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(54) **HEAT EXCHANGER AND REFRIGERATION CYCLE APPARATUS**

WÄRMETAUSCHER UND KÜHLZYKLUS-VORRICHTUNG

ECHANGEUR THERMIQUE ET APPAREIL À CYCLE DE RÉFRIGÉRATION

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(56) References cited:
EP-A1- 1 705 432
DE-A1-102005 007 516
JP-A- 2000 199 656
JP-A- 2001 149 735
JP-A- 2002 001 106
JP-A- 2006 349 342
WO-A1-2006/079448
JP-A- 2000 140 625
JP-A- 2001 104 744
JP-A- 2002 001 106
JP-A- 2005 134 099
JP-A- 2007 240 128

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Description

Technical Field

[0001] The present invention relates to a heat exchanger for adsorbing water content in the air, and a refrigeration cycle apparatus having the same.

[0002] JP 2000-140625 A discloses a heat exchanger for an air-conditioner for vehicles. The heat exchanger comprises corrugated heat transfer fins which include, in corresponding crevices, an adsorbent which has fine pores of a pore diameter between 0.6 nm and 1.6 nm.

[0003] DE 10 2005 007 516 A1 describes an adsorption heat pump wherein a tube connection is connected on its one end with a porous body of zeolite. On the other end of the tube connection, a container is arranged which acts as an evaporator. The zeolite is used to add saltwater. The pore size of the zeolite is smaller than 1.5 nm.

[0004] JP 2001-149735 A discloses a dehumidifying element which can be used to adsorb humidity. This element is built from corrugated paper. The corrugated paper has a nano-hole structure. The holes can have a diameter between 0.8 and 3.0 nm.

[0005] JP 2006-349342 discloses a heat exchanger according to the preamble of claim 1.

Background Art

[0006] In a refrigeration cycle apparatus using a refrigeration cycle such as an air conditioning system and a refrigeration system, a compressor, a condenser (heat exchanger), an expansion valve, and an evaporator (heat exchanger) are basically connected with the pipe to form a refrigerant circuit for circulating a refrigerant such as a geotropic refrigerant mixture, a pseudo azeotropic refrigerant mixture, and a single refrigerant. Utilizing that the refrigerant absorbs and radiates heat against air subjected to the heat exchange upon evaporation and condensation, the air-conditioning and cooling operations are performed while changing the pressure of the refrigerant passing through a pipe.

[0007] The heat exchanger functioning as the evaporator and the condenser allows the refrigerant to pass through the pipe therein to perform heat exchange. Since in the heat exchanger serving as the evaporator, the low temperature refrigerant passes through the pipe to absorb the heat in the air, the water content (vapor) in the air is condensed on the surface of the pipe to be deposited thereon as frost. When the frost is deposited (formed), the frost exists between the refrigerant and air. The deposited frost narrows a gap through which air passes, thus interfering with the air flow. Heat exchange between the refrigerant and air cannot be appropriately performed, thus deteriorating operation efficiency. Therefore, defrosting for removing the frost adhered to the evaporator is performed on a regular basis, or when it is judged that efficiency is deteriorated.

[0008] The defrosting may remove the frost adhered to the evaporator, but consumes an extra energy, thus failing to improve the efficiency of the air conditioning system. Immediately after finishing the defrosting, the temperature in the freezer and refrigerator is increased although it is required to maintain a predetermined temperature range. In such a case, the load for cooling up to a required temperature range is increased, thus further consuming electric power, resulting in efficiency deterioration.

[0009] Meanwhile, in the case of such as an air conditioning system for cooling/heating, in the in-between period of cooling (rainy season, autumn) for example, a cooling load tends to be small. In such a case, an operation frequency of the compressor is usually controlled to decrease the flow rate of the refrigerant (per unit time) circulated in the refrigerant circuit. At this time, the evaporating temperature in the evaporator increases to remove a sensible heat. However, a latent heat (water content in the air (vapor)) may not be removed. If the latent heat in the room cannot be removed, the relative humidity in the air in the room will be increased, which cause discomfort of the person in the room to increase.

[0010] A device for removing the water content in the air has been disclosed to solve the aforementioned problem (for example, see Patent Document 1). The dehumidifying device uses zeolite which is a porous inorganic oxide, as a water adsorbing material (hereinafter referred to as an adsorbing material) for example, to make the fin for performing heat exchange between the air and refrigerant support it.

[0011] [Patent Document 1] Japanese Unexamined Patent Application Publication No. 2004-353887 (Fig. 1)

Disclosure of Invention

Problems to be Solved by the Invention

[0012] In the aforementioned device, the thermal expansion may cause distortion between the fan and the adsorbing material, so that the adsorbing material may peel off to be contained in the air and dispersed. In the case where such device is employed in the freezer and refrigerator for storing foods, it is required to prevent the adsorbing material from being peeled off for quality control of foods. The aforementioned requirement may be applied to the heat exchanger for

air conditioning in the living space. It is therefore difficult to control the heat exchanger using the adsorbing material. The loss in thermal conduction may lower the energy efficiency.

[0013] If the zeolite is employed as the adsorbing material on the surface of the desiccant rotor for removing a water content in the air which flows into the heat exchanger, the temperature required for desorption is high. Therefore, it is difficult to desorb the adsorbed water using the temperature of the refrigerant flowing through the refrigerant circuit and to re-use the water.

[0014] The present invention is made to solve the aforementioned problem, and an object of the present invention is to provide a heat exchanger and a refrigeration cycle apparatus such as an air conditioning device and refrigerator capable of more efficiently adsorbing water performing the heat exchange.

Means for Solving the Problems

[0015] The aforementioned problems are solved by a heat exchanger according to claim 1.

[0016] A heat exchanger according to the present invention includes fins for heat transfer in the heat exchanger for exchanging heat between a refrigerant and air. The fin has fin pores on the surface for adsorbing the water in the air by capillary condensation. Each of the fine pores has a different diameter depending on the position on the surface of the fins.

[0017] The fine pores are in the range from 1 to 3.5 nm. The fin is made of a material which contains aluminum, titanium, zirconium, niobium, or tantalum. The fine pores are formed by an anodic oxidation method. After forming recess portions at predetermined intervals on the material to be the fin, fin pores can be formed by the anodic oxidation method.

Advantages

[0018] The fine pores are formed on the surface of the fin of the heat exchanger such that the fin itself functions as the water adsorbing unit using the capillary condensation phenomenon. The heat exchanger capable of making the fins adsorb water contained in air in the subject space may be provided without requiring special means and materials. The peeling-off of the supported adsorbing member never occurs, thus it is safe from the viewpoint of sanitation and easy to manage. No pressure loss of air caused by the adsorbing material occurs, so that efficient heat exchange can be performed in view of the energy consumption. Since it is not necessary to provide adsorbing material, the apparatus can be made compact. By making each diameter of the fine pores different depending on the position on the fin surface, it is possible to obtain the heat exchanger appropriately performing adsorption/desorption while being suitably adapted to the environment.

Brief Description of the Drawings

[0019]

[Fig. 1] Fig. 1 is a configuration view showing an essential portion of the structure of a heat exchanger according to Example 1.

[Fig. 2a] Fig. 2a is a view showing a relationship between the fine pore diameter of the fin which forms the heat exchanger and the relative humidity.

[Fig. 2b] Fig. 2b is a view schematically showing how the water being adsorbed in the fine pore.

[Fig. 3] Fig. 3 is a view showing characteristics of the fine pore diameter and adsorption.

[Fig. 4] Fig. 4 is a view showing an example of the water adsorbing characteristics of the fine pores on the surface of a fin 45.

[Fig. 5] Fig. 5 is an enlarged view of the surface of the fin 45.

[Fig. 6] Fig. 6 is a structural view showing the heat exchanger having a fine-pores distribution of along an air flow direction (column direction).

[Fig. 7a] Fig. 7a is a view showing the relationship between the air flow direction (column direction) and the relative humidity.

[Fig. 7b] Fig. 7b is a view showing the relationship between the air flow direction and the adsorption amount.

[Fig. 8a] Fig. 8a is a view showing the relationship between the air flow direction (column direction) and the relative humidity.

[Fig. 8b] Fig. 8b is a view showing the relationship between the air flow direction and the desorption amount.

[Fig. 9a] Fig. 9a is a view showing the method for manufacturing the fin having fine pores with the same diameter.

[Fig. 9b] Fig. 9b is a view showing the method for manufacturing the fin having fine pores with a different diameter.

[Fig. 10] Fig. 10 shows operation points on the psychrometric diagram.

[Fig. 11] Fig. 11 schematically shows an exemplary structure of a refrigeration cycle apparatus according to Embodiment 4.

[Fig. 12] Fig. 12 is a relationship view showing a relationship between the evaporating temperature and COP.
[Fig. 13] Fig. 13 is a schematic view showing an exemplary structure of the refrigeration cycle apparatus according to Embodiment 5.
[Fig. 14] Fig. 14 is a schematic view three-dimensionally showing a humidifying unit of the refrigeration cycle apparatus.
[Fig. 15] Fig. 15 is an illustrative view showing the state where an air passage in an indoor unit is switched.
[Fig. 16] Fig. 16 is a P-h diagram showing the state of the refrigerant in the refrigeration cycle.
[Fig. 17] Fig. 17 is a phsychrometric diagram for explaining the operation of the refrigeration cycle apparatus.
[Fig. 18] Fig. 18 is a schematic view showing an exemplary structure of the refrigeration cycle apparatus according to Embodiment 6.
[Fig. 19] Fig. 19 is a schematic view showing the structure of the indoor unit having a built-in evaporator.
[Fig. 20] Fig. 20 is an illustrative view showing the state where the air passage in the indoor unit is switched.
[Fig. 21] Fig. 21 is a P-h diagram showing the state of the refrigerant in the refrigeration cycle.
[Fig. 22] Fig. 22 is a psychrometric diagram for explaining the operation of the refrigeration cycle apparatus.
[Fig. 23] Fig. 23 is a schematic view showing an exemplary structure of the refrigeration cycle apparatus according to Embodiment 7.

Reference Numerals

[0020]

1, 1a, 1b, 1c, 1d refrigerant pipe
2,2a,3,3a bypass pipe
10 compressor
20 condenser
30,31,32,33,34,35,36,37 on-off valve
38,39 three-way valve
40 heat exchanger
40a heat exchanger with distributed fine pores
41,41a,41b,41c,41d,41e,41f heat exchanger for dehumidification/humidification
45 fin
45a fine pores
45b porous layer
45c barrier layer
45aa fin at the first column
45ab fin at the second column
45ac fin at the third column
46 heat transfer pipe
50,51 back-flow prevention member
60,61,62,63,64,85 throttle device
70 evaporator
80,80a,80b control unit
81 temperature/humidity detection unit
90,91 blower
100,100a,100b,100c refrigeration cycle apparatus
300,300a indoor unit
301a,301b,302a,302b,303a,303b,304a,304b,311a,311b,312a, 312b, 313a, 313b, 314a, 314b air passage switching unit
305a,305b,315a,315b air passage switching unit
400 refrigerated warehouse
401 interior
500 external air
610 dc power source
620 electrolyte
630 electrolyte vessel
640 carbon electrode
650 fin

Best Mode for Carrying Out the Invention

EMBODIMENT 1

[0021] Fig. 1 shows an essential portion of a heat exchanger 40 according to Embodiment 1 of the present invention. The structure of the heat exchanger 40 as the featuring part of the present invention will be described referring to Fig. 1. The heat exchanger of a fin tube type which has been widely used for the refrigerator and air conditioning system will be explained as an example.

[0022] The heat exchanger 40 is mainly composed of plural fins 45 for heat exchanger (hereinafter referred to as a fin 45), and plural heat transfer pipes 46. The fin 45 of the present embodiment is a flat plate made of the material with a high thermal conductivity (thermal conductivity: approximately 230 W/mK) such as aluminum. The fin 45 has fine pores on the surface as described later. The plural fins 45 are laminated at a predetermined interval. The heat transfer pipes 46 are provided, for example, at a predetermined interval so as to penetrate through holes formed in each fin 45. Each heat transfer pipe 46 becomes part of a refrigerant circuit, allowing the refrigerant to flow therethrough. The heat of the refrigerant flowing through the heat transfer pipe 46 and the heat of the air flowing outside are transferred via the fins 45 to expand the heat transfer area, thus the heat exchange between the refrigerant and the air is efficiently performed. The path of the heat transfer pipes 46 in the heat exchanger 40 is not especially limited. For example, the flow path may be formed to be branched to allow the refrigerant to flow into the plural heat transfer pipes 46 which penetrate the fins 45, and then to be joined. The refrigerant flow path may also be formed to make the laminated fins 45 bent at the end of the heat exchanger 40 or connected by the bent pipe reciprocated. Referring to Fig. 1, the heat transfer pipes 46 penetrate the fins 45 at 6 points, however, the number of the heat transfer pipes 46 is not limited thereto.

[0023] The fin 45 is made of materials containing aluminum, titanium, zirconium, niobium, or tantalum, and includes a plurality of fine pores having diameters ranging from 1 to 3.5 nm. The fine pores of the fin 45 may be formed by the anodic oxidation method. The fine pores may be formed by the anodic oxidation method after forming recesses at a predetermined interval in the material for forming the fin 45 in advance.

[0024] Fig. 2a is a view showing the relationship between the diameter (hereinafter referred to as fine pore diameter) of the fine pores of the fin 45 and the relative humidity at which the capillary condensation occurs. The horizontal axis represents the fine pore diameter [nm (nanometer)], and the vertical axis represents the relative humidity [%] of the air in the subject space (assuming that current humidity is P, and saturated humidity at the current humidity being PO, the relative humidity may be expressed as P/PO). Fig. 2a shows a graph calculated based on the formula of Kelvin.

[Formula 1]

$$\text{Relative humidity} \quad : \frac{P}{P_0} = \exp\left(-\frac{2 v_1 \gamma \cos\theta}{rRT}\right)$$

[0025] Here, v1 denotes a condensed molecule volume, y a surface tension, θ an angle when in contact with capillary, R a gas constant (8.31[J/mol·°K]), T the absolute temperature, and r the radius of the fine pore. The relationship may hold in the case of water vapor. So, the radius r of the fine pore required for water vapor to cause capillary condensation may be theoretically obtained for a certain relative humidity P/P0.

[0026] As shown in Fig. 2a, capillary condensation (a phenomenon in which the vapor (water content) inside the fine pore is liquefied) occurs at the relative humidity corresponding to the fine pore diameter. In Fig. 2a, water molecules may be maintained in the fine pore in the zone A, while water molecules cannot be maintained in the fine pore in the zone B. That is, the water content in the air may be adsorbed in the zone A. On the contrary, the water content can be removed from the fine pore by making the air conditions of the zone B.

[0027] Fig. 2b shows the image of adsorption of water into the fine pore. As shown in Fig. 2b, water is gradually adsorbed into the fine pore. The equilibrium adsorption amount may be sharply increased (changed) at the boundary of the predetermined narrow range in the vicinity of the relative humidity when a number of fine pores having the diameter in accordance with the relative humidity uniformly are formed.

[0028] The relationship between the fine pore diameter and the adsorption isothermal may be obtained in reference to Fig. 2a. Fig. 3 is a characteristic view showing the relationship between the water content (adsorbing characteristics) of the fine pore diameter of the fin 45 according to Embodiment 1 of the present invention and the relative humidity showing a sharp change (hereinafter referred to as rising edge). As shown in Fig. 3, the relative humidity at the rising edge becomes relatively low by making the fine pore diameter of the fin 45 relatively small (line (a) in Fig. 3). On the other hand ,the relative humidity at the rising edge becomes relatively high by making the fine pore diameter large (line

(b) in Fig. 3).

[0029] For example, the adsorbing material may demonstrate the rising edge feature at the relative humidity of approximately 30% as shown by the line (c) in Fig. 3 by setting the fine pore diameter d at 2.0 nm. The adsorbing material may demonstrate the sharp rising edge feature at the relative humidity of approximately 90% by setting the fine pore diameter of 20 nm, such diameter not falling within the scope of the invention. Adsorbing characteristics of the fin 45 may be freely controlled using Fig. 3, within the scope of the attached claims.

