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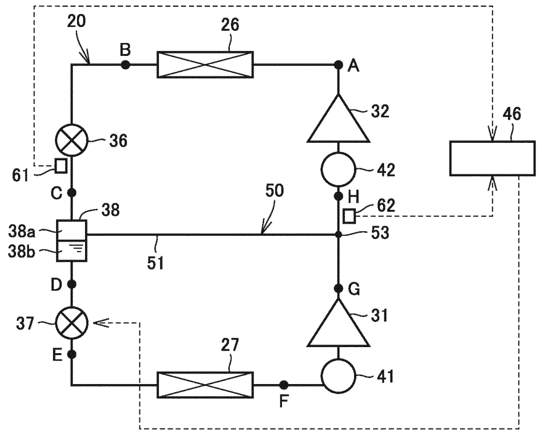
(54) ## HEAT PUMP-TYPE HEATING DEVICE

(57) A heat pump-type heating device includes an indoor heat exchanger (26), a low-pressure side compressor (31), a high-pressure side compressor (32), a first decompressor (36) configured to decompress the refrigerant delivered from the indoor heat exchanger (26), a gas-liquid separator (38) configured to separate the refrigerant delivered from the first decompressor (36) into a gas phase and a liquid phase, a second decompressor (37) connected to the liquid-phase side of the gas-liquid separator (38) and configured to decompress the refrigerant delivered form the gas-liquid separator (38) and deliver the decompressed refrigerant to an outdoor heat exchanger (27), an injection pipeline (51) connected to the gas-phase side of the gas-liquid separator (38) and configured to guide the refrigerant delivered from the gas-liquid separator (38) to a pipeline between the low-pressure side compressor (31) and the high-pressure side compressor (32), and a control unit (46) configured to control a decompression ratio of the refrigerant in the second decompressor (37) so as to render the refrigerant that flows into the high-pressure side compressor (32) into a superheated gas state or a saturated vapor state. Provided with such configuration, it is expected to sufficiently improve the heating ability of the heat pump-type heating device.

FIG.1



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Description

TECHNICAL FIELD

[0001] The present invention generally relates to a heat pump-type heating device, and more particularly relates to a two-stage compression heat pump-type heating device provided with two compressors on a heat pump cycle.

BACKGROUND ART

[0002] As a conventional heat pump-type heating device, for example, Japanese Patent Laid-Open No. 8-210709 (PTD 1) discloses a heat pump air conditioner suitable for cold areas so as to achieve a heating operation even when the outdoor air temperature is for example as low as -20 °C.

[0003] In the heat pump air conditioner disclosed in PTD 1, a scroll compressor, a four-way valve, an indoor air-heat exchanger, a receiver, an outdoor refrigerant control valve and an outdoor air-heat exchanger are sequentially connected to one another via a pipe. A bypass flow path equipped with a refrigerant injection control valve is provided between the scroll compressor and the receiver for injecting liquid refrigerant into the scroll compressor. The refrigerant injection control valve is controlled in accordance with a difference between a temperature at the discharge side of the compressor and a target discharge temperature thereof. The outdoor refrigerant control valve is controlled so that the difference between temperatures obtained by temperature sensors that are respectively provided before and after the outdoor air-heat exchanger is equal to the degree of superheat of the refrigerant at the refrigerant outlet of the outdoor air-heat exchanger.

[0004] Meanwhile, Japanese Patent Laying-Open No. 11-132575 (PTD 2) discloses an air conditioner designed to prevent a liquid refrigerant from being mixed into a gas refrigerant flowing from a gas-liquid separator, which is interposed in a liquid refrigerant pipe, back to the compressor through a gas injection bypass pipe, and thereby prevent the reliability of the compressor from being deteriorated.

[0005] In the air conditioner disclosed in PTD 2, an outdoor heat exchanger and an indoor heat exchanger are sequentially connected to the compressor, thereby forming a refrigerant circulation circuit. The gas-liquid separator is interposed in the liquid refrigerant pipe between the outdoor heat exchanger and the indoor heat exchanger. The gas injection bypass pipe for flowing the gas refrigerant in the gas-liquid separator back to the compressor and an on-off valve for opening or closing the flow path through the bypass pipe are provided between the gas-liquid separator and the suction side of the compressor. The on-off valve is closed as a difference between the discharge temperature of the compressor and the condensing temperature of the refrigerant circulating in the refrigerant circulation path becomes smaller than a reference temperature difference. The reference temperature difference is set larger as the operation frequency of the compressor becomes higher.

[0006] Moreover, Japanese Patent Laying-Open No. 2007-263440 (PTD 3) discloses an air conditioner designed to appropriately adjust an injection amount of the refrigerant into the compressor in the compression process during the heating operation so as to render the air conditioner to operate at a high operating efficiency under a low load and operate at an improved heating ability under a high load.

[0007] The air conditioner disclosed in PTD 3 is provided with an injection pipe configured to inject a part of the refrigerant that flows out from an indoor heat exchanger to the compressor in the compression process via an injection decompressor, a rotational speed control means configured to control the rotational speed of compressor in response to the magnitude of load, and an injection control means configured to control the injection decompressor such that the degree of superheat or the discharge temperature of gas discharged at the outlet of the compressor becomes equal to a target value. The target value is set smaller as the rotational speed of the compressor controlled by the rotational speed control means becomes higher, and is set larger as the rotational speed of the compressor becomes lower.

CITATION LIST

PATENT DOCUMENT

[0008]

PTD 1: Japanese Patent Laying-Open No. 8-210709
PTD 2: Japanese Patent Laying-Open No. 11-132575
PTD 3: Japanese Patent Laying-Open No. 2007-263440

SUMMARY OF INVENTION

TECHNICAL PROBLEM

[0009] As a heat pump-type heating device such as an air conditioner or a water heater, there has been disclosed a two-stage compression heat pump-type heating device provided with two compressors, i.e., a low-pressure side compressor and a high-pressure side compressor, on a heat pump cycle. However, in the two-stage compression heat pump-type heating device, the suction temperature and the discharge temperature of the refrigerant in the high-pressure side compressor may rise to exceed the operating range of the compressor, disadvantageously. To solve such a problem, there has been suggested such a method that an injection pipe is provided to connect a pipeline between the low-pressure side compressor and the high-pressure side compressor and a pipeline between the indoor heat exchanger (condenser) and the outdoor heat exchanger (evaporator) so that a part of the refrigerant flowing in the pipeline between the indoor heat exchanger and the outdoor heat exchanger is injected into the pipeline between the low-pressure side compressor and the high-pressure side compressor through the intermediary of the injection pipe. In this manner, the suction temperature of the refrigerant in the high-pressure side compressor is lowered, thereby achieving an operation with maintained reliability.

[0010] In such a heat pump-type heating device employing the injection pipe, an optimal amount of refrigerant is required to be injected into the pipeline between the low-pressure side compressor and the high-pressure side compressor so as to improve the heating ability. Moreover, in each of the various types of devices employing the injection pipe as disclosed in PTD 1 to PTD 3 described above, the injection of the refrigerant is controlled by the on-off valve or the decompressor provided on the pipeline of the injection pipe. In this case, it is disadvantageous that the heat pump-type heating device cannot be manufactured at low cost.

[0011] The present invention has been accomplished in view of the aforementioned problems, and it is therefore an object of the present invention to provide a heat pump-type heating device having a simple structure and having sufficiently improved heating ability.

SOLUTION TO PROBLEM

[0012] The heat pump-type heating device according to the present invention is provided with a first heat exchanger configured to perform heat exchange between a refrigerant and a heat-receiving fluid, a second heat exchanger configured to perform heat exchange between the refrigerant and outdoor air, a low-pressure side compressor configured to compress the refrigerant delivered from the second heat exchanger, a high-pressure side compressor configured to compress the refrigerant delivered from the low-pressure side compressor and deliver the compressed refrigerant to the first heat exchanger, a first decompressor configured to decompress the refrigerant delivered from the first heat exchanger, a gas-liquid separator configured to separate the refrigerant delivered from the first decompressor into a gas phase and a liquid phase, a second decompressor connected to the liquid-phase side of the gas-liquid separator and configured to decompress the refrigerant delivered from the gas-liquid separator and deliver the decompressed refrigerant to the second heat exchanger, an injection pipeline connected to the gas-phase side of the gas-liquid separator and configured to guide the refrigerant delivered from the gas-liquid separator to a pipeline between the low-pressure side compressor and the high-pressure side compressor, and a control unit configured to control a decompression ratio of the refrigerant in the second decompressor so as to render the refrigerant that flows into the high-pressure side compressor into a superheated gas state or a saturated vapor state.

[0013] According to the heat pump-type heating device thus configured, the refrigerant flowing in the injection pipeline is merged with the gas-phase refrigerant which is discharged from the low-pressure side compressor and has a high temperature and a high pressure so as to render the refrigerant that flows into the high-pressure side compressor into a superheated gas state or a saturated vapor state. Accordingly, it is possible to sufficiently improve the heating ability of the high-pressure side compressor. In this case, since the second decompressor is used to decompress the refrigerant delivered from the gas-liquid separator so as to render the refrigerant flowing into the high-pressure side compressor into a superheated gas state or a saturated vapor state, it is possible to simplify the structure of the heat pump-type heating device.

[0014] Preferably, the heat pump-type heating device is further provided with a first temperature detector provided on the pipeline between the first decompressor and the gas-liquid separator and configured to detect a temperature T1 of the refrigerant, and a second temperature detector provided on the pipeline between the low-pressure side compressor and the high-pressure side compressor and configured to detect a temperature T2 of the refrigerant after having been merged with the refrigerant flowing in the injection pipeline. The control unit controls the decompression ratio of the refrigerant in the second decompressor based on a comparison between temperature T1 of the refrigerant detected by the first temperature detector and temperature T2 of the refrigerant detected by the second temperature detector.

