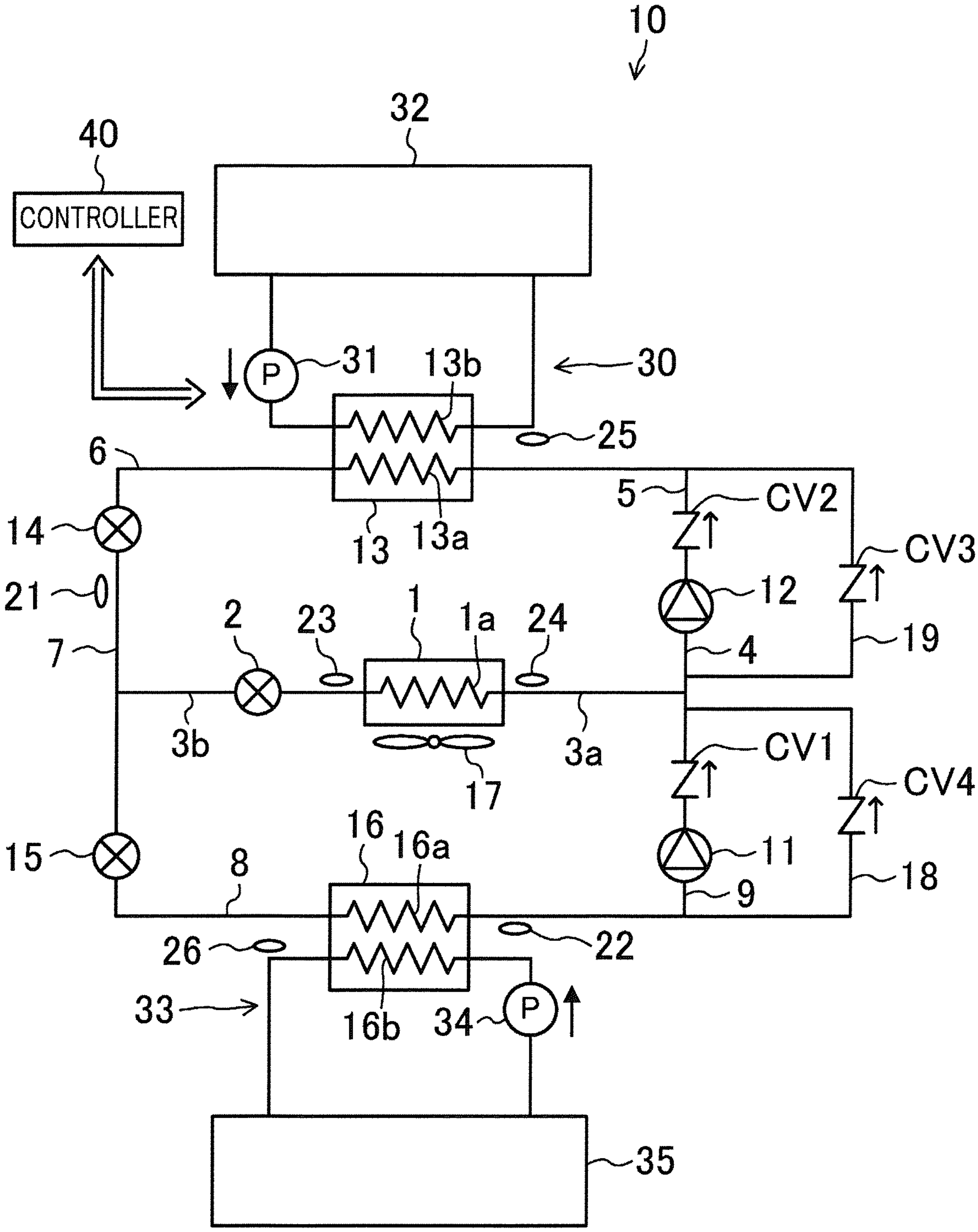
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FIG.9