[0030] Since the adsorbing material is used for the fin 45 for the purpose of dehumidification, an upper limit of the relative humidity is less than 100%. The line (b) shown in Fig. 3 indicating a sharp rising edge at the relative humidity of approximately 90% is an upper limit value of the adsorbing characteristics, and the fine pore diameter is approximately 20 nm, the upper limit of the fine pore diameter according to the invention being 3.5 nm. Therefore, when used for the air-conditioning device (including the refrigerator), the upper limit of the fine pore diameter of the fin 45 is set at 3.5 nm.

[0031] In the manufacturing, selection of the fine pore diameter in accordance with the usage may reduce the production volume to increase the cost of the adsorbing material. Since the fine pore of nano scale is invisible to human eyes, it is not possible to identify the fin with a different fine pore diameter. As a result, there may be a risk of mounting the adsorbing material with improper fine pore diameter on the product, resulting in a poor quality. Therefore, it is preferable to consolidate the fine pore diameter of the adsorbing material to one kind in view of the cost and quality. However, in order to be effective as the fin 45 for dehumidification, the fine pore diameter have to be consolidated into a fine pore diameter capable of dehumidification in the most usages (most humidity conditions) and according to the features defined by the attached claims.

[0032] Table 1 shows an example of the fine pore diameters (relative humidity) required for the respective usages. Table 1 shows the relative humidity in the subject space and the fine pore diameter of the fin 45 required for the relative humidity. Table 1 is quoted from the collection of papers of "Latest humidity control technology" in the lecture meeting of Japan Society of Refrigerating and Air conditioning Engineers (pp. 5-6, published on May 25, 2005).

[Table 1]

Field	Fine pore diameter [nm]	Relative humidity condition [%]
Fruits and vegetables	6-20	70-95
Stock farm products	4-20	65-95
Pharmaceutical factory	2-4	40-50
Library	2-4	40-50
Art museum/museum	2-5	40-55
Photographic plant	1-6	24-70
Human	1-2	20-30

[0033] For example, in the warehouse for storing fruits and vegetables, the relative humidity is required to be in the range from approximately 70 to 95%, therefore, the fine pore diameter may be set so that the relative humidity sharply rises at approximately 70 to 95% in the adsorbing characteristics shown in Fig. 2a. That is, the fine pore diameter of the adsorbing material may be designed to be 6 to 20 nm, thereby not falling within the scope of the invention as defined by the attached claims, in reference to Fig. 2a (showing the relationship between the fine pore diameter and capillary phenomenon). In an air conditioned space (living space for the human), the relative humidity is generally said to be kept in the range from 20 to 30% or higher.

[0034] Generally, the lower limit value of the relative humidity is considered to be in the range from approximately 20 to 30% except for a special usage. Accordingly, the use of the fin 45 with the adsorbing characteristics (the fine pore diameter is in the range from 1.0 nm to 3.5 nm) which sharply rises at the relative humidity around 20% to 50% as shown in Fig. 3 may cover almost all the usage (wide range). Increase of the use of the fin 45 having the same specification (the same fine pore diameter) may produce a mass production effect to reduce the cost of the fin 45, thus improving the manufacturing quality.

[0035] The refrigeration cycle of the refrigerator is designed to have the condensation pressure corresponding to the condensation temperature of approximately 65°C. Because of the restriction, in the condenser, the air of an external air side 100a is heated up to about 65°C. So that, it is realistic to think that a lower limit value of the relative humidity produced by the exhaust heat of the condenser of the refrigeration cycle be approximately 10% (an external air of 32°C and the relative humidity 60% is heated up to 65°C and the relative humidity 10%). The fine pore diameter at that time is approximately 1 nm by Fig. 2a. Accordingly, the lower limit of the fine pore diameter of the fin 45 for dehumidification is made to be 1 nm.

[0036] Fig. 4 is a view showing an example of the water adsorbing characteristics of the fine pore formed on the surface of the fin 45. Next, the fine pore formed on the surface of the fin 45 will be described. Fig. 4 shows the adsorption

isothermal of the fine pore with the diameter of approximately 2 nm. The horizontal axis represents the relative humidity [%] of the air in the space to be cooled, and vertical axis represents the water content (adsorbed water amount/weight of fin 45, which is proportional to the equilibrium adsorption amount).

[0037] As shown in Fig. 4, the fin 45 capable of adsorbing water at the relative humidity of approximately 30% or higher no longer keeps adsorbing the water when the relative humidity is decreased to approximately 30% or lower. Therefore, the adsorbed water may be desorbed by lowering the relative humidity to approximately 30% or smaller.

[0038] Fig. 5 is an enlarged view of the surface of the fin 45. In the present embodiment, as described above, the fine pores 45a for adsorbing/desorbing the water are formed on both surfaces of the fin 45 simultaneously using the anodic oxidation (anodization). When performing a direct current electrolysis under the environment of in the acidic solution such as sulfuric acid, oxalic acid, phosphoric acid, chromic acid, and alkaline solution such as sodium phosphate with the fin (aluminum) being an anode, the aluminum ion (Al^{3+}) dissolved from the fin (aluminum) reacts with the water (H_2O) to generate an aluminum oxide (alumina) film (hereinafter, referred to as the anodic oxide film) on the aluminum, which is a substrate metal. Here, the fine pore 45a may be formed by a through hole, because an effect remains unchanged that the fine pore 45a can adsorb/desorb water even if it is a through hole.

[0039] The anodic oxide film is formed of a porous layer 45b where the vertical fine pore 45a is formed and a barrier layer 45c at the bottom wall portion in contact with the substrate metal, having a so-called hexagonal cell structure. When forming the fine pore 45a, since the thickness of the barrier layer 45c is basically kept constant, the depth of the fine pore 45a may be controlled by substantially controlling the film thickness. Since the film forming rate and film thickness depend on the current or potential to be supplied between the both electrodes and the anodic oxidation period, the current or potential to be supplied between the both electrodes and the anodic oxidation period are controlled when forming the fine pore 45a with a predetermined depth. Since the number of fine pores per unit area (density) and the fine pore diameter depend on the potential between the both electrodes, the potential therebetween is controlled so as to form a predetermined number and diameter of the fine pores. A mold (metal mold) having protrusions formed at an interval corresponding to that of the fine pores 45a is pressed against an aluminum surface, which is to be a fin 45, to form regular recess portions on the surface. Thereafter, when the anodic oxidation is conducted, the fine pores 45a are formed centering around the recess portion, so that the fine pores 45a are regularly arranged and it is possible to perform high accuracy control on a surface with a constant density. In order to prevent the fine pore 45a formed by the anodic oxidation from reacting with the water contained in air to be blocked, the fin 45 is heated by hot air at the temperature 100 to 200°C immediately after the formation of the fine pores to remove the water contained in the film to perform an operation of changing into a stable oxide. The heat transfer pipes 46 are inserted into the thus formed plural through holes of the fins 45 to make the heat exchanger 40.

[0040] In Embodiment 1, the fine pores are formed on the surface of the fin 45 of the heat exchanger 40 and the fin 45 is made to function as water adsorbing member, so that it is possible to make the water in the air in the subject space adsorb without requiring the special member and materials. Thus, the adsorbing material can be prevented from peeling off due to the distortion caused by the temperature swing between adsorbing materials having different heat expansion coefficients. Since no thermal resistance exists between the fin 45 and the adsorbing material such as silica gel, the heat transfer efficiency may be improved. The thermal conductivity of the silica gel is small, such as approximately in the range from 0.05 to 0.17 W/mK, so that the heat transfer efficiency may be deteriorated. However, since direct heat exchange can be performed between the fin 45 with good thermal conductivity and the air, the heat exchange between the air and the refrigerant can be more efficiently conducted.

[0041] The fin 45 does not have to be thick to accommodate the adsorbing material so that the interval between the fins 45 of the heat exchanger 40 may be increased. As a result, the pressure loss of the flowing air is reduced, so that the input of the fan for making the air flow into the heat exchanger 40 can be decreased. Even if the interval is not changed, the heat exchanger 40 may be made compact by the amount corresponding to the thickness of the adsorbing material. Since the fine pores with a fine pore diameter within the range from approximately 1 to 3.5 nm in accordance with the relative humidity are arranged to be formed on the surface of the fin 45, the adsorbed water may be desorbed and reproduced (making the water adsorbed again) using the heat (exhaust heat) of the refrigerant flowing in a common refrigeration cycle apparatus.

[0042] The fine pores having a regular pattern are formed by the anodic oxidation method vertically to the surface of the fin 45. Therefore, unlike the adsorbing material with the fine pores arranged in no regular pattern, for example, the flow directionality of the adsorbed water may be aligned to efficiently transfer the heat of the fin 45.

EXAMPLE 2

[0043] Fig. 6 is a configuration view showing an essential portion of a heat exchanger 40a according to Embodiment 2 of the present invention. In Fig. 6, column 45aa, 45ab, 45ac are made to have fine pores with different diameters for each column along the air flow direction. That is, against the air flow direction upon the adsorption, the fine pore diameter at an upwind side is made large, and that at a downwind side is made small. Therefore, in the case of Fig. 6, the fine

pores are distributed in such an aspect that fine pore diameter on the first column 45aa > fine pore diameter on the second column 45ab > fine pore diameter on the third column 45ac.

[0044] Fig. 7a shows a simulated result of the relative humidity distribution of the air flow direction (column direction of the heat transfer pipe) of the heat exchanger using fins having the same fine pore diameter. The water content of the fin 45 has a feature to rise up at the relative humidity of approximately 30%. From the heat transfer pipe at the upwind side upon the adsorption, it is referred to as the first and the second, and the third columns. As can be understood from Fig. 7a, as the air flows from the upwind side to the downwind side, the water is adsorbed by the fin 45, so that the relative humidity around the fin is decreased.

[0045] Meanwhile, the embodiment according to the present invention employs the fin which has the fine pore distribution with different fine pore diameters along the flow direction. That is, viewing from the upwind side, the fine pore diameter on the second column is smaller than that on the first column, and the fine pore diameter on the third column is smaller than that on the second column. The fine pore diameters of the fin 45 is changed for each column. The number of the columns is not especially limited, though, as the number of the columns is increased, the diameter of the fine pores distributed on the fin may be gradually changed.

[0046] For example, the first column (45aa) has the fine pores diameter for which the water content rises at the relative humidity of approximately 50% (3.5 nm), the second column (45ab) at the relative humidity of approximately 40% (2.5 nm), and the third column (45ac) at the relative humidity of approximately 30% (2 nm). The change in the adsorption amount with respect to the air flow direction (column direction) is shown in Fig. 7b. As can be understood from Fig. 7b, the total adsorption amount of the fin with the fine pores made to have a distribution along the column direction is larger than that of the fin having the same fine pores diameter. So that the fin may be effectively used.

[0047] In the case of the fin having the same fine pores diameter, not falling within the scope of the claims, the more downstream side the fin is located, the smaller the difference becomes between the relative humidity of air around the fin and the relative humidity 30 %, where the water content of the fin rises. So that the adsorbing speed of the fin is reduced, and resultantly, the adsorption amount decreases toward the downstream side.

[0048] The fin having the fine pores of relatively larger diameter may be employed, so that it is possible to reduce the total cost of the fin. (the larger the fine pore diameter becomes, the less manufacturing period, and accordingly, the production cost may be reduced).

[0049] Fig. 8a shows the relative humidity to the air flow direction upon desorption when using the fin with fine pores having the same diameter, not falling within the scope of the claims. The air flow direction is reversed to that upon adsorption. While on desorption, the fin at the third column is at the upwind side, on adsorption it is at the downwind side. From Fig. 8a, it is found that the relative humidity around the fin becomes larger toward the downwind side.

[0050] The embodiment according to the present invention employs the fins having a fine pore distribution making the fine pore diameter changed along the flow direction. For example, the first column (45aa) of the fin has the fine pore diameter for which the water content rises at the relative humidity of approximately 50% (about 3.5 nm), the second column (45ab) of the fin the relative humidity of approximately 40% (about 2.5 nm), and the third column (45ac) of the fin relative humidity of approximately 30% (about 2 nm), respectively. As shown in Fig. 8b, the fine pore distribution in the column direction provides a larger desorption amount in total compared with the use of the fin having the fine pores of the same diameter, resulting in the effective use of the fin. In the case of the fins having the fine pores of the same diameter, which embodiment does not fall within the scope of the claims, the difference between the relative humidity of air around the fin and the relative humidity 30%, where the water content around the fin 45 rises, becomes smaller toward the downstream side to decelerate adsorbing rate of the fin, an desorption amount is decreased toward the downwind side.

[0051] The distribution of the fine pores diameter of the fin 45 formed along the column direction allows effective use of the fin 45 to improve the adsorption/desorption performance. Resultantly, it is possible to make the heat exchanger compact.

[0052] When the fin includes two columns (Fig. 1 shows two columns, however, three or more columns may be possible) as shown in Fig. 1, the relative humidity is lowered at the downwind side, it is preferable to set the fine pore diameter to be relatively smaller. As shown in Fig. 7a, the relative humidity around the fin adjacent to the outlet is approximately 35%. The fine pore diameter may be set to approximately 1 to 3.5 nm because the fin is required to have characteristics in which the water content abruptly rises at the relative humidity of about 20 to 40%.

[0053] In the description, the fine pore diameter of the fin at the downwind side with respect to the air flow direction in the water absorption is smaller than the fine pore diameter of the fin at the upwind side. However, the fine pore diameter of the fin at the downwind side with respect to the air flow direction in the water absorption may be larger than that of the fin at the more downwind side than that so far as the fine pore diameters are different for each column and in the range approximately from 1 to 3.5 nm. For example, when the fine pore diameter of the fin outside the heat exchanger may be set to be larger than the fine pore diameter of the fin inside the heat exchanger, it is possible to perform efficient dehumidification by making the air in contact with the heat exchanger flow from two directions rather than a single direction.

[0054] A method for manufacturing the heat exchanger having a fine pore distribution using an anodic oxidation method

will be briefly described. As shown in Fig. 9a, a three-column fin 650 is collectively submerged in an electrolysis vessel 630 to form a fin having homogeneous fine pores. Meanwhile, each column of the fin 650 is submerged to form predetermined fine pores as shown in Fig. 9b when manufacturing the heat exchanger having fine pore distribution along the column direction. The anodic oxidation is conducted three times under the different conditions to produce three fins having different fine pores. Thereafter, the heat transfer pipes 43 are inserted for each column to be finally combined together by connecting a U-type pipe for providing the three-column heat exchanger. In Figs. 9(a) and 9(b), reference numerals 610, 620 and 640 denote a direct current power supply, an electrolyte, and a cathode, respectively.

[0055] The effect derived from operating the heat exchangers 40, 40a using the fins with fine pores while being cooled with the refrigerant supplied thereto will be described. Fig. 10 shows operations on the psychrometric diagram when the refrigerant is supplied to the heat exchanger and when the refrigerant is not supplied. At the inlet of the heat exchangers 40 and 40a, the dry-bulb temperature, the relative humidity, the absolute humidity, and the dew point of air are 25°C, 60%, 0.0119[kg/kg], and 16.7°C, respectively. In the case where the refrigerant is not supplied, the relative humidity is lowered while the air temperature is increased by the adsorbing heat to finally become the air of the state (b)(dry-bulb temperature: 32.2°C, relative humidity: 30%, absolute humidity: 0.0119kg/kg, and dew point: 12.44°C). Meanwhile, in the case where the heat exchanger 40 is operated while cooling with the refrigerant supplied thereto, the refrigerant removes the adsorbing heat to substantially realize isothermal adsorption to finally become the state (c) (dry-bulb temperature: 25°C, relative humidity: 30%, absolute humidity: 0.0058 kg/kg, and dew point: 6.24°C).

[0056] The difference in the absolute humidity for the case where the refrigerant is supplied and the case where the refrigerant is not supplied shows that the absolute humidity difference in the case where the refrigerant is supplied is twice higher than the absolute humidity difference in the case where the refrigerant is not supplied (absolute humidity difference when the refrigerant is not supplied: 0.0029[kg/kg], absolute humidity difference when the refrigerant is supplied: 0.00601[kg/kg]). That is, adsorption conducted while supplying the refrigerant to the heat exchangers 40,40a may largely improve the absorbing performance. The isothermal adsorption allows the dew point to be decreased from 12.44°C to 6.24°C, thus applicable for use at the low dew point.