[0015] Preferably, the control unit decreases the decompression ratio of the refrigerant in the second decompressor when a state where temperature T2 of the refrigerant detected by the second temperature detector is equal to temperature T1 of the refrigerant detected by the first temperature detector has lasted for a predetermined time.

[0016] Preferably, the heat pump-type heating device is further provided with a first temperature detector provided on the pipeline between the first decompressor and the gas-liquid separator and configured to detect a temperature T1 of the refrigerant, and a third temperature detector and a pressure detector both provided on the pipeline between the high-pressure side compressor and the first heat exchanger and configured to detect a temperature T3 and a pressure P of the refrigerant, respectively. The control unit determines on a p-h (pressure-specific enthalpy) diagram a specific enthalpy H' as an intersection point between an intermediate pressure line defined according to temperature T1 of the refrigerant detected by the first temperature detector and an isentropic line passing through a point defined according to temperature T3 of the refrigerant detected by the third temperature detector and pressure P of the refrigerant detected by the pressure detector, and controls the decompression ratio of the refrigerant in the second decompressor based on a comparison between specific enthalpy H' and a specific enthalpy H of a saturated vapor in the intermediate pressure line.

[0017] According to the heat pump-type heating device thus configured, temperature T1 detected by the first temperature detector is considered as a temperature at which the refrigerant is in a gas-liquid two-phase state to control the decompression ratio of the refrigerant in the second decompressor.

[0018] Preferably, the heat pump-type heating device is further provided with a first temperature detector provided on the pipeline between the first decompressor and the gas-liquid separator and configured to detect a temperature T1 of the refrigerant, a second temperature detector provided on the pipeline between the low-pressure side compressor and the high-pressure side compressor and configured to detect a temperature T2 of the refrigerant after having been merged with the refrigerant flowing in the injection pipeline, and a fourth temperature detector provided on the injection pipeline and configured to detect a temperature T4 of the refrigerant. When the relationship of T1>T4 is satisfied, the control unit controls the decompression ratio of the refrigerant in the second decompressor based on a comparison between temperature T1 of the refrigerant detected by the first temperature detector and temperature T2 of the refrigerant detected by the second temperature detector, and when the relationship of T1<T4 is satisfied, the control unit controls the decompression ratio of the refrigerant in the second decompressor based on a comparison between temperature T4 of the refrigerant detected by the fourth temperature detector and temperature T2 of the refrigerant detected by the second temperature detector.

[0019] According to the heat pump-type heating device thus configured, when the relationship of T1>T4 is satisfied, temperature T1 of the refrigerant detected by the first temperature detector is considered as the temperature at which the refrigerant is in a gas-liquid two-phase state to control the decompression ratio of the refrigerant in the second decompressor, and when the relationship of T1<T4 is satisfied, temperature T4 of the refrigerant detected by the fourth temperature detector is considered as the temperature at which the refrigerant is in a gas-liquid two-phase state to control the decompression ratio of the refrigerant in the second decompressor.

[0020] Preferably, the heat pump-type heating device is further provided with a buffer unit being provided on the pipeline which is between the low-pressure side compressor and the high-pressure side compressor and in which the refrigerant after having been merged with the refrigerant flowing in the injection pipeline flows and being configured to store liquid refrigerant.

[0021] According to the heat pump-type heating device thus configured, it is possible to render the refrigerant that flows into the high-pressure side compressor into a superheated gas state or a saturated vapor state more certainly.

ADVANTAGEOUS EFFECTS OF INVENTION

[0022] As described in the above, according to the present invention, it is possible to provide a heat pump-type heating device having a simple structure and having sufficiently improved heating ability.

BRIEF DESCRIPTION OF DRAWINGS

[0023]

Fig. 1 is a circuit diagram illustrating a heat pump-type heating device according to a first embodiment of the present invention;
Fig. 2 is a Mollier diagram illustrating a refrigeration cycle by the heat pump-type heating device illustrated in Fig. 1;
Fig. 3 illustrates a flowchart of controlling the state of refrigerant at a suction position of a high-pressure side compressor in the heat pump-type heating device in Fig. 1;
Fig. 4 is a circuit diagram illustrating a modification of the heat pump-type heating device illustrated in Fig. 1;
Fig. 5 is a graph illustrating the variation of heating ability and the variation of COP in accordance with the ratio of an injection amount of the refrigerant relative to the amount of the refrigerant in a gas-liquid separator before being branched;
Fig. 6 is a circuit diagram illustrating a heat pump-type heating device according to a second embodiment of the present invention;
Fig. 7 illustrates a flowchart of controlling the state of refrigerant at a suction position of a high-pressure side compressor in the heat pump-type heating device in Fig. 6;
Fig. 8 is a Mollier diagram for explaining a control to be executed in the heat-pump-type heating device in Fig. 6; and
Fig. 9 is a circuit diagram illustrating a heat pump-type heating device according to a third embodiment of the present invention.

DESCRIPTION OF EMBODIMENTS

[0024] Embodiments of the present invention will be described with reference to the drawings. In the drawings, the same or equivalent members to be referenced below will be assigned with the same reference numbers.

(First Embodiment)

[0025] Fig. 1 is a circuit diagram illustrating a heat pump-type heating device according to a first embodiment of the present invention. With reference to Fig. 1, the heat pump-type heating device of the present embodiment is typically applied to a heat pump-type water heater or a heat pump-type heating system. The heat pump-type heating device includes a refrigeration circuit 20 and an injection circuit 50 as a circuit configuration. A refrigerant such as R410A is enclosed in refrigeration circuit 20 and injection circuit 50.

[0026] Refrigeration circuit 20 extends annularly to form a heat pump cycle. On the path of refrigeration circuit 20, an indoor heat exchanger (condenser) 26 and an outdoor heat exchanger (evaporator) 27 are provided. Indoor heat exchanger 26 performs heat exchange between the refrigerant circulating in the heat pump cycle and a heat-receiving fluid (water or air). Outdoor heat exchanger 27 performs heat exchange between the refrigerant circulating in the heat pump cycle and the outside air (outdoor air).

[0027] On the path of refrigeration circuit 20, a first decompressor 36, a gas-liquid separator 38 and a second decompressor 37 are further provided. First decompressor 36, gas-liquid separator 38, and second decompressor 37 are provided between indoor heat exchanger 26 and outdoor heat exchanger 27. First decompressor 36, gas-liquid separator 38 and second decompressor 37 are arranged in series in the flow direction of the refrigerant in refrigeration circuit 20. On the path of refrigeration circuit 20 from indoor heat exchanger 26 to outdoor heat exchanger 27, first decompressor 36, gas-liquid separator 38 and second decompressor 37 are arranged in the same order as described.

[0028] First decompressor 36 decompresses the refrigerant delivered from indoor heat exchanger 26. First decompressor 36 is provided as a decompression device configured to control the supercooling of the refrigerant in indoor heat exchanger 26. Gas-liquid separator 38 separates the refrigerant delivered from first decompressor 36 into the gas-phase refrigerant and the liquid-phase refrigerant (liquid refrigerant). Gas-liquid separator 38 has a gas-phase refrigerant space 38a for storing the gas-phase refrigerant and a liquid-phase refrigerant space 38b for storing the liquid-phase refrigerant. Second decompressor 37 is connected to liquid-phase refrigerant space 38b of gas-liquid separator 38 via a pipe. Second decompressor 37 decompresses the liquid refrigerant delivered from gas-liquid separator 38. Second decompressor 37 is provided as a decompression device configured to control the degree of superheat of the refrigerant in outdoor heat exchanger 27 and an amount of injection refrigerant provided by injection circuit 50 which will be described later. In the present embodiment, expansion valves are used as first decompressor 36 and second decompressor 37.

[0029] On the path of refrigeration circuit 20, a low-pressure side compressor 31 and a high-pressure side compressor 32 are further provided. Low-pressure side compressor 31 and high-pressure side compressor 32 are provided between outdoor heat exchanger 27 and indoor heat exchanger 26. Low-pressure side compressor 31 and high-pressure side compressor 32 are arranged in series in the flow direction of the refrigerant in refrigeration circuit 20. On the path of refrigeration circuit 20 from outdoor heat exchanger 27 to indoor heat exchanger 26, low-pressure side compressor 31 and high-pressure side compressor 32 are arranged in the same order as described. Low-pressure side compressor 31 compresses the refrigerant, which is delivered from outdoor heat exchanger 27 and has a low pressure, to an intermediate pressure. High-pressure side compressor 32 further compresses the refrigerant which is delivered from low-pressure side compressor 31 and has an intermediate pressure to a high pressure.

[0030] In the present embodiment, low-pressure side compressor 31 is a displacement-variable compressor capable of controlling the discharge displacement of the refrigerant (for example, a compressor driven by a inverter capable of changing the rotational speed), and high-pressure side compressor 32 is a constant speed compressor. It should be noted that at least one of low-pressure side compressor 31 and high-pressure side compressor 32 may be a displacement-variable compressor, or may be a combination of a low-pressure side compressor running at constant speed and a high-pressure side compressor having a variable displacement, or may be a combination of a low-pressure side compressor having a variable displacement and a high-pressure side compressor having a variable displacement. Note that in the case where the low-pressure side compressor is a displacement-variable compressor, it provides a wider operable range under a high load.