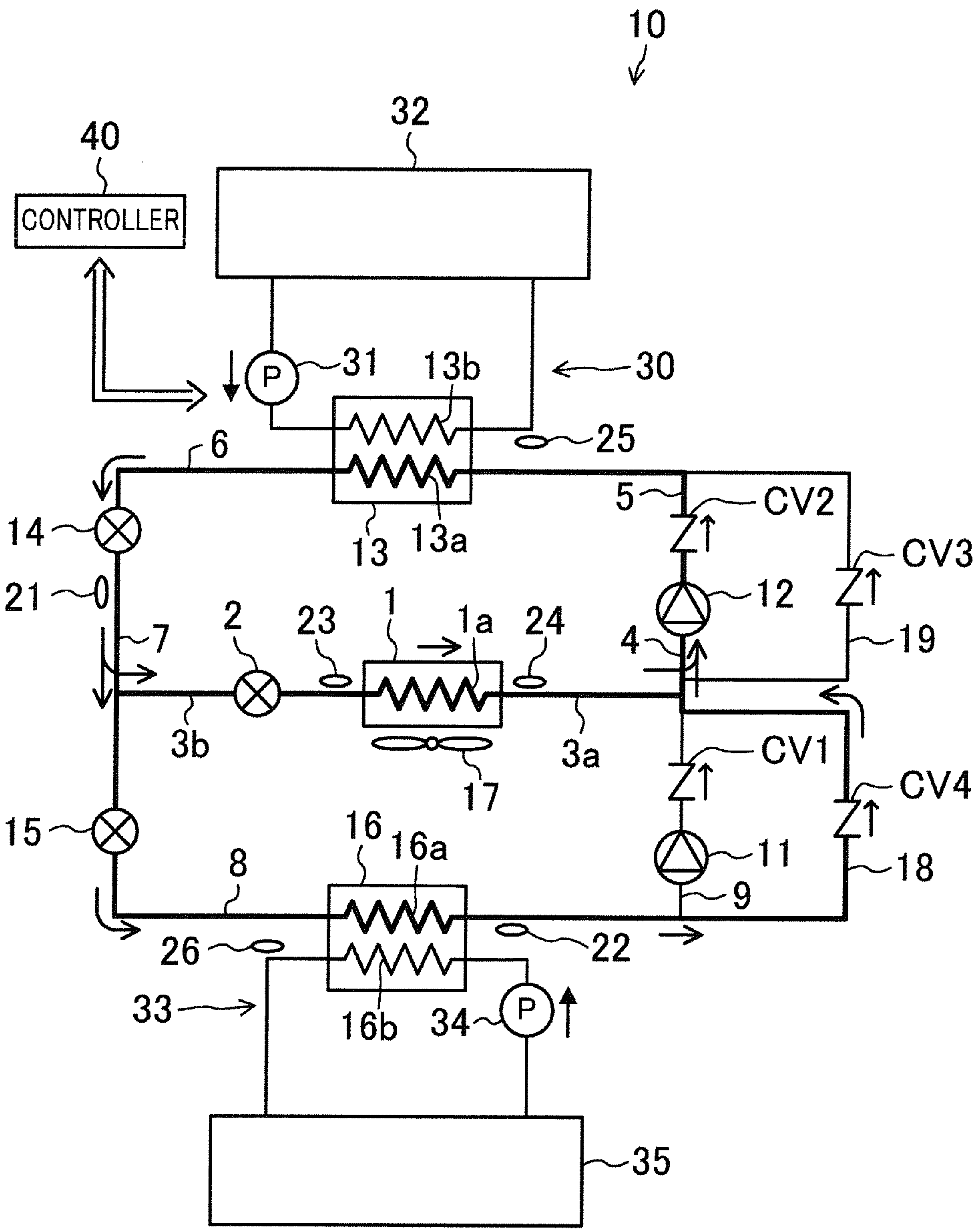
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FIG.10