[0057] By preparing the fine pores which cause the capillary condensation on the fin of the heat exchanger, and supplying the refrigerant to the heat transfer pipes 46, it is possible to provide a latent heat exchanger having the largely improved adsorbing performance.

EMBODIMENT 3

[0058] In the above Embodiment 1, the fin 45 is made of aluminum, however, the material is not limited to the aluminum. For example, the so-called valve metal may be used as the material of the fin 45 to form fine pores on the surface through the anodic oxidation. The valve metal refers to a generic name of a metal which forms an oxide film showing an electrolytic rectifying operation through the anodic oxidation such as aluminum, tantalum, niobium, titanium, hafnium, zirconium, zinc, tungsten, bismuth and antimony. Among them, metals such as aluminum, titanium, zirconium, niobium, and tantalum may be practically used as the fin 45. The use of those metals may provide the same effect as aluminum.

EMBODIMENT 4

[0059] Fig. 11 shows an exemplary structure of a refrigeration cycle apparatus 100 according to Embodiment 4 of the present invention. The basic structure of the refrigerant circuit composed by the refrigeration cycle apparatus 100 will be described based on Fig. 11. The refrigeration cycle apparatus 100 is operated for cooling, refrigerating, and air conditioning by circulating the refrigerant.

[0060] The refrigeration cycle apparatus 100 is formed by sequentially connecting the compressor 10, the condenser 20, the first throttle device 60, the heat exchanger 41 for dehumidification/humidification, the second throttle device 61, and the evaporator 70 with a refrigerant pipe 1. Here, the explanation will be given on the assumption that the compressor 10 and the condenser 20 are built into an outdoor unit (unit at the heat source side) which is disposed outside the space to be cooled/air-conditioned, and the first throttle device 60, the heat exchanger 41 for dehumidification/humidification, the second throttle device 61, and the evaporator 70 are built into the indoor unit (unit at the load side) disposed inside the subject space. Here, the condenser 20 is disposed at the outdoor unit, and the evaporator 70 is disposed at the indoor unit for explaining such operations for cooling and air-conditioning, however, these roles will be switched in the case of the heating operation. The switching is performed by the control unit for controlling a four-way valve (not shown).

[0061] The refrigerant pipe 1 includes a refrigeration pipe at the gas side which allows communication of a gaseous refrigerant, and the refrigeration pipe at the liquid side which allows communication of the liquid refrigerant. The refrigeration pipe at the liquid side communicates the refrigerant which has been condensed and liquefied, and the refrigeration pipe at the gas side communicates the refrigerant which has been evaporated and gasified. A blower (not shown) such as the fan for feeding air outside the subject space (hereinafter, referred to an external air) into the condenser 20 to promote heat exchange is disposed around the condenser 20. A blower (not shown) such as the fan is also disposed

around the evaporator 70. The refrigerant to be enclosed into the refrigerant pipe 1 will be described later.

[0062] The compressor 10 sucks and compresses the refrigerant into a gaseous state of high temperature/pressure to supply to the refrigerant pipe 1. The condenser 20 is a heat exchanger to perform heat exchange between the refrigerant and external air to condensate/liquefy the refrigerant. The first throttle device 60 is generally composed of a decompression valve and an expansion valve such as an electronic expansion valve for decompressing and expanding the refrigerant.

[0063] The heat exchanger 41 for dehumidification/humidification is composed of the heat exchangers 40, 40a (hereinafter represented by the heat exchanger 40) as described in Embodiments 1 to 3, having fine pores on the surface of the fin 45. The description will be given, not limited to, on the assumption hereinafter that the heat exchanger 41 for dehumidification/humidification includes the fine pores with diameters for increasing the adsorption amount at the relative humidity of approximately 30% to desorbe the adsorbed water. The heat exchanger 41 for dehumidification/humidification mainly as an apparatus to remove the latent heat so as to supply air within the dehumidified subject space (hereinafter referred to simply the air) to the evaporator 70 by adsorbing the water. It is not limited to the usage, however, humidification unit may be disposed in the refrigeration cycle apparatus 100 to humidify the subject space using the heat exchanger 41 for dehumidification/humidification.

[0064] The second throttle device 61 is generally composed of the decompression valve and the expansion valve such as the electronic expansion valve to decompress and expand the refrigerant. The evaporator 70 evaporates and gasifies the refrigerant through the heat exchange between the refrigerant and the air. The blower disposed adjacent to the evaporator 70 sucks the air and supplies the cooled air through the heat exchange in the evaporator 70 to a region to be cooled (interior space, in the refrigerator, and refrigerated warehouse). For example, the control unit 80 composed of such as microcomputers controls the drive frequency of the compressor 10, and opening of the first throttle device 60 and the second throttle device 61. In the embodiment, description is given as a single control unit 80, however, the control units may be provided for both the outdoor unit and the indoor unit, respectively and each controller controls the apparatus (unit) that each unit possesses. Thereby, an associated control is possible by enabling signal communication.

[0065] The refrigerant used for the refrigeration cycle apparatus 100 will be described. As for the refrigerant used for the refrigeration cycle apparatus 100, there are the zeotropic refrigerant mixture, the quasi-azeotropic refrigerant mixture, and the single refrigerant. Regarding the zeotropic refrigerant mixture, there is such as R407C (R32/R125/R134a), which is an HFC (hydrofluoro carbon) refrigerant. Since the zeotropic refrigerant mixture is a mixture of refrigerants having different boiling points, it has characteristics that composition ratios of the liquid-phase refrigerant and gas-phase refrigerant are different. For the quasi-azeotropic mixture refrigerant, there are such as R410A (R32/R125), R404A(R125/R143a/R134a), that are HFC refrigerants.

[0066] The single refrigerant has the type R22 as the HCFC (hydro chlorofluorocarbon) refrigerant, and R134a as the HFC refrigerant. The single refrigerant has characteristics that it is not a mixture, so that it may be easily handled. Such natural refrigerants as carbon dioxide, propane, isobutene, and ammonia may also be employed. The R22, R32, R125, R134a, and R143a denote chlorodifluoromethane, difluoromethane, pentafluoroethane, 1,1,1,2-tetrafluoroetahne, and 1,1,1-trifluoroethane, respectively. Accordingly, the refrigerant suitable for the usage and object of the refrigeration cycle apparatus 100 may be employed.

[0067] The operation of the refrigeration cycle apparatus 100 will be described. The heat exchanger 41 for dehumidification/humidification will be described with respect to the operation for adsorbing the water in the air. The refrigerant of high temperature/pressure compressed by the compressor 10 is condensed/liquefied to be the liquid refrigerant while releasing heat through the heat exchange with the external air in the condenser 20. The liquid refrigerant flows into the first throttle device 60 and decompressed therein to become the low pressure gas-liquid two-phase refrigerant. The gas-liquid two-phase refrigerant having the temperature lower than the air flowed into the heat exchanger 41 for dehumidification/humidification is made to cool the fin 45 and air passing therearound through the heat exchange, and part of the air is evaporated and discharged. At this time, the fin 45 adsorbs the water of the passing air. The gas-liquid two-phase refrigerant discharged from the heat exchanger 41 for dehumidification/humidification passes through the fully opened second throttle device 61 to flow into the evaporator 70. All the gas-liquid two-phase refrigerant is evaporated/gasified through the heat exchange in the evaporator 70 to be a gaseous refrigerator, and sucked by the compressor 10 again and discharged.

[0068] The operation of the heat exchanger 41 for dehumidification/humidification to desorb the adsorbed water will be described. The refrigerant of high temperature/pressure compressed by the compressor 10 becomes the gas-liquid two-phase refrigerant while releasing the heat to the external air in the condenser 20. The gas-liquid two-phase refrigerant under the high pressure state passes through the fully opened first throttle device 60 and flows into the heat exchanger 41 for dehumidification/humidification. The gas-liquid two-phase refrigerant having the higher temperature than the air flowed into the heat exchanger 41 for dehumidification/humidification heats to liquefy the fin 45 and the ambient air. The liquefied refrigerant is decompressed by the second throttle device 61 to become the low pressure gas-liquid two-phase refrigerant. The gas-liquid two-phase refrigerant flows into the evaporator 70, evaporated and gasified entirely into the gaseous refrigerant, and arranged to be sucked by the compressor 10 again.

[0069] Fig. 12 is a view showing the relationship between the evaporating temperature and COP (Coefficient of Performance: energy consumption efficiency). Fig. 12 shows the proportional relation between the evaporating temperature and the COP. For example, when the evaporating temperature is 11[°C], the COP is approximately 3.1 (shown by (A)). When the evaporating temperature is increased to 20[°C], the COP increases up to approximately 3.9 (shown by (B)). The increase in the evaporating temperature may improve the COP accordingly.

[0070] In the refrigeration cycle apparatus 100 according to the present embodiment, the latent heat of the water contained in the air and the sensible heat may be processed by the heat exchanger 41 for dehumidification/humidification and the evaporator 70 respectively, so that the division of roles is achieved. Unlike the case where the evaporator 70 processes the latent heat and the sensible heat, the evaporating temperature of the refrigerant can be set higher. Thus, the air conditioning system can prevent deposition of the frost and needs no defrosting operation even when conventionally the evaporating temperature has to set to be the value lower than the dew point in the evaporator 70 and the frost deposited.

[0071] The use of the condensed exhaust heat in the condenser 20 may desorb the water adsorbed in the heat exchanger 41 for dehumidification/humidification (fin 45). The desorbed water may be disposed, or used for humidification. The heating device such as the heater for desorbing the water is no longer necessary, thus requiring no power for the heating device. This makes it possible to largely reduce the power consumption.

[0072] When the refrigeration cycle apparatus 100 according to the present embodiment is applied to the refrigerated warehouse under the conditions where external air is maintained at the dry-bulb temperature of 30[°C], the relative humidity of 60[%], and absolute humidity of 16.04[g/kg], the control unit 80 may control each apparatus so that the refrigeration cycle apparatus is operated for the refrigerated room (air-conditioned space) in the refrigerated warehouse to be maintained and continued under the conditions of the dry-bulb temperature 10[°C], the relative humidity 60[%], and the absolute humidity 4.56[g/kg].

[0073] In the refrigeration cycle apparatus according to Embodiment 4, by making the water adsorbed by the fin 45 using the heat exchanger 41 for dehumidification/humidification as described in Embodiments 1 to 3 as the dehumidifying/humidifying device, the evaporating temperature of the refrigerant in the heat exchange with the air in the evaporator 70 does not have to be set in consideration of the latent heat caused by the water, so that the refrigerant may be controlled to the temperature in consideration of the sensible heat. This makes it possible to make the compression ratio in the compressor of the refrigeration cycle apparatus small, thus improving the energy performance represented by the COP in the refrigeration cycle apparatus as an index.

EMBODIMENT 5

[0074] Fig. 13 shows an exemplary structure of the refrigeration cycle apparatus 100a according to Embodiment 3 of the present invention. The refrigeration cycle apparatus 100a of the present embodiment is formed, but is not limitedto, as an air conditioning system for heating/cooling operations. The apparatuses shown in Fig. 13 with the same reference numerals as those described in Embodiment 4 perform the same functions, so that explanations will be omitted.

[0075] The refrigeration cycle apparatus 100a is formed by the compressor 10, the condenser 20, the first on-off valve 30 and the second on-off valve 31 which are provided in parallel, the heat exchangers 41a and 41b for dehumidification/humidification which are provided in parallel, back-flow prevention members 50 and 51 which are provided in parallel, the throttle device 62, and the evaporator 70, being sequentially connected with the refrigerant pipe 1. As described in Embodiments 1 to 3, the refrigeration cycle apparatus 100a includes heat exchangers 41a (a first heat exchanger) and 41b (a second heat exchanger) for dehumidification/humidification that are the heat exchangers 40 having the fin 45 with fine pores formed on the surface. These two heat exchangers 41a and 41b for dehumidification/humidification are independently built into the indoor unit independently.

[0076] The refrigerant pipe 1 is branched into refrigerant pipes 1a and 1b. After the on-off valve 30, the heat exchanger 41a for dehumidification/humidification, and the back-flow prevention member 50 are connected by refrigerant pipe 1a, and the on-off valve 31, the heat exchanger 41b for dehumidification/humidification and the back-flow prevention member 51 being connected by refrigerant pipe 1b respectively, these are joined together again. The refrigerant flowing in the refrigerant pipe 1 may employ the one described in Embodiment 2. The refrigeration cycle apparatus 100a is provided with a temperature/humidity detection unit 81 (a first temperature/humidity detection unit) at the inlet of the air passage of the evaporator 70 for detecting the temperature and humidity of air flowing into the evaporator 70.

[0077] The temperature/humidity detection unit 81 may be of any type so far as the temperature and the humidity are detected and types are not limited in particular. For example, the temperature sensor such as the therimstor, thermometer, humidity sensor, and hygrometer may be employed.

[0078] The on-off valves 30 and 31 function as flow passage selecting units for selecting the refrigerant circuit that are not limited to the particular type. The back-flow prevention members 50 and 51 prevent back-flow of the refrigerant flowing through the refrigerant pipes 1a and 1b. Such as a check valve may be employed, but is not limited to the particular type. The throttle device 62 is generally composed of a decompression valve and expansion valve for decompressing the refrigerant to expand. The electronic expansion valve may be employed, for example. The control unit 80

according to the present embodiment controls the respective apparatuses including the on-off valves 30, 31 in addition to the controlling operations described above. The control unit further performs the air passage control by switching the air passage switching units 301a to 304a, and 301b to 304b described later, and calculates the relative humidity of the air in the evaporator 70 based on the information from the temperature/humidity detection unit 81 to convert the relative humidity into the dew point (dew-point temperature).

[0079] Fig. 14 shows a structure of an indoor unit 300 where the evaporator 70 and the like are built-in. The indoor unit 300 shown in Fig. 14 is partially disposed in the refrigerated warehouse (air-conditioned space) 400 and the rest portion is disposed at the external air side 500. In the indoor unit 300, the heat exchangers 41a and 41b for dehumidification/humidification and the evaporator 70 as shown in Fig. 13 are built-in. Blowers 90 and 91 such as a centrifugal fan and axial flow fan are disposed adjacent to the heat exchangers 41a and 41b for dehumidification/humidification. The indoor unit 300 is provided with a duct 310 which not only feeds air from the evaporator 70 to the refrigerated warehouse 400 but also sucks the air.

[0080] The indoor unit 300 is structured to disconnect the air passage (air flow) between the heat exchangers 41a and 41b for dehumidification/humidification. The indoor unit 300 is capable of switching the air passage. By switching the air passage the heat exchangers 41a and 41b for dehumidification/humidification can be communicated with the inside of the refrigerated warehouse 400 and the external air 500. The operation for switching the air passage is performed by the air passage switching units 301a and 301b, 302a and 302b, 303a and 303b, and 304a and 304b, respectively. The air passage may be finely adjusted by the air passage adjustment units 305a and 305b.

[0081] The air flow in the indoor unit 300 will be described. Referring to Fig. 14, the air passage switching units 301a, 302b, 303b, and 304a are opened, and the air passage switching units 301b, 302a, 303a, and 304b are closed. When the respective air passage switching units are in the aforementioned state, the built-in space of the heat exchanger 41a for dehumidification/humidification is communicated with the external air 500 to allow the air to flow from outside (arrow A). The built-in space of the heat exchanger 41b for dehumidification/humidification is communicated with the inside of the refrigerated warehouse 400 via the duct 310 to allow the air (for example, the temperature 10[°C] and relative humidity 60[%])(arrow B) to flow in.

[0082] In the above structured air passages, the heat exchanger 41a for dehumidification/humidification performs desorption, and the heat exchanger 41b for dehumidification/humidification performs adsorption. Thereby, the latent heat may be processed by the heat exchanger 41b for dehumidification/humidification and the sensible heat may be processed by the evaporator 70 individually. Meanwhile, when the open/closed states of each air passage switching unit are inverted, the air flows into the built-in space of the heat exchanger 41a for dehumidification/humidification, and the expanded air flows into the built-in space of the heat exchanger 41b for dehumidification/humidification. The heat exchanger 41a for dehumidification/humidification performs adsorption, and the heat exchanger 41b for dehumidification/humidification performs desorption, respectively.