[0031] Injection circuit 50 is composed of an injection pipeline 51 where the refrigerant flows. Injection pipeline 51 is provided to guide a part of the gas refrigerant separated into gas-phase refrigerant space 38a of gas-liquid separator 38 to refrigeration circuit 20 between low-pressure side compressor 31 and high-pressure side compressor 32.

[0032] More specifically, injection pipeline 51 is provided with one end thereof connected to gas-phase refrigerant space 38a of gas-liquid separator 38 and the other end thereof connected to refrigeration circuit 20 between low-pressure side compressor 31 and high-pressure side compressor 32. Injection pipeline 51 has a refrigerant inlet connected to gas-phase refrigerant space 38a of gas-liquid separator 38, and has a refrigerant outlet connected to refrigeration circuit 20 between low-pressure side compressor 31 and high-pressure side compressor 32.

[0033] In the present embodiment, no on-off valve configured to allow or block the flowing of the refrigerant or no flow-regulating valve capable of adjusting the flow rate of the refrigerant is provided on injection pipeline 51.

[0034] A buffer unit 41 and a buffer unit 42 are further provided on the path of refrigeration circuit 20. Each of buffer unit 41 and buffer unit 42 is constituted by an accumulator capable of accumulating therein liquid refrigerant. Buffer 41 is provided on the path of refrigeration circuit 20 between outdoor heat exchanger 27 and low-pressure side compressor 31. Buffer unit 42 is provided on the path of refrigeration circuit 20 between low-pressure side compressor 31 and high-pressure side compressor 32. In the case where injection pipeline 51 is connected to refrigeration circuit 20 at a connection portion 53, buffer unit 42 is provided between connection portion 53 and high-pressure side compressor 32. Buffer unit 41 and buffer unit 42 are provided to prevent the reliability of the compressors from being deteriorated due to the introduction of liquid refrigerant into low-pressure side compressor 31 and high-pressure side compressor 32.

[0035] Fig. 2 is a Mollier diagram illustrating a refrigeration cycle by the heat pump-type heating device illustrated in Fig. 1.

[0036] The Mollier diagram is also referred to as a P-h diagram with the vertical axis representing the pressure [MPa] and the horizontal axis representing the specific enthalpy [kJ/kg]. The Mollier diagram depicts refrigerant-specific properties such as pressure, specific enthalpy, temperature, phase state, enthalpy and specific volume of the refrigerant used in the refrigeration cycle. The refrigerant states denoted by A to H in Fig. 2 correspond to the refrigerant states denoted by A to H in Fig. 1, respectively.

[0037] With reference to Figs. 1 and 2, firstly, the gas refrigerant (state A) discharged from high-pressure side compressor 32 flows into indoor heat exchanger (condenser) 26 and is condensed into a high-temperature liquid refrigerant (state B). After the high-temperature liquid refrigerant passes through first decompressor 36, the pressure and the temperature of the liquid refrigerant are decreased (state C). If a device such as an internal heat exchanger is not provided, the refrigerant is generally in a gas-liquid two-phase state.

[0038] Next, the refrigerant flows into gas-liquid separator 38 and is separated into the gas phase and the liquid phase. After the separated liquid refrigerant (state D) passes through second decompressor 37, the pressure and the temperature of the refrigerant are further decreased (state E). Thereafter, when the refrigerant passes through outdoor heat exchanger 27, the refrigerant absorbs heat from the outside air to evaporate (state F). The refrigerant in state F flows into low-pressure side compressor 31 and is compressed to have an intermediate pressure (state G).

[0039] Meanwhile, the gas-phase refrigerant (injection refrigerant) separated into gas-phase refrigerant space 38a of gas-liquid separator 38 passes through injection pipeline 51, and is then merged with the refrigerant discharged from low-pressure side compressor 31. Since the temperature of the injection refrigerant is lower than the temperature of the refrigerant discharged from low-pressure side compressor 31, the temperature of the refrigerant after having been merged with the injection refrigerant is decreased (state H).

[0040] Although during the heating operation, when the outside air temperature becomes low, the evaporation pressure is decreased, and thereby the compression ratio is increased, by adding the injection refrigerant into the refrigerant having an intermediate pressure at a stage posterior to the compression process performed by low-pressure side compressor 31 and prior to the compression process performed by high-pressure side compressor 32 so as to increase the flow rate of the refrigerant, it is possible to secure the heating (room-warming) ability without causing the discharge temperature to rise abnormally. Thus, with the effect provided by the injection refrigerant, even when the outside air temperature is for example as low as about - 20 °C, it is possible to obtain sufficient heating ability.

[0041] The heat pump-type heating device further includes a control unit 46, a temperature detector 61 serving as a first temperature detector, and a temperature detector 62 serving as a second temperature detector.

[0042] Temperature detector 61 is provided between first decompressor 36 and gas-liquid separator 38, and specifically, temperature detector 61 is provided at a discharge position of first decompressor 36. Temperature detector 61 detects a temperature T1 of the refrigerant which is discharged from first decompressor 36 and flows into gas-liquid separator 38. Temperature detector 62 is provided between low-pressure side compressor 31 and high-pressure side compressor 32, and specifically, temperature detector 62 is provided at a position between connection portion 53 and high-pressure side compressor 32, and more specifically, temperature detector 62 is provided at a position between connection portion 53 and buffer unit 42. Temperature detector 62 detects a temperature T2 of the refrigerant which is discharged from low-pressure side compressor 31 and merged with the refrigerant flowing in injection pipeline 51. Temperature detector 62 detects temperature T2 of the refrigerant at a suction position of high-pressure side compressor 32.

[0043] Control unit 46 controls the decompression ratio of the refrigerant in second decompressor 37 based on a comparison between temperature T1 of the refrigerant detected by temperature detector 61 and temperature T2 of the refrigerant detected by temperature detector 62. Specifically, control unit 46 controls the decompression ratio of the refrigerant in second decompressor 37 based on a difference between temperature T1 of the refrigerant detected by temperature detector 61 and temperature T2 of the refrigerant detected by temperature detector 62.

[0044] The heat-pump-type heating device according to the present embodiment, based on temperature T1 of the refrigerant detected by temperature detector 61 and temperature T2 of the refrigerant detected by temperature detector 62, controls the opening degree of second decompressor 37 so as to render the refrigerant at the suction position of high-pressure side compressor 32 into a state between the superheated gas state containing the gas phase only and the saturated vapor state. Thereby, the refrigerant to be sucked into high-pressure side compressor 32 can be maintained at a desired state regardless of the magnitude of the heating load or the rotational speed of the compressor so as to improve the heating ability of the heat pump-type heating device or to maintain high the COP (Coefficient Of Performance).

[0045] Furthermore, in the heat pump type heating device according to the present embodiment, second decompressor 37 provided at the upstream side of outdoor heat exchanger 27 in the flow direction of the refrigerant during the heating operation is used instead of providing, on injection pipeline 51, a device configured to control the state of the refrigerant at the suction position of high-pressure side compressor 32. Accordingly, it is possible to obtain an effect comparable to that in the case where a decompressor is provided on the injection pipeline to directly control the flow rate of the injection refrigerant. In addition, since no on-off valve or decompressor needs to be disposed on the injection pipeline, the device can be constructed at low cost.

[0046] The following specifically describes the above-described method of controlling the state of the refrigerant at the suction position of high-pressure side compressor 32. First, since the rotational speed of a compressor is a manipulated variable which can be used to adjust the heating ability most directly, the rotational speed of low-pressure side compressor 31 having a variable displacement is controlled in accordance with a load. For example, the rotational speed of low-pressure side compressor 31 is increased or decreased in accordance with a deviation between a target heating temperature set by a user or a target heating temperature set in advance in the device and a measured heating temperature.

[0047] Fig. 3 illustrates a flowchart of control over the state of the refrigerant at the suction position of the high-pressure side compressor in the heat pump-type heating device illustrated in Fig. 1. The control flow illustrated in the figure is performed by control unit 46.

[0048] With reference to Figs. 1 and 3, prior to the control over the state of the refrigerant at the suction position of high-pressure side compressor 32, at the start of operation, the following controls are performed: the control over the rotational speed of low-pressure side compressor 31; the control over the supercooling at the outlet of indoor heat exchanger 26 through the adjustment of the opening degree of first decompressor 36; and the control over the degree of superheating at the outlet of outdoor heat exchanger 27 through the adjustment of the opening degree of second decompressor 37. The enclosed refrigerant is configured in such a manner that the injection refrigerant is brought into the gas phase state when the series of controls are completed.

[0049] In the control over the state of the refrigerant at the suction position of high-pressure side compressor 32, firstly, temperature T1 of the refrigerant between first decompressor 36 and gas-liquid separator 38 is detected by temperature detector 61, temperature T2 of the refrigerant at the suction position of high-pressure side compressor 32 is detected by temperature detector 62, and the detected temperatures T1 and T2 are stored in control unit 46 (S101).

[0050] Next, a target refrigerant temperature $T2_{SP}$ at the suction position of high-pressure side compressor 32 is determined (S102).

[0051] The refrigerant in refrigeration circuit 20 between first decompressor 36 and gas-liquid separator 38 where temperature detector 61 is provided is in the gas-liquid two-phase state. Since no decompressor is present between the position where temperature detector 61 is provided and the position where temperature detector 62 is provided, the refrigerant pressures at both positions are identical to each other. In addition, when the refrigerant is in an intermediate state between the saturated vapor state and the gas-liquid two-phase state, the degree of dryness of the refrigerant will change in accordance with the ratio between the gas-phase refrigerant and the liquid-phase refrigerant, but the temperature of the refrigerant remains constant. Thus, in the case of rendering the refrigerant at the suction position of the high-pressure side compressor 32 into the saturated vapor state, target refrigerant temperature $T2_{SP}$ may be set equal to T1.

