(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 2 995 885 A1

(12) # EUROPEAN PATENT APPLICATION
published in accordance with Art. 153(4) EPC

(43) Date of publication:
16.03.2016 Bulletin 2016/11

(21) Application number: **13883949.3**

(22) Date of filing: **08.05.2013**

(51) Int Cl.:
F25B 7/00 *(2006.01)* ***F25B 6/04*** *(2006.01)*

(86) International application number:
PCT/JP2013/062931

(87) International publication number:
WO 2014/181399 (13.11.2014 Gazette 2014/46)

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA ME

(71) Applicant: **Mitsubishi Electric Corporation**
Tokyo 100-8310 (JP)

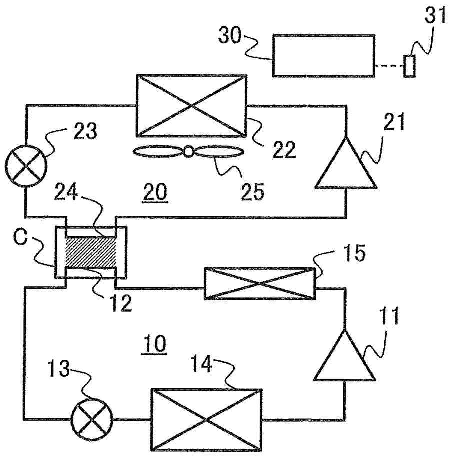
(72) Inventors:
- **ISHIKAWA, Tomotaka**
  Tokyo 100-8310 (JP)
- **NOMOTO, So**
  Tokyo 100-8310 (JP)
- **SUGIMOTO, Takeshi**
  Tokyo 100-8310 (JP)
- **YAMASHITA, Tetsuya**
  Tokyo 100-8310 (JP)
- **IKEDA, Takashi**
  Tokyo 100-8310 (JP)

(74) Representative: **Moore, Graeme Patrick et al**
Mewburn Ellis LLP
City Tower
40 Basinghall Street
London EC2V 5DE (GB)

(54) **BINARY REFRIGERATION DEVICE**

(57) A two-stage refrigeration cycle system includes a higher-stage compressor 21, a higher-stage condenser 22, a higher-stage expansion device 23, and a higher-stage evaporator 24 connected in a higher-stage refrigeration cycle 20 by pipes through which refrigerant circulates therein, a lower-stage compressor 11, an auxiliary radiator 15, a lower-stage condenser 12, a lower-stage expansion device 13, and a lower-stage evaporator 14 connected in a lower-stage refrigeration cycle 10 by pipes through which refrigerant circulates therein, a cascade condenser C including the higher-stage evaporator 24 and the lower-stage condenser 12 and configured to heat exchange between the refrigerant flowing in the higher-stage refrigeration cycle 20 and the refrigerant flowing in the lower-stage refrigeration cycle 10, and a controller 30 configured to switch between a two-stage operation in which both the higher-stage refrigeration cycle 20 and the lower-stage refrigeration cycle 10 operate and a single-stage operation in which the higher-stage refrigeration cycle 20 stops and the lower-stage refrigeration cycle 10 operates, so as to select one of the two-stage operation and the single-stage operation having a higher COP.

FIG. 1



EP 2 995 885 A1

Description

Technical Field

[0001] The present invention relates to a two-stage refrigeration cycle system.

Background Art

[0002] A two-stage refrigeration cycle system including a higher-stage refrigeration cycle that is a refrigeration cycle apparatus for circulating a higher-temperature refrigerant therein and a lower-stage refrigeration cycle that is a refrigeration cycle apparatus for circulating a lower-temperature refrigerant therein has been conventionally used as a system for cooling at a low temperature of several tens of minus degrees C. In such a two-stage refrigeration cycle system, for example, a lower-stage refrigeration cycle and a higher-stage refrigeration cycle are coupled to each other by a cascade condenser configured to perform heat exchange between a lower-stage condenser in the lower-stage refrigeration cycle and a higher-stage evaporator in the higher-stage refrigeration cycle.

[0003] In a two-stage refrigeration cycle system, an auxiliary radiator is disposed at a stage previous to a cascade condenser in a lower-stage refrigeration cycle (see Patent Literature 1). In this two-stage refrigeration cycle system, refrigerant discharged from a lower-temperature compressor is caused to reject heat by the auxiliary radiator to enhance the operating efficiency.

[0004] Since the two-stage refrigeration cycle system can be used in a range with a high efficiency in terms of a high compression ratio, this system is generally advantageous for energy conservation. However, in a range where the two-stage refrigeration cycle system has a low compression ratio in, for example, an operation at a low outdoor-air temperature such as winter, even a single-stage refrigeration cycle system can easily perform cooling to a necessary temperature. Thus, the use of a two-stage refrigeration cycle system does not have an advantage in energy conservation. In view of this, a known technique is proposed in order to avoid performance degradation by switching to a single-stage operation in which a higher-stage refrigeration cycle stops and only a lower-stage refrigeration cycle operates in a case where a two-stage refrigeration cycle system operates at a low compression ratio (see, for example, Patent Literature 2).

[0005] In this technique, the operation mode is switched between the single-stage operation and a two-stage operation by connecting part of a higher-stage heat exchanger that is a radiator of the higher-stage refrigeration cycle in parallel to a condenser of the lower-stage refrigeration cycle so that the higher-stage heat exchanger can partially serve as a radiator of the lower-stage refrigeration cycle. That is, in the single-stage operation, the lower-stage refrigeration cycle transfers heat to outdoor air by using the part of the higher-stage heat exchanger. In the two-stage operation, the lower-stage refrigeration cycle prevents refrigerant from flowing into the part of the higher-stage heat exchanger and allows refrigerant to flow toward a cascade condenser, whereas the higher-stage refrigeration cycle transfers heat to outdoor air by using another part of the higher-stage heat exchanger.

Citation List

Patent Literature

[0006]

Patent Literature 1: Japanese Patent No. 3604973 (pages 2 and 3, Fig. 1)
Patent Literature 2: Japanese Unexamined Patent Application Publication No. 2000-274848 (page 5, Fig. 1)

Summary of Invention

Technical Problem

[0007] The two-stage refrigeration cycle system of Patent Literature 1 is intended to enhance the operating efficiency by using the auxiliary radiator. However, in an operation a low compression ratio such as an operation at a low outdoor-air temperature, for example, compressor performance inevitably degrades. In addition, as a configurational feature of the two-stage refrigeration cycle system, a temperature difference between a lower-stage condensing temperature and a higher-stage evaporating temperature in a cascade condenser leads to a loss. In an operation at a high compression ratio, this temperature difference does not cause any problem. On the other hand, in an operation at a low compression ratio, the influence of the temperature difference is larger than that in the operation at the high compression ratio, and thus, the operating efficiency inevitably degrades. That is, in the two-stage refrigeration cycle system of Patent Literature 1, although the use of the auxiliary radiator can enhance the operating efficiency, the operating efficiency inevitably degrades in a certain period, and it is difficult to obtain an energy conservation effect through the year.

[0008] The lower-stage refrigeration cycle of the two-stage refrigeration cycle system described in Patent Literature 2 can avoid performance degradation to a certain degree by switching from the two-stage operation to the single-stage operation in an operation at a low compression ratio at a low outdoor-air temperature. However, the cascade condenser rejects heat in the two-stage operation (where both the lower-stage refrigeration cycle and the higher-stage refrigeration cycle operate), whereas the higher-stage condenser rejects heat in the single-stage operation. In this manner, in Patent Literature 2, the channel needs to be switched between the two-stage operation and the single-stage operation, and part of the higher-stage condenser is wastefully unused in the two-stage operation. Accordingly, Patent Literature 2 has problems of an increase in costs due to addition of air-cooled radiators and switching valves arranged in parallel and addition of bypass channels.

[0009] The present invention has been made in view of the problems described above, and provides a two-stage refrigeration cycle system that can enhance operating efficiency by using an auxiliary radiator, avoid performance degradation in an operation at a low compression ratio, and obtain an energy conservation effect through the year without an increase in costs.

Solution to Problem

[0010] A two-stage refrigeration cycle system according to the present invention includes: a higher-stage compressor, a higher-stage condenser, a higher-stage expansion device, and a higher-stage evaporator connected in a higher-stage refrigeration cycle by pipes through which refrigerant circulates therein, a lower-stage compressor, an auxiliary radiator, a lower-stage condenser, a lower-stage expansion device, and a lower-stage evaporator connected in a lower-stage refrigeration cycle by pipes through which refrigerant circulates therein, a cascade condenser including the higher-stage evaporator and the lower-stage condenser and configured to heat exchange between the refrigerant flowing in the higher-stage refrigeration cycle and the refrigerant flowing in the lower-stage refrigeration cycle, and a controller configured to switch between a two-stage operation in which both the higher-stage refrigeration cycle and the lower-stage refrigeration cycle operate and a single-stage operation in which the higher-stage refrigeration cycle stops and the lower-stage refrigeration cycle operates, so as to select one of the two-stage operation and the single-stage operation having a higher COP.

Advantageous Effects of Invention

[0011] According to the present invention, the lower-stage refrigeration cycle includes the auxiliary radiator so that the auxiliary radiator is used as an aid for the lower-stage condenser in the two-stage operation and the auxiliary radiator is used as a main radiator in the single-stage operation, and an operation with a higher COP is selected from the two-stage operation and the single-stage operation. In this manner, a high operating efficiency can be obtained through the year, and energy conservation can be achieved. In the single-stage operation, the auxiliary radiator is used as a main radiator so that the lower-stage refrigeration cycle can be used without a change in channel. Thus, no components need to be added in order to perform a single-stage operation, and costs can be reduced.

Brief Description of Drawings

[0012]

[Fig. 1] Fig. 1 illustrates a configuration of a two-stage refrigeration cycle system according to Embodiment 1 of the present invention.

[Fig. 2] Fig. 2 showing a relationship between an enthalpy and a saturation temperature in the two-stage refrigeration cycle system illustrated in Fig. 1.

[Fig. 3] Fig. 3 shows a relationship between a lower-stage condensing temperature and a compressor input.

[Fig. 4] Fig. 4 is a flowchart showing a flow before a rotation speed of a higher-stage compressor 21 is determined in accordance with an outdoor-air temperature in a two-stage operation of the two-stage refrigeration cycle system illustrated in Fig. 1.

[Fig. 5] Fig. 5 shows Mollier charts of heat transfer amounts in a case where a lower-stage condensing temperature is lower than the outdoor-air temperature and a case where the lower-stage condensing temperature higher than the outdoor-air temperature.

[Fig. 6] Fig. 6 is a diagram for describing a relationship between the amount of heat transfer by an auxiliary radiator 15 and a COP.

[Fig. 7] Fig. 7 shows an outdoor-air temperature to COP characteristic in a single-stage operation and a two-stage operation (with an auxiliary radiator) in the two-stage refrigeration cycle system illustrated in Fig. 1.

[Fig. 8] Fig. 8 is a flowchart showing an operation of the two-stage refrigeration cycle system illustrated in Fig. 1.

[Fig. 9] Fig. 9 is a graph showing a relationship between an outdoor-air temperature and a threshold outdoor-air temperature Tca in the two-stage refrigeration cycle system according to the Embodiment 1 of the present invention, where the abscissa represents the outdoor-air temperature and the ordinate represents a temperature (outdoor-air temperature - Tc) obtained by subtracting a target lower-stage condensing temperature Tc from the outdoor-air temperature.

[Fig. 10] Fig. 10 illustrates a configuration of the two-stage refrigeration cycle system in a case where a higher-stage condenser and an auxiliary radiator illustrated in Fig. 1 form an integrated radiator.

[Fig. 11] Fig. 11 is an illustration of an example configuration in the case where the higher-stage condenser and the auxiliary radiator shown in Fig. 1 form the integrated radiator.

[Fig. 12] Fig. 12 illustrates an example configuration 1 of a two-stage refrigeration cycle system according to Embodiment 2 of the present invention.

[Fig. 13] Fig. 13 illustrates an example configuration 2 of the two-stage refrigeration cycle system according to Embodiment 2 of the present invention.

[Fig. 14] Fig. 14 shows a relationship between an enthalpy and a saturation temperature in the two-stage refrigeration cycle system according to Embodiment 2 of the present invention.

Description of Embodiments

[0013] Preferred embodiments of a two-stage refrigeration cycle system according to the present invention will be hereinafter described with reference to the drawings.

Embodiment 1

[0014] Fig. 1 illustrates a configuration of a two-stage refrigeration cycle system according to Embodiment 1 of the present invention. As illustrated in Fig. 1, the two-stage refrigeration cycle system according to Embodiment 1 includes a lower-stage refrigeration cycle 10 and a higher-stage refrigeration cycle 20, and the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20 constitute refrigerant circuits in which refrigerant circulates, independently of each other. To configure the two refrigerant circuits in multiple stages, a cascade condenser (inter-refrigerant heat exchanger) C is formed by coupling a higher-stage evaporator 24 and a lower-stage condenser 12 in such a manner that heat can be exchanged between refrigerant passing through the higher-stage evaporator 24 and refrigerant passing through the lower-stage condenser 12. The two-stage refrigeration cycle system also includes a controller 30 that controls an operation of the entire two-stage refrigeration cycle system. Here, the levels of, for example, the temperature and pressure are not determined based on specific absolute values, and are determined relative to the states, operations, and other factors in, for example, a system or a device.

[0015] In Fig. 1, the lower-stage refrigeration cycle 10 constitutes a refrigerant circuit by sequentially connecting a lower-stage compressor 11, an auxiliary radiator 15, a lower-stage condenser 12, a lower-stage expansion valve (lower-stage expansion device) 13, and a lower-stage evaporator 14 by refrigerant pipes. On the other hand, the higher-stage refrigeration cycle 20 constitutes a refrigerant circuit by sequentially connecting a higher-stage compressor 21, a higher-stage condenser 22, a higher-stage expansion valve (higher-stage expansion device) 23, and a higher-stage evaporator 24 by refrigerant pipes.

[0016] The lower-stage compressor 11 of the lower-stage refrigeration cycle 10 sucks refrigerant, compresses the refrigerant into a high-temperature high-pressure state, and discharges the resulting refrigerant. This compressor is of a type whose rotation speed is controlled by, for example, an inverter circuit so that the discharging amount of higher-stage refrigerant can be adjusted.

[0017] The auxiliary radiator 15 serves as, for example, a gas cooler, and cools gas refrigerant discharged from the lower-stage compressor 11 through heat exchange with air in the outside (outdoor air), water, or brine, for example. The auxiliary radiator 15 of Embodiment 1 performs heat exchange between outdoor air (ambient air) and refrigerant.

[0018] The lower-stage condenser 12 condenses refrigerant that has passed through the auxiliary radiator 15, through heat exchange with refrigerant that has passed from the higher-stage expansion valve 23 in the higher-stage refrigeration cycle 20, and obtains liquid refrigerant (condenses and liquefies the refrigerant). Specifically, in this example, in the cascade condenser C, a heat transfer tube through which refrigerant flowing in the lower-stage refrigeration cycle 10 passes, for example, serves as the lower-stage condenser 12.