[0083] Fig. 15 is an explanatory view showing the state where the air passages in the indoor unit 300 is switched. Referring to Fig. 15(a), the air passage switching units 301a, 302b, 303b and 304a are closed, and the air passage switching units 301b, 302a, 303a and 304b are opened.

[0084] As shown in Fig. 15(a), the built-in space of the heat exchanger 41b for dehumidification/humidification is communicated with the external air 500 to allow the external air to flow in (arrow C). The built-in space of the heat exchanger 41a for dehumidification/humidification is communicated with the inside of the refrigerated warehouse 400 via the duct 310 to allow the air to flow in (arrow D). At this time, the heat exchanger 41b for dehumidification/humidification desorbs the water, and the heat exchanger 41a for dehumidification/humidification adsorbs the water. Fig. 15(b) shows the same as what is shown in Fig. 14, so that the explanation will be omitted.

[0085] Fig. 16 is a P-h diagram (Mollier diagram) which represents the refrigerant state in the refrigeration cycle. The refrigerant state in the refrigeration cycle will be described based on Fig. 16. The vertical axis of the diagram denotes an absolute pressure (P), and the horizontal axis denotes enthalpy (h). Referring to Fig. 16, the region surrounded by the saturated liquid line and the saturated vapor line represents the refrigerant in the gas-liquid two-phase state. The region to the left of the saturated liquid line represents the liquefied refrigerant, and the region to the right of the saturated vapor line represents the gaseous refrigerant. That is, in the states (1) and (5), the refrigerant is gaseous, and in the states (2) and (4), the refrigerant is in the gas-liquid two-phase state. In the state (3), the refrigerant is liquefied.

[0086] The operation of the refrigeration cycle apparatus 100a will be described based on Figs. 13 and 16. Descriptions will be given to when the on-off valve 30 is opened, the on-off valve 31 is closed, the heat exchanger 41a for dehumidification/humidification is operated for desorbing the water, and the heat exchanger 41b for dehumidification/humidification is operated for adsorbing the water. Since the on-off valve 31 is closed, the refrigerant does not flow into the heat exchanger 41b for dehumidification/humidification.

[0087] The refrigerant in the gaseous state of high temperature/pressure state compressed by the compressor 10 (in the state (1) shown in Fig. 16) flows into the condenser 20. The refrigerant in the aforementioned state turns into the gas-liquid two-phase state while partially releasing heat to the external air in the condenser 20 (the state (2) shown in Fig. 16). The gas-liquid two-phase refrigerant of high pressure state flows into the heat exchanger 41a for dehumidifi-

cation/humidification, and passes through the heat transfer pipe 46. At this time, a heat exchange between the refrigerant and the air is conducted to increase temperature of the fin 45 and the ambient air to reduce the relative humidity. As a result, the water adsorbed on the fin 45 is desorbed. The gas-liquid two-phase refrigerant turns into the liquefied refrigerant (the state (3) shown in Fig. 16).

[0088] The refrigerant flows in the back-flow prevention member 50 to be decompressed in the throttle device 62. The decompressed refrigerant becomes a low pressure gas-liquid two-phase refrigerant (the state (4) shown in Fig. 16). The gas-liquid two-phase refrigerant flows into the evaporator 70 and is evaporated by removing heat from the air to become a low pressure gaseous refrigerant (the state (5) shown in Fig. 16). The air here adsorbs the water by the heat exchanger 41b for dehumidification/humidification as described later. The air is cooled to flow out into the refrigerated warehouse 400. Then, the gaseous refrigerant is sucked by the compressor 10 again to circulate in the refrigerant circuit. The cooling/refrigerating operations are conducted by circulating the refrigerant in the refrigerant circuit while changing the states of the refrigerant by repeating the heat absorbing/releasing operations.

[0089] Fig. 17 is a psychrometric diagram for explaining the operation of the heat exchanger 41b for dehumidification/humidification in the refrigeration cycle apparatus 100a. The operation of the above refrigeration cycle apparatus 100a will be described using the psychrometric diagram and the structural view of Fig. 14. In Figs. 14 and 17, for the air passing through the heat exchanger 41b for dehumidification/humidification made to be communicated with the inside of the refrigerated warehouse 400, descriptions are given to the state (1) shown in Fig. 17 representing the state of the air before passing through the heat exchanger 41b for dehumidification/humidification, the state (2) shown in Fig. 17 representing the state of the air immediately after passing through the heat exchanger 41b for dehumidification/humidification, and the state (3) shown in Fig. 17 representing the state of the air immediately after the heat exchange with the evaporator 70.

[0090] Descriptions will be given to when the heat exchanger 41b for dehumidification/humidification adsorbs the water content of the air inside the refrigerated warehouse 400. The air in the state (1) is the dry-bulb temperature 10[°C], the relative humidity 60[%], and the absolute humidity 4.56[g/kg]. When the air in the aforementioned state flows into the heat exchanger 41b for dehumidification/humidification, the air is brought into the state (2) along an equi-enthalpy line to be fed to the evaporator 70, where the relative humidity being reduced from 60[%] to 30[%], the absolute humidity being reduced from 4.56[g/kg] to 2.96[g/kg], and the dry-bulb temperature being increased from 10[°C] to 14[°C].

[0091] Since the amount of the water adsorbed by the heat exchanger 41b for dehumidification/humidification becomes large in the region where the relative humidity is equal to or higher than approximately 30%, it is possible to dehumidify the air in the state (1). The air in the state (2) is cooled by the removal of the sensible heat through the heat exchange of the evaporator 70 in the state of constant absolute humidity to turn into the air of the state (3) where the relative humidity is lower than 100[%] and the dry-bulb temperature is -2[°C].

[0092] In most cases, the inside of the refrigerated warehouse 400 is generally kept at the temperature range lower than 10[°C], and the evaporating temperature is required to set lower than 0[°C]. However, the refrigeration cycle apparatus 100a is capable of setting the evaporating temperature of the evaporator 70 (14[°C] of the state (2)) to be higher than the dew-point temperature (for example, the dew-point temperature -2.9[°C] of the state (2)) so as not to allow the refrigeration cycle to execute a defrosting operation for removing the frost formed on the evaporator 70.

[0093] The control unit 80 may be configured to adjust the evaporating temperature of the evaporator 70 to increase by controlling the opening of the throttle device 62, the drive frequency of the compressor 10, the rotating speed of the blower 91 and the like. As described in Fig. 12, if the evaporating temperature is set high, the COP may be improved by that amount. Since the evaporating temperature of the evaporator 70 may be higher than the dew point, no drain occurs. That is, no drain pipe is required, thus reducing the manufacturing cost.

[0094] The control unit 80 calculates the relative humidity of the air in the evaporator 70 based on the information from the temperature/humidity detection unit 81. The calculated relative humidity is then converted into the dew point. The dew point may be detected based on the converted result. Air in the state (3) is diffused into the refrigerated warehouse 400 to maintain the dry-bulb temperature at 10[°C] or lower. The amount of the water content which can be adsorbed by the heat exchanger 41b for dehumidification/humidification is limited. When it is determined that the relative humidity of the heat exchanger 41g for dehumidification/humidification at the outlet of the air passage becomes equal to or larger than a predetermined threshold value based on the detection information from the temperature/humidity detection unit 81, the control unit 80 switches the on-off valve 30 from the open to the closed state, and the on-off valve 31 from the closed to the open state to switch the refrigerant flow. The gaseous refrigerant of high temperature/pressure is made to flow into the heat exchanger 41b for dehumidification/humidification to increase temperatures of the fin 45 and the ambient air.

[0095] That is, the operation of the heat exchanger 41b for dehumidification/humidification which has been adsorbing the water is switched to desorb the water. When temperatures of the fin 45 of the heat exchanger 41b for dehumidification/humidification and the ambient air are increased, the relative humidity is decreased to release the adsorbed water for reproduction. Meanwhile, the refrigerant flow passage is switched, so that the heat exchanger 41a for dehumidification/humidification comes to adsorb the water in the air. The heat exchanger 41a for dehumidification/humidification is

structured to adsorb the water in the air such that the air inside the warehouse 400 is dehumidified from the state (1) to (2) as shown in Fig. 17.

[0096] The amount of the water which can be adsorbed by the heat exchanger 41a for dehumidification/humidification is limited. When it is determined that the relative humidity of the heat exchanger 41a for dehumidification/humidification at the outlet side of the air passage becomes equal to or higher than a predetermined threshold value based on the detection information from the temperature/humidity detection unit 81, the control unit 80 switches the on-off valve 30 from the closed to the open state, and the on-off valve 31 from the open to the closed state to switch the refrigerant flow. The gaseous refrigerant of high temperature/pressure is fed into the heat exchanger 41a for dehumidification/humidification to increase temperatures of the fin 45 and the ambient air such that the relative humidity is lowered to desorb the water.

[0097] As mentioned above, when one of the heat exchangers for dehumidification/humidification (heat exchanger 41b for dehumidification/humidification) adsorbs the water, the refrigeration cycle apparatus 100a is structured to allow the other heat exchanger for dehumidification/humidification (heat exchanger 41a for dehumidification/humidification) to desorb the water. The operations of the heat exchangers are alternately switched depending on the amount of the adsorbed water. Switching of the air passage to select the refrigerant flow passage allows the humidity (latent heat) of the air in the refrigerated warehouse 400 to be removed continuously.

[0098] Table 2 collectively shows control states of the on-off valves 30 and 31 (flow passage switching unit) and air passage switching units 301a to 304b, and functions of the heat exchangers 41a and 41b for dehumidification/humidification. In Table 2, pattern 1 represents that the heat exchanger 41a for dehumidification/humidification adsorbs water, and the heat exchanger 41b for dehumidification/humidification desorbs the adsorbed water as shown in Fig. 15(a). Patten 2 represents that the heat exchanger 41a for dehumidification/humidification desorbs the adsorbed water, and the heat exchanger 41b for dehumidification/humidification adsorbs the water as shown in Fig. 15 (b). The continuous operation may be performed by switching the patterns 1 and 2.

[Table 2]

Pattern	Heat exchanger		On-off valve		Air passage switching unit		Air passage switching unit		Air passage switching unit		Air passage switching unit	
	41a	41b	30	31	301a	301b	302a	302b	303a	303b	304a	304b
1	Adsorb	Desorb	Close	Open	Close	Open	Open	Close	Open	Close	Close	Open
2	Desorb	Adsorb	Open	Close	Open	Close	Close	Open	Close	Open	Open	Close

[0099] As described above, the refrigeration cycle apparatus 100a according to Embodiment 5 is structured to allow the heat exchangers 41a and 41b for dehumidification/humidification composed of the heat exchanger 40 according to Embodiment s 1 to 3 to alternately adsorb the water in air in the refrigerated warehouse 400 continuously. This makes it possible to eliminate the defrosting operation conventionally frequently performed to further reduce the power consumption for the defrosting operation. The evaporating temperature of the evaporator 70 may be set higher than the dew-point temperature to enable an efficient operation of the refrigeration cycle.

[0100] Since the water adsorbed by the heat exchangers 41a and 41b for dehumidification/humidification is configured to be desorbed using the heat (exhaust heat which is not required for cooling the inside of the refrigerated warehouse 400) of the refrigerant condensed by the condenser 20, no specific heating device for the desorption is required, and the space for accommodation can be saved, so that no electric power is required for heating by the heating unit.

[0101] The refrigeration cycle apparatus 100a does not require a high pressure in excess of a critical pressure. That is, the compressor 10, the condenser 20 and the refrigerant pipe 1 (including refrigerant pipes 1a and 1b) connecting those may be low in pressure-resistant performance, so that manufacturing costs can be reduced. The compression ratio of the refrigerant in the compressor 10 may be suppressed, thus improving the operation efficiency of the compressor 10. That is, COP can be significantly improved and energy saving can be achieved.

EMBODIMENT 6

[0102] Fig. 18 shows an exemplary structure of a refrigeration cycle apparatus 100b according to Embodiment 6 of the present invention. Descriptions will be given, but not limited in particular, to that the refrigeration cycle apparatus 100b of the present embodiment is, for example, an air conditioning system for cooling/heating operations. In Fig. 18, since the apparatuses designated with the same reference numerals as those described in Embodiments 4 and 5 perform the same functions, explanations will be omitted.

[0103] The refrigeration cycle apparatus 100b is formed by the compressor 10, the condenser 20, the first and the second on-off valves 32 and 33 provided in parallel, the heat exchangers 41c and 41d for dehumidification/humidification provided in parallel, the on-off valves 34 and 35 provided in parallel, the throttle device 85 (third throttle device), and the evaporator 70,being sequentially connecting with the refrigerant pipe 1. The refrigeration cycle apparatus 100b is also provided with the heat exchangers 41c (a first heat exchanger) and 41d (a second heat exchanger) for dehumidification/humidification that are the heat exchanger 40 having the fin 45 with the fine pores formed on the surface. Those two heat exchangers 41c and 41d for dehumidification/humidification are built into the indoor unit separately.

[0104] The refrigerant pipe 1 is branched into the refrigerant pipes 1c and 1d. After the on-off valve 32, the heat exchanger 41c for dehumidification/humidification, and the on-off valve 34 are connected by the refrigerant pipe 1c, and the on-off valve 33, the heat exchanger 41d for dehumidification/humidification, and the on-off valve 35 are connected by the refrigerant pipe 1d, these are then joined again. The refrigerant flowing in the refrigerant pipe 1 may employ the refrigerant described above. The refrigerant pipes 1c and 1d include a bypass pipe 2 (a first bypass pipe) branched from the refrigerant pipe 1c between the on-off valve 32 and the heat exchanger 41c for dehumidification/humidification to join with the refrigerant pipe 1d between the heat exchanger 41d for dehumidification/humidification and the on-off valve 35, and a bypass pipe 3 (a second bypass pipe) branched from the refrigerant pipe 1d between the on-off valve 33 and the heat exchanger 41d for dehumidification/humidification to join with the refrigerant pipe 1c between the heat exchanger 41c for dehumidification/humidification and the on-off valve 34.

[0105] The bypass pipe 3 includes a throttle device 63 (a first throttle device) and an on-off valve 36 (a third on-off valve). The bypass pipe 2 includes a throttle device 64 (a second throttle device) and an on-off valve 37 (a fourth on-off valve). The refrigeration cycle apparatus 100b includes a temperature/humidity detection unit 81 for detecting the temperature/humidity of the evaporator 70 at the inlet side of the air passage thereof, and a temperature/humidity detection unit 82 (a second temperature/humidity detection unit) for detecting the temperature/humidity of the heat exchangers 41c and 41d for dehumidification/humidification at the outlet side of the air passage of the heat exchanger 41c for dehumidification/humidification, respectively.

[0106] The temperature/humidity detection units 81 and 82 may be of any type so far as the temperature and the humidity are detected and types are not limited in particular. For example, the temperature sensor such as the thermistor, thermometer, humidity sensor, and hygrometer may be employed. In the example, the apparatus employs a single unit of the temperature/humidity detection units 81 and 82, respectively, however, plural units may be employed without being limited to the above. The temperature/humidity detection unit 82 may be disposed at the outlet sides of the respective air passages of the heat exchangers 41c and 41d for dehumidification/humidification.

[0107] The refrigeration cycle apparatus 100b is provided with a control unit 80a which controls a drive frequency of the compressor 10, opening of the on-off valves 32 to 37, and opening of the throttle devices 63, 64 and 85. The on-off valves 32 to 37 are operated for switching the flow passages, not limited to a specific type. The throttle devices 63, 64 and 85 are generally composed of the decompression and expansion valves to decompress and expand the refrigerant, and may be composed of an electronic expansion valve and the like.

[0108] In addition to controlling each apparatus, the control unit 80a calculates the relative humidity of the heat exchanger 41c for dehumidification/humidification at the outlet side of the air passage based on the signal which contains data from the temperature/humidity detection unit 82 to convert the calculated relative humidity into the dew point (dew-point temperature). The control unit 80 also controls the relative humidity in the evaporator 70 based on the information from the temperature/humidity detection unit 81 to convert and the calculated relative humidity into the dew point (dew-point temperature). When the desorption/adsorption function is switched between the heat exchangers 41c and 41d for dehumidification/humidification, the control unit 80a calculates the relative humidity of the heat exchanger 41d for dehumidification/humidification at the outlet side of the air passage to convert the calculated relative humidity into the dew point (dew-point temperature).