[0052] In the present embodiment, in order to render the refrigerant at the suction position of high-pressure side compressor 32 into a superheated gas state which is close to the saturated vapor state, target refrigerant temperature $T2_{SP}$ is set so that $T2_{SP}= T1+\alpha$ (α is any predetermined value) (S102). By way of example, α is set to 5 °C. Preferably, α is set greater than 0 °C and equal to or less than 10 °C, and more preferably, α is set greater than 0 °C and equal to or less than 5 °C.

[0053] Thereafter, temperature T2 is compared with target refrigerant temperature $T2_{SP}$ (S103). When the relationship of $T2>T2_{SP}$ ($T2-T1>\alpha$) is satisfied, it can be considered that the refrigerant at the suction position of high-pressure side compressor 32 has been shifted to the side of the superheating state further than the target state, the opening degree of second decompressor 37 is decreased (i.e., increase the compression ratio of the refrigerant in second decompressor 37) (S104). The amount of the liquid refrigerant in gas-liquid separator 38 is increased, and consequently the liquid refrigerant overflows from gas-liquid separator 38 into injection pipeline 51. Accordingly, the refrigerant in injection pipeline 51 changes from the gas phase state into the gas-liquid two-phase state, and thus, the flow density of the refrigerant flowing from injection pipeline 51 into the pipeline between low-pressure side compressor 31 and high-pressure side compressor 32 increases. On the other hand, when the relationship of $T2\leq T2_{SP}$ ($T2-T1\leq\alpha$) is satisfied, it can be considered that the refrigerant at the suction position of high-pressure side compressor 32 has been shifted to the side of the saturated vapor state further than the target state, the opening degree of second decompressor 37 is increased (i.e., decrease the decompression ratio of the refrigerant in second decompressor 37) (S105).

[0054] The heat pump-type heating device remains idle for t seconds after the operation of second decompressor 37 (S106). Thereafter, the step of detecting temperature T1 and temperature T2 (S101), the step of S102 and the subsequent steps are repeated so as to maintain the refrigerant at the suction position of high-pressure side compressor 32 to the desired state.

[0055] In the step of S 102 as described above, in order to render the refrigerant at the suction position of high-pressure side compressor 32 into the saturated vapor state, in the case where target refrigerant temperature $T2_{SP}$ is set so that $T2_{SP}$= T1, decreasing the opening degree of second decompressor 37 will increase the ratio of the liquid refrigerant in injection pipeline 51. However, since the temperature of the refrigerant will not change even if the refrigerant is rendered from the saturated vapor state into the gas-liquid two-phase state, the temperature of the refrigerant at the suction position of high-pressure side compressor 32 will not change despite the introduction of the liquid refrigerant from injection pipeline 51. In such a case, as a method of setting the lower limit of the opening degree of second decompressor 37, when the state of $T2_{SP}$(T1)= T2 has lasted for a predetermined time of β seconds, it is determined that the refrigerant has been transferred across the saturated vapor state into the gas-liquid two-phase state. According to the method, after the state of $T2_{SP}$(T1)= T2 has lasted for β seconds, increasing the opening degree of second decompressor 37 will transfer the refrigerant at the suction position of high-pressure side compressor 32 back into the superheating state. By way of example, β is set to 10 seconds.

[0056] Fig. 4 is a circuit diagram illustrating a modification of the heat pump-type heating device in Fig. 1. With reference to Fig. 4, in the heat pump-type heating device according to the present modification, on the path of refrigeration circuit 20, an internal heat exchanger 43 is further provided. Internal heat exchanger 43 is provided between indoor heat exchanger 26 and first decompressor 36. Injection pipeline 51 is provided so as to pass through internal heat exchanger 43. Internal heat exchanger 43 performs heat exchange between the refrigerant flowing out from indoor heat exchanger 26 and the refrigerant flowing in injection pipeline 51.

[0057] According to the abovementioned configuration, when the refrigerant in liquid phase flows into injection pipeline 51, the liquid-phase refrigerant is heated by internal heat exchanger 43 to vaporize, and as a result, the flow rate of the injection refrigerant is increased, making it possible to improve the heating ability.

[0058] Hereinafter, the description will be carried on explaining why the improvement in the heating ability can be achieved in the heat pump-type heating device according to the present embodiment.

[0059] The supply of the injection refrigerant can increase the heating ability by increasing the flow rate of the refrigerant at the heating side and increase the limit of the operating pressure ratio of high-pressure side compressor 32 by decreasing the discharge temperature of high-pressure side compressor 32. For the purpose of increasing the flow rate of the refrigerant at the heating side, the injection refrigerant in the gas-liquid two-phase state is more effective than the injection refrigerant in the gas phase state. However, when the injection refrigerant in the gas-liquid two-phase state is supplied too much, the COP may be deteriorated and the reliability of the compressor may be decreased due to liquid compression. Hence, it is preferable to perform injection to such an extent that the liquid phase in the injection refrigerant is phase-changed into a saturated vapor state by the high-temperature gas refrigerant discharged from low-pressure side compressor 31 and the refrigerant sucked into high-pressure side compressor 32 is at the saturated vapor state.

[0060] Therefore, in the present embodiment, the opening degree of second decompressor 37 is adjusted on the basis of temperature T1 of the refrigerant between first decompressor 36 and gas-liquid separator 38 and temperature T2 of the refrigerant at the suction position of high-pressure side compressor 32 to keep the refrigerant at the suction position of high-pressure side compressor 32 at the saturated vapor state to a superheated gas state close to the saturated vapor state so as to maintain the cycle involving the increased refrigerant flow rate at the heating side. When the condensation temperature, the evaporating temperature, and the rotational speed of the compressor are in the same conditions, it can be said that the major factor that determines the ability of the cycle is the suction pressure of high-pressure side compressor 32 (state H in Fig. 2). As the flow rate of the injection refrigerant increases, the pressure of the sucked refrigerant becomes higher, and the density thereof becomes greater, and thereby, the flow rate of the refrigerant in indoor heat exchanger 26 is increased. In the present embodiment, by setting the suction pressure at a value comparable to the conventional value, increasing the flow rate of the refrigerant at the heating side can achieve the heating ability as good as or better than the conventional heating ability.

[0061] In order to render the refrigerant at the suction position of high-pressure side compressor 32 into the saturated vapor state, the temperature of the refrigerant at the suction position of high-pressure side compressor 32 should be set to the saturated vapor temperature relative to the pressure at the same position. Thus, it is required to set the refrigerant at the suction position of high-pressure side compressor 32 to the saturated vapor temperature, and according to the present embodiment, it is easy to set the refrigerant at the suction position of high-pressure side compressor 32 to the saturated vapor temperature by using temperature T1 of the refrigerant between first decompressor 36 and gas-liquid separator 38.

[0062] If the liquid-phase refrigerant is supplied just before the suction position of high-pressure side compressor 32, it is possible to approach the starting point of the compression process by high-pressure side compressor 32 to the saturated vapor state. Therefore, in the present embodiment, the refrigerant discharged from low-pressure side compressor 31 is merged with the injection refrigerant to reliably decrease the refrigerant temperature, and thereby, the temperature of the refrigerant discharged from high-pressure side compressor 32 is kept low. Accordingly, the limit of the operating pressure ratio in the compressor can be increased.

[0063] Hereinafter, the description will be carried out on explaining why the improvement in controllability can be achieved in the heat pump-type heating device according to the present embodiment.

[0064] The control method in the present embodiment includes the control over the rotational speed of the compressor in accordance with the heating load, the control over the injection in accordance with the temperature of the refrigerant discharged from low-pressure side compressor 31, and the control means configured to control the decompressor subsequent to indoor heat exchanger 26 in the flow direction of the refrigerant. The cycle can be controlled by using the two decompressors. Accordingly, the number of the decompressors and control means, which are control factors, can be suppressed to the minimum, and consequently, the controllability can be improved.

[0065] Under a low load, for example, when the outside air temperature is high, a large amount of the injection refrigerant may disadvantageously cause the COP to decrease excessively. Such a problem can be solved by providing an on-off valve in injection pipeline 51 and opening or closing it based on the outside air temperature or the like.

[0066] Fig. 5 is a graph illustrating the variation of heating ability and the variation of COP in accordance with the ratio of an injection amount of the refrigerant relative to the amount of the refrigerant in a gas-liquid separator before being branched. In Fig. 5, the horizontal axis represents the ratio of the injection amount whereas the vertical axis represents experimental values of the heating ability and the COP.

[0067] With reference to Fig. 5, when increasing the flow rate of the injection refrigerant by decreasing the opening degree of second decompressor 37, the heating ability is improved. Furthermore, when the injection state is brought into the gas-liquid two-phase state, the flow rate of the injection refrigerant is increased further, and thereby the heating ability becomes higher. However, as a large amount of the liquid refrigerant flows into injection pipeline 51, the heating ability decreases. On the other hand, the COP decreases gradually as the injection amount increases.

[0068] In summary, the above-described heat pump type heating device according to the first embodiment of the present invention includes: indoor heat exchanger 26 serving as a first heat exchanger performing heat exchange between refrigerant and heat-receiving fluid; outdoor heat exchanger 27 serving as a second heat exchanger performing heat exchange between the refrigerant and outdoor air; low-pressure side compressor 31 compressing the refrigerant delivered from outdoor heat exchanger 27; high-pressure side compressor 32 compressing the refrigerant delivered from low-pressure side compressor 31; first decompressor 36 decompressing the refrigerant delivered from indoor heat exchanger 26; gas-liquid separator 38 separating the refrigerant delivered from first decompressor 36 into a gas phase and a liquid phase; second decompressor 37 connected to a liquid phase side of gas-liquid separator 38 and decompressing the refrigerant delivered from gas-liquid separator 38; injection pipeline 51 connected to a gas phase side of gas-liquid separator 38 and guiding the refrigerant delivered from gas-liquid separator 38 to a pipeline between low-pressure side compressor 31 and high-pressure side compressor 32; and control unit 46 controlling a decompression ratio of the refrigerant in second decompressor 37 such that the refrigerant flowing in injection pipeline 51 is brought into a gas-liquid two-phase state.