[0019] The lower-stage expansion valve 13 serving as a pressure reducing device or an expansion device, for example, reduces the pressure of refrigerant flowing in the lower-stage refrigeration cycle 10 and expands the refrigerant. For example, the lower-stage expansion valve 13 is constituted by a flow rate controlling means such as an electronic expansion valve, a capillary tube (capillary), and a refrigerant flow rate adjusting means such as a themosensitive expansion valve. The lower-stage evaporator 14 causes refrigerant flowing in the lower-stage refrigeration cycle 10 to exchange heat with, for example, a cooling target and evaporate as a gaseous refrigerant (evaporates and gasifies the refrigerant). The cooling target is directly or indirectly cooled through heat exchange with refrigerant.

[0020] On the other hand, the higher-stage compressor 21 of the higher-stage refrigeration cycle 20 sucks refrigerant flowing in the higher-stage refrigeration cycle 20, compresses the refrigerant into a high-temperature high-pressure state, and discharges the refrigerant. The higher-stage compressor 21 is, for example, a compressor of a type including, for example, an inverter circuit and capable of adjusting the discharging amount of refrigerant. The higher-stage condenser 22 performs heat exchange between outdoor air, water, or brine, for example, and refrigerant flowing in the higher-stage refrigeration cycle 20, and condenses and liquefies the refrigerant. In Embodiment 1, the higher-stage condenser 22 performs heat exchange between outdoor air (ambient air) and refrigerant and incudes a higher-stage condenser fan 25 for promoting heat exchange. The higher-stage condenser fan 25 is also a fan capable of adjusting an airflow rate.

[0021] The higher-stage expansion valve 23 serving as, for example, a pressure reducing device or an expansion device reduces the pressure of refrigerant flowing in the higher-stage refrigerant circuit and expands the refrigerant. For example, the higher-stage expansion valve 23 is a flow rate controlling means such as the electronic expansion valve described above or a refrigerant flow rate adjusting means such as a capillary tube. The higher-stage evaporator 24 evaporates and gasifies refrigerant flowing in the higher-stage refrigeration cycle 20 through heat exchange. For example, a heat transfer tube through which refrigerant flowing in the higher-stage refrigeration cycle 20 passes in the cascade condenser C serves as the higher-stage evaporator 24 and performs heat exchange with refrigerant flowing in the lower-stage refrigeration cycle 10.

[0022] The cascade condenser C is an inter-refrigerant heat exchanger having the functions of the higher-stage evaporator 24 and the lower-stage condenser 12 and enables heat exchange between higher-stage refrigerant and lower-stage refrigerant. The higher-stage refrigeration cycle 20 and the lower-stage refrigeration cycle 10 have a multi-stage configuration using the cascade condenser C and perform heat exchange between refrigerants, thereby causing these independent refrigerant circuits to cooperate with each other. The controller 30 controls operations of devices constituting the two-stage refrigeration cycle system, for example. An outdoor-air temperature detecting means 31 is a temperature sensor for detecting an outdoor-air temperature. Hereinafter, the outdoor-air temperature is a temperature used for detecting the outdoor-air temperature detecting means 31.

[0023] In the thus-configured two-stage refrigeration cycle system, refrigerant used in the lower-stage refrigeration cycle 10 is CO_2 (carbon dioxide) because of the following reasons. The lower-stage refrigeration cycle 10 is connected to, for example, a loaded device in a room, such as a showcase in a supermarket, and a refrigerant circuit is opened by rearranging the showcase so that leakage of refrigerant is likely to occur. In consideration of the refrigerant leakage, CO_2 (carbon dioxide) whose influence on global warming is small is used.

[0024] On the other hand, refrigerant used in the higher-stage refrigeration cycle 20 is preferably a refrigerant whose influence on global warming is small, such as an HFO refrigerant (e.g., HFO1234yf, HFO1234ze), HC refrigerant, CO_2, ammonia, and water. In the higher-stage refrigeration cycle 20, however, since the refrigerant circuit is not open, HFC refrigerant having a high global warming potential, for example, may be used. Thus, in Embodiment 1, R32, which is an HFC refrigerant, is used as refrigerant circulating in the higher-stage refrigeration cycle 20.

[0025] Operations of components in a cooling operation of the two-stage refrigeration cycle system as described above, for example, will be described with reference to a flow of refrigerant circulating in the refrigerant circuits will be described. First, an operation of the higher-stage refrigeration cycle 20 will be described and then an operation of the lower-stage refrigeration cycle 10 will be described.

[0026] (Operation of Higher-stage Refrigeration Cycle)

[0027] The higher-stage compressor 21 sucks higher-stage refrigerant, compresses the refrigerant into a high-temperature high-pressure state, and discharges the refrigerant. The discharged refrigerant flows into the higher-stage condenser 22. The higher-stage condenser 22 performs heat exchange between outdoor air supplied from the higher-stage condenser fan 25 and the higher-stage refrigerant, and condenses and liquefies the higher-stage refrigerant. The pressure of the condensed and liquefied refrigerant is reduced by the higher-stage expansion valve 23. The higher-stage refrigerant having a reduced pressure flows into the higher-stage evaporator 24 (cascade condenser C). The higher-stage evaporator 24 performs heat exchange between the higher-stage refrigerant and lower-stage refrigerant passing through the lower-stage condenser 12, and evaporates and gasifies the higher-stage refrigerant. The evaporated gaseous higher-stage refrigerant is sucked into the higher-stage compressor 21.

(Operation of Lower-stage Refrigeration Cycle)

[0028] The lower-stage compressor 11 sucks CO_2 refrigerant, compresses the refrigerant into a high-temperature high-pressure state, and discharges the refrigerant. The discharged refrigerant is cooled by the auxiliary radiator 15 and flows into the lower-stage condenser 12 (cascade condenser C). The lower-stage condenser 12 performs heat exchange between the lower-stage refrigerant and the higher-stage refrigerant passing through the higher-stage evaporator 24 and condenses and liquefies the lower-stage refrigerant. The pressure of the condensed and liquefied lower-stage refrigerant is reduced by the lower-stage expansion valve 13. The lower-stage refrigerant having a reduced pressure flows into the lower-stage evaporator 14. The lower-stage evaporator 14 evaporates and gasifies the lower-stage refrigerant through heat exchange with a cooling target. The evaporated gaseous lower-stage refrigerant is sucked into the higher-stage compressor 21.

[0029] In the two-stage refrigeration cycle system according to Embodiment 1, the higher-stage compressor 21 controls the frequency of a driving motor so as to control the cooling capacity of the higher-stage refrigeration cycle 20, thereby adjusting a pressure (high pressure) at a discharge side of the lower-stage refrigerant circuit. This will be specifically described.

[0030] Fig. 2 showing a relationship between an enthalpy and a saturation temperature in the two-stage refrigeration cycle system of the present invention. In the upper and lower graphs of Fig. 2, a low pressure Ps and a high pressure Pd are fixed (that is, Fig. 2 shows a relationship between an enthalpy and a saturation temperature at a certain outdoor-air temperature). Referring to Fig. 2, it will be described how inputs of each of the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20 change in accordance with the lower-stage condensing temperature.

[0031] In the cascade condenser C of the two-stage refrigeration cycle system, a temperature difference ΔT arises between the lower-stage condensing temperature and the higher-stage evaporating temperature. The temperature difference ΔT depends on the size (performance) of the cascade condenser C, and is, for example, about 5 degrees C. In the following description, the low pressure Ps refers to an evaporating pressure of the lower-stage refrigeration cycle 10 in each of a two-stage operation and a single-stage operation. The high pressure Pd refers to a condensing pressure of the higher-stage refrigeration cycle 20 in the two-stage operation and to a condensing pressure of the lower-stage refrigeration cycle 10 in the single-stage operation.

[0032] For example, when operating frequency of the higher-stage compressor 21 is increased from an operating state so as to increase the cooling capacity of the higher stage, the higher-stage evaporating temperature decreases, and the lower-stage condensing temperature (lower-stage high pressure) decreases accordingly. In contrast, when the cooling capacity of the higher stage is reduced, the lower-stage high pressure increases.

[0033] As clearly shown in Fig. 2, when the operating frequency of the higher-stage compressor 21 is increased so as to reduce the lower-stage high pressure of the lower-stage refrigeration cycle 10, an input of the higher-stage compressor 21 (hereinafter referred to as a higher-stage compressor input) increases (WH1 < WH2). On the other hand, an input of the lower-stage compressor 11 (hereinafter referred to as a lower-stage compressor input) decreases (WL1 > WL2). Here, the relationship of refrigeration capacity Q = Gr (refrigerant flow rate) $\times$ ΔH (enthalpy difference in compressor) is established.

[0034] In the two-stage refrigeration cycle system, the cooling load changes in accordance with the outdoor-air temperature, and a refrigeration capacity (corresponding to the evaporating capacity of the lower-stage refrigeration cycle 10) is determined with respect to the cooling load. To maintain the determined refrigeration capacity at a constant level, a Gr (refrigerant flow rate) is controlled by the lower-stage compressor 11. For example, the lower-stage compressor 11 is controlled in such a manner that the Gr (refrigerant flow rate) is constant if the ΔH (enthalpy difference) is constant.

[0035] Specifically, in the two-stage refrigeration cycle system of Embodiment 1, CO_2 refrigerant used for the lower-stage refrigeration cycle 10 has a smaller refrigeration cycle effect than R32 used in the higher-stage refrigeration cycle 20. Thus, a large compressor power is needed, and the operating efficiency is lower than R32 used in the higher-stage refrigeration cycle 20. In view of this, the volume of the higher-stage compressor 21 is increased so as to reduce the lower-stage high pressure, thereby reducing power consumption in the lower-stage refrigeration cycle 10. Even when power consumption in the higher-stage refrigeration cycle 20 using R32 showing a high operating efficiency increases, a work of the higher-stage refrigeration cycle 20 is increased so that the operating efficiency of the entire two-stage refrigeration cycle system is enhanced. By increasing a power consumption ratio of the highly efficient higher-stage refrigeration cycle 20, the operating efficiency of the entire two-stage refrigeration cycle system can be optimized. Thus, CO_2 is often out of a supercritical state at the lower-stage high pressure of the lower-stage refrigeration cycle 10, and a saturation temperature (lower-stage condensing temperature) at which a phase change occurs in the lower-stage condenser 12 is fixed.

[0036] Fig. 3 shows a relationship between a lower-stage condensing temperature and a compressor input. In Fig. 3, the abscissa represents a lower-stage condensing temperature, and the ordinate represents a compressor input. Fig. 3 shows an input of the higher-stage compressor 21, an input of the lower-stage compressor 11, and a total input thereof (a total input of the entire two-stage refrigeration cycle system). As shown in Fig. 3, when the lower-stage condensing temperature is equal to or less than the outdoor-air temperature and the higher-stage compressor 21 and the lower-stage compressor 11 have substantially the same compressor input, the total input is at the minimum, and the coefficient of performance (COP) = refrigeration capacity / (higher-stage compressor input + lower-stage compressor input) is at the maximum.

[0037] As described above, the higher-stage compressor 21 of the two-stage refrigeration cycle system controls an operation in such a manner that the higher-stage compressor input and the lower-stage compressor input are substantially the same so as to maximize the COP. Specifically, referring to Fig. 2, the controller 30 controls an operation in such a manner that the higher-stage compressor input (= enthalpy difference WH1 × higher-stage refrigerant flow rate Grh) is substantially equal to a lower-stage compressor input (= enthalpy difference WL1 × lower-stage refrigerant flow rate Grl).

[0038] In another aspect of Fig. 3, when the lower-stage condensing temperature of the lower-stage refrigeration cycle 10 is Tc, the total input is at the minimum, and the COP is at the maximum. Thus, with the operation control for making the higher-stage compressor input and the lower-stage compressor input substantially equal to each other, the lower-stage refrigeration cycle 10 is controlled in such a manner that the lower-stage condensing temperature is kept at a target lower-stage condensing temperature Tc. At this time, in the higher-stage refrigeration cycle 20, control is performed in such a manner that a temperature lower than the target lower-stage condensing temperature Tc by ΔT degrees C (which is 5 degrees C in this example as described above) is kept constant as a target higher-stage evaporating temperature. With such a control, the COP can be maximized.

[0039] (Rotation Speed Control of Higher-stage Compressor 21 in accordance with Outdoor-air Temperature)

[0040] Fig. 4 is a flowchart showing a flow before the rotation speed of the higher-stage compressor 21 is determined in accordance with the outdoor-air temperature in a two-stage operation of the two-stage refrigeration cycle system illustrated in Fig. 1.

(S1)

[0041] First, preconditions will be described. The refrigeration capacity is determined based on a request (a cooling load of use side equipment such as a cold room used by a user) (e.g., 10 kW for 10 horsepower). The target lower-stage condensing temperature Tc at which the COP is at the maximum depends on a lower-stage evaporating temperature (e.g., -40 degrees C) determined based on a request from a user and a higher-stage condensing temperature uniquely determined in accordance with the outdoor-air temperature. The higher-stage condensing temperature tends to increase as the outdoor-air temperature increases. In other words, the target lower-stage condensing temperature Tc is determined based on the low pressure Ps converted from a lower-stage evaporating temperature ET and the high pressure Pd converted from a higher-stage condensing temperature CT in Fig. 2. As described above, the temperature difference ΔT is 5 degrees C and depends on the size (performance) of the cascade condenser C.

(S2)

[0042] Based on the refrigeration cycle capacity, a necessary heat transfer amount Q1 required for the lower-stage condenser 12 in order to obtain this refrigeration capacity is determined.

(S3)

[0043] Since the lower-stage condenser 12 and the higher-stage evaporator 24 exchange heat in the cascade condenser C, the heat transfer amount in the lower-stage condenser 12 is equal to the heat absorption amount in the higher-stage evaporator 24. Thus, based on the necessary heat transfer amount Q1 in the lower-stage condenser 12, a refrigeration capacity Q2 of the higher-stage refrigeration cycle 20 is determined.

(S4)

[0044] Based on the lower-stage evaporating temperature and the higher-stage condensing temperature determined in accordance with the outdoor-air temperature, the target lower-stage condensing temperature Tc is determined. The controller 30 previously holds an approximation for obtaining the target lower-stage condensing temperature Tc using the lower-stage evaporating temperature and the higher-stage condensing temperature as variables and maximizing the COP and a corresponding map, and based on these information items, the target lower-stage condensing temperature Tc is determined. Since the higher-stage condensing temperature varies in accordance with the outdoor-air temperature, the target lower-stage condensing temperature Tc for maximizing the COP also varies in accordance with the outdoor-air temperature. Specifically, as the outdoor-air temperature increases, the higher-stage condensing temperature tends to increase, and the target lower-stage condensing temperature Tc also tends to increase.

(S5)

[0045] A higher-stage evaporating temperature is determined by subtracting a temperature difference ΔT from the target lower-stage condensing temperature Tc. In the higher-stage refrigeration cycle 20, the rotation speed of the higher-stage compressor 21 is controlled in such a manner that the higher-stage evaporating temperature becomes equal to the target higher-stage evaporating temperature (= target lower-stage condensing temperature Tc - ΔT degrees C).