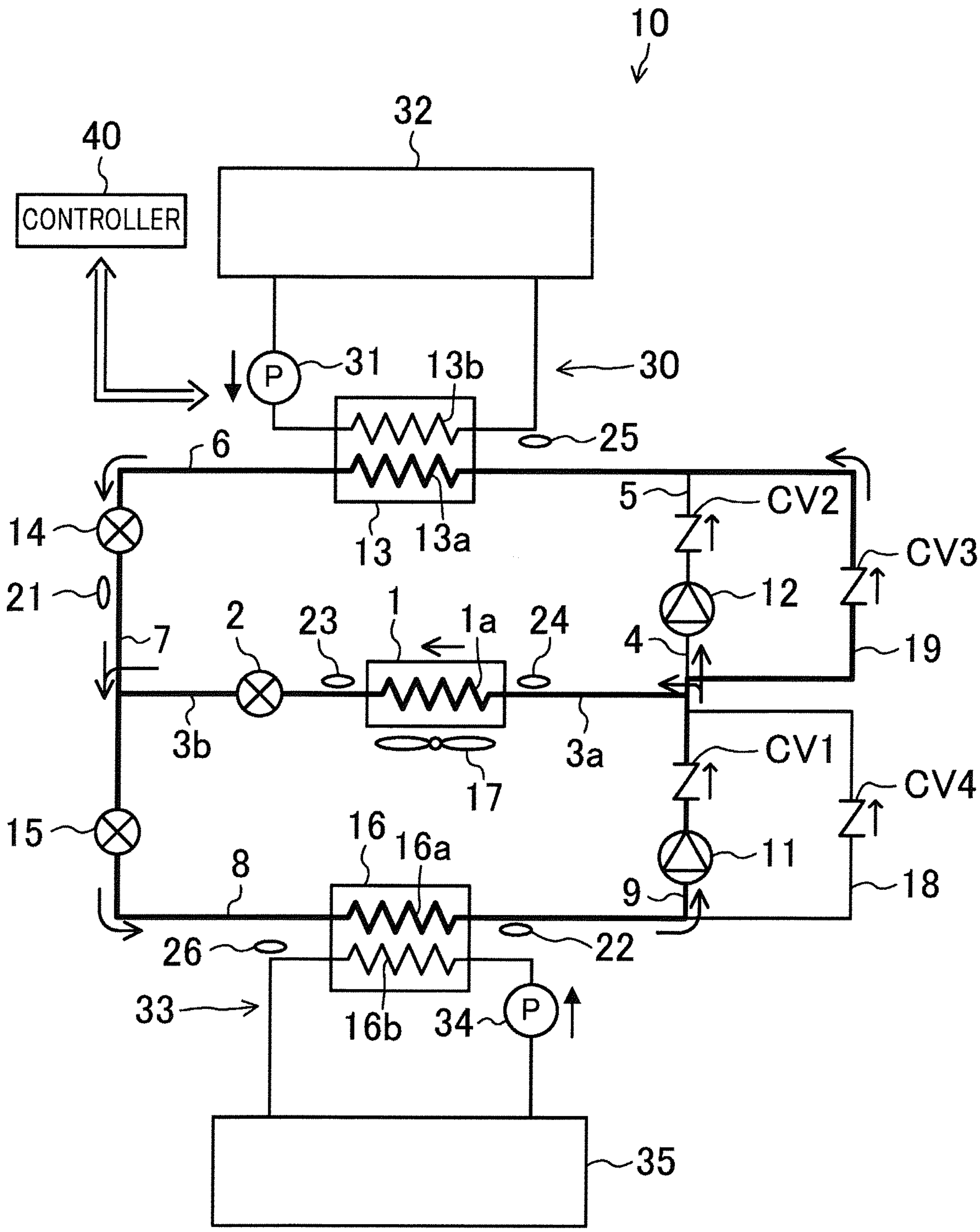
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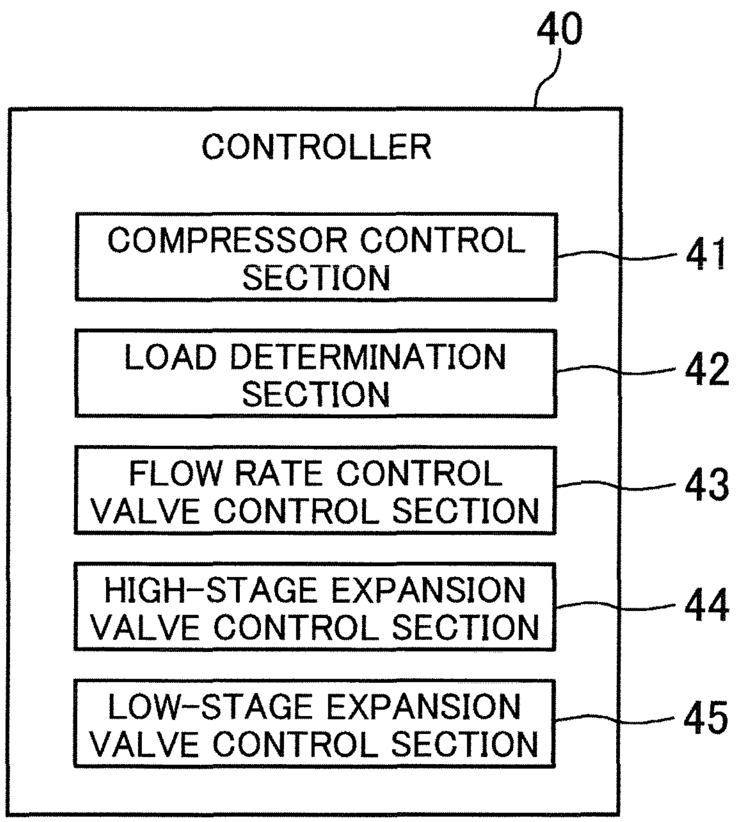
FIG.11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LOAD DETERMINATION SECTION
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FLOW RATE CONTROL VALVE CONTROL SECTION
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LOW-STAGE EXPANSION VALVE CONTROL SECTION
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FIG.12