[0069] According to the heat pump-type heating device thus configured in the first embodiment of the present invention, it is possible to make the heat pump-type heating device excellent in controllability while having the heating ability improved sufficiently.

[0070] In the present embodiment, in order to prevent liquid refrigerant from flowing into high-pressure side compressor 32 more certainly, buffer unit 42 is provided at the suction position of high-pressure side compressor 32. In such a configuration, second decompressor 37 may be controlled by control unit 46 such that the refrigerant at the suction position of high-pressure side compressor 32 is brought into the state just after it is transferred from the gas phase state to the gas-liquid two-phase state. In this case, even though the refrigerant at the suction position of high-pressure side compressor 32 contains some liquid refrigerant, the liquid refrigerant is captured by buffer unit 42, and thereby the refrigerant to be sucked into high-pressure side compressor 32 is kept at the saturated vapor state or the superheated gas state.

[0071] Alternatively, buffer unit 42 may not be provided at the suction position of high-pressure side compressor 32. In this case, preferably, α in the equation of target refrigerant temperature $T2_{SP} = T1+\alpha$ is set so as to shift the state of the refrigerant at the suction position of high-pressure side compressor 32 slightly to the superheated gas side. Thereby, it is possible to prevent liquid refrigerant from flowing into high-pressure side compressor 32 more certainly.

(Second Embodiment)

[0072] Fig. 6 is a circuit diagram illustrating a heat pump-type heating device according to a second embodiment of the present invention. The heat pump-type heating device in the present embodiment has basically the same structures as those in the heat pump-type heating device in the first embodiment. In the description below, the same structures as those in the heat pump-type heating device in the first embodiment will not be described repeatedly.

[0073] With reference to Fig. 6, the heat pump-type heating device according to the present embodiment, in addition to temperature detector 61, further includes a temperature detector 66 serving as a third temperature detector and a pressure detector 67.

[0074] Temperature detector 66 is provided between high-pressure side compressor 32 and indoor heat exchanger 26, and specifically, temperature detector 66 is provided at a discharge position of high-pressure side compressor 32. Temperature detector 66 detects a temperature T3 of the refrigerant that is discharged from high-pressure side compressor 32 and then flows into indoor heat exchanger 26. Pressure detector 67 is provided between high-pressure side compressor 32 and indoor heat exchanger 26, and specifically, pressure detector 67 is provided at a discharge position of high-pressure side compressor 32. Pressure detector 67 detects a pressure P of the refrigerant that is discharged from high-pressure side compressor 32 and then flows into indoor heat exchanger 26.

[0075] Fig. 7 illustrates a flowchart of controlling the state of the refrigerant at a suction position of a high-pressure side compressor in the heat pump-type heating device of Fig. 6. Fig. 8 is a Mollier diagram for explaining a control to be executed in the heat-pump-type heating device of Fig. 6.

[0076] With reference to Figs. 6 to 8, in the present embodiment, firstly, temperature T1 of the refrigerant between first decompressor 36 and gas-liquid separator 38 is detected by temperature detector 61, and temperature T3 and pressure P of the refrigerant at the discharge side of high-pressure side compressor 32 are detected by temperature detector 66 and pressure detector 67, respectively. The detected temperature T1, temperature T3 and pressure P are stored in control unit 46 (S201).

[0077] Next, temperature T1 of the refrigerant between first decompressor 36 and gas-liquid separator 38 is considered as the temperature at which the refrigerant is in the gas-liquid two-phase state, and based on temperature T1, an intermediate pressure line 301 (D→C→H→G) is defined on the p-h diagram by using the physical properties of the refrigerant. Moreover, an isentropic line X' passing through a point A' defined according to temperature T3 and pressure P is defined on the p-h diagram. An intersection point between intermediate pressure line 301 and isentropic line X' represents a specific enthalpy H' at the suction position of high-pressure side compressor 32 in the current cycle, and an intersection point between intermediate pressure line 301 and saturated vapor curve 302 of the refrigerant represents a specific enthalpy H of the refrigerant in the saturated vapor state (S202).

[0078] In the present embodiment, in the case where isentropic line X' defined on the p-h diagram overlaps with isentropic line X passing through specific enthalpy H of the refrigerant in the saturated vapor state and thus the relationship of H'= H is satisfied, the refrigerant at the suction position of high-pressure side compressor 32 is in the saturated vapor state. In the case where isentropic line X' (isentropic line X'1 in Fig. 8) defined on the p-h diagram shifts to the outer side of saturated vapor curve 302 than isentropic line X and thus the relationship of H'>H is satisfied, the refrigerant at the suction position of high-pressure side compressor 32 is in the superheated gas state. In the case where isentropic line X' (isentropic line X'2 in Fig. 8) defined on the p-h diagram shifts to the inner side of saturated vapor curve 302 than isentropic line X and thus the relationship of H'<H is satisfied, the refrigerant at the suction position of high-pressure side compressor 32 is in the gas-liquid two-phase state at which the liquid phase is dominant.

[0079] Next, control unit 46 compares H' and H+α (α is any predetermined value) (S203). When the relationship of H+$\alpha \leq$H' is satisfied, it is considered that the refrigerant at the suction position of high-pressure side compressor 32 has been shifted to the side of the superheating state further than the target state, the opening degree of second decompressor 37 is decreased (S205). When the relationship of H+α>H' is satisfied, control unit 46 compares H and H' (S204). When the relationship of H-H'$\geq$0 (H$\geq$H'), it is considered that the refrigerant at the suction position of high-pressure side compressor 32 has been shifted to the side of the saturated vapor state further than the target state, the opening degree of second decompressor 37 is increased (S206).

[0080] When the relationship of H-H'<0 (H<H') is satisfied, the heat pump-type heating device remains idle for t seconds after the operation of second decompressor 37 (S207). In other words, after the step of S205 or S206, the heat pump-type heating device remains idle for t seconds after operation of second decompressor 37 (S207). Thereafter, the step of detecting temperatures T1, T3 and pressure P1 (S201), the step of S202 and the subsequent steps are repeated so as to maintain the refrigerant at the suction position of high-pressure side compressor 32 to the desired state.

[0081] Thus, in the present embodiment, the opening degree of second decompressor 37 is controlled on the basis of temperature T1 of the refrigerant detected by temperature detector 61, temperature T3 detected by temperature detector 63 and pressure P detected by the pressure detector so as to maintain the refrigerant at the suction position of high-pressure side compressor 32 in an intermediate state between the superheated gas state and the saturated vapor state.

[0082] According to the heat pump-type heating device thus configured in the second embodiment of the present invention, it is possible to obtain the same effects as those described in first embodiment.

[0083] Moreover, in the present embodiment, the refrigerant is supplied just after it is transferred from the gas phase state to the gas-liquid two-phase state so as to increase the injection flow rate in the circuit configuration where buffer unit 42 is provided at the suction position of high-pressure side compressor 32 , which is the same as that in the first embodiment. Especially in the present embodiment, since the ratio of the liquid-phase refrigerant relative to the refrigerant in the gas-liquid two-phase state can be determined according to the specific enthalpy, it is possible to supply the injection refrigerant having a larger percentage of liquid phase in accordance with the capacity of buffer unit 42.

(Third Embodiment)

[0084] Fig. 9 is a circuit diagram illustrating a heat pump-type heating device according to a third embodiment of the present invention. The heat pump-type heating device in the present embodiment has basically the same structures as those in the heat pump-type heating device in the first embodiment. In the description below, the same structures as those in the heat pump-type heating device in the first embodiment will not be described repeatedly.

[0085] With reference to Fig. 9, the heat pump-type heating device according to the present embodiment, in addition to temperature detector 61 and temperature detector 62, further includes a temperature detector 71 as a fourth temperature detector. Temperature detector 71 is provided on injection pipeline 51. Temperature detector 71 detects a temperature T4 of the refrigerant flowing in injection pipeline 51.

[0086] The method of controlling the opening degree of second decompressor 37 is basically identical to that shown in the flowchart in Fig. 3. However, since the pipeline may be cooled when the outside air temperature is low, as the refrigerant discharged from first decompressor 36 performs heat exchange with the injection refrigerant in the cycle including an internal heat exchanger (see Fig. 4), it may be transferred into a supercooled state and thereby becomes the liquid refrigerant. In this case, the refrigerant temperature detected by temperature detector 61 cannot be considered as the refrigerant temperature at the suction position of high-pressure side compressor 32. According to the present embodiment, even in such a situation, by providing temperature detector 71 to detect the temperature of the refrigerant flowing in injection pipeline 51 and comparing temperature T4 of the refrigerant flowing in injection pipeline 51 and temperature T1 of the refrigerant at the discharge side of first decompressor 36, it is possible to determine the temperature of the refrigerant in the gas-liquid two-phase state.