(S6)

[0046] Based on the higher-stage evaporating temperature, an evaporating pressure of the higher-stage refrigeration cycle 20 is uniquely determined.

(S7)

[0047] Based on the evaporating pressure of the higher-stage refrigeration cycle 20, a refrigerant density p and an enthalpy difference ΔH of the higher-stage refrigeration cycle are determined.

(S8)

[0048] The following equations are obtained:

Refrigeration capacity Q2 of higher-stage refrigeration cycle 20 = Gr (refrigerant flow rate of higher-stage refrigeration cycle) × ΔH (enthalpy difference in higher-stage cascade condenser C) (1)

Refrigerant flow rate Gr of higher-stage refrigeration cycle 20 = ρ (refrigerant density of higher-stage refrigeration cycle 20) × Vst (displacement amount of higher-stage compressor 21) × N (rotation speed of higher-stage compressor 21) (2)

where although the "refrigeration capacity Q2 of higher-stage refrigeration cycle 20" and the "enthalpy difference ΔH" are already known, a "refrigerant flow rate Gr of higher-stage refrigeration cycle 20" is obtained from Equation (1). The displacement amount Vst of the higher-stage compressor 21 is a unique value to the compressor and is already known. Thus from Equation (2), a "rotation speed of higher-stage compressor 21" for setting a higher-stage evaporating temperature at a target higher-stage evaporating temperature (= target lower-stage condensing temperature Tc - ΔT degrees C) in the higher-stage refrigeration cycle 20 is determined.

[0049] The higher-stage compressor 21 is operated at the thus-determined rotation speed so that an operation can be controlled with substantially the same degree of the higher-stage compressor input and the lower-stage compressor input, thereby maximizing the COP. Here, in controlling the higher-stage refrigeration cycle 20, the target higher-stage evaporating temperature is determined so as to control the higher-stage evaporating temperature. Alternatively, the lower-stage condensing temperature may be directly detected for control. The higher-stage compressor input and the lower-stage compressor input may also be directly detected or computed so as to control the higher-stage refrigeration cycle 20. The higher-stage refrigeration cycle 20 may be controlled in accordance with the map or the approximation. As shown in Fig. 4, the control of the two-stage refrigeration cycle system is not limited to the control method of calculating the rotation speed of the higher-stage compressor 21 satisfying a target value based on principles of the refrigeration cycle, and may be a feedback control method based on a deviation between a target value (target lower-stage condensing temperature Tc) and a current value (current lower-stage condensing temperature) (a similar method can be applied to the lower-stage evaporating temperature).

[0050] The foregoing description is directed to the flow before the determination of the "rotation speed of higher-stage compressor 21." A flow before determination of a "rotation speed of lower-stage compressor 11" is similar to the flow described above. That is, the determination is made in the order of a lower-stage evaporating temperature, a lower-stage evaporating pressure, a lower-stage refrigerant density and a lower-stage enthalpy difference, a lower-stage flow rate, and the rotation speed of the lower-stage compressor 11.

[0051] In the foregoing description, the lower-stage high pressure (lower-stage condensing temperature) is reduced in order to reduce power consumption of the lower-stage refrigeration cycle 10 having a low efficiency. However, this description is only in control principle, and does not mean that the lower-stage high pressure is reduced in an actual operation. In the actual operation, the target lower-stage condensing temperature Tc is maintained at a constant value as described above.

[0052] Control principle of reducing the lower-stage high pressure will be additionally described. R32 used in the higher-stage refrigeration cycle 20 is a refrigerant with a higher efficiency (refrigerant for obtaining a higher COP) than CO_2 refrigerant used in the lower-stage refrigeration cycle 10. Thus, in the higher-stage refrigeration cycle 20, a slope θh on Mollier charts of Fig. 2 derived by an operation of the higher-stage compressor 21 is larger than a slope θl derived by an operation of the lower-stage compressor 11. Thus, as clearly shown in Fig. 3, even when the higher-stage compressor input is increased so as to reduce the lower-stage condensing temperature, the higher-stage compressor input does not exceed the lower-stage compressor input until the lower-stage condensing temperature reaches the target lower-stage condensing temperature Tc. Then, at the target lower-stage condensing temperature Tc, the higher-stage compressor input becomes equal to the lower-stage compressor input.

[0053] Then, the operating efficiency of refrigerant will be specifically described. If a theoretical COP (= enthalpy difference of evaporator / enthalpy difference in compression process) as an index of the operating efficiency is high, large evaporative latent heat can be obtained with small compression power, and this refrigerant is highly efficient refrigerant. For example, in an operation state of a typical single-stage refrigeration cycle system operating at an outdoor-air temperature of 32 degrees C, that is, an operation state in which the evaporating temperature is -40 degrees C, the condensing temperature is 40 degrees C (where a supercritical CO_2 high pressure is 8.8 MPa), the degree of suction superheat is 5 degrees C, and the degree of liquid subcooling is 5 degrees C, theoretical COPs for individual refrigerants are CO_2:1.25, R32: 1.98, HFO1234yf: 1.84, HFO1234ze: 1.97, propane: 1.99, isobutane: 2.05, ammonia: 2.07, R134a: 2.01, R410A: 1.91, R407C: 1.98, and R404A: 1.76. CO_2 has a lower COP and a lower efficiency than the other refrigerants such as HFO refrigerant, HFC refrigerant, and HC refrigerant.

[0054] In Embodiment 1, CO_2 is used as refrigerant in the lower-stage refrigeration cycle 10. In this case, in a high-temperature outdoor air condition of 32 degrees C, for example, the target lower-stage condensing temperature Tc is about 20 degrees C, and the target lower-stage condensing temperature Tc is lower than the outdoor-air temperature. Since the lower-stage compressor input in the lower-stage refrigeration cycle 10 having a low operating efficiency can be reduced by reducing the lower-stage high pressure (lower-stage condensing temperature) as described above, the target lower-stage condensing temperature Tc is located in a temperature range lower than the outdoor-air temperature.

[0055] Here, the target lower-stage condensing temperature Tc is located in the temperature range lower than the outdoor-air temperature in a case where the CO_2 refrigerant having a low efficiency is applied to the lower-stage refrigeration cycle 10. However, the locational relationship is not limited to this case depending on the combination of refrigerant types used in the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20. For example, the target lower-stage condensing temperature Tc is higher than the outdoor-air temperature in an operation at a low outdoor-air temperature, and the target lower-stage condensing temperature Tc is lower than the outdoor-air temperature in an operation at a high outdoor-air temperature. As such, the correlation between the target lower-stage condensing temperature Tc and the outdoor-air temperature changes with respect to a change in the outdoor-air temperature with some combinations of refrigerants.

(Difference in Heat Transfer Amount of Auxiliary Radiator 15 between Case where Lower-stage Condensing Temperature is Lower than Outdoor-air Temperature and Case where Lower-stage Condensing Temperature is Higher than Outdoor-air Temperature)

[0056] Then, the amount of heat transfer of the auxiliary radiator 15 will be described. In the two-stage refrigeration cycle system of Embodiment 1, since CO_2 refrigerant having a low operating efficiency is used in the lower-stage refrigeration cycle 10, the target lower-stage condensing temperature Tc is lower than the outdoor-air temperature at a high-temperature outdoor air condition of 32 degrees C. The auxiliary radiator 15 transfers heat of lower-stage refrigerant to the outdoor air. Thus, even when the auxiliary radiator 15 exchanges heat between lower-stage refrigerant discharged from the lower-stage compressor 11 and the outdoor air, the temperature of the lower-stage refrigerant decreases only to the outdoor-air temperature at the lowest. However, between a case where the lower-stage condensing temperature of the lower-stage refrigeration cycle 10 is lower than the outdoor-air temperature and a case where the lower-stage condensing temperature of the lower-stage refrigeration cycle 10 is higher than the outdoor-air temperature, the amount of heat transfer differs even when the auxiliary radiator 15 reduces the discharge temperature of the lower-stage refrigerant to the same outdoor-air temperature.

[0057] Fig. 5 shows Mollier charts of heat transfer amounts in the case where the lower-stage condensing temperature is lower than the outdoor-air temperature and the case where the lower-stage condensing temperature is higher than the outdoor-air temperature. Specifically, Fig. 5(1) shows a heat transfer enthalpy difference in the case where the lower-stage condensing temperature is higher than the outdoor-air temperature, and Fig. 5(2) shows a heat transfer enthalpy difference in the case where the lower-stage condensing temperature is lower than the outdoor-air temperature.

(1) Case where Lower-stage Condensing Temperature is Higher than Outdoor-air Temperature

[0058] A case where the temperature (temperature at point a) of refrigerant discharged from the lower-stage compressor 11 is, for example, 80 degrees C to 90 degrees C, the outdoor-air temperature is 20 degrees C, and the lower-stage condensing temperature is 25 degrees C will be described. Since the auxiliary radiator 15 transfers heat to the outdoor air, the temperature (point a) of refrigerant at 80 degrees C to 90 degrees C first deceases to 25 degrees C (point b) that is a condensing temperature while remaining in a gaseous state, through heat exchange with the outdoor air in the auxiliary radiator 15 as shown in Fig. 5(1). The refrigerant is condensed into a liquid state (point c) while remaining at 25 degrees C. Since the outdoor-air temperature is 20 degrees C, heat of the refrigerant can be further transferred, and the temperature of the refrigerant decreases to 20 degrees C (point d) in a liquid state. In this manner, since the refrigerant is condensed when the condensing temperature is higher than the outdoor-air temperature, cooling with a phase change can be performed, and the amount of heat transfer is larger than that of cooling without a phase change.

(2) Case where Lower-stage Condensing Temperature is Lower than Outdoor-air Temperature

[0059] A case where the temperature (temperature at point a) of refrigerant discharged from the lower-stage compressor 11 is, for example, 80 degrees C to 90 degrees C, the outdoor-air temperature is 20 degrees C, and the lower-stage condensing temperature is 10 degrees C will be described. Since the auxiliary radiator 15 transfers heat to the outdoor air, the temperature of refrigerant at 80 degrees C to 90 degrees C decreases only to 20 degrees C that is the outdoor-air temperature to the lowest, through heat exchange with the outdoor air in the auxiliary radiator 15. Specifically, as shown in Fig. 5(2), the temperature (point a) of refrigerant at 80 degrees C to 90 degrees C decreases to 20 degrees C (point b) in the auxiliary radiator 15 while remaining in a gaseous state. That is, in a case where the lower-stage condensing temperature is lower than the outdoor-air temperature, the auxiliary radiator 15 cannot perform cooling with a phase change but performs gas cooling without a phase change. That is, the auxiliary radiator 15 is used in a gas cooling range.

[0060] Here, since heat transfer from point a to point b in Fig. 5(2) is heat transfer in a gaseous state, the amount of heat transfer by the auxiliary radiator 15 in reducing the refrigerant temperature to an identical outdoor-air temperature of 20 degrees C cannot be increased as compared to the case of (1) in which the refrigerant is condensed so as to reduce the temperature to 20 degrees C. Thus, in the case where the lower-stage condensing temperature is lower than the outdoor-air temperature, even when the airflow rate of the auxiliary radiator 15 is increased or a radiator having a large heat transfer area is employed as the auxiliary radiator 15, the amount of heat transfer by the auxiliary radiator 15 cannot be increased and is, at most, increased to the amount of heat transferred until the discharged refrigerant has its temperature decrease to the outdoor-air temperature while maintaining a gaseous state.

(Relationship between Heat Transfer Amount of Auxiliary Radiator 15 and COP)

[0061] Fig. 6 is a diagram for showing a relationship between the amount of heat transfer by the auxiliary radiator 15 and a COP. Fig. 6 is a Mollier chart of the lower-stage refrigeration cycle 10. In constituting the lower-stage refrigeration cycle 10, a comparison between a case where the amount of heat transfer by the auxiliary radiator 15 is Qsub1 in Fig. 6 and a case where the amount of heat transfer by the auxiliary radiator 15 is Qsub2 in Fig. 6 shows that a heat transfer amount Qc2 (< Qc1) of the corresponding lower-stage condenser 12 in the case of Qsub2 is smaller than that in the other case. In the cascade condenser C, the amount of heat exchange of the higher-stage evaporator 24 is equal to that of the lower-stage condenser 12. Thus, the higher-stage refrigeration cycle 20 only needs to obtain a balance with the heat transfer amount Qc2 in the lower-stage condenser 12, and thus, the higher-stage compressor input in the case where the heat transfer amount of the auxiliary radiator 15 is Qsub2 than that in the case where the heat transfer amount of the auxiliary radiator 15 is Qsub1. As the heat transfer amount of the auxiliary radiator 15 increases, the COP value increases.

[0062] In the two-stage refrigeration cycle system, since control with a constant refrigeration capacity is performed and the relationship of COP = refrigeration capacity / (higher-stage compressor input + lower-stage compressor input) is established, if the higher-stage compressor input can be reduced, the COP can be increased.

[0063] As described above, in the two-stage refrigeration cycle system of Embodiment 1, the auxiliary radiator 15 is used in the gas cooling range. Thus, irrespective of the configuration such as the size of the heat transfer area of the auxiliary radiator 15, the temperature of discharged refrigerant is reduced to the outdoor-air temperature even at the maximum heat transfer. As described above, as the heat transfer amount of the auxiliary radiator 15 increases, the COP can be increased. Thus, the heat transfer amount of the auxiliary radiator 15 is obtained to such a degree that the discharge temperature of refrigerant at the auxiliary radiator 15 can be reduced to a temperature around the outdoor-air temperature. The "heat transfer amount of the auxiliary radiator 15 in reducing the discharge temperature of refrigerant to a temperature around the outdoor-air temperature through heat transfer in the auxiliary radiator 15" will be hereinafter referred to as a necessary heat transfer amount. To obtain the necessary heat transfer amount, the airflow rate of the auxiliary radiator 15 is controlled or the configuration of the auxiliary radiator 15 itself is designed, for example. In this manner, the heat transfer amount of the auxiliary radiator 15 is set at the necessary heat transfer amount, thereby increasing the COP as compared to the case of a heat transfer amount smaller than the necessary heat transfer amount.

[0064] The necessary heat transfer amount varies in accordance with the outdoor-air temperature. Thus, to obtain a high COP through the year, a necessary heat transfer amount in a low-temperature outdoor air condition and a necessary heat transfer amount in a high-temperature outdoor air condition need to be known. In the two-stage refrigeration cycle system according to Embodiment 1, the auxiliary radiator 15 is used in the gas cooling range and the necessary heat transfer amount is smaller as described above. However, as described above, with some combinations of refrigerant types in the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20, the target lower-stage condensing temperature Tc is higher in an operation at a low outdoor-air temperature and the target lower-stage condensing temperature Tc is lower in an operation at a high outdoor-air temperature, for example. Thus, the correlation of the target lower-stage condensing temperature Tc with the outdoor-air temperature changes, and the necessary heat transfer amount changes.