[0109] Fig. 19 shows a structure of an indoor unit 300a where the evaporator 70 and the like are built-in. The basic structure of the indoor unit 300a will be described based on Fig. 19. Descriptions will be given to differences from the indoor unit 300 shown in Fig. 14. In Fig. 19, a part of the indoor unit 300a is disposed inside (air-conditioned space) 401 of the room, and the rest is disposed at the side of the external air 500. In the indoor unit 300a, the heat exchangers 41c and 41d for dehumidification/humidification and the evaporator 70 shown in Fig. 18 are built-in.

[0110] The indoor unit 300a is structured to disconnect the air passage between the heat exchangers 41c and 41d for dehumidification/humidification. The indoor unit 300a is allowed to switch the air passage to communicate the heat exchangers 41c and 41d for dehumidification/humidification with the interior 401 and the external air 500. The switching of the air passage may be performed by air passage switching units 311a and 311b, 312a and 312b, 313a and 313b, and 314a and 314b, respectively. The fine adjustment of the air passage may be performed by the air passage adjustment units 315a and 315b.

[0111] The air flow in the indoor unit 300a will be described. Fig. 19 shows that the air passage switching units 311a, 312b, 313b and 314a are opened, and the air passage switching units 311b, 312a, 313a and 314b are closed. In the aforementioned state of the air passage switching units, the built-in space of the heat exchanger 41c for dehumidification/humidification is communicated with the external air 500 to allow the air to flow from outside (arrow A). The built-in space of the heat exchanger 41d for dehumidification/humidification is communicated with the interior 401 via the duct 310 to allow the air (for example, temperature 26[°C] and relative humidity 60[%]) to flow in (arrow B).

[0112] When the air passages are formed as described above, the heat exchanger 41c for dehumidification/humidification performs the desorption, and the heat exchanger 41d for dehumidification/humidification performs the adsorption. Thereby, the heat exchanger 41d for dehumidification/humidification processes latent heat, and the evaporator 70 processes sensible heat separately. Meanwhile, when the open-close states of the respective air passage switching units are inverted, the heat exchanger 41c for dehumidification/humidification performs adsorption, and the heat exchanger 41d for humidification/humidification performs desorption.

[0113] Fig. 20 is an explanatory view showing the state where the air passage of the indoor unit 300a is switched. Referring to Fig. 20, a portion of the indoor unit 300a is disposed in the interior 401, and the rest is disposed at the side of the external air 500. Fig. 20(a) shows that the air passage switching units 311a, 312b, 313b and 314a are closed, and the air passage switching units 311b, 312a, 313a and 314b are closed.

[0114] Referring to Fig. 20(a), the built-in space of the heat exchanger 41d for dehumidification/humidification is communicated with the external air 500 to allow the external air to flow in (arrow C). The built-in space of the heat exchanger 41c for dehumidification/humidification is communicated with the interior 401 via the duct 310 to allow the air to flow in (arrow D). At this time, the heat exchanger 41d for dehumidification/humidification desorbs water, and the heat exchanger 41c for dehumidification/humidification adsorbs the water, respectively. Fig. 20(b) shows the same as what is shown in Fig. 19, so that the explanation will be omitted.

[0115] In the case where the refrigeration cycle apparatus 100b is applied to an air-conditioning apparatus such as the room air-conditioner and all-in-one air conditioning system with the condition of the external air 500 being kept at the dry-bulb temperature 30 [°C], the relative humidity 60 [%], and the absolute humidity 16.04 [g/kg], the control unit 80a should control the respective apparatuses to operate the refrigeration cycle apparatus 100b while maintaining and continuing the interior 401 (air-conditioned space) under conditions of the dry-bulb temperature 26 [°C], the relative humidity 60 [%], and the absolute humidity 8.74 [g/kg].

[0116] Fig. 21 is a P-h diagram (Mollier diagram) which represents the refrigerant state in the refrigeration cycle. The refrigerant state in the refrigeration cycle will be described based on Fig. 21. Referring to Fig. 21, it is configured to be able to understand that the refrigerant is gaseous in the states (1) and (7). The refrigerant is in the gas-liquid two-phase state in the states (2), (4), (5) and (6). The refrigerant is liquefied in the state (3).

[0117] The operation of the refrigeration cycle apparatus 100b will be described based on Figs. 18 and 21. Descriptions will be given on the operation of the refrigeration cycle apparatus 100b when the on-off vales 32, 34 and 35 are opened, the on-off valves 33, 37 and 36 are closed, the heat exchanger 41c for dehumidification/humidification is operated as the heat exchanger for desorption, and the heat exchanger 41d for dehumidification/humidification is operated as the heat exchanger for adsorption.

[0118] The gaseous refrigerant of high temperature/pressure compressed by the compressor 10 (the state (1) shown

in Fig. 21) flows into the condenser 20. The refrigerant in the aforementioned state turns into the gas-liquid two-phase state (state (2) shown in Fig. 21) while partially releasing the heat to the external air by the condenser 20. The high pressure gas-liquid two-phase refrigerant flows into the heat exchanger 41c for dehumidification/humidification. The gas-liquid two-phase refrigerant flowing into the heat exchanger 41c for dehumidification/humidification increases the temperatures of the fin 45 and the ambient air to reduce the relative humidity. Thereby, the water adsorbed in the fin 45 is desorbed. The gas-liquid two-phase refrigerant turns into the liquefied refrigerant (state (3) shown in Fig. 21).

[0119] The aforementioned refrigerant flows through the on-off valve 36 to be decompressed by the throttle device 63. The decompressed refrigerant turns into the low pressure gas-liquid two-phase state (state (4) shown in Fig. 21, here, the first evaporating temperature). Then the gas-liquid two-phase refrigerant flows into the heat exchanger 41d for dehumidification/humidification to lower the temperatures of the fin 45 and the ambient air with the first evaporating temperature lower than the air and enhance the adsorbing performance. The gas-liquid two-phase refrigerant flowing into the heat exchanger 41d for dehumidification/humidification partially evaporates to turn into the low pressure gas-liquid two-phase refrigerant (state (5) shown in Fig. 21). The gas-liquid two-phase refrigerant is further decompressed by the throttle device 85 to be a second evaporating temperature (state (6) shown in Fig. 21), then flows into the evaporator 70 to turn into a low pressure gaseous refrigerant by absorbing the sensible heat of the air through the heat exchange (state (7) shown in Fig. 21). The gaseous refrigerant is sucked by the compressor 10 again to circulate in the refrigerant circuit.

[0120] The refrigeration cycle apparatus 100b is structured to allow the refrigerant which has passed through one of the heat exchangers for dehumidification/humidification (heat exchanger 41c for dehumidification/humidification) to flow into the other heat exchanger for dehumidification/humidification (heat exchanger 41d for dehumidification/humidification) via a bypass pipe (bypass pipe 3). As a result, the heat exchanger 41 for dehumidification/humidification efficiently desorbs the water using the heat of the refrigerant related to condensation, and the other heat exchanger 41 for dehumidification/humidification efficiently adsorbs the water using the heat of the refrigerant related to evaporation to enhance adsorption/desorption performance and improve the performance of the refrigeration cycle apparatus.

[0121] Fig. 22 is a psychrometric diagram for explaining the operation of the heat exchanger 41d for dehumidification/humidification of the refrigeration cycle apparatus 100b. The operation of the above-mentioned refrigeration cycle apparatus 100b will be described referring to the psychrometric diagram and the structure shown in Fig. 19. Referring to Figs. 19 and 22, for the air passing through the heat exchanger 41d for dehumidification/humidification communicating with the interior 401, the state (1) shown in Fig. 22 represents the state of air before passing through the heat exchanger 41d for dehumidification/humidification , the point (2) shown in Fig. 22 represents the state of air immediately after passing through the heat exchanger 41d for dehumidification/humidification, and the point (3) shown in Fig. 22 represents the state of air immediately after the heat exchange with the evaporator 70.

[0122] The operation of the heat exchanger 41d for dehumidification/humidification when adsorbing the water of air in the interior 401 will be described. The air in the state (1) is the dry-bulb temperature of 26 [°C] and the relative humidity 60 [%] when the air in this state flows into the heat exchanger 41d for dehumidification/humidification, the air is subjected to isothermal or cooling adsorption in the heat exchanger 41d to turn into the state (2) to flow into the evaporator 70. Same the amount of water which can be adsorbed by the heat exchanger 41b for dehumidification/humidification is increased in the region of the relative humidity of 30% or higher, the air in the state (1) can be dehumidified.

[0123] The air in the state (2) is subjected to heat exchange by the evaporator 70 to turn into the air in the state (3). The air in the state (2) is cooled with only the sensible heat being removed at a constant absolute humidity by the evaporator 70 to turn into the state (3) where the relative humidity is lower than 100 [%] and the dry-bulb temperature is 14 [°C]. The air in the state (3) is supplied to the interior 401.

[0124] The control unit 80a controls the opening of the throttle devices 63 and 85, the drive frequency of the compressor 10, and the rotating speed of the blower 91, and adjusts the first evaporating temperature to be equal to or higher than the dew point (in the present embodiment, 18 [°C]) of the intake air in the heat exchanger 41d for dehumidification/humidification. The control unit controls the second evaporating temperature to be equal to or higher than the dew point (in the present embodiment, 14 [°C]) of air at the outlet of the heat exchanger 41d for dehumidification/humidification. The control unit 80a converts the data of temperature and humidity detected by the temperature/humidity detection units 81 and 82 into the dew point. Fig. 22 shows the first evaporating temperature of 18 [°C], and the second evaporating temperature of 14 [°C].

[0125] The amount of water is limited which can be adsorbed by the heat exchanger 41d for dehumidification/humidification functioning as a heat adsorption exchanger. When it is determined that the relative humidity in the evaporator 70 becomes equal to or higher than a predetermined threshold value based on the data detected by the temperature/humidity detection unit 81, the control unit 80a switches on-off valves 32, 36 and 35 from the open to the closed state, and on-off valves 33, 37 and 34 from the closed to the open state to change the refrigerant flow. A high temperature/pressure gaseous refrigerant is fed into the heat exchanger 41d for dehumidification/humidification to increase the temperatures of the fin 45 and the ambient air for desorption and reproduction.

[0126] Since the refrigerant flow passage is switched, the heat exchanger 41c for dehumidification/humidification is

operated as an adsorption heat exchanger. In the heat exchanger 41c for dehumidification/humidification, the water contained in the air is adsorbed. The refrigeration cycle apparatus 100b, alternately switches heat exchangers according to the water adsorption amount such that when one of the heat exchangers for dehumidification/humidification (heat exchanger 41d for dehumidification/humidification) is adsorbs the water, the other heat exchanger for dehumidification/humidification (heat exchanger 41c for dehumidification/humidification) desorbs the water. The air in the interior 401 may be continuously dehumidified (latent heat may be removed) by switching the air passages.

[0127] The external air for example, the dry-bulb temperature 32 [°C] and the relative humidity 60 [%], is supplied from the external air 500 side to the heat exchanger 41c for dehumidification/humidification by the blower 90. The heat exchanger 41c for dehumidification/humidification desorbs the adsorbed water. Then, the absolute humidity is increased through the desorption, and the air is discharged to the external air 500 again. The air is merely discharged here, however, the desorbed water may be used for humidification. In this way, the latent heat may be removed by the heat exchangers 41c, 41d for dehumidification/humidification, and the sensible heat may be removed by the evaporator 70. The exhaust heat generated by condensation in the condenser 20 may be used for desorbing the adsorbed water, air-conditioning and refrigerating performance is significantly improved.

[0128] Table 3 shows control states of the on-off valves 32 to 37 and functions of the heat exchangers 41c, 41d for dehumidification/humidification. Referring to Table 3, the pattern 1 shows that the heat exchanger 41d for dehumidification/humidification adsorbs the water, and the heat exchanger 41c for dehumidification/humidification desorbs the adsorbed water as shown in Fig. 20(b). Then, the on-off valves 32, 36 and 35 are opened, and the on-off valves 33, 37 and 34 are closed. The pattern 2 shows that the heat exchanger 41d for dehumidification/humidification desorbs the adsorbed water, and the heat exchanger 41c for dehumidification/humidification adsorbs the water as shown in Fig. 20(a). Then, the on-off valves 33, 37 and 34 are opened, and the on-off valves 32, 36 and 35 are closed. Continuous operations may be performed by alternately switching the patterns 1 and 2.

[Table 3]

Pattern	Heat Exchanger		On-off valve		On-off valve		On-off valve	
	41c	41d	32	33	36	37	34	35
1	Desorption	Adsorption	Open	Close	Open	Close	Close	Open
2	Adsorption	Desorption	Close	Open	Close	Open	Open	Close

[0129] When either of the heat exchangers 41c and 41d for dehumidification/humidification desorbs the water, the refrigerant is condensed therein. When adsorbing the water, the refrigerant is evaporated. Functions of the heat exchangers 41c and 41d for dehumidification/humidification may be switched by controlling opening of each on-off valve to switch the refrigerant flow and continuous operation is possible while switching adsorption and desorption alternately.

[0130] When the air-conditioned space is at the dry-bulb temperature of 26 [°C] and the relative humidity of 60 [%], and the external air is at the dry-bulb temperature of 32 [°C] and the relative humidity of 60 [%], the conventional refrigeration cycle apparatus is used to adjust the balance of the condenser 20 at the condensation temperature of about 47 [°C] and the evaporator 70 at the evaporating temperature of about 11 [°C] to process both the sensible heat (cooling operation) and the latent heat (dehumidifying operation) of the air-conditioned space simultaneously. Such a refrigeration cycle apparatus requires the evaporating temperature to be set low, so that the operation efficiency is poor.

[0131] The refrigeration cycle apparatus 100b allows the sensible heat processing (cooling operation) and the latent heat processing (dehumidifying operation) in the air-conditioned space to be conducted separately. The evaporator 70 is operated only for processing the sensible heat, so that the evaporating temperature can be set high. The evaporating temperature conventionally set at 11 [°C] may be increased to be as high as approximately 14 [°C]. As a result, the refrigerant cycle efficiency may be largely improved.

[0132] As shown in Fig. 12, the evaporating temperature is proportional to the COP. In Embodiment 4, when the evaporating temperature is 11 [°C], the COP is approximately 3.1 ((A) in the drawing). When the evaporating temperature is increased to 14 [°C], the COP is increased up to approximately 3.3 (shown as the point (B) in the drawing). The increase in the evaporating temperature by 3 [°C] may improve the COP by approximately 14%.

[0133] Likewise Embodiment 5, the refrigeration cycle apparatus 100b according to Embodiment 6 is not required to conduct defrosting, and allows the evaporating temperature of the evaporator 70 to be set high. In the present embodiment, since the refrigerant of the first evaporating temperature lower than the air is made to lower the temperature of the fins 45 of the heat exchanger 41c or 41d for dehumidification/humidification on the side of adsorbing the water and the ambient air to increase the relative humidity so as to promote the water adsorption, it is possible to realize a higher performance operation.

EXAMPLE 7

[0134] Fig. 23 shows an exemplary structure of a refrigeration cycle apparatus 100c according to Embodiment 7 of the present invention. The refrigeration cycle apparatus 100c of the present embodiment is described as, for example, but not limited to, the air conditioning system for cooling/heating operations. Referring to Fig. 23, since what is designated with the same reference numerals as those described in Embodiments 4, 5 and 6 have the same functions, explanations will be omitted.

[0135] The refrigeration cycle apparatus 100c is formed by sequentially connecting the compressor 10, the condenser 20, the on-off valve 32 as the first on-off valve and the on-off valve 33 as the second on-off valve that are provided in parallel, the heat exchangers 41e and 41f for dehumidification/humidification provided in parallel, three-way valves 38 and 39 provided in parallel, the on-off valves 34 and 35 provided in parallel, throttle device 85, and evaporator 70 with the refrigerant pipe 1. Here, the refrigeration cycle apparatus 100c is also provided with the heat exchanger 41e (a first heat exchanger) for dehumidification/humidification and the heat exchanger 41f (a second heat exchanger) for dehumidification/humidification that are the heat exchanger 40 including the fin 45 which fine pores are formed on the surface. Those two heat exchangers 41e and 41f for dehumidification/humidification are built into the indoor unit separately. The control unit 80b controls the three-way valves 38 and 39 to switch the refrigerant flow passage.