[0087] Specifically, in the case where the relationship of T4=T1 is satisfied, the refrigerant at either detection position is in the gas-liquid two-phase state, and thereby, either T1 or T4 may be used in step S102 of Fig. 3. In the case where the relationship of T1>T4 is satisfied, the refrigerant flowing in injection pipeline 51 is likely in the overcooled state, T1 is used in step S102 of in Fig. 3. In the case where the relationship of T1<T4 is satisfied, due to the influence of the outside air temperature and the internal heat exchanger, the refrigerant after first decompressor 36 is likely to be overcooled into the liquid refrigerant, T4 is used in step S102 of in Fig. 3.

[0088] According to the heat pump-type heating device thus configured in the third embodiment of the present invention, it is possible to obtain the same effects as those described in first embodiment.

[0089] It is acceptable to combine the heat-pump-type heating device according to each of the first to third embodiments as described above to achieve a new heat pump-type heating device.

[0090] It should be understood that the embodiments disclosed herein have been presented for the purpose of illustration and description but not limited in all aspects. It is intended that the scope of the present invention is not limited to the description above but defined by the scope of the claims and encompasses all modifications equivalent in meaning and scope to the claims.

INDUSTRIAL APPLICABILITY

[0091] The present invention, for example, is applied to a heat pump-type hot water supplier or a heat pump-type heating system.

REFERENCE SIGNS LIST

[0092]

20: refrigeration circuit; 26: indoor heat exchanger; 27: outdoor heat exchanger; 31: low-pressure side compressor; 32: high-pressure side compressor; 36: first decompressor; 37: second decompressor; 38: gas-liquid separator; 38a: gas-phase refrigerant space; 38b: liquid-phase refrigerant space; 41, 42: buffer unit; 43: internal heat exchanger; 46: control unit; 50: injection circuit; 51: injection pipeline; 53: connection portion; 61, 62, 63, 66, 71: temperature detector; 67: pressure detector; 301: intermediate pressure line; 302: saturated vapor curve

Claims

1. A heat pump-type heating device comprising:

a first heat exchanger configured to perform heat exchange between a refrigerant and a heat-receiving fluid;

a second heat exchanger configured to perform heat exchange between the refrigerant and outdoor air;

a low-pressure side compressor configured to compress the refrigerant delivered from said second heat exchanger;

a high-pressure side compressor configured to compress the refrigerant delivered from said low-pressure side compressor and deliver the compressed refrigerant to said first heat exchanger;

a first decompressor configured to decompress the refrigerant delivered from said first heat exchanger;

a gas-liquid separator configured to separate the refrigerant delivered from said first decompressor into a gas phase and a liquid phase;

a second decompressor connected to the liquid-phase side of said gas-liquid separator, and configured to decompress the refrigerant delivered from said gas-liquid separator and deliver the decompressed refrigerant to said second heat exchanger;

an injection pipeline connected to the gas-phase side of said gas-liquid separator, and configured to guide the refrigerant delivered from said gas-liquid separator to a pipeline between said low-pressure side compressor and said high-pressure side compressor; and

a control unit configured to control a decompression ratio of the refrigerant in said second decompressor so as to render the refrigerant that flows into said high-pressure side compressor into a superheated gas state or a saturated vapor state.

2. The heat pump-type heating device according to claim 1 further comprising:

a first temperature detector provided on the pipeline between said first decompressor and said gas-liquid separator, and configured to detect a temperature T1 of the refrigerant; and

a second temperature detector provided on the pipeline between said low-pressure side compressor and said high-pressure side compressor, and configured to detect a temperature T2 of the refrigerant after having been merged with the refrigerant flowing in said injection pipeline,

wherein said control unit controls the decompression ratio of the refrigerant in said second decompressor based on a comparison between temperature T1 of the refrigerant detected by said first temperature detector and temperature T2 of the refrigerant detected by the second temperature detector.

3. The heat pump-type heating device according to claim 2, wherein

said control unit decreases the decompression ratio of the refrigerant in said second decompressor when a state where temperature T2 of the refrigerant detected by said second temperature detector is equal to temperature T1 of the refrigerant detected by said first temperature detector has lasted for a predetermined time.

4. The heat pump-type heating device according to any one of claims 1 to 3 further comprising:

a first temperature detector provided on the pipeline between said first decompressor and said gas-liquid separator, and configured to detect a temperature T1 of the refrigerant; and

a third temperature detector and a pressure detector both provided on the pipeline between said high-pressure side compressor and said first heat exchanger, and configured to detect a temperature T3 and a pressure P of the refrigerant, respectively,

wherein said control unit determines on a p-h (pressure-specific enthalpy) diagram a specific enthalpy H' as an intersection point between an intermediate pressure line defined according to temperature T1 of the refrigerant detected by said first temperature detector and an isentropic line passing through a point defined according to temperature T3 of the refrigerant detected by said third temperature detector and pressure P of the refrigerant detected by said pressure detector, and controls the decompression ratio of the refrigerant in said second decompressor based on a comparison between said specific enthalpy H' and a specific enthalpy H of a saturated vapor in said intermediate pressure line.

5. The heat pump-type heating device according to any one of claims 1 to 4 further comprising:

a buffer unit being provided on the pipeline which is between said low-pressure side compressor and said high-pressure side compressor and in which the refrigerant after having been merged with the refrigerant flowing in said injection pipeline flows, and being configured to store the liquid refrigerant.

FIG.1

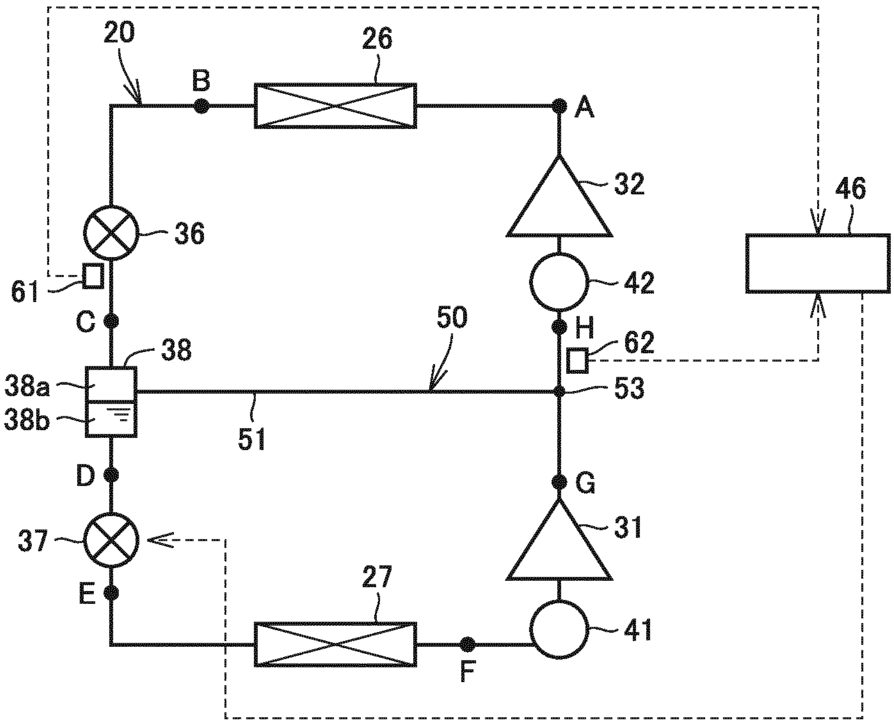

20
26
B
A
32
46
36
42
61
C
H
62
50
38
38a
53
38b
51
D
G
31
37
27
E
41
F

FIG.2

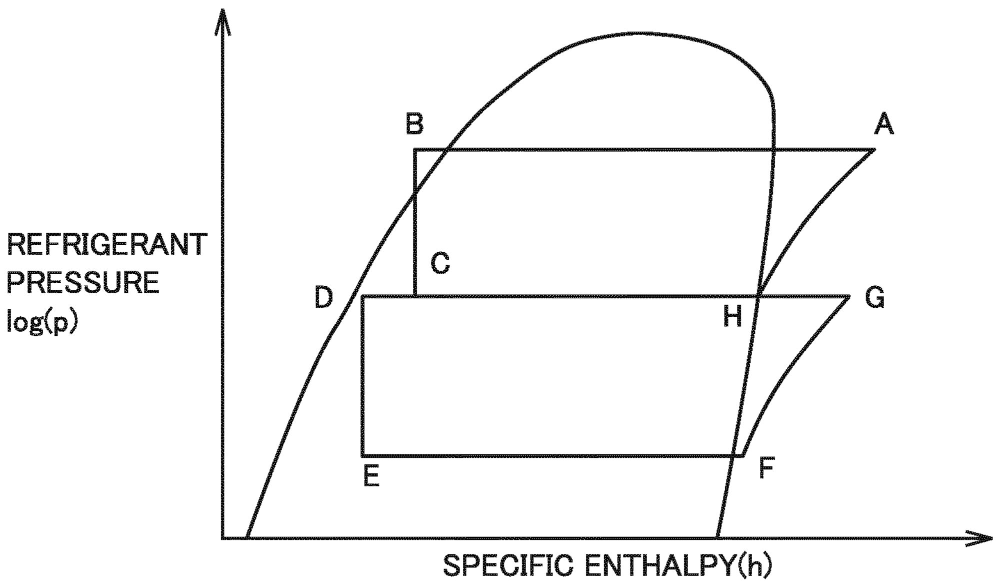

REFRIGERANT
PRESSURE
log(p)
B
A
C
D
H
G
E
F
SPECIFIC ENTHALPY(h)