[0065] For example, in a case where the lower-stage condensing temperature is lower than the outdoor-air temperature, the auxiliary radiator 15 cannot perform cooling with a phase change, and the necessary heat transfer amount decreases, as described above. The heat transfer amount of the auxiliary radiator 15 is, at most, the amount of heat transferred until the discharge refrigerant has its temperature reduced to the outdoor-air temperature while remaining in the gaseous state. Thus, even when the airflow rate of the auxiliary radiator 15 is increased, the heat transfer amount of the auxiliary radiator 15 cannot increase. In contrast, unless the airflow rate of the auxiliary radiator 15 is reduced and optimized, fan input is unnecessarily consumed, leading to a COP decrease. Accordingly, the heat transfer amount of the auxiliary radiator 15 is obtained to such a degree that the auxiliary radiator 15 can reduce the discharge temperature of refrigerant to a temperature around the outdoor-air temperature without wastefully increasing fan input. In this manner, by reducing the airflow rate of the auxiliary radiator 15, fan input can be optimized, and the COP of the entire two-stage refrigeration cycle system can be enhanced.

[0066] On the other hand, in a case where the lower-stage condensing temperature is higher than the outdoor-air temperature, the auxiliary radiator 15 performs cooling with a phase change, and the necessary heat transfer amount increases. At this time, the airflow rate of the auxiliary radiator 15 is continuously increased as the necessary heat transfer amount increases, and the COP of the entire two-stage refrigeration cycle system can be enhanced by increasing the heat transfer amount of the auxiliary radiator 15.

[0067] In the case where the lower-stage condensing temperature is lower than the outdoor-air temperature, the airflow rate control of the auxiliary radiator 15 with respect to a change in the necessary heat transfer amount is performed in the following manner. Specifically, the control is performed in such a manner that a temperature difference between an outlet refrigerant temperature of the auxiliary radiator 15 and the outdoor-air temperature is a predetermined value (which is about 2 degrees C in this example). In this manner, the airflow rate of the auxiliary radiator 15 can be appropriately adjusted, thereby enhancing the COP of the entire two-stage refrigeration cycle system.

[0068] As described above, the heat transfer amount of the auxiliary radiator 15 with respect to the outdoor-air temperature can be appropriately controlled by controlling the airflow rate of the auxiliary radiator 15 so that a high degree of energy conservation can be obtained through the year.

[0069] In a case where the auxiliary radiator 15 is supposed to be used in a gas cooling range where the necessary heat transfer amount is small, a sufficient heat transfer area of the auxiliary radiator 15 is about 10 to 20% of the heat transfer area of the higher-stage condenser 22, and is sufficient. On the other hand, In a case where the auxiliary radiator 15 is supposed to be used in a gas cooling range where the necessary heat transfer amount is large, the heat transfer area of the auxiliary radiator 15 is enlarged to a degree substantially equal to that of the higher-stage condenser 22, and the COP of the entire two-stage refrigeration cycle system can be enhanced by significantly increasing the heat transfer amount of the auxiliary radiator 15. The auxiliary radiator 15 and the higher-stage condenser 22 can have similar shapes so that components can be commonly used, thereby enabling cost reduction.

[0070] In the case where the heat transfer area of the auxiliary radiator 15 is substantially equal to that of the higher-stage condenser 22, the airflow rate of the auxiliary radiator 15 is increased as the necessary heat transfer amount increases, and the heat transfer amount of the auxiliary radiator 15 significantly increases. As the heat transfer amount of the auxiliary radiator 15 significantly increases, the heat transfer amount in the lower-stage condenser 12 of the cascade condenser C decreases, and the cooling capacity at the higher-stage refrigeration cycle also decreases. Thus, the lower-stage condensing temperature cannot be controlled by promoting heat transfer in the lower-stage condenser 12 through control of the cooling capacity of the higher-stage refrigeration cycle. That is, when the heat transfer amount of the auxiliary radiator 15 greatly exceeds the heat transfer amount of the lower-stage condenser 12, the lower-stage condensing temperature depends on the heat transfer amount of the auxiliary radiator 15.

[0071] When the airflow rate of the auxiliary radiator 15 is increased, fan input increases but the lower-stage condensing temperature decreases. Thus, the lower-stage compressor input can be reduced. However, after the lower-stage condensing temperature has decreased and approaches the outdoor-air temperature, the lower-stage condensing temperature does not decrease even by increasing the airflow rate of the auxiliary radiator 15, resulting in wasted consumption of the fan input. Thus, in the case where the heat transfer area of the auxiliary radiator 15 is substantially equal to that of the higher-stage condenser 22, the lower-stage condensing temperature is higher than the outdoor-air temperature by a predetermined temperature (about 10 degrees C in this example) so that the lower-stage compressor input and the fan input can be optimized and the COP of the entire two-stage refrigeration cycle system can be enhanced.

(Features of the Present Invention)

[0072] Features of the present invention will now be described. In the present invention, the auxiliary radiator 15 provided in the lower-stage refrigeration cycle 10 is used as an aid for the lower-stage condenser 12 so as to enhance the operating efficiency in a two-stage operation, whereas the auxiliary radiator 15 is used as a main radiator in a single-stage operation. That is, a feature of the present invention is that the auxiliary radiator 15 is used in both the two-stage operation and the single-stage operation. In an operation at a low outdoor-air temperature, the operation mode is switched to the single-stage operation whose COP is higher than that of the two-stage operation. In this manner, as another feature of the present invention, an operation with a higher COP is selected from the two-stage operation and the single-stage operation. In addition, as still another feature of the present invention, in performing the single-stage operation, the auxiliary radiator 15 is used as a main radiator so that the lower-stage refrigeration cycle 10 can be used without switching the channel, unlike Patent Literature 2.

[0073] With the foregoing features of the two-stage refrigeration cycle system of the present invention, the operating efficiency can be enhanced by using the auxiliary radiator 15, and in addition, performance degradation in an operation at a low compression ratio is avoided so that an energy conservation effect can be achieved through the year. These advantages will be hereinafter described more specifically.

[0074] When the two-stage refrigeration cycle system operates at a low outdoor-air temperature, the higher-stage condensing temperature decreases, and the lower-stage condensing temperature decreases accordingly. In this case, each of the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20 operates at a low compression ratio, the compressor performance thereof degrades, and operations of these cycles deviate from operation ranges defined as standards. Thus, reliability cannot be maintained. In an operation at a low compression ratio, the ratio of the temperature difference ΔT (= lower-stage condensing temperature - higher-stage evaporating temperature) of the cascade condenser C increases with respect to the temperature difference between the lower-stage evaporating temperature and the higher-stage condensing temperature. Thus, the influence of performance degradation increases.

[0075] In general, compressor performance changes depending on the compression ratio and the rotation speed. A compressor is designed such that the compressor performance is at the maximum at a compression ratio at which the compressor is supposed to be most frequently used. Thus, when the compression ratio during operation becomes extremely lower or higher than this compression ratio, performance degrades significantly.

[0076] In view of this, in order to avoid such performance degradation and reliability degradation, the two-stage refrigeration cycle system of the Embodiment 1 performs the following control. Specifically, if it is determined that the outdoor-air temperature has decreased to the "predetermined outdoor-air temperature" or less, which might cause a COP decrease if the two-stage operation continues without change, in the two-stage operation, the controller 30 stops the higher-stage refrigeration cycle 20 and causes only the lower-stage refrigeration cycle 10 to operate. That is, the two-stage operation is switched to the single-stage operation. A specific flow of refrigerant in the single-stage operation is as follows. That is, refrigerant compressed in the lower-stage compressor 11 and discharged therefrom is subjected to heat transfer and cooling only by using the auxiliary radiator 15, and the pressure of the refrigerant cooled by the auxiliary radiator 15 is reduced by the lower-stage expansion valve 13. Then, the resulting refrigerant is evaporated by the lower-stage evaporator 14 and is fed back to the lower-stage compressor 11. The "predetermined outdoor-air temperature," that is, a threshold outdoor-air temperature used for determining whether a COP decrease is induced or not will be described later.

[0077] As described above, in the low outdoor-air temperature, the two-stage operation is switched to the single-stage operation, and only the lower-stage refrigeration cycle 10 operates. Thus, the lower-stage compressor 11 is kept at an appropriate compression ratio so that sufficient performance and reliability can be obtained. In addition, structural problems typically found in the two-stage refrigeration cycle system can be avoided at the same time. Specifically, it is possible to avoid performance degradation caused by an increased ratio of the temperature difference ΔT of the cascade condenser C with respect to the temperature difference between the lower-stage evaporating temperature and the higher-stage condensing temperature. In addition, in performing the single-stage operation, the auxiliary radiator 15 is used as a main radiator without change including switching of channel. Thus, it is unnecessary to additionally provide a heat exchanger, a switching valve, and a bypass pipe, and an increase in costs can also be avoided. As a result, operation control at the low outdoor-air temperature can be optimized without an increase in costs, and a high degree of energy conservation can be achieved through the year.

[0078] A threshold outdoor-air temperature as a switching point between the two-stage operation and the single-stage operation will be described.

[0079] Fig. 7 shows an outdoor-air temperature to COP characteristic in each of a single-stage operation and a two-stage operation (with an auxiliary radiator) in the two-stage refrigeration cycle system illustrated in Fig. 1. Fig. 7 also shows an outdoor-air temperature to COP characteristic (two-stage operation (without an auxiliary radiator)) in a two-stage operation of a conventional two-stage refrigeration cycle system including no auxiliary radiator 15 for comparison. Fig. 7 is a graph showing outdoor-air temperature to COP characteristics in a case where a low pressure Ps is fixed at a certain pressure value and a high pressure Pd is determined in accordance with the outdoor-air temperature.

[0080] As shown in Fig. 7, in any one of the operations, the COP tends to decrease as the outdoor-air temperature increases, that is, the cooling load increases. A temperature Tca at a point at which a characteristic of the single-stage operation and a characteristic of the two-stage operation (with an auxiliary heat exchanger) intersect each other is a threshold temperature for determining whether a COP decrease occurs or not. That is, if the outdoor-air temperature is higher than the threshold outdoor-air temperature Tca, the COP in the two-stage operation is higher than that in the single-stage operation, whereas if the outdoor-air temperature is less than or equal to the threshold outdoor-air temperature Tca, the COP of the single-stage operation is higher than that of the two-stage operation. Thus, in the two-stage refrigeration cycle system of Embodiment 1, an operation is performed with switching the operation mode to an operation with a higher COP based on a comparison result between the outdoor-air temperature and the threshold outdoor-air temperature Tca. The characteristics shown in Fig. 7 are obtained beforehand by experiment or simulation, and thus, the threshold outdoor-air temperature Tca can also be obtained beforehand.

[0081] In Fig. 7, Tc is a target lower-stage condensing temperature in the two-stage operation (with an auxiliary radiator). In a case where the outdoor-air temperature is Tc or less, the rate of a COP increase in the two-stage operation (with an auxiliary radiator) becomes high from the Tc. This is because if the outdoor-air temperature is less than or equal to the lower-stage condensing temperature Tc as described with reference to Fig. 5, lower-stage refrigerant is condensed in the auxiliary radiator 15 so that cooling with a phase change can be performed, and the amount of heat transfer is larger than that in cooling without a phase change. As described with reference to Fig. 6, as the heat transfer amount in the auxiliary radiator 15 increases, the COP increases. For the foregoing reasons, a comparison between a case where the outdoor-air temperature is less than or equal to the lower-stage condensing temperature Tc and a case where the outdoor-air temperature is higher than the lower-stage condensing temperature Tc shows that the rate of the COP increase is higher in the case where the outdoor-air temperature is less than or equal to the lower-stage condensing temperature Tc than that in the case where the outdoor-air temperature is higher than the lower-stage condensing temperature Tc.

[0082] Similarly, switching from the single-stage operation to the two-stage operation is also based on the threshold outdoor-air temperature Tca, and an operation mode with a high COP is selected. To prevent frequent switchings, the switching direction from the two-stage operation to the single-stage operation and switching in the reverse direction are provided with a hysteresis or switching is not performed in a predetermined time. Since the threshold outdoor-air temperature Tca changes with the lower-stage evaporating temperature, the threshold outdoor-air temperature Tca may be set by using an approximation using the lower-stage evaporating temperature as a variable or a map.

[0083] In Embodiment 1, CO_2 refrigerant is used in the lower-stage refrigeration cycle 10 as described above. In a case where the high pressure exceeds a critical pressure in a single-stage operation, performance significantly degrades. Thus, the threshold outdoor-air temperature Tca used for switching from the two-stage operation to the single-stage operation in the case where CO_2 refrigerant is used in the lower-stage refrigeration cycle 10 is set in such a manner that the high pressure does not exceed the critical pressure after switching to the single-stage operation, which will be described below.

[0084] In consideration of an example of a design guideline of a heat exchanger in which a radiator is designed such that the condensing temperature is higher than the outdoor-air temperature by about 10 degrees C, the threshold outdoor-air temperature Tca for use in switching from the two-stage operation to the single-stage operation in the case where CO_2 refrigerant is used in the lower-stage refrigeration cycle 10 is 21 degrees C, which is lower by 10 degrees C than 31 degrees C that is a critical saturation temperature. That is, when the outdoor-air temperature decreases to 21 degrees C or less in the two-stage operation, the operation mode is switched to the single-stage operation. The timing of returning from the single-stage operation to the two-stage operation is the time when a refrigerant temperature at the higher-pressure side becomes a critical saturation temperature of 31 degrees C. By performing such a switching operation, even in the case where CO_2 refrigerant is used in the lower-stage refrigeration cycle 10, the high pressure does not exceed the critical pressure in the single-stage operation, and occurrence of significant performance degradation can be avoided.

[0085] Fig. 8 is a flowchart showing an operation of the two-stage refrigeration cycle system illustrated in Fig. 1. The process shown in the flowchart of Fig. 8 is repeatedly performed at every control interval, for example.

[0086] The controller 30 compares an outdoor-air temperature detected by the outdoor-air temperature detecting means 31 and a previously determined threshold outdoor-air temperature Tca (S11). If the outdoor-air temperature is less than or equal to the threshold outdoor-air temperature Tca, the controller 30 performs a single-stage operation (S12), whereas if the outdoor-air temperature is higher than the threshold outdoor-air temperature Tca, the controller 30 performs a two-stage operation (S13). Thus, if the outdoor-air temperature decreases to the threshold outdoor-air temperature Tca or less in the two-stage operation, the two-stage operation is switched to the single-stage operation. If the outdoor-air temperature exceeds the threshold outdoor-air temperature Tca in the single-stage operation, the single-stage operation is switched to the two-stage operation.

[0087] In the above description, the outdoor-air temperature is used as a threshold value for switching from the two-stage operation to the single-stage operation. Alternatively, the switching may be based on the following threshold values.

(1) Switching based on Compression Ratio

[0088] Compressor performance generally depends on a compression ratio, and thus, an operation switching may be performed based on the compression ratio of the lower-stage refrigeration cycle 10 or the higher-stage refrigeration cycle 20. Specifically, if the compression ratio of the lower-stage refrigeration cycle 10 or the higher-stage refrigeration cycle 20 is less than or equal to a threshold compression ratio that is a threshold value for switching, the single-stage operation is performed, whereas if the compression ratio of the lower-stage refrigeration cycle 10 or the higher-stage refrigeration cycle 20 is higher than the threshold compression ratio, the two-stage operation is performed. The threshold compression ratio is set (at, for example, a compression ratio of 2.0) based on compressor performance or reliability.