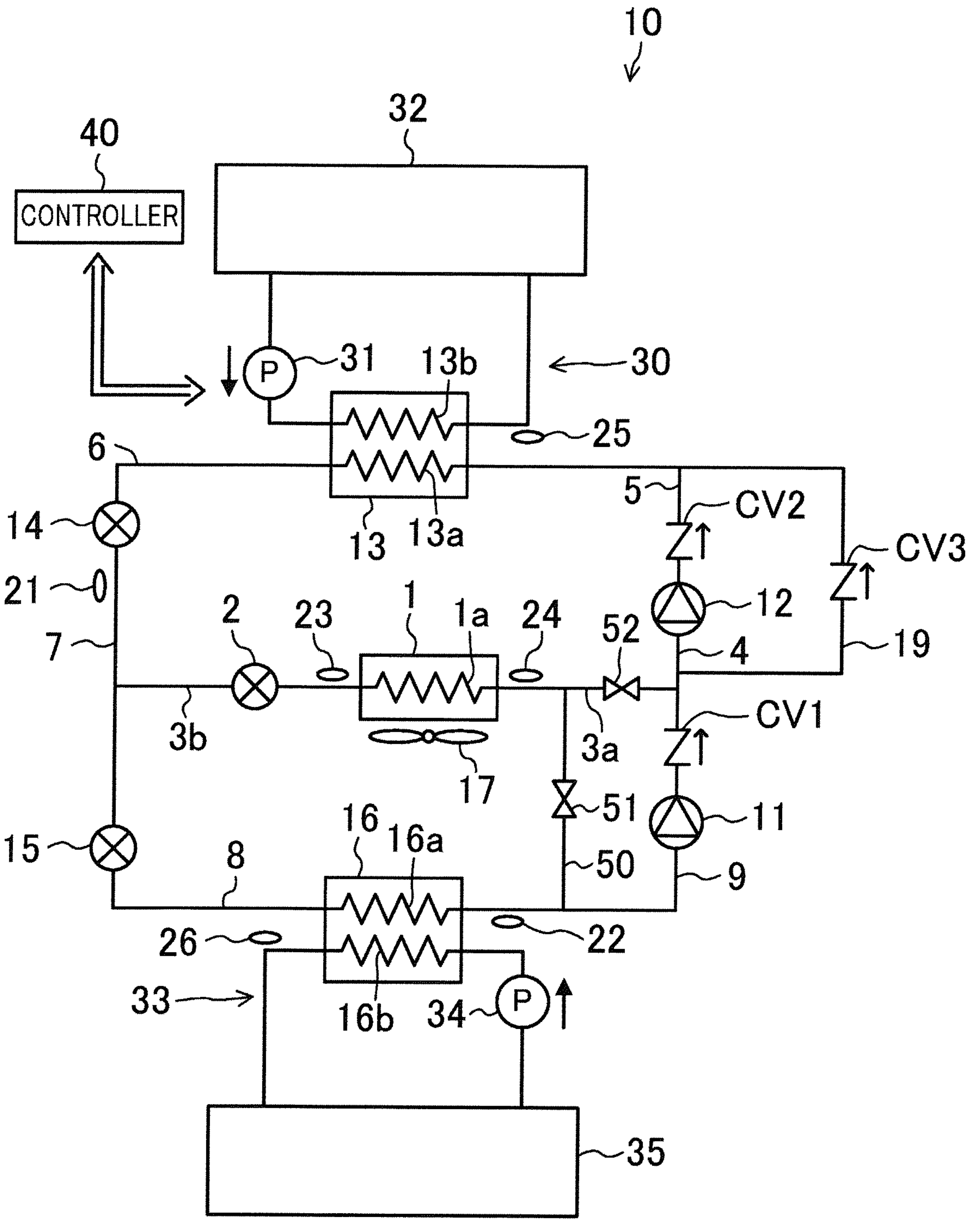
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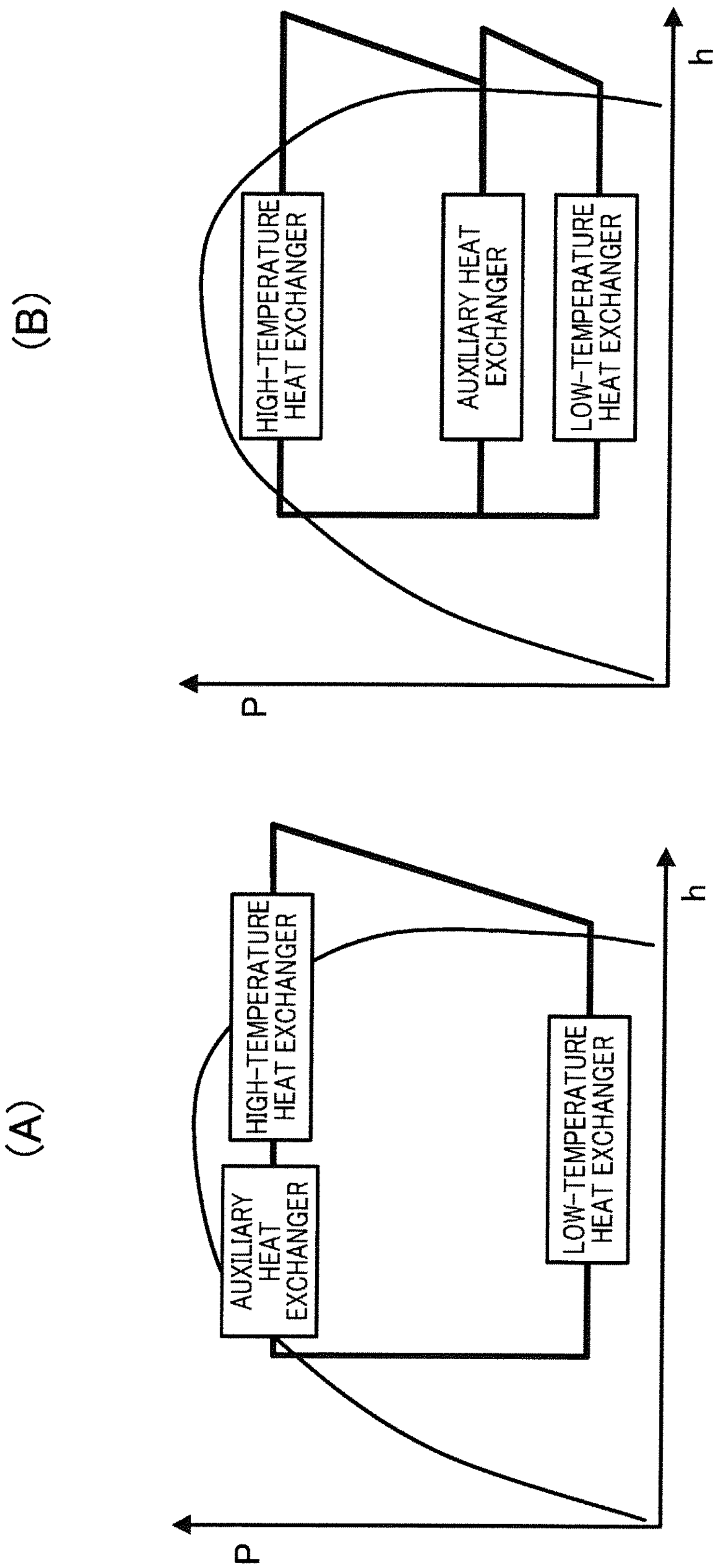
FIG.13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AUXILIARY HEAT EXCHANGER
LOW-TEMPERATURE HEAT EXCHANGER