FIG.3

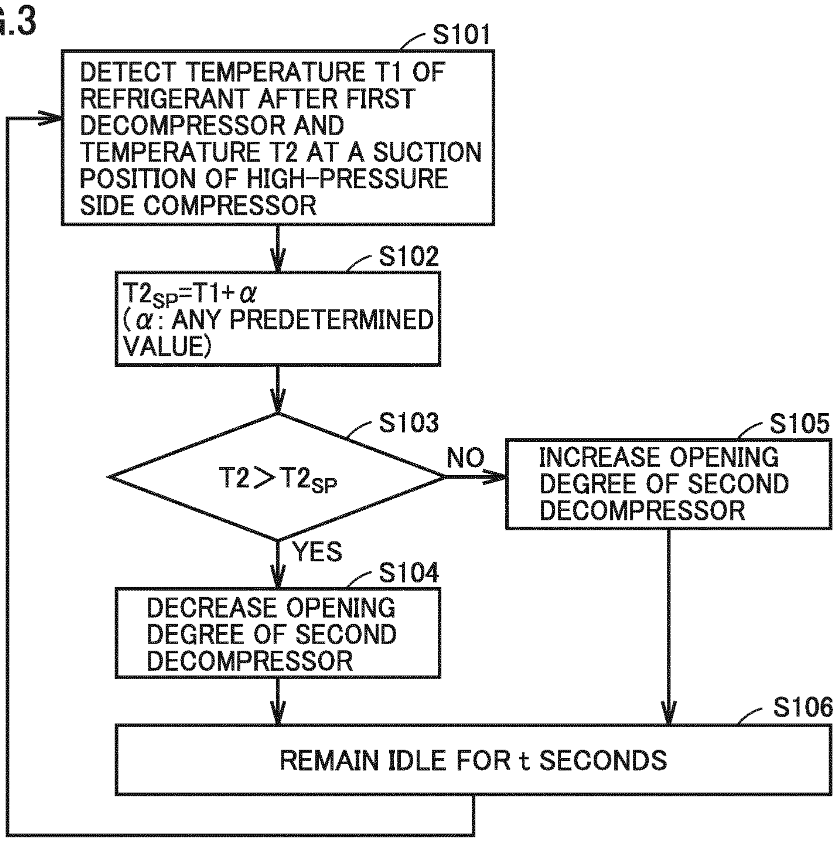

S101
DETECT TEMPERATURE T1 OF REFRIGERANT AFTER FIRST DECOMPRESSOR AND TEMPERATURE T2 AT A SUCTION POSITION OF HIGH-PRESSURE SIDE COMPRESSOR
S102
$T2_{SP}=T1+\alpha$
(α: ANY PREDETERMINED VALUE)
S103
$T2>T2_{SP}$
NO
S105
INCREASE OPENING DEGREE OF SECOND DECOMPRESSOR
YES
S104
DECREASE OPENING DEGREE OF SECOND DECOMPRESSOR
S106
REMAIN IDLE FOR t SECONDS