[0089] Thus, if the compression ratio of the lower-stage refrigeration cycle 10 or the higher-stage refrigeration cycle 20 in the two-stage operation becomes less than or equal to the threshold compression ratio, the two-stage operation is switched to the single-stage operation. This ensures prevention of performance degradation of the operation at a low compression ratio and prevention of reliability degradation. Similarly, in the reverse direction from the single-stage operation to the two-stage operation, if the compression ratio of the lower-stage refrigeration cycle 10 becomes higher than the threshold compression ratio in the single-stage operation, the single-stage operation is switched to the two-stage operation. In this manner, it is possible to ensure prevention of compressor performance degradation caused by a high compression ratio. In addition, the operation switching may be performed based on a compression ratio (compression ratio based on an evaporating pressure of the lower-stage refrigeration cycle 10 and a condensing pressure of the higher-stage refrigeration cycle 20) of the entire two-stage refrigeration cycle in the two-stage operation. In this case, if the compression ratio of the entire two-stage refrigeration cycle is less than or equal to the threshold compression ratio (e.g., a compression ratio of 4.0), the two-stage operation is switched to the single-stage operation.

(2) Switching based on Temperature Difference between Outdoor-air Temperature and Lower-stage Condensing Temperature

[0090] In the case of using CO_2 as lower-stage refrigerant, the target lower-stage condensing temperature Tc at which the COP is at the maximum is lower than the outdoor-air temperature, and as the outdoor-air temperature decreases, the target lower-stage condensing temperature Tc also decreases while maintaining this relationship. That is, as the outdoor-air temperature decreases, the target lower-stage condensing temperature Tc tends to decrease while being kept lower than the outdoor-air temperature. As the outdoor-air temperature decreases, the temperature difference between the outdoor-air temperature and the target lower-stage condensing temperature Tc decreases. Thus, while the temperature difference between the outdoor-air temperature and the target lower-stage condensing temperature Tc is larger than a predetermined value a, the two-stage operation is performed, whereas when the temperature difference between the outdoor-air temperature and the target lower-stage condensing temperature Tc becomes less than or equal to the predetermined value a, the two-stage operation may be switched to the single-stage operation. This will be described with reference to Fig. 9.

[0091] Fig. 9 is a graph showing a relationship between an outdoor-air temperature and a threshold outdoor-air temperature Tca in the two-stage refrigeration cycle system according to the Embodiment 1 of the present invention, where the abscissa represents the outdoor-air temperature and the ordinate represents a temperature (outdoor-air temperature - Tc) obtained by subtracting the target lower-stage condensing temperature Tc from the outdoor-air temperature.

[0092] As shown in Fig. 9, the threshold outdoor-air temperature Tca may be replaced by the predetermined value a obtained by subtracting the target lower-stage condensing temperature Tc from the outdoor-air temperature. Thus, the operation mode may be switched in the following manner. Specifically, if the temperature obtained by subtracting the target lower-stage condensing temperature Tc from the outdoor-air temperature is higher than the predetermined value a, the two-stage operation may be performed, whereas if the temperature obtained by subtracting the target lower-stage condensing temperature Tc from the outdoor-air temperature is less than or equal to the predetermined value a, the single-stage operation may be performed.

[0093] Fig. 9 shows a temperature range in which the "outdoor-air temperature - Tc" decreases to a negative value, that is, Tc becomes higher than the outdoor-air temperature. This is because of the following reasons. A system has a lower limit of the compression ratio. Thus, even when the outdoor-air temperature decreases, the target lower-stage condensing temperature Tc does not decrease below a certain temperature. Accordingly, the relationship between the outdoor-air temperature and the Tc is reversed so that the Tc becomes higher than the outdoor-air temperature in some cases.

[0094] In this manner, prevention of performance degradation can also be ensured by switching from the two-stage operation to the single-stage operation based on the temperature obtained by subtracting the target lower-stage condensing temperature Tc from the outdoor-air temperature.

(3) Switching (Single-stage Operation to Two-stage Operation) based on High Pressure

[0095] Switching from the single-stage operation to the two-stage operation is performed in the following manner. Specifically, when a high pressure Pd in the single-stage operation exceeds a high pressure Pd immediately after switching from the two-stage operation to the single-stage operation, the single-stage operation is switched to the two-stage operation. In this manner, it is possible to ensure selection of an operation mode with a high COP.

[0096] The threshold value used in switching from the single-stage operation to the two-stage operation in the forward direction (from the two-stage operation to the single-stage operation) and the threshold value used in switching from the two-stage operation to the single-stage operation in the reverse direction (from the single-stage operation to the two-stage operation) are not limited to the same threshold value, but may be different from each other. That is, the threshold outdoor-air temperature Tca in (1) may be used for switching in the forward direction whereas the high pressure Pd in (3) being used for switching in the reverse direction, for example.

[0097] Here, heat exchangers used for the higher-stage condenser 22 and the auxiliary radiator 15 will be described. The higher-stage condenser 22 and the auxiliary radiator 15 are plate-fin-tube type heat exchangers in which heat transfer tubes penetrate heat transfer fins having flat-plate shapes. As shown in Fig. 10, the higher-stage condenser 22 and the auxiliary radiator 15 may share heat transfer fins 40 so as to form an integrated radiator 42, or heat transfer fin parts may be divided. If the heat transfer fins 40 are integrated, fabrication is easily performed in terms of the configuration of the heat exchangers.

[0098] Fig. 10 illustrates a configuration of the two-stage refrigeration cycle system in a case where the higher-stage condenser and the auxiliary radiator illustrated in Fig. 1 form an integrated radiator.

[0099] In Fig. 10, reference numeral 43 denotes an air-sending device for sending air to the integrated radiator 42 obtained by integrating the higher-stage condenser 22 and the auxiliary radiator 15.

[0100] Since a high-temperature refrigerant gas discharged from the lower-stage compressor 11 passes through the auxiliary radiator 15, the auxiliary radiator 15 is at a high temperature. Thus, in a case where the heat transfer fins are divided between the auxiliary radiator 15 at the high temperature and the higher-stage condenser 22, an excellent heat insulating effect is obtained, and both the auxiliary radiator 15 and the higher-stage condenser 22 can more efficiently transfer heat.

[0101] In the case where the higher-stage condenser 22 and the auxiliary radiator 15 form the integrated radiator 42, the auxiliary radiator 15 is disposed in an upper portion (above in the direction of gravity) and the higher-stage condenser 22 is disposed in a lower portion (below in the direction of gravity) as shown in Fig. 10. Since the auxiliary radiator 15 at a high temperature is disposed in the upper portion of the heat exchangers, the heat transfer of the auxiliary radiator 15 does not interfere with the higher-stage condenser 22. That is, heat transfer target fluid heated by the auxiliary radiator 15 does not move toward the higher-stage condenser 22, and both the auxiliary radiator 15 and the higher-stage condenser 22 can efficiently transfer heat.

[0102] In the case where the air-sending device 43 is shared by the higher-stage condenser 22 and the auxiliary radiator 15 as shown in Fig. 10, in switching from the two-stage operation to the single-stage operation, the flow rate of air passing through the higher-stage condenser 22 is wasted. In view of this, the integrated radiator 42 may be configured as illustrated in Fig. 11.

[0103] Fig. 11 is an illustration of an example configuration in the case where the higher-stage condenser and the auxiliary radiator shown in Fig. 1 form the integrated radiator. Fig. 11 schematically illustrates a configuration of the heat transfer fins.

[0104] The integrated radiator 42 includes a plurality of heat transfer fins 40 spaced from each other so as to allow air to pass therebetween, and a plurality of heat transfer tubes 41 penetrating the heat transfer fins 40. The heat transfer tubes 41 are arranged in a plurality of columns in a direction perpendicularly to an air passage direction (vertically in Fig. 11) and in a plurality of rows in the air passage direction (laterally in Fig. 11). Among the heat transfer tubes 41, the heat transfer tubes 41 constituting the auxiliary radiator 15 are collected in one row. This configuration can obtain a flow rate of the auxiliary radiator 15 with no waste in the single-stage operation. As a result, a high flow rate can be obtained in the single-stage operation, thereby enhancing performance.

[0105] It is known that the heat transfer area of the auxiliary radiator 15 is about 10 to 20% of the heat transfer area of the higher-stage condenser 22 in order to maximize a COP in a two-stage operation in a region where the auxiliary radiator 15 is frequently used in a gas range and the outdoor-air temperature is high through the year. On the other hand, in consideration of energy conservation through the year including a single-stage operation in a region where the outdoor-air temperature is low, the heat transfer area of the auxiliary radiator 15 is preferably increased to be substantially equal to that of the higher-stage condenser 22 so as to enhance a COP in a single-stage operation at a low outdoor-air temperature. At this time, components can be commonly used by setting the heat transfer areas of the auxiliary radiator 15 and the higher-stage condenser 22 at substantially the same degree, thereby enabling cost reduction.

[0106] As described above, in the two-stage operation, an operation is performed using the target lower-stage condensing temperature Tc at which the COP is at the maximum as a target value. In a case where different refrigerants are used in the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20, the lower-stage condensing temperature is used as a target value in such a manner that the compression ratio in the case of using a refrigerant with a lower theoretical COP is lower and the compression ratio in the case of using a refrigerant with a higher theoretical COP is higher. Since the compression ratios differ between the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20, the compression ratio might be extremely low or high. In particular, in the case of using CO_2 refrigerant with a low theoretical COP or mixed refrigerant including CO_2 in one of the refrigeration cycles, difference in the compression ratio becomes conspicuous.

[0107] As described above, since the compression ratios of both the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20 do not become appropriate at the same time, at least one of these cycles shows degradation of compressor performance. Thus, in the two-stage refrigeration cycle system using different types of refrigerant, the single-stage operation capable of avoiding performance degradation at a low compression ratio or a high compression ratio is especially effective, thereby significantly enhancing an energy conservation effect through the year.

[0108] In particular, in the case of using a refrigerant with a low theoretical COP in the lower-stage refrigeration cycle 10 and a refrigerant with a high theoretical COP in the higher-stage refrigeration cycle 20 as in Embodiment 1 in which CO_2 is used in the lower-stage refrigeration cycle 10, the lower-stage condensing temperature with which the compression ratio of the lower-stage refrigeration cycle is low is used as a target. Thus, the low-pressure side compression ratio significantly decreases at the low outdoor-air temperature. In view of this, the single-stage operation of the lower-stage refrigeration cycle 10 that avoids performance degradation due to a compression ratio decrease of the lower-stage refrigeration cycle is especially effective, and an energy conservation effect through the year can be significantly enhanced.

[0109] Examples of the refrigerant with a high theoretical COP include R32, R410A, R134a, R404A, R407C, HFO1234yf, HFO1234ze, ammonia, propane, and isobutane. The present invention includes a configuration in which the refrigerant with a high theoretical COP is used in at least one of the lower-stage refrigeration cycle 10 and the higher-stage refrigeration cycle 20.

[0110] As described above, in Embodiment 1, the lower-stage refrigeration cycle 10 includes the auxiliary radiator 15 so that the auxiliary radiator 15 is used as an aid for the lower-stage condenser 12 in the two-stage operation, whereas the auxiliary radiator 15 is used as a main radiator in the single-stage operation. Switching between the two-stage operation and the single-stage operation is performed in such a manner than an operation with a higher COP is performed. In this manner, a high operating efficiency can be obtained through the year, thereby achieving energy conservation. Since the auxiliary radiator 15 is used as a main radiator in the single-stage operation, the lower-stage refrigeration cycle 10 can be used without change including switching of channel. Thus, none of an air-cooled radiator, a switching valve, and a bypass channel needs to be additionally provided for the single-stage operation, and thus, cost reduction can be achieved.

Embodiment 2

[0111] In a two-stage refrigeration cycle system according to Embodiment 2, CO_2 is used in a lower-stage refrigeration cycle 10 and a design pressure of the lower-stage refrigeration cycle 10 is reduced to about 4.15 MPa, which is a design pressure approximately equal to that of HFC refrigerant, corresponding to R410A, for example.

[0112] A refrigerant operating pressure of CO_2 is higher than that of a conventional HFC refrigerant such as R404A or R410A. Thus, to use CO_2 in the lower-stage refrigeration cycle 10 designed on the assumption that the conventional HFC refrigerant such as R404A or R410A is used therein, an additional component with a high design pressure needs to be used, which leads to a significant cost increase. Thus, in order to reduce costs, component of the conventional lower-stage refrigeration cycle using an operating refrigerant as the HFC refrigerant are diverted. In view of this, in Embodiment 2, the design pressure of the lower-stage refrigeration cycle 10 is not increased but is reduced to about 4.15 MPa, which is a design pressure approximately equal to that of the HFC refrigerant, corresponding to R41 0A, for example. A configuration enabling the design pressure of the lower-stage refrigeration cycle to be reduced to about 4.15 MPa will be described with reference to Figs. 12 and 13 later.

[0113] In the following description, aspects of Embodiment 2 different from those of Embodiment 1 will be mainly described. Variations applied to components similar to those of Embodiment 1 will be similarly applicable to Embodiment 2.

[0114] Here, reasons for necessity of increase in the design pressure of the lower-stage refrigeration cycle 10 in the case of using CO_2 in the lower-stage refrigeration cycle 10 will be described again.

[0115] Since a single-stage operation is performed in an operation with a low load, the lower-stage compressor 11 is supposed to be repeatedly started and stopped in the single-stage operation. When a lower-stage compressor 11 of the lower-stage refrigeration cycle 10 stops, refrigerant is heated to a temperature around an outdoor-air temperature and becomes gas so that the pressure in the lower-stage refrigeration cycle 10 increases. For example, in a case where the ambient temperature is high and refrigerant reaches a critical state while the lower-stage compressor 11 stops, the pressure in the lower-stage refrigeration cycle 10 might exceeds the design pressure with some inner capacity and amount of enclosed refrigerant in the lower-stage refrigeration cycle 10.

[0116] Such a pressure increase while the lower-stage refrigeration cycle 10 stops can be solved by starting a higher-stage refrigeration cycle 20 so as to cool the lower-stage refrigeration cycle 10. However, alternate starting and stopping of the higher-stage refrigeration cycle 20 and the lower-stage refrigeration cycle 10 do not lead to energy conservation because of the influence of an ON/OFF loss. In addition, the system cannot cope with a blackout and an abnormal situation such as a failure in the higher-stage compressor 21. In view of this, the two-stage refrigeration cycle system is configured as illustrated in Fig. 12.

[0117] Fig. 12 illustrates an example configuration 1 of the two-stage refrigeration cycle system according to Embodiment 2 of the present invention. In Fig. 12, the same reference characters designate the same parts in Fig. 1.

[0118] In the two-stage refrigeration cycle system illustrated in Fig. 12, an expansion tank 32 is connected to a line between the lower-stage compressor 11 and the lower-stage evaporator 14 of the lower-stage refrigeration cycle 10 illustrated in Fig. 1 through a solenoid valve 33. The inner capacity of the lower-stage refrigeration cycle 10 can be enlarged by opening the solenoid valve 33 so that the expansion tank 32 communicates with the lower-stage refrigeration cycle 10.