[0136] Likewise the refrigeration cycle apparatus 100b according to the above Embodiment 6, the refrigerant pipe 1 is branched into the refrigerant pipes 1c and 1d, and after connecting the on-off valve 32, the heat exchanger 41e for dehumidification/humidification, and the three-way valve 38 with the refrigerant pipe 1c connecting the on-off valve 33, the heat exchanger 41f for dehumidification/humidification, and the three-way valve 39 with the refrigerant pipe 1b, respectively, they are joined again. The aforementioned refrigerant may be employed as the one flowing through the refrigerant pipe 1. The refrigerant pipes 1c and 1d include a bypass pipe 2a (a first bypass pipe) which is branched from the refrigerant pipe 1c between the on-off valve 32 and the heat exchanger 41e for dehumidification/humidification to join with the refrigerant pipe 1d between the heat exchanger 41f for dehumidification/humidification and the three-way valve 39, and a bypass pipe 3a (a second bypass pipe) which is branched from the refrigerant pipe 1d between the on-off valve 33 and the heat exchanger 41f for dehumidification/humidification to join with the refrigerant pipe 1c between the heat exchanger 41e for dehumidification/humidification and the three-way valve 38.

[0137] The bypass pipe 3a is provided with the throttle device 63. The bypass pipe 2a is provided with the throttle device 64. The refrigeration cycle apparatus 100c is provided with a temperature/humidity detection unit (not shown) for detecting the temperature and humidity of the evaporator 70 at the inlet of the air passage of the evaporator 70. The temperature/humidity detection unit is not limited to the specific type so far as the temperature and the humidity may be detected. For example, a temperature sensor such as a thermistor, thermometer, humidity sensor, and hygrometer may be employed.

[0138] The refrigeration cycle apparatus 100c is provided with a control unit (not shown) for controlling the drive frequency of the compressor 10, opening of the on-off valves 32, 33, opening of the throttle devices 63, 64 and 85, and opening of the three-way valves 38, 39. The three-way valves 38 and 39 switch the flow of the refrigerant flowing through the refrigerant pipes 1a and 1b to switch the functions (adsorption and desorption) of the heat exchangers 41e and 41f for dehumidification/humidification.

[0139] The operation of the refrigeration cycle apparatus 100c will be described based on Fig. 23 on the assumption that the on-off valve 32 is opened, the on-off valve 33 is closed, the heat exchanger 41e for dehumidification/humidification is operated to desorb the water, and the heat exchanger 41f for dehumidification/humidification is operated to adsorb the water.

[0140] A high temperature/pressure gaseous refrigerant compressed by the compressor 10 flows into the condenser 20. The refrigerant in the aforementioned state becomes the gas-liquid two-phase refrigerant in the condenser 20 while partially releasing the heat to the external air. The high pressure gas-liquid two-phase refrigerant flows into the heat exchanger 41e for dehumidification/humidification. The incoming gas-liquid two-phase refrigerant passes through the heat transfer pipe 46 to allow the heat exchange between the refrigerant and air and then increase the temperature of the fins 45 and the ambient air to lower the relative humidity. Thereby, the water adsorbed in the fin 45 is desorbed. The gas-liquid two-phase refrigerant is liquefied into the liquid refrigerant.

[0141] The refrigerant flowing from the heat exchanger 41e for dehumidification/humidification has a flow direction determined by the three-way valve 38 under the control of the control unit 80b. If the refrigerant is controlled to flow through the bypass pipe 3a, the low pressure gas-liquid two-phase refrigerant flows into the heat exchanger 41f for dehumidification/humidification to lower the temperatures of the fins 45 and the ambient air like the Embodiment 6. This makes it possible to enhance the adsorbing performance of the heat exchanger 41f for dehumidification/humidification. The gas-liquid two-phase refrigerant flowing out from the heat exchanger 41f for dehumidification/humidification is decompressed by the throttle device 85 via the three-way valve 38, on-off valve 35, and bypass pipe 1d, flowing into the evaporator 70 to become the low pressure gaseous refrigerant while removing the sensible heat of the air by the heat exchange, then being sucked by the compressor 10 to be circulate in the refrigerant circuit. The operation of the refrig-

eration cycle apparatus 100c is reversed compared with that of the three-way valves 38 and 39 when closing the on-off valve 32, opening the on-off valve 33, making the heat exchanger 41f for dehumidification/humidification desorb the water, and making the heat exchanger 41e for dehumidification/humidification adsorb the water.

[0142] In Embodiment 7, the apparatus is provided with the three-way valves 38, 39 to facilitate water adsorption by lowering the temperatures of the fins 45 of the heat exchanger 41e or 41f for dehumidification/humidification on the side of the water adsorbtion and the ambient air to increase the relative humidity, so that it is possible to operate the apparatus with higher performance.

EMBODIMENT 8

[0143] In Embodiments 4 to 7, the heat exchanger 40 according to Embodiments 1 to 3 is applied only to the heat exchanger 41 for dehumidification/humidification, however, it is not limited thereto. For example, the condenser 20 and the evaporator 70 which serve as the heat exchangers may be provided with the fin 45 for adsorbing the water.

[0144] In Embodiments 4 to 7 as described above, two heat exchangers 41a, 41b for dehumidification/humidification are used for alternate desorption and adsorption, however, the number of the heat exchangers 41 for dehumidification/humidification is not limited.

[0145] In Embodiments 4 to 7, the compressor 10 is not limited to a specific type. For example, the inverter compressor capable of controlling the capacity, and the constant rate compressor which performs the compression at a constant rate may be employed. In the respective embodiments, a single unit of the compressor 10 is disposed for the refrigeration cycle, however, the number of the compressors is not limited and plural compressor may be provided. In the aforementioned case, the control unit 80 may be configured to execute the multiple control of as many as the provided compressors.

EMBODIMENT 9

[0146] In the respective embodiments, the control units 80 and 80a control the opening of the on-off valves, drive frequency of the compressor 10, the opening of the respective throttle devices and the three-way valves, however, it is not limited thereto. The control unit may be provided for each of the apparatuses. In the respective embodiments, a single temperature/humidity detection unit 81 is disposed at the inlet side of the air passage of the evaporator 70, however, it is not limited thereto. For example, the temperature detection unit and the humidity detection unit may be separately provided, or plural units may be provided. The pressure detection units for detecting the refrigerant pressure may be provided adjacent to the respective apparatuses.

[0147] In the embodiments, the refrigeration cycle apparatuses 100 to 100c are applied to the refrigerator, room air-conditioner, all-in-one air conditioning system and the like, however, it is not limited thereto. For example, the refrigeration cycle apparatuses 100 to 100c may be applied to the refrigerated warehouse, humidifier, humidity control unit and the like. The type of the refrigerant, the air passage and the flow passage in the refrigeration cycle may be determined in accordance with the intended use and application.

Claims

1. A heat exchanger (40) for exchanging heat between a refrigerant and air, comprising a plurality of fins (45) for transferring the heat, and heat transfer pipes (46) that are inserted into through holes formed on the fins (45) and disposed in a plurality of columns (45aa, 45ab, 45ac), **characterised in that** the fins (45) include fine pores (45a) with diameters ranging from 1 to 3.5 nm for adsorbing water contained in the air by a capillary condensation phenomenon, wherein
said fine pores are formed on a surface of fin material itself such that the fin itself functions as a water absorbing unit, said diameters differ according to positions on the surface of said fin (45),
said pores are distributed so as to have smaller pore diameters from an upwind side to a downwind side along a air flow direction when moisture is absorbed, and each column (45aa, 45ab, 45ac) of the heat transfer tubes (46) has different diameters of said fine pores.

2. The heat exchanger of Claim 1, wherein the fin (45) includes a first column which has the fine pore with a first predetermined diameter, and a second column which has the fine pore positioned either upwind or downwind of the first fin in an air flow direction upon the water adsorption, and has a second predetermined diameter different from the first predetermined diameter.

3. The heat exchanger of Claim 2, wherein the second column is positioned downwind of the first fin in an air flow direction upon the water adsorption, and the second predetermined diameter is smaller than the first predetermined

diameter.

4. A refrigeration cycle apparatus which includes a refrigerant circuit formed by connecting a compressor (10) for compressing a refrigerant, a condenser (20) for condensing the refrigerant through a heat exchange, a throttle device (61,62,85) for adjusting a pressure of the condensed refrigerant, and an evaporator (70) for evaporating the refrigerant through the heat exchange between the refrigerant decompressed by the throttle device and air, further comprising the heat exchanger (40) of any one of Claims 1 to 3 as a heat exchanger (41) for dehumidification/humidification, wherein the throttle device adjusts a refrigerant pressure for controlling adsorption and desorption performed by the heat exchanger for dehumidification/humidification.

5. The refrigeration cycle apparatus of Claim 4, further comprising:

plural units (41,41a-41f) of the heat exchangers for dehumidification/humidification;
a flow passage switching unit (30-33) for forming a flow passage through which the refrigerant selectively flows into or out of the heat exchanger (41,41a-41f) for dehumidification/humidification; and
a control unit (80,80a,80b) for controlling the flow passage switching unit (30-31, 32-33) to switch between the heat exchanger for dehumidification/humidification for adsorbing water contained in air subjected to a heat exchange in the evaporator by flowing the refrigerant decompressed by the throttle device and the heat exchanger for dehumidification/humidification for desorbing the adsorbed water by flowing the refrigerant condensed by the condenser.

6. A refrigeration cycle apparatus which includes a refrigerant circuit formed by connecting a compressor (10) for compressing a refrigerant, a condenser (20) for condensing the refrigerant through a heat exchange, a throttle device (61,62,85) for adjusting a pressure of the condensed refrigerant, and an evaporator (70) for evaporating the refrigerant through the heat exchange between the refrigerant decompressed by the throttle device and air, further comprising:

plural heat exchangers (41,41a-41f) for dehumidification/humidification each formed as the heat exchanger (40) of any one of Claims 1 to 3;
a flow passage switching unit (30-33) for forming a flow passage through which the refrigerant condensed by the condenser selectively flows into the heat exchanger for dehumidification/humidification; and
a control unit (80,80a,80b) for switching between the heat exchanger for dehumidification/humidification for adsorbing the water contained in the air subjected to a heat exchange in the evaporator and the heat exchanger for dehumidification/humidification for desorbing the adsorbed water by controlling the flow passage switching unit to flow the refrigerant.

7. The refrigeration cycle apparatus of Claim 5 or 6, further comprising a first temperature/humidity detection unit (81) for detecting temperature and/or humidity of the evaporator (70) at an inlet of an air passage of the evaporator, wherein the control unit (80, 80a, 80b) calculates a relative humidity of the air in the evaporator (70) based on data detected by the first temperature/humidity detection unit, and controls the flow passage switching unit (30-31, 32-33) to switch between the heat exchanger for dehumidification/humidification for adsorbing water and the heat exchanger for dehumidification/humidification for desorbing the water based on the calculated relative humidity.

8. The refrigeration cycle apparatus of Claim 5, wherein the control unit (80,80a,80b) further controls a drive frequency of the compressor (10), and converts the relative humidity into a dew point to control the drive frequency of the compressor (10) and switching of the heat exchanger for dehumidification/humidification under the control of the flow passage switching unit (30-33) so that the evaporating temperature of the refrigerant in the evaporator (70) is equal to or higher than the dew point.

9. The refrigeration cycle apparatus of any one of Claims 5 to 8, further comprising a second temperature/humidity detection unit (82) for detecting the temperature and/or the humidity of the heat exchanger (41,41a-41f) for dehumidification/humidification at an inlet of the air passage of the heat exchanger for dehumidification/humidification, wherein the control unit (80,80a,80b) calculates the relative humidity of the air in the heat exchanger for dehumidification/humidification based on the detection data of the second temperature/humidity detection unit to convert the calculated relative humidity into a dew point, and controls the drive frequency of the compressor (10) and switching of the heat exchanger for dehumidification/humidification under the control of the flow passage switching unit (30-33) so that a temperature of a refrigerant which passes in the heat exchanger becomes higher than or equal to the dew point.

10. The refrigeration cycle apparatus of any one of Claims 5 to 8, further comprising an air passage switching unit (301a,b-304a,b;311a,b-314a,b),for switching the air passages between the plural heat exchangers (41,41a-41f) for dehumidification/humidification and the evaporator (70), wherein the control unit (80, 80a, 80b) controls the air passage switching unit to form the air passages for the heat exchanger for adsorbing water and the evaporator.

Patentansprüche

1. Wärmetauscher (40) zum Austauschen von Wärme zwischen einem Kältemittel und Luft, umfassend eine Vielzahl von Rippen (45) zum Übertragen der Wärme, und Wärmeübertragungsrohre (46), die in Durchgangslöcher eingesetzt sind, die auf den Rippen (45) ausgebildet sind, und in einer Vielzahl von Säulen (45aa, 45ab, 45ac) angeordnet sind,
dadurch gekennzeichnet, dass die Rippen (45) enthalten:

Feinporen (45a) mit Durchmessern im Bereich von 1 bis 3,5 nm zum Adsorbieren von in der Luft enthaltenem Wasser durch ein Kapillarkondensationsphänomen, wobei
die Feinporen auf einer Oberfläche des Rippenmaterials selbst ausgebildet sind, so dass die Rippe selbst als eine wasserabsorbierende Einheit wirkt,
wobei sich die Durchmesser entsprechend Positionen auf der Oberfläche der Rippe (45) unterscheiden,
die Poren so verteilt sind, dass sie kleinere Porendurchmesser von einer windzugewandten Seite zu einer windabgewandten Seite entlang einer Luftströmungsrichtung aufweisen, wenn Feuchtigkeit absorbiert wird, und jede Säule (45aa, 45ab, 45ac) der Wärmeübertragungsrohre (46) unterschiedliche Durchmesser der Feinporen aufweist.

2. Wärmeübertrager nach Anspruch 1, wobei die Rippe (45) eine erste Säule, welche die Feinpore mit einem ersten vorherbestimmten Durchmesser aufweist, und eine zweite Säule, welche die Feinpore aufweist, welche entweder windzugewandt von oder windabgewandt zur ersten Rippe in einer Luftströmungsrichtung auf die Wasseradsorption positioniert ist, enthält, und einen zweiten vorherbestimmten Durchmesser aufweist, der sich vom ersten vorherbestimmten Durchmesser unterscheidet.

3. Wärmetauscher nach Anspruch 2, wobei die zweite Säule windabgewandt zur ersten Rippe in einer Luftströmungsrichtung auf die Wasseradsorption positioniert ist, und der zweite vorherbestimmte Durchmesser kleiner ist als der erste vorherbestimmte Durchmesser.

4. Kältemittelkreislaufvorrichtung, welche einen Kältemittelkreislauf, der durch Verbinden eines Verdichters (10) zum Verdichten eines Kältemittels, einen Kondensator (20) zum Kondensieren des Kältemittels durch einen Wärmeaustausch, eine Drosseleinrichtung (61, 62, 85) zum Einstellen eines Druckes des kondensierten Kältemittels und einen Verdampfer (70) zum Verdampfen des Kältemittels durch den Wärmeaustausch zwischen dem Kältemittel, das durch die Drosseleinrichtung entspannt wurde, und Luft, enthält, ferner umfassend den Wärmetauscher (40) nach einem der Ansprüche 1 bis 3 als einen Wärmetauscher (41) zum Entfeuchten/Befeuchten, wobei die Drosseleinrichtung einen Kältemitteldruck zum Steuern von Adsorption und Desorption, die durch den Wärmetauscher zum Entfeuchten/Befeuchten durchgeführt werden, einstellt.