FIG.4

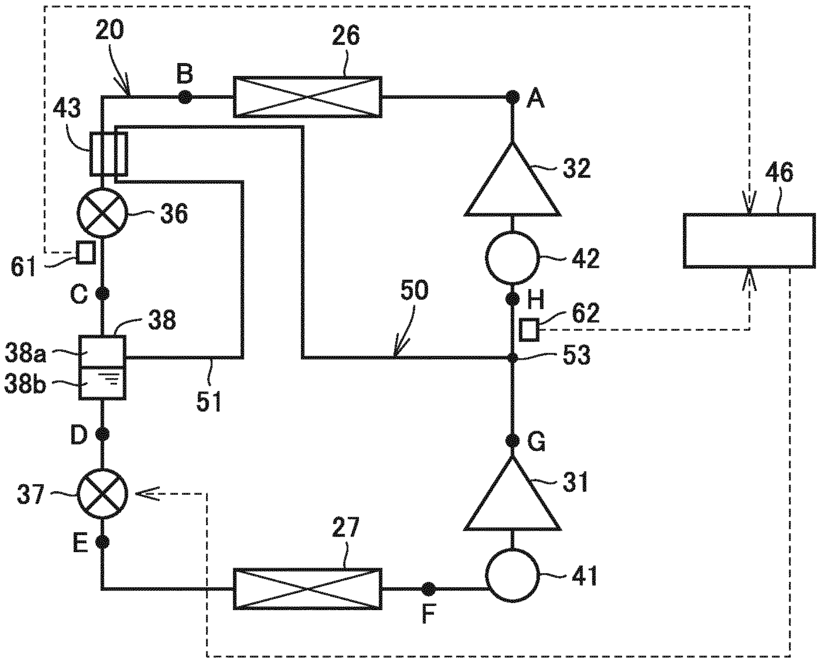

20
B
26
A
43
32
46
36
61
42
C
H
62
50
38
38a
53
38b
51
D
G
37
31
E
27
41
F

FIG.5

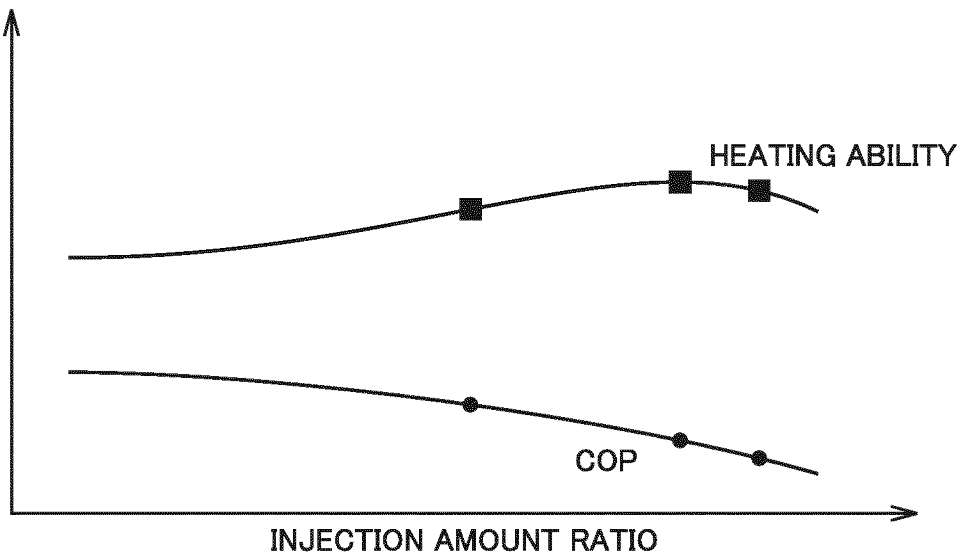

HEATING ABILITY
COP
INJECTION AMOUNT RATIO

FIG.6

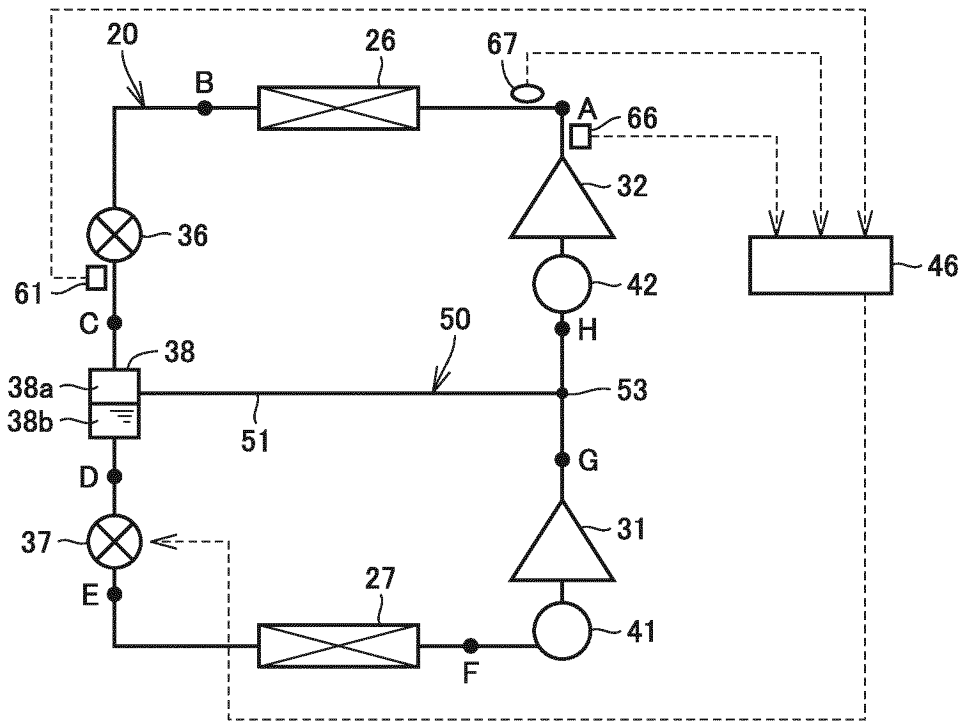

20
B
26
67
A
66
32
36
61
42
46
C
H
38
50
38a
53
38b
51
D
G
37
31
E
27
F
41

FIG.7

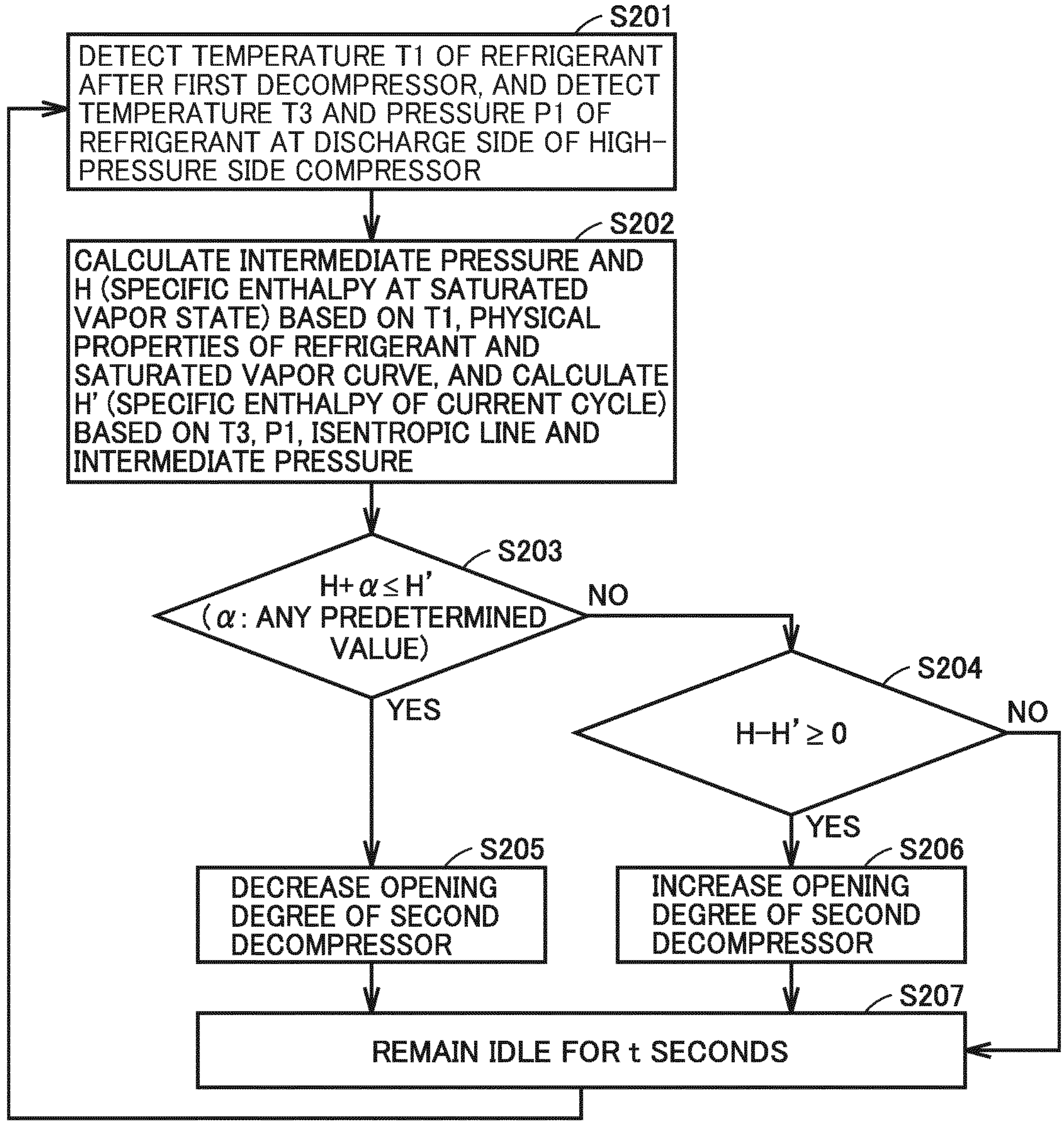

S201
DETECT TEMPERATURE T1 OF REFRIGERANT AFTER FIRST DECOMPRESSOR, AND DETECT TEMPERATURE T3 AND PRESSURE P1 OF REFRIGERANT AT DISCHARGE SIDE OF HIGH-PRESSURE SIDE COMPRESSOR
S202
CALCULATE INTERMEDIATE PRESSURE AND H (SPECIFIC ENTHALPY AT SATURATED VAPOR STATE) BASED ON T1, PHYSICAL PROPERTIES OF REFRIGERANT AND SATURATED VAPOR CURVE, AND CALCULATE H' (SPECIFIC ENTHALPY OF CURRENT CYCLE) BASED ON T3, P1, ISENTROPIC LINE AND INTERMEDIATE PRESSURE
S203
H+α≤H' (α: ANY PREDETERMINED VALUE)
NO
YES
S204
H−H'≥0
NO
YES
S205
DECREASE OPENING DEGREE OF SECOND DECOMPRESSOR
S206
INCREASE OPENING DEGREE OF SECOND DECOMPRESSOR
S207
REMAIN IDLE FOR t SECONDS

FIG.8

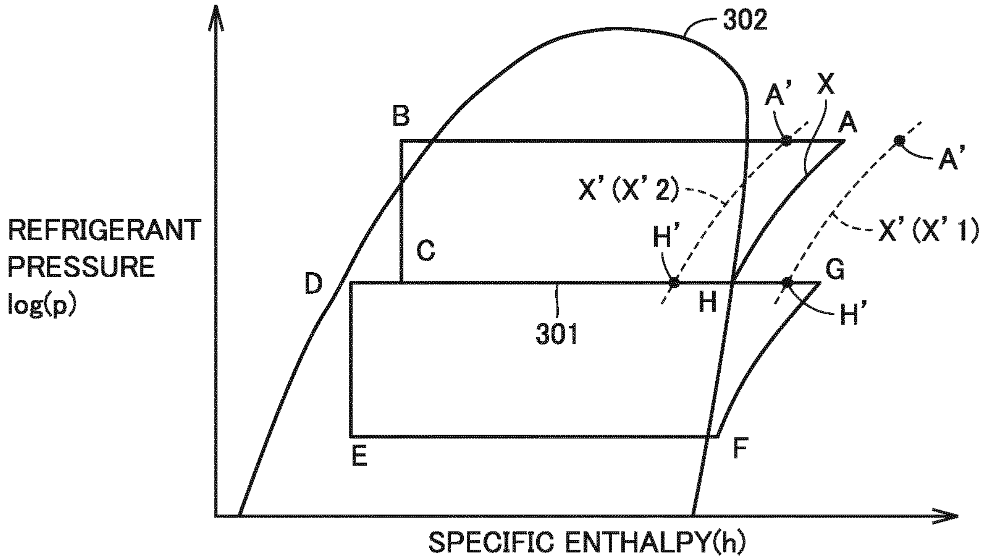
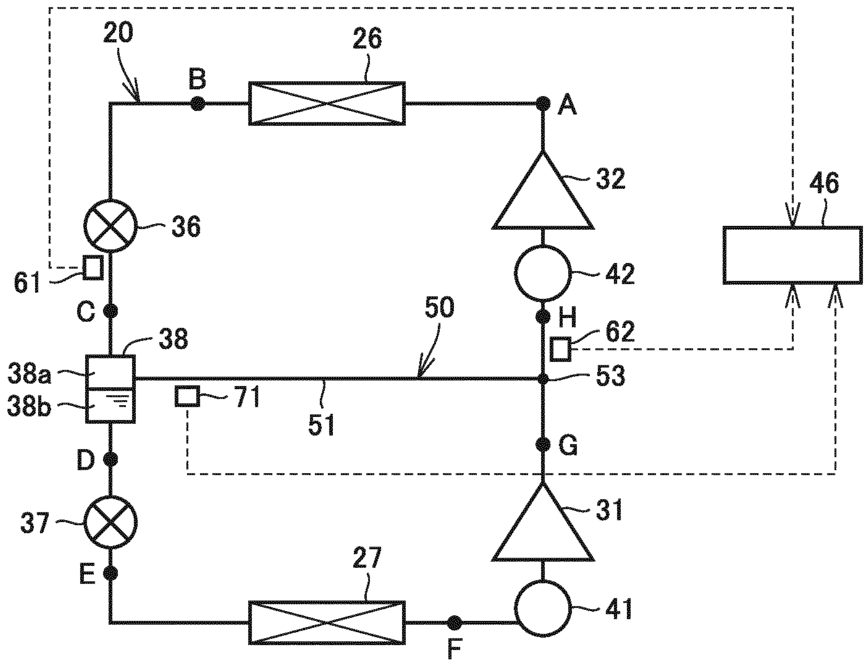

302
A'
X
B
A
A'
X' (X' 2)
REFRIGERANT
PRESSURE
log(p)
X' (X' 1)
H'
C
G
D
H
H'
301
E
F
SPECIFIC ENTHALPY(h)

FIG.9


20
26
B
A
32
46
36
42
61
C
H
62
50
38
38a
53
71
38b
51
D
G
37
31
E
27
41
F

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2013/081612

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

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

B. FIELDS SEARCHED

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

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2014
Kokai Jitsuyo Shinan Koho	1971-2014	Toroku Jitsuyo Shinan Koho	1994-2014

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y A	JP 2006-177597 A (Hitachi Home & Life Solution, Inc.), 06 July 2006 (06.07.2006), paragraphs [0034], [0065]; fig. 1, 2 & CN 1793755 A	1,2 5 3,4
Y	JP 2007-178042 A (Mitsubishi Electric Corp.), 12 July 2007 (12.07.2007), paragraphs [0056] to [0057], [0063]; fig. 14, 18 (Family: none)	5

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

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04 February, 2014 (04.02.14)	18 February, 2014 (18.02.14)
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INTERNATIONAL SEARCH REPORT

International application No.
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C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	WO 2012/032699 A1 (Panasonic Corp.), 15 March 2012 (15.03.2012), claims & US 2012/0167604 A1 & CN 102523753 A	1-5
A	JP 2008-082653 A (Mitsubishi Electric Corp.), 10 April 2008 (10.04.2008), paragraph [0043] (Family: none)	4
A	JP 2008-039332 A (Mitsubishi Heavy Industries, Ltd.), 21 February 2008 (21.02.2008), paragraph [0040] (Family: none)	4

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 8210709 A **[0002] [0008]**
- JP 11132575 A **[0004] [0008]**
- JP 2007263440 A **[0006] [0008]**