[0119] In the thus-configured two-stage refrigeration cycle system, while the lower-stage refrigeration cycle 10 stops, the solenoid valve 33 is opened so that refrigerant in the lower-stage refrigeration cycle 10 is recovered to the expansion tank 32. The solenoid valve 33 is of an exciting close type, and is open even in a blackout so that refrigerant can be recovered to the expansion tank 32. In this manner, since the refrigerant in the lower-stage refrigeration cycle 10 can be recovered to the expansion tank 32, it is possible to prevent the pressure in the lower-stage refrigeration cycle 10 from exceeding a set pressure.

[0120] In addition, the expansion tank 32 is provided in the low-pressure side, especially a suction port of the lower-stage compressor 11, in such a manner that refrigerant in the expansion tank 32 can be recovered to the lower-stage refrigeration cycle 10 when the lower-stage refrigeration cycle 10 is restarted. To enable refrigerant recovery from the lower-stage refrigeration cycle 10 to the expansion tank 32 when the solenoid valve 33 is open, the inside of the expansion tank 32 is always kept at a low pressure. If the expansion tank 32 has been cooled, refrigerant recovery from the lower-stage refrigeration cycle 10 to the expansion tank 32 can be further promoted.

[0121] Another configuration as illustrated in Fig. 13 may also be employed.

[0122] Fig. 13 illustrates an example configuration 2 of the two-stage refrigeration cycle system according to Embodiment 2 of the present invention. In Fig. 13, the same reference characters designate the same parts in Fig. 1.

[0123] A second expansion valve (second expansion device) 34 is provided in an upstream part of a liquid pipe 16 between a cascade condenser C and a lower-stage expansion valve 13 in the lower-stage refrigeration cycle 10. Two-phase gas-liquid flows in the liquid pipe 16 so that the amount of refrigerant in the liquid pipe 16 is reduced. In this manner, the amount of enclosed refrigerant in the lower-stage refrigeration cycle 10 is reduced, and even when CO_2 becomes a critical state, it is possible to prevent the pressure in the lower-stage refrigeration cycle 10 from exceeding the design pressure. Although Fig. 13 shows the configuration including no expansion tank 32, the expansion tank 32 may be provided. In this case, the use of the second expansion valve 34 can reduce the volume of the expansion tank 32 as compared to the configuration illustrated in Fig. 12, and the size of the expansion tank 32 can be reduced.

[0124] Fig. 14 shows a relationship between an enthalpy and a saturation temperature in the two-stage refrigeration cycle system according to Embodiment 2 of the present invention.

[0125] In the case of using CO_2 in the lower-stage refrigeration cycle 10, each of the two-stage operation and the single-stage operation is performed in such a manner that the lower-stage condensing temperature is less than or equal to 8 degrees C, which is a CO_2 saturation temperature corresponding to a design pressure of 4.15 MPa. In the case of using CO_2 in the lower-stage refrigeration cycle 10, since the radiator is designed in such a manner that the lower-stage condensing temperature is higher than the outdoor-air temperature by about 10 degrees C as described above, switching from the two-stage operation to the single-stage operation is performed at -2 degrees C, which is an outdoor-air temperature lower than the lower-stage condensing temperature by 10 degrees C. That is, if the outdoor-air temperature is less than or equal to -2 degrees C, the single-stage operation is performed, and if the outdoor-air temperature is higher than -2 degrees C, the two-stage operation is performed.

[0126] In each of the two-stage operation and the single-stage operation, a target value (lower-stage condensing temperature) is determined in the following manner. Specifically, in a case where the target value (lower-stage condensing temperature) with a maximum COP is less than or equal to 8 degrees C, an operation is performed using this target value, whereas in a case where the target value (lower-stage condensing temperature) with a maximum COP is higher than 8 degrees C, an operation is performed with the target value being controlled to 8 degrees.

[0127] As described above, in Embodiment 2, advantages similar to those of Embodiment 1 are obtained, and the configuration in which the expansion tank 32 is provided in the lower-stage refrigeration cycle 10 or the configuration in which the amount of enclosed refrigerant is reduced by flowing refrigerant in a two-phase state in the liquid pipe with the second expansion valve 34 achieves the following advantages. Specifically, even when the higher-stage refrigeration cycle 20 stops, the pressure in the lower-stage refrigeration cycle 10 using CO_2 can be reduced to be less than or equal to a design pressure of 4.15 MPa corresponding to that of HFC refrigerant so that components of a conventional HFC refrigeration cycle system can be used. Thus, costs can be reduced. In addition, even when the lower-stage refrigeration cycle 10 frequently starts and stops in a single-stage operation in an operation with a low load, the higher-stage refrigeration cycle 20 can always stop so that a loss due to ON/OFF can be avoided, thereby achieving an energy conservation effect.

Industrial Applicability

[0128] The two-stage refrigeration cycle systems according to Embodiments 1 and 2 are widely applicable to cold storages or refrigeration cycle systems such as showcases, refrigerator-freezers for commercial use, and vending machines that need CFC elimination of refrigerant, reduction of CFC refrigerant, and energy conservation of the systems.

Reference Signs List

[0129] 10 lower-stage refrigeration cycle, 11 lower-stage compressor, 12 lower-stage condenser, 13 lower-stage expansion valve (lower-stage expansion device), 14 lower-stage evaporator, 15 auxiliary radiator, 16 liquid pipe, 20 higher-stage refrigeration cycle, 21 higher-stage compressor, 22 higher-stage condenser, 23 higher-stage expansion valve (higher-stage expansion device), 24 higher-stage evaporator, 25 higher-stage condenser fan, 30 controller, 31 outdoor-air temperature detecting means, 32 expansion tank, 33 solenoid valve (exciting close type), 34 second expansion valve (second expansion device), 40 heat transfer fin, 41 heat transfer tube, 42 integrated radiator, 43 air-sending device, C cascade condenser.

Claims

1. A two-stage refrigeration cycle system comprising:

a higher-stage compressor, a higher-stage condenser, a higher-stage expansion device, and a higher-stage evaporator connected in a higher-stage refrigeration cycle by pipes through which refrigerant circulates;
a lower-stage compressor, an auxiliary radiator, a lower-stage condenser, a lower-stage expansion device, and a lower-stage evaporator connected in a lower-stage refrigeration cycle by pipes through which refrigerant circulates;
a cascade condenser including the higher-stage evaporator and the lower-stage condenser and configured to heat exchange between the refrigerant flowing in the higher-stage refrigeration cycle and the refrigerant flowing in the lower-stage refrigeration cycle; and
a controller configured to switch between a two-stage operation in which both the higher-stage refrigeration cycle and the lower-stage refrigeration cycle operate and a single-stage operation in which the higher-stage refrigeration cycle stops and the lower-stage refrigeration cycle operates, so as to select one of the two-stage operation and the single-stage operation having a higher COP.

2. The two-stage refrigeration cycle system of claim 1, wherein
each of the higher-stage condenser and the auxiliary radiator is a heat exchanger that transfers heat to outdoor air, and the controller switches from the two-stage operation to the single-stage operation in a case where an outdoor-air temperature in the two-stage operation is less than or equal to a threshold outdoor-air temperature serving as a threshold value for determining a level of the COP.

3. The two-stage refrigeration cycle system of claim 1, wherein
the controller switches from the two-stage operation to the single-stage operation in a case where a compression ratio of the lower-stage refrigeration cycle or the higher-stage refrigeration cycle in the two-stage operation is less

than or equal to a threshold compression ratio serving as a threshold value for determining a level of the COP.

4. The two-stage refrigeration cycle system of claim 1, wherein
the controller switches from the two-stage operation to the single-stage operation in a case where a compression ratio determined based on an evaporating pressure of the lower-stage refrigeration cycle and a condensing pressure of the higher-stage refrigeration cycle in the two-stage operation is less than or equal to a threshold compression ratio serving as a threshold value for determining a level of the COP.

5. The two-stage refrigeration cycle system of any one of claims 1 to 4, wherein
each of the higher-stage condenser and the auxiliary radiator is a heat exchanger that transfers heat to outdoor air, and the controller switches from the single-stage operation to the two-stage operation in a case where an outdoor-air temperature in the single-stage operation is higher than a threshold outdoor-air temperature serving as a threshold value for determining a level of the COP.

6. The two-stage refrigeration cycle system of any one of claims 1 to 4, wherein
the controller switches from the single-stage operation to the two-stage operation in a case where a compression ratio of the lower-stage refrigeration cycle in the single-stage operation is higher than a threshold compression ratio serving as a threshold value for determining a level of the COP.

7. The two-stage refrigeration cycle system of any one of claims 1 to 4, wherein
the controller switches from the single-stage operation to the two-stage operation in a case where a high pressure that is a condensing pressure of the lower-stage refrigeration cycle in the single-stage operation exceeds a high pressure that is a condensing pressure of the lower-stage refrigeration cycle when the two-stage operation switches to the single-stage operation.

8. The two-stage refrigeration cycle system of any one of claims 1 to 7, wherein
the higher-stage condenser and the auxiliary radiator are integrated and form an integrated radiator, and
the two-stage refrigeration cycle system further includes an air-sending device configured to send air to the integrated radiator.

9. The two-stage refrigeration cycle system of claim 8, wherein
the integrated radiator includes a plurality of heat transfer fins spaced from each other so as to allow air to pass therebetween and a plurality of heat transfer tubes arranged in a manner of extending through the plurality of heat transfer fins,
the plurality of heat transfer tubes are arranged in a plurality of columns aligning perpendicularly to an air passage direction and a plurality of rows aligning in the air passage direction, and
among the plurality of heat transfer tubes, a plurality of heat transfer tubes constituting the auxiliary radiator are integrated in one row.

10. The two-stage refrigeration cycle system of any one of claims 1 to 9, wherein
the higher-stage condenser and the auxiliary radiator have a substantially identical heat transfer area.

11. The two-stage refrigeration cycle system of any one of claims 1 to 10, wherein
the refrigerant used in the lower-stage refrigeration cycle is different from the refrigerant used in the higher-stage refrigeration cycle.

12. The two-stage refrigeration cycle system of any one of claims 1 to 11, wherein
refrigerant showing an efficiency higher than that of the refrigerant used in the lower-stage refrigeration cycle is used in the higher-stage refrigeration cycle.

13. The two-stage refrigeration cycle system of any one of claims 1 to 12, wherein
CO_2 refrigerant or mixed refrigerant including CO_2 is used in at least one of the lower-stage refrigeration cycle and the higher-stage refrigeration cycle.

14. The two-stage refrigeration cycle system of claim 13, wherein
refrigerant selected from R32, R410A, R134a, R404A, R407C, HFO1234yf, HFO1234ze, ammonia, propane, and isobutane, or mixed refrigerant is used in at least one of the lower-stage refrigeration cycle and the higher-stage refrigeration cycle.

15. The two-stage refrigeration cycle system of any one of claims 1 to 14, wherein CO_2 refrigerant is used in the lower-stage refrigeration cycle, and the controller sets a high pressure that is a condensing pressure of the lower-stage refrigeration cycle in the single-stage operation at a critical pressure or less.

16. The two-stage refrigeration cycle system of any one of claims 1 to 15, wherein CO_2 refrigerant is used in the lower-stage refrigeration cycle, each of the higher-stage condenser and the auxiliary radiator is a heat exchanger that transfers heat to outdoor air, and the controller switches from the two-stage operation to the single-stage operation in a case where an outdoor-air temperature in the two-stage operation is less than or equal to 21 degrees C.

17. The two-stage refrigeration cycle system of any one of claims 1 to 16, wherein CO_2 refrigerant is used in the lower-stage refrigeration cycle, and an expansion tank is disposed in the lower-stage refrigeration cycle.

18. The two-stage refrigeration cycle system of any one of claims 1 to 17, wherein CO_2 refrigerant is used in the lower-stage refrigeration cycle, and a second expansion device is disposed in an upstream portion of a liquid pipe between the lower-stage condenser and the lower-stage expansion device so that the refrigerant passing through the liquid pipe is changed to two-phase gas-liquid refrigerant.

ＦＩＧ．１

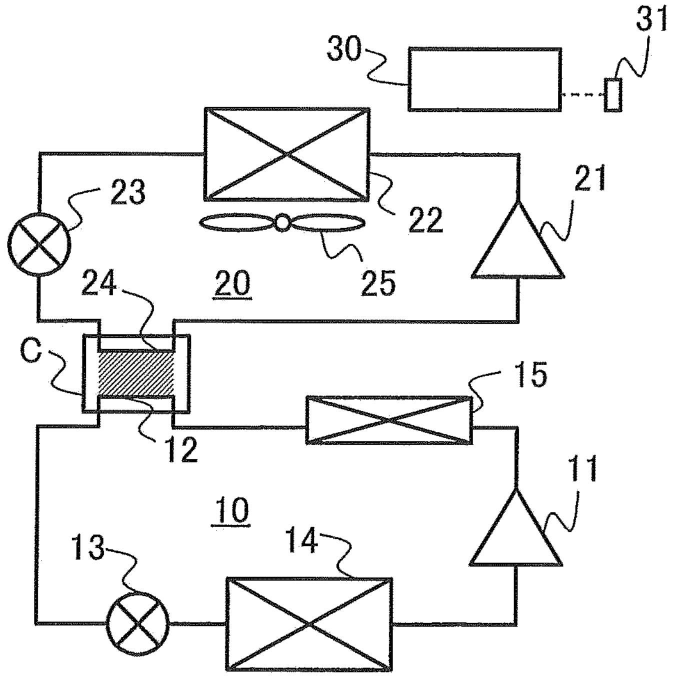


FIG. 2

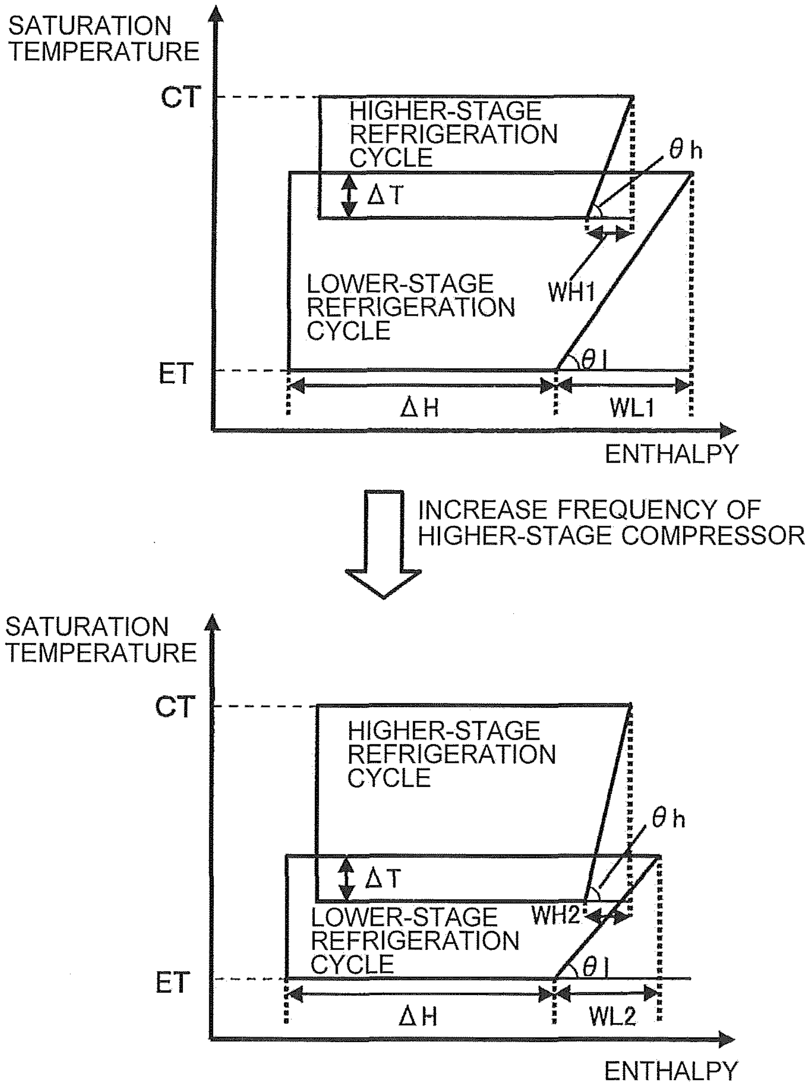

SATURATION TEMPERATURE
CT
HIGHER-STAGE REFRIGERATION CYCLE
θh
ΔT
LOWER-STAGE REFRIGERATION CYCLE
WH1
θl
ET
ΔH
WL1
ENTHALPY
INCREASE FREQUENCY OF HIGHER-STAGE COMPRESSOR
SATURATION TEMPERATURE
CT
HIGHER-STAGE REFRIGERATION CYCLE
θh
ΔT
LOWER-STAGE REFRIGERATION CYCLE
WH2
θl
ET
ΔH
WL2
ENTHALPY