5. Kältemittelkreislaufvorrichtung nach Anspruch 4, ferner umfassend:

mehrere Einheiten (41, 41a-41f) der Wärmetauscher zum Entfeuchten/Befeuchten;
eine Strömungsdurchlassschalteinheit (30-33) zum Bilden eines Strömungsdurchlasses, durch welchen das Kältemittel selektiv hinein in oder heraus aus dem Wärmetauscher (41, 41a-41f) zum Entfeuchten/Befeuchten strömt; und
eine Steuereinheit (80, 80a, 80b) zum Steuern der Strömungsdurchlassschalteinheit (30-31, 32-33), um zwischen dem Wärmetauscher zum Entfeuchten/Befeuchten zum Adsorbieren von Wasser, das in Luft enthalten ist, die im Verdampfer einem Wärmeaustausch unterzogen wurde durch Strömenlassen des durch die Drosseleinrichtung entspannten Kältemittels, und dem Wärmetauscher zum Entfeuchten/Befeuchten zum Desorbieren des adsorbierten Wassers durch Strömenlassen des durch den Kondensator kondensierten Kältemittels umzuschalten.

6. Kältemittelkreislauf, welcher einen Kältemittelkreislauf, der durch Verbinden eines Verdichters (10) zum Verdichten eines Kältemittels, eines Kondensator (20) zum Kondensieren des Kältemittels durch einen Wärmeaustausch, einer

Drosseleinrichtung (61, 62, 85) zum Einstellen eines Druckes des kondensierten Kältemittels und eines Verdampfers (70) zum Verdampfen des Kältemittels durch den Wärmeaustausch zwischen dem Kältemittel, das durch die Drosseleinrichtung entspannt wurde, und Luft, gebildet ist, enthält, ferner umfassend:

mehrere Wärmetauscher (41, 41a-41f) zum Entfeuchten/Befeuchten, die jeweils als der Wärmetauscher (40) nach einem der Ansprüche 1 bis 3 ausgebildet sind;
eine Strömungsdurchlassschalteinheit (30-33) zum Bilden eines Strömungsdurchlasses, durch welchen das durch den Kondensator kondensierte Kältemittel selektiv in den Wärmetauscher zum Entfeuchten/Befeuchten hineinströmt; und
eine Steuereinheit (80, 80a, 80b) zum Schalten zwischen dem Wärmetauscher zum Entfeuchten/Befeuchten zum Adsorbieren des Wassers, das in der Luft enthalten ist, die im Verdampfer einem Wärmeaustausch unterzogen wurde, und dem Wärmetauscher zum Entfeuchten/Befeuchten zum Desorbieren des adsorbierten Wassers durch Steuerung der Strömungsdurchlassschalteinheit, um das Kältemittel strömen zu lassen.

7. Kältemittelkreislaufvorrichtung nach Anspruch 5 oder 6, ferner umfassend eine erste Temperatur/Feuchtigkeit-Erfassungseinheit (81) zum Erfassen einer Temperatur und/oder Feuchtigkeit des Verdampfers (70) an einem Einlass eines Luftdurchlasses des Verdampfers, wobei die Steuereinheit (80, 80a, 80b) eine relative Feuchtigkeit der Luft im Verdampfer (70) auf Grundlage von Daten berechnet, die durch die erste Temperatur/Feuchtigkeit-Erfassungseinheit erfasst wurden, und die die Strömungsdurchlassschalteinheit (30-31, 32-33) steuert, um zwischen dem Wärmetauscher zum Entfeuchten/Befeuchten zum Adsorbieren von Wasser und dem Wärmetaucher zum Entfeuchten/Befeuchten zum Desorbieren des Wassers umzuschalten auf Grundlage der berechneten relativen Feuchtigkeit.

8. Kältemittelkreislaufvorrichtung nach Anspruch 5, wobei die Steuereinheit (80, 80a, 80b) ferner eine Antriebsfrequenz des Verdichters (10) steuert und die relative Feuchtigkeit in einen Taupunkt umwandelt, um die Antriebsfrequenz des Verdichters (10) und Schalten des Wärmetauschers zum Entfeuchten/Befeuchten unter der Steuerung der Strömungsdurchlassschalteinheit (30-33) zu steuern, so dass die Verdampfungstemperatur des Kältemittels im Verdampfer (70) gleich ist wie oder höher ist als der Taupunkt.

9. Kältemittelkreislaufvorrichtung nach einem der Ansprüche 5 bis 8, ferner umfassend eine zweite Temperatur/Feuchtigkeit-Erfassungseinheit (82) zum Erfassen der Temperatur und/oder der Feuchtigkeit des Wärmetauschers (41, 41a-41f) zum Entfeuchten/Befeuchten an einem Einlass des Luftdurchlasses des Wärmetauschers zum Entfeuchten/Befeuchten, wobei die Steuereinheit (80, 80a, 80b) die relative Feuchtigkeit der Luft im Wärmetauscher zum Entfeuchten/Befeuchten berechnet auf Grundlage der Erfassungsdaten der zweiten Temperatur/Feuchtigkeit-Erfassungseinheit, um die berechnete relative Feuchtigkeit in einen Taupunkt umzuwandeln, und die Antriebsfrequenz des Verdichters (10) und Schalten des Wärmetauschers zum Entfeuchten/Befeuchten unter der Steuerung der Strömungsdurchlassschalteinheit (30-33) steuert, so dass eine Temperatur eines Kältemittels, welches im Wärmetauscher strömt, höher wird als oder gleich wird wie der Taupunkt.

10. Kältemittelkreislaufvorrichtung nach einem der Ansprüche 5 bis 8, ferner umfassend eine Luftdurchlassschalteinheit (301a,b-304a,b;311a,b-314a,b) zum Schalten der Luftdurchlässe zwischen den mehreren Wärmetauschern (41,41a-41f) zum Entfeuchten/Befeuchten, und den Verdampfer (70), wobei die Steuereinheit (80, 80a, 80b) die Luftdurchlassschalteinheit, um die Luftdurchlässe für den Wärmetauscher zum Adsorbieren von Wasser zu bilden, und den Verdampfer steuert.

Revendications

1. Échangeur thermique (40) destiné à échanger de la chaleur entre un réfrigérant et de l'air, comprenant une pluralité d'ailettes (45) destinées à transférer de la chaleur, et des conduits de transfert de chaleur (46) qui sont insérés dans des orifices traversants formés sur les ailettes (45) et disposés dans une pluralité de colonnes (45aa, 45ab, 45ac), **caractérisé en ce que** les ailettes (45) comprennent
des pores fins (45a) avec des diamètres compris entre 1 et 3,5 nm pour adsorber l'eau contenue dans l'air grâce à un phénomène de condensation capillaire, dans lequel
lesdits pores fins sont formés sur une surface d'un matériau d'ailette lui-même de sorte que l'ailette elle-même fonctionne comme une unité d'absorption de l'eau,
lesdits diamètres diffèrent selon les emplacements sur la surface de ladite ailette (45),
lesdits pores sont répartis de façon à avoir des diamètres de pores plus petits d'un côté amont jusqu'à un côté aval le long d'une direction de flux d'air lorsque l'humidité est absorbée, et

chaque colonne (45aa, 45ab, 45ac) des conduits de transfert de chaleur (46) possède des diamètres différents pour lesdits pores fins.

2. Échangeur thermique selon la revendication 1, dans lequel l'ailette (45) comprend une première colonne qui possède le pore fin avec un premier diamètre prédéterminé, et une seconde colonne qui possède le pore fin positionné soit en amont, soit en aval de la première ailette dans une direction de flux d'air lors de l'adsorption de l'eau, et possède un second diamètre prédéterminé différent du premier diamètre prédéterminé.

3. Échangeur thermique selon la revendication 2, dans lequel la seconde colonne est positionnée en aval de la première ailette dans une direction de flux d'air lors de l'adsorption de l'eau, et le second diamètre prédéterminé est plus petit que le premier diamètre prédéterminé.

4. Appareil à cycle de réfrigération qui comprend un circuit de réfrigérant formé en reliant un compresseur (10) destiné à comprimer un réfrigérant, un condenseur (20) destiné à condenser le réfrigérant par le biais d'un échange thermique, un dispositif d'étranglement (61, 62, 85) destiné à régler une pression du réfrigérant condensé, et un évaporateur (70) destiné à évaporer le réfrigérant par le biais de l'échange thermique entre le réfrigérant décompressé par le dispositif d'étranglement et l'air, comprenant en outre l'échangeur thermique (40) selon l'une quelconque des revendications 1 à 3 en tant qu'échangeur thermique (41) destiné à la déshumidification/l'humidification, dans lequel le dispositif d'étranglement ajuste une pression de réfrigérant pour contrôler l'adsorption et la désorption réalisées par l'échangeur thermique pour la déshumidification/l'humidification.

5. Appareil à cycle de réfrigération selon la revendication 4, comprenant en outre :

plusieurs unités (41, 41a-41f) d'échangeurs thermiques pour la déshumidification/l'humidification ;
une unité de commutation de passage d'écoulement (30-33) destinée à former un passage d'écoulement par lequel le réfrigérant circule de manière sélective dans ou hors de l'échangeur thermique (41, 41a-41f) pour la déshumidification/l'humidification ; et
une unité de commande (80, 80a, 80b) destinée à contrôler l'unité de commutation de passage d'écoulement (30-31, 32-33) afin de basculer entre l'échangeur thermique pour la déshumidification/l'humidification destiné à adsorber l'eau contenue dans l'air soumis à un échange de chaleur dans l'évaporateur en faisant circuler le réfrigérant décompressé par le dispositif d'étranglement et l'échangeur thermique pour la déshumidification/l'humidification destiné à désorber l'eau adsorbée en faisant circuler le réfrigérant condensé par le condenseur.

6. Appareil à cycle de réfrigération qui comprend un circuit de réfrigérant formé en reliant un compresseur (10) destiné à comprimer un réfrigérant, un condenseur (20) destiné à condenser le réfrigérant par le biais d'un échange de chaleur, un dispositif d'étranglement (61, 62, 85) destiné à ajuster une pression du réfrigérant condensé, et un évaporateur (70) destiné à évaporer le réfrigérant par le biais de l'échange de chaleur entre le réfrigérant décompressé par le dispositif d'étranglement et l'air, comprenant en outre :

plusieurs échangeurs thermiques (41, 41a-41f) pour la déshumidification/l'humidification, chacun formé comme l'échangeur thermique (40) selon l'une quelconque des revendications 1 à 3 ;
une unité de commutation de passage d'écoulement (30-33) destiné à former un passage d'écoulement par lequel le réfrigérant condensé par le condenseur s'écoule de manière sélective vers l'échangeur thermique pour la déshumidification/l'humidification ; et
une unité de commande (80, 80a, 80b) destiné à basculer entre l'échangeur thermique pour la déshumidification/l'humidification pour adsorber l'eau contenue dans l'air soumis à un échange thermique dans l'évaporateur et l'échangeur thermique pour la déshumidification/l'humidification destiné à désorber l'eau adsorbée en contrôlant l'unité de commutation de passage d'écoulement afin de faire circuler le réfrigérant.

7. Appareil à cycle de réfrigération selon la revendication 5 ou 6, comprenant en outre une première unité de détection de température/d'humidité (81) destinée à détecter la température et/ou l'humidité de l'évaporateur (70) au niveau d'une entrée d'un passage d'air de l'évaporateur, dans lequel l'unité de commande (80, 80a, 80b) calcule une humidité relative de l'air dans l'évaporateur (70) en fonction des données détectées par la première unité de détection de température/d'humidité, et contrôle l'unité de commutation de passage d'écoulement (30-31, 32-33) afin de basculer entre l'échangeur thermique pour la déshumidification/l'humidification destiné à adsorber l'eau et l'échangeur thermique pour la déshumidification/l'humidification destiné à désorber l'eau en fonction de l'humidité relative calculée.

8. Appareil à cycle de réfrigération selon la revendication 5, dans lequel l'unité de commande (80, 80a, 80b) contrôle en outre une fréquence d'entraînement du compresseur (10), et convertit l'humidité relative en un point de rosée afin de contrôler la fréquence d'entraînement du compresseur (10) et la commutation de l'échangeur thermique pour la déshumidification/l'humidification sous le contrôle de l'unité de commutation de passage d'écoulement (30-33) de sorte que la température d'évaporation du réfrigérant dans l'évaporateur (70) soit égale ou supérieure au point de rosée.

9. Appareil à cycle de réfrigération selon l'une quelconque des revendication 5 à 8, comprenant en outre une seconde unité de détection de température/d'humidité (82) destinée à détecter la température et/ou l'humidité de l'échangeur thermique (41, 41a-41f) pour la déshumidification/l'humidification au niveau d'une entrée du passage d'air de l'échangeur thermique pour la déshumidification/l'humidification, dans lequel l'unité de commande (80, 80a, 80b) calcule l'humidité relative de l'air dans l'échangeur thermique pour la déshumidification/l'humidification en fonction des données de détection de la seconde unité de détection de température/d'humidité afin de convertir l'humidité relative calculée en un point de rosée, et contrôle la fréquence d'entraînement du compresseur (10) et la commutation de l'échangeur thermique pour la déshumidification/l'humidification sous le contrôle de l'unité de commutation de passage d'écoulement (30-33) de sorte qu'une température d'un réfrigérant qui passe dans l'échangeur thermique devienne supérieure ou égale au point de rosée.

10. Appareil à cycle de réfrigération selon l'une quelconque des revendications 5 à 8, comprenant en outre une unité de commutation de passage d'air (301a, b-304a, b ; 311a, b-314a, b), pour commuter les passages d'air entre les multiples échangeurs thermiques (41, 41a-41f) pour la déshumidification/l'humidification et l'évaporateur (70), dans lequel l'unité de commande (80, 80a, 80b) contrôle l'unité de commutation de passage d'air afin de former les passages d'air pour l'échangeur thermique destiné à adsorber l'eau et l'évaporateur.

FIG. 1

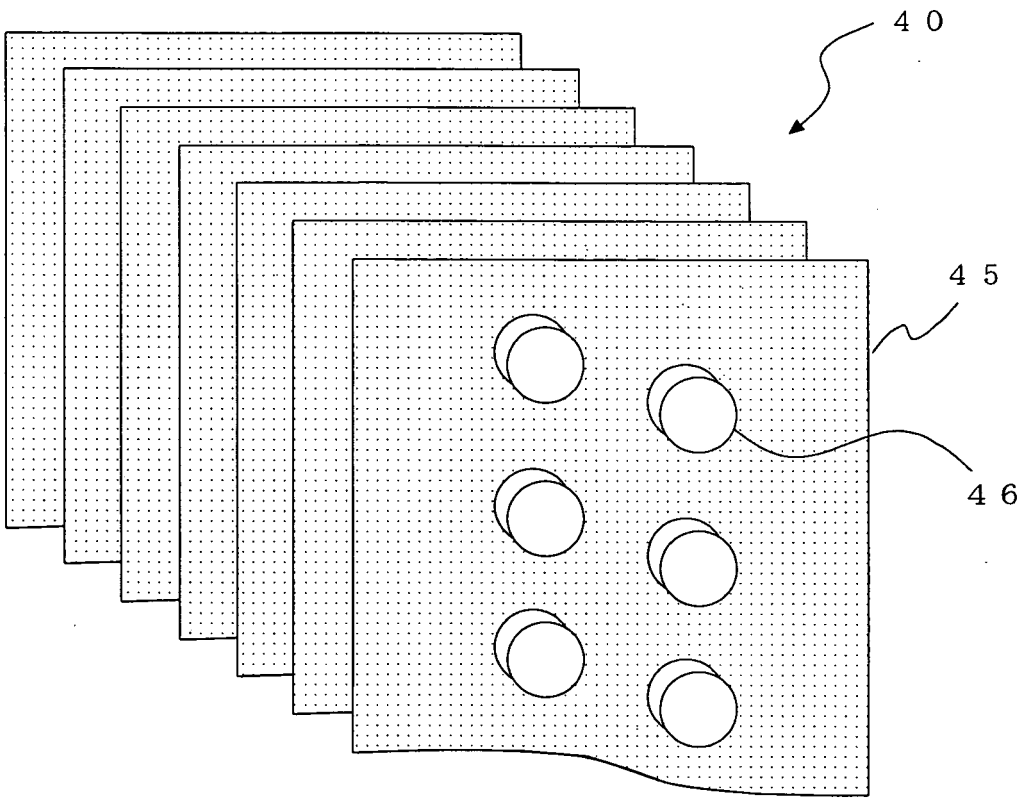


FIG. 2a

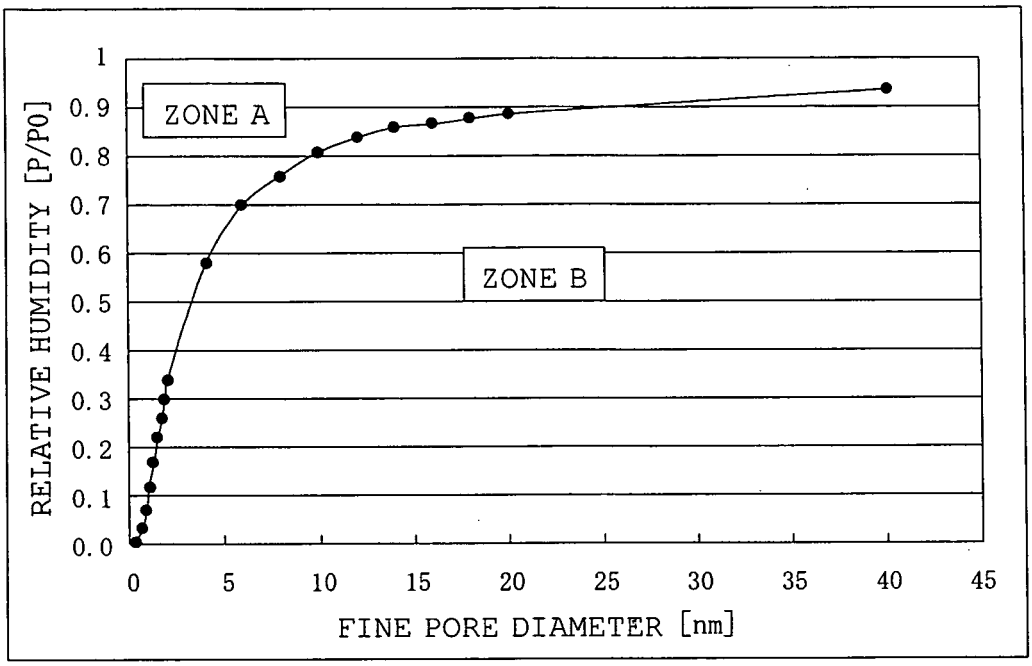

RELATIVE HUMIDITY [P/P0]
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
ZONE A
ZONE B
0
5
10
15
20
25
30
35
40
45
FINE PORE DIAMETER [nm]