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2012/006102

A. CLASSIFICATION OF SUBJECT MATTER
F25B1/10(2006.01)i, *F25B29/00*(2006.01)i

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F25B1/10, F25B29/00

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2012
Kokai Jitsuyo Shinan Koho	1971-2012	Toroku Jitsuyo Shinan Koho	1994-2012

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y A	JP 63-87559 A (Toshiba Corp.), 18 April 1988 (18.04.1988), page 3, upper right column, line 5 to page 4, upper right column, line 14; fig. 1 to 4 (Family: none)	1 2,4-10 3
Y A	JP 2005-308375 A (Mitsubishi Electric Corp.), 04 November 2005 (04.11.2005), paragraphs [0031], [0068] (Family: none)	2,4-10 3
Y	JP 2009-229055 A (Daikin Industries, Ltd.), 08 October 2009 (08.10.2009), paragraph [0102]; fig. 6, 8 (Family: none)	4-10

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

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Date of the actual completion of the international search	Date of mailing of the international search report
03 December, 2012 (03.12.12)	11 December, 2012 (11.12.12)

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INTERNATIONAL SEARCH REPORT

International application No. PCT/JP2012/006102

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP 63-25388 A (Coopland Corp.), 02 February 1988 (02.02.1988), page 6, upper right column, line 15 to lower left column, line 10; fig. 1 & US 4787211 A & GB 2192735 A & DE 3716393 A & FR 2598788 A	5-10
Y	JP 2-309157 A (Matsushita Electric Industrial Co., Ltd.), 25 December 1990 (25.12.1990), page 3, lower right column, line 10 to page 4, upper left column, line 3; figures (Family: none)	7-10
Y	JP 58-48824 B2 (Mitsubishi Electric Corp.), 31 October 1983 (31.10.1983), column 3, line 24 to column 4, line 3; fig. 4 (Family: none)	7-10
Y	JP 2010-156493 A (Daikin Industries, Ltd.), 15 July 2010 (15.07.2010), paragraphs [0046], [0049], [0053] (Family: none)	7-10
Y	JP 2009-180492 A (Daikin Industries, Ltd.), 13 August 2009 (13.08.2009), paragraph [0038] (Family: none)	9
A	JP 9-300940 A (Denso Corp.), 25 November 1997 (25.11.1997), paragraphs [0023] to [0034]; fig. 4 to 8 (Family: none)	1-10

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