FIG. 3

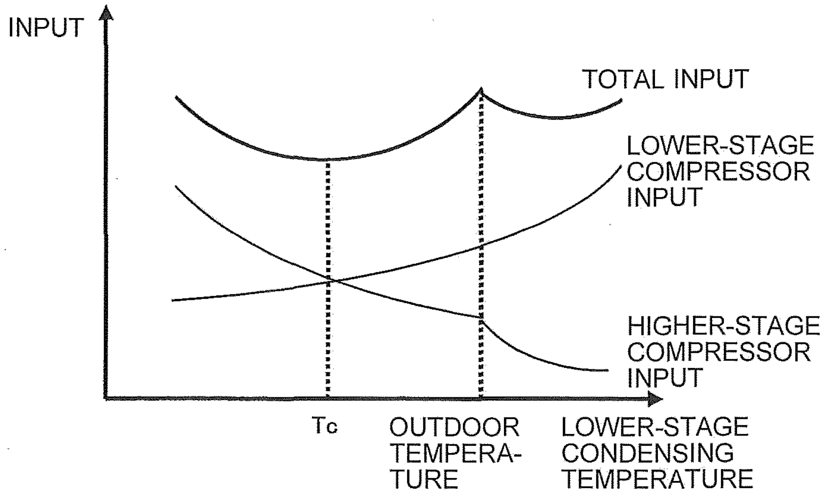
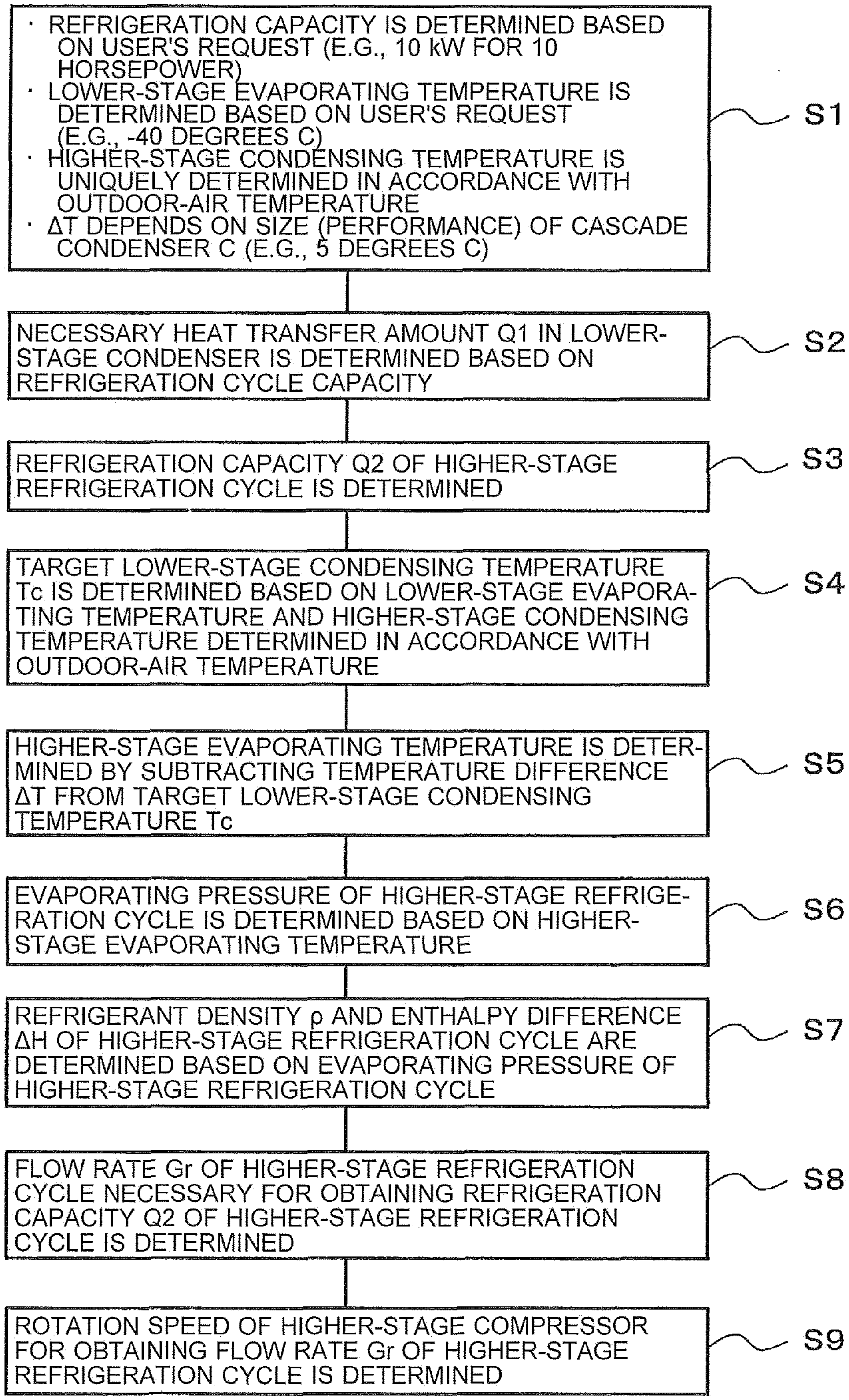
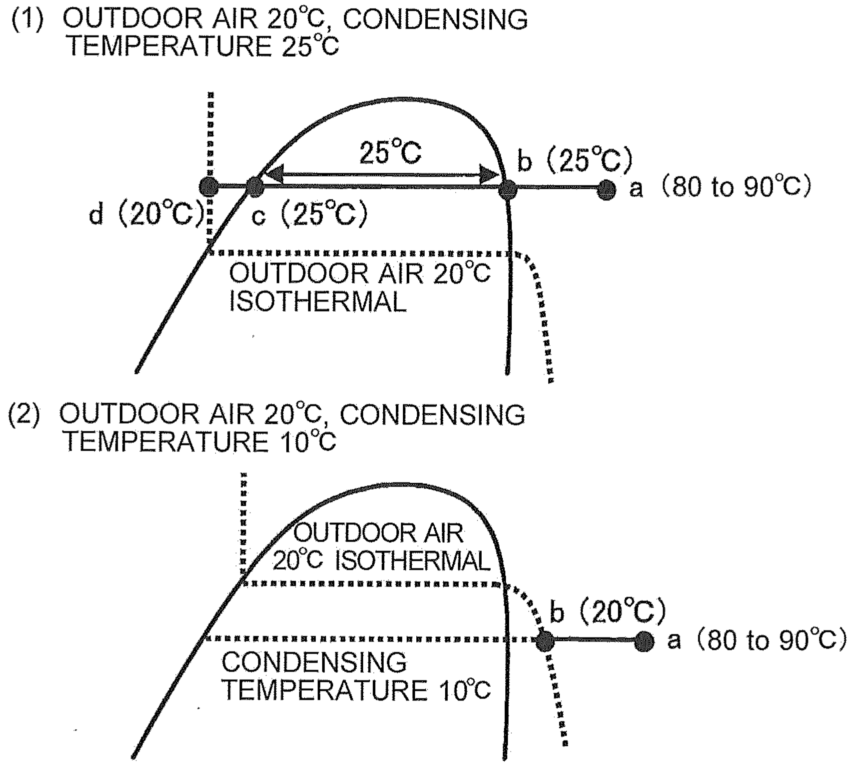

INPUT
TOTAL INPUT
LOWER-STAGE COMPRESSOR INPUT
HIGHER-STAGE COMPRESSOR INPUT
Tc
OUTDOOR TEMPERA-TURE
LOWER-STAGE CONDENSING TEMPERATURE

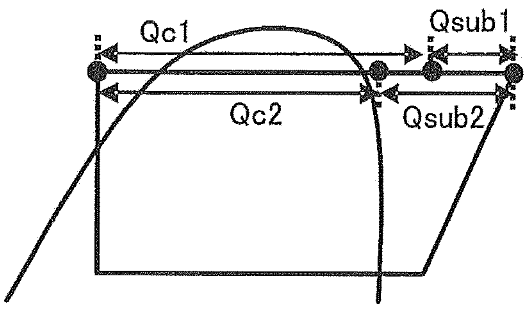
FIG. 4


· REFRIGERATION CAPACITY IS DETERMINED BASED ON USER'S REQUEST (E.G., 10 kW FOR 10 HORSEPOWER)
· LOWER-STAGE EVAPORATING TEMPERATURE IS DETERMINED BASED ON USER'S REQUEST (E.G., -40 DEGREES C)
· HIGHER-STAGE CONDENSING TEMPERATURE IS UNIQUELY DETERMINED IN ACCORDANCE WITH OUTDOOR-AIR TEMPERATURE
· ΔT DEPENDS ON SIZE (PERFORMANCE) OF CASCADE CONDENSER C (E.G., 5 DEGREES C)
S1
NECESSARY HEAT TRANSFER AMOUNT Q1 IN LOWER-STAGE CONDENSER IS DETERMINED BASED ON REFRIGERATION CYCLE CAPACITY
S2
REFRIGERATION CAPACITY Q2 OF HIGHER-STAGE REFRIGERATION CYCLE IS DETERMINED
S3
TARGET LOWER-STAGE CONDENSING TEMPERATURE Tc IS DETERMINED BASED ON LOWER-STAGE EVAPORATING TEMPERATURE AND HIGHER-STAGE CONDENSING TEMPERATURE DETERMINED IN ACCORDANCE WITH OUTDOOR-AIR TEMPERATURE
S4
HIGHER-STAGE EVAPORATING TEMPERATURE IS DETERMINED BY SUBTRACTING TEMPERATURE DIFFERENCE ΔT FROM TARGET LOWER-STAGE CONDENSING TEMPERATURE Tc
S5
EVAPORATING PRESSURE OF HIGHER-STAGE REFRIGERATION CYCLE IS DETERMINED BASED ON HIGHER-STAGE EVAPORATING TEMPERATURE
S6
REFRIGERANT DENSITY ρ AND ENTHALPY DIFFERENCE ΔH OF HIGHER-STAGE REFRIGERATION CYCLE ARE DETERMINED BASED ON EVAPORATING PRESSURE OF HIGHER-STAGE REFRIGERATION CYCLE
S7
FLOW RATE Gr OF HIGHER-STAGE REFRIGERATION CYCLE NECESSARY FOR OBTAINING REFRIGERATION CAPACITY Q2 OF HIGHER-STAGE REFRIGERATION CYCLE IS DETERMINED
S8
ROTATION SPEED OF HIGHER-STAGE COMPRESSOR FOR OBTAINING FLOW RATE Gr OF HIGHER-STAGE REFRIGERATION CYCLE IS DETERMINED
S9

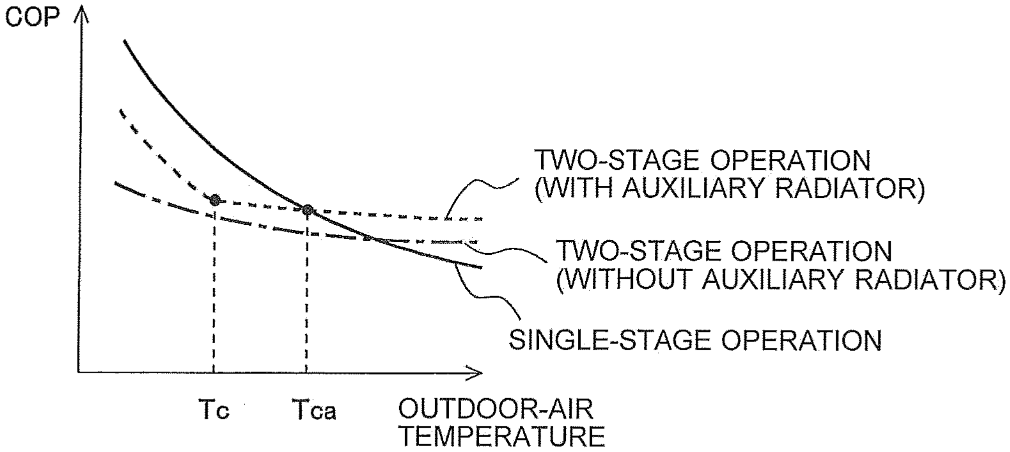
F I G. 5


(1) OUTDOOR AIR 20℃, CONDENSING TEMPERATURE 25℃
25℃
b (25℃)
a (80 to 90℃)
d (20℃)
c (25℃)
OUTDOOR AIR 20℃ ISOTHERMAL
(2) OUTDOOR AIR 20℃, CONDENSING TEMPERATURE 10℃
OUTDOOR AIR 20℃ ISOTHERMAL
b (20℃)
a (80 to 90℃)
CONDENSING TEMPERATURE 10℃

F I G. 6


Qc1
Qsub1
Qc2
Qsub2

F I G. 7


COP
TWO-STAGE OPERATION
(WITH AUXILIARY RADIATOR)
TWO-STAGE OPERATION
(WITHOUT AUXILIARY RADIATOR)
SINGLE-STAGE OPERATION
Tc
Tca
OUTDOOR-AIR
TEMPERATURE