FIG. 2b

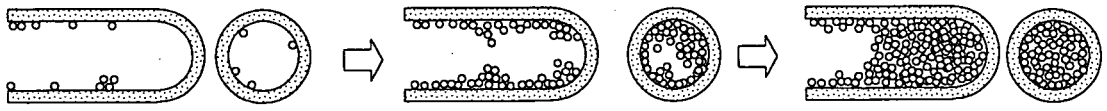

INCREASE IN HUMIDITY
INCREASE IN HUMIDITY

FIG. 3

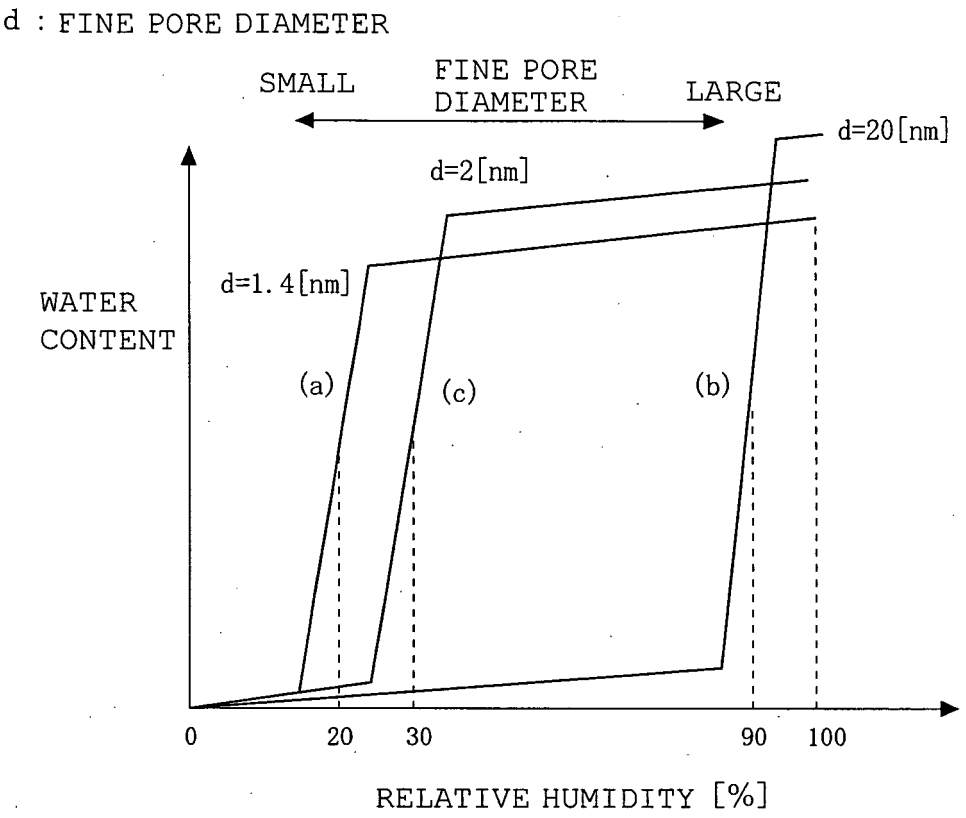

d : FINE PORE DIAMETER
SMALL
FINE PORE DIAMETER
LARGE
d=20[nm]
d=2[nm]
d=1.4[nm]
WATER CONTENT
(a)
(c)
(b)
0
20
30
90
100
RELATIVE HUMIDITY [%]

FIG. 4

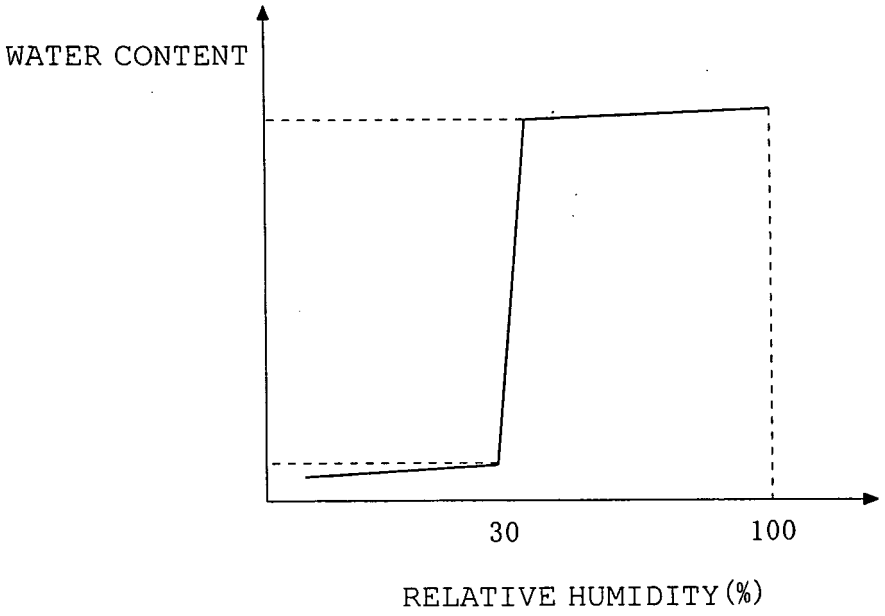


FIG. 5

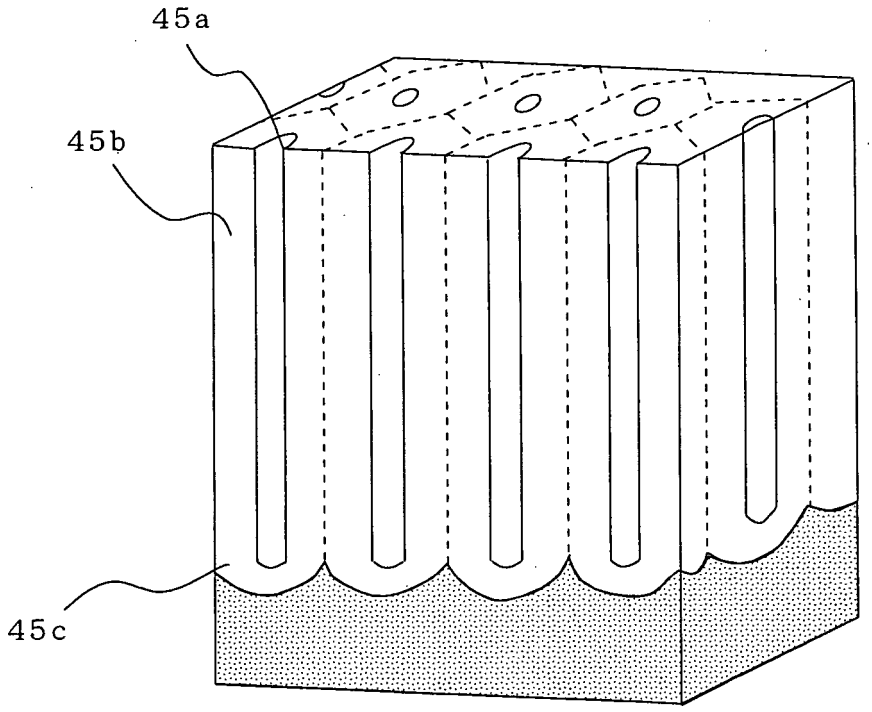


FIG. 6

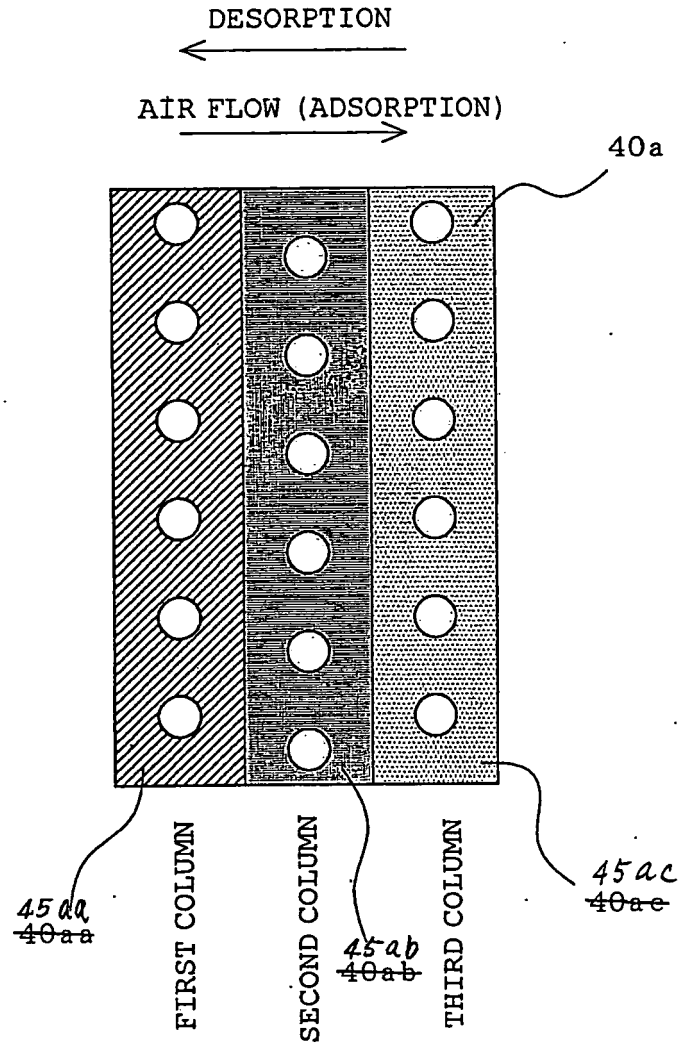

DESORPTION
AIR FLOW (ADSORPTION)
40a
45aa
40aa
45ab
40ab
45ac
40ac
FIRST COLUMN
SECOND COLUMN
THIRD COLUMN

FIG. 7a

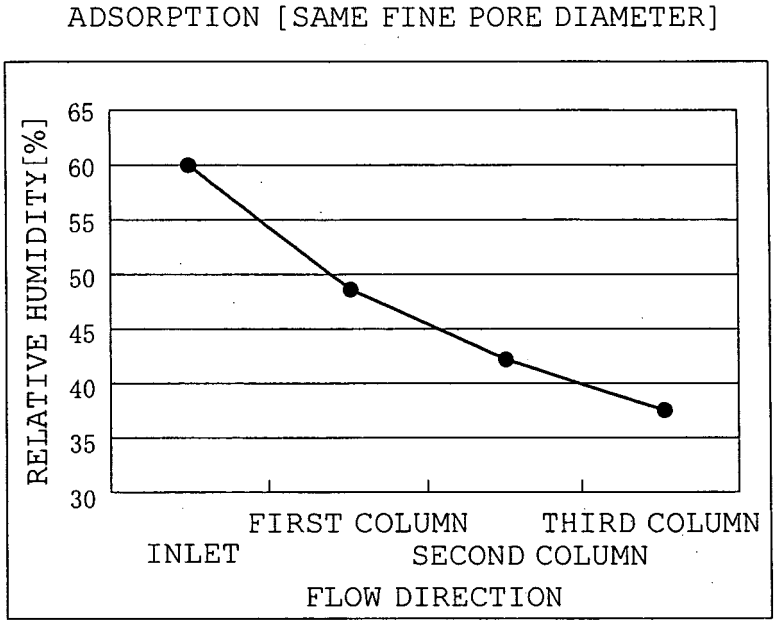

ADSORPTION [SAME FINE PORE DIAMETER]
RELATIVE HUMIDITY[%]
65
60
55
50
45
40
35
30
INLET
FIRST COLUMN
SECOND COLUMN
THIRD COLUMN
FLOW DIRECTION

FIG. 7b

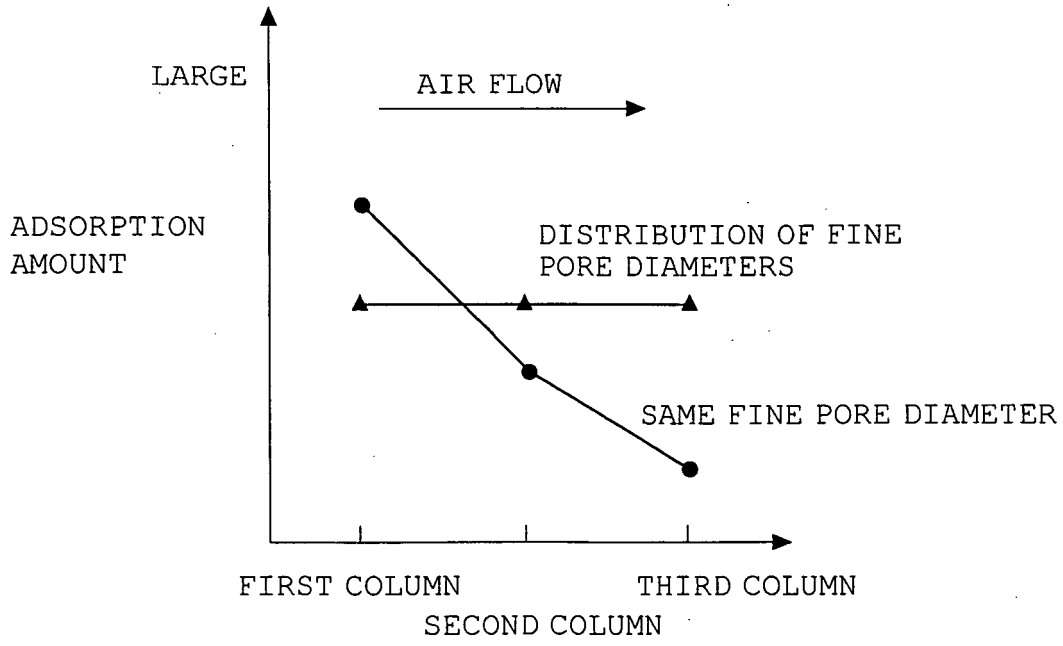

LARGE
AIR FLOW
ADSORPTION AMOUNT