F I G. 8

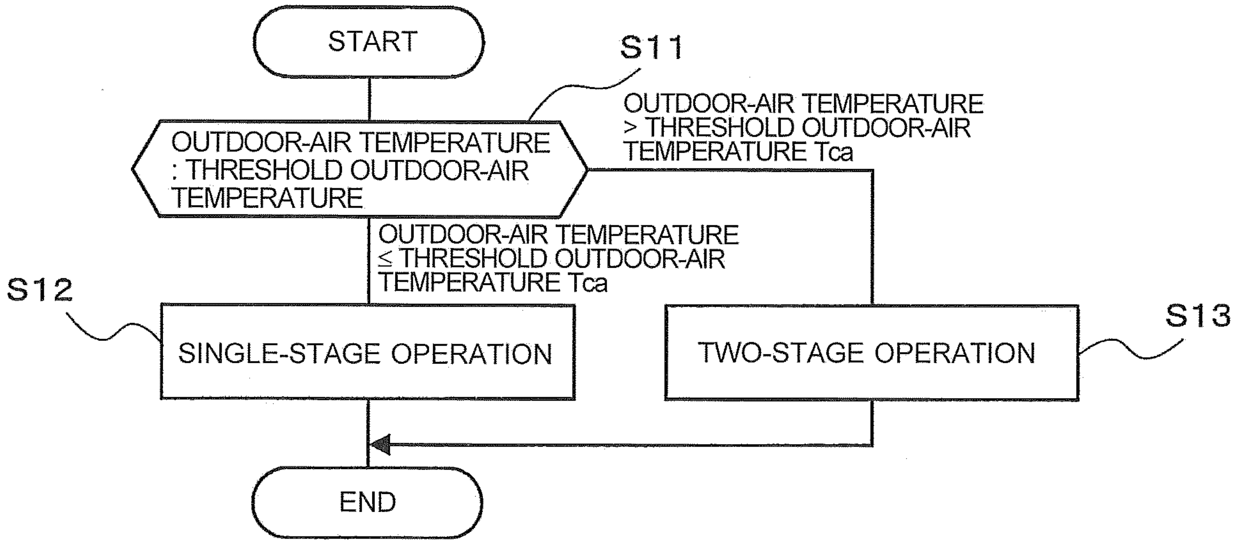

START
S11
OUTDOOR-AIR TEMPERATURE
> THRESHOLD OUTDOOR-AIR
TEMPERATURE Tca
OUTDOOR-AIR TEMPERATURE
: THRESHOLD OUTDOOR-AIR
TEMPERATURE
OUTDOOR-AIR TEMPERATURE
≤ THRESHOLD OUTDOOR-AIR
TEMPERATURE Tca
S12
SINGLE-STAGE OPERATION
TWO-STAGE OPERATION
S13
END

FIG. 9

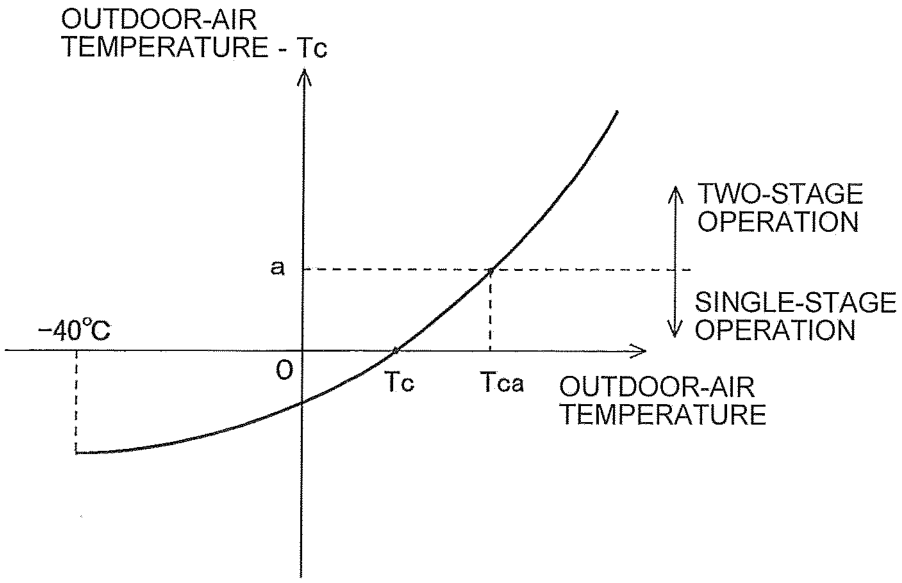
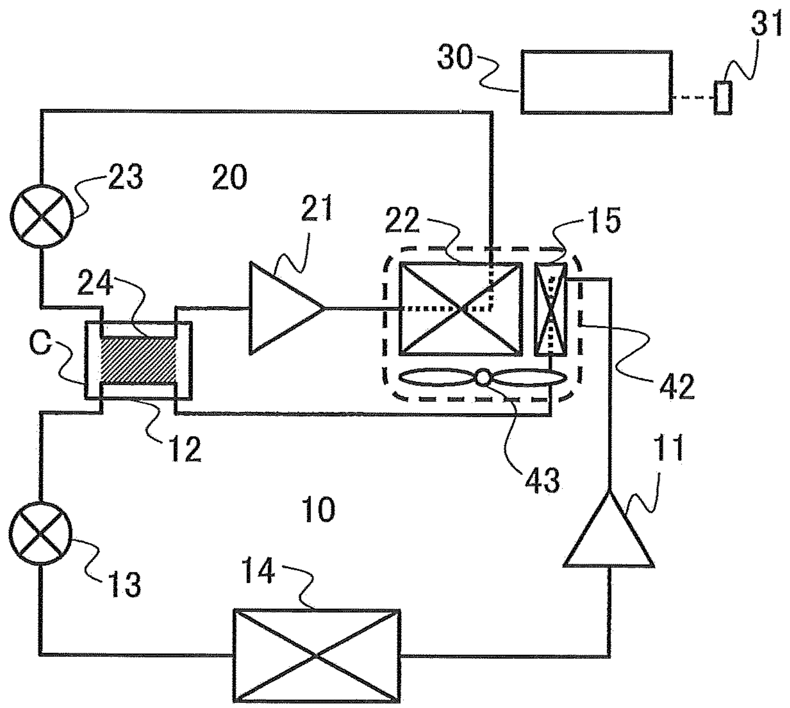
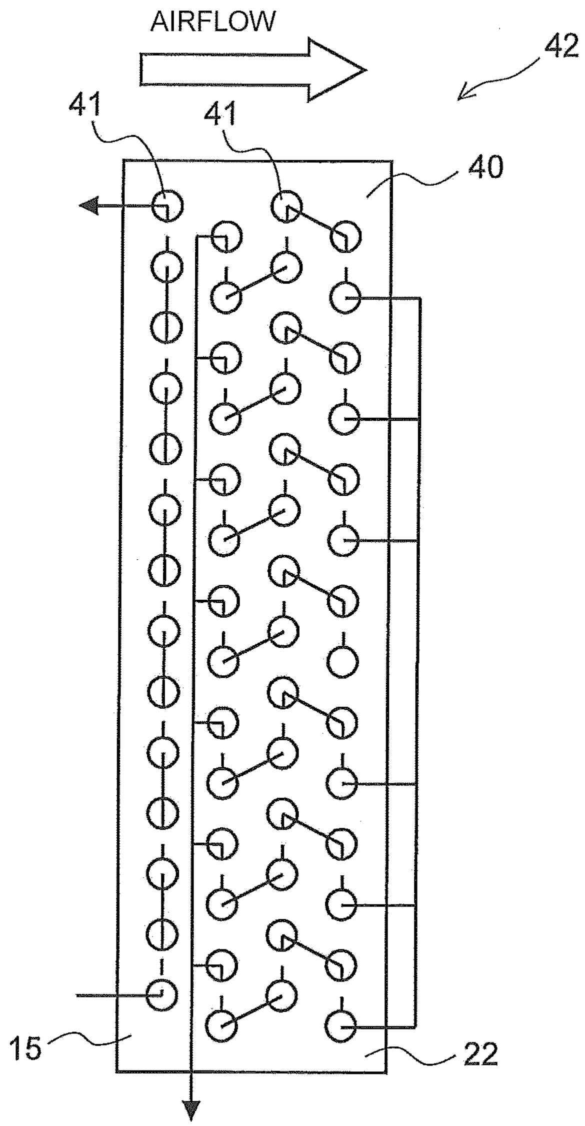

OUTDOOR-AIR TEMPERATURE - Tc
TWO-STAGE OPERATION
a
SINGLE-STAGE OPERATION
−40°C
0
Tc
Tca
OUTDOOR-AIR TEMPERATURE

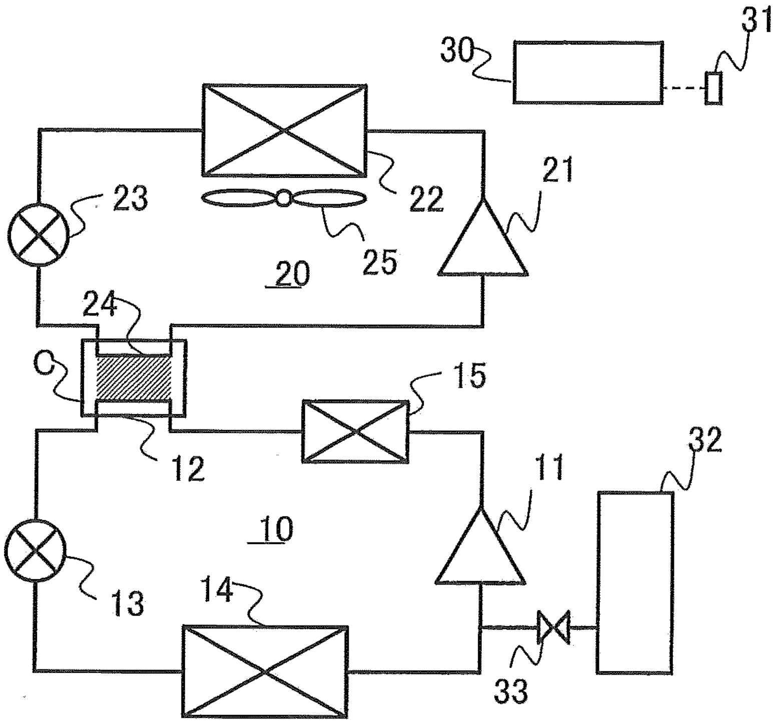
FIG. 10


31
30
23
20
21
22
15
24
C
42
12
11
43
10
13
14

ＦＩＧ. １１



ＦＩＧ．１２


31
30
22
21
23
25
20
24
C
15
32
12
11
10
13
14
33

ＦＩＧ．１３

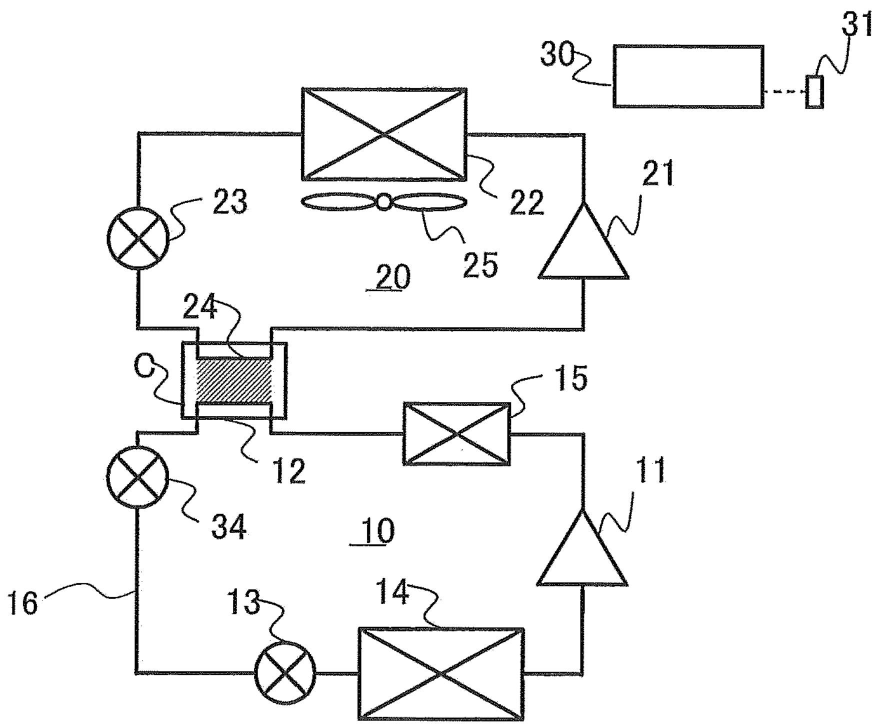
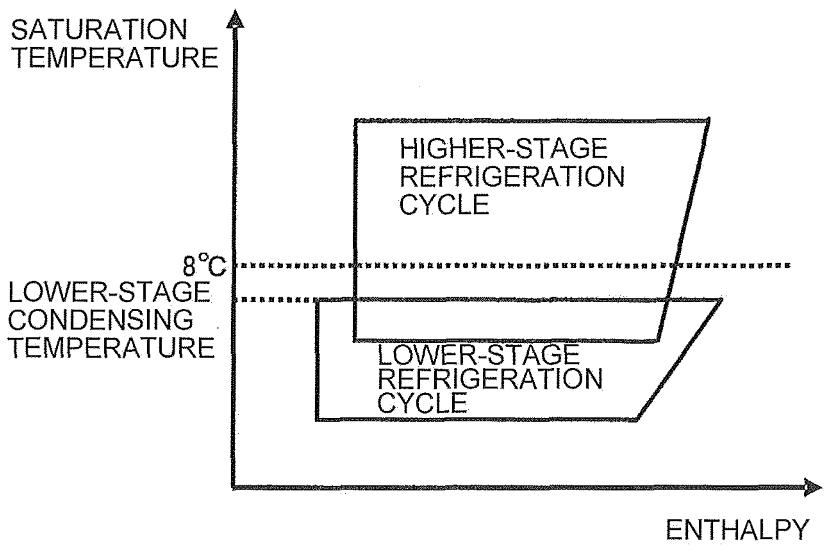

31
30
22
21
23
25
20
24
C
15
12
11
34
10
16
13
14

FIG. 14


SATURATION
TEMPERATURE
HIGHER-STAGE
REFRIGERATION
CYCLE
8℃
LOWER-STAGE
CONDENSING
TEMPERATURE
LOWER-STAGE
REFRIGERATION
CYCLE
ENTHALPY

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2013/062931

A. CLASSIFICATION OF SUBJECT MATTER
F25B7/00(2006.01)i, *F25B6/04*(2006.01)i

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F25B7/00, F25B6/04

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2013
Kokai Jitsuyo Shinan Koho	1971-2013	Toroku Jitsuyo Shinan Koho	1994-2013

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 A	JP 2000-274848 A (Daikin Industries, Ltd.), 06 October 2000 (06.10.2000), paragraph [0033]; fig. 1 to 5 (Family: none)	1-16 17-18
A	JP 2001-91074 A (Sanyo Electric Co., Ltd.), 06 April 2001 (06.04.2001), fig. 1 (Family: none)	1-18

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

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier application or patent but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
14 June, 2013 (14.06.13)	25 June, 2013 (25.06.13)

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

Form PCT/ISA/210 (second sheet) (July 2009)

REFERENCES CITED IN THE DESCRIPTION

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

Patent documents cited in the description

- JP 3604973 B **[0006]**
- JP 2000274848 A **[0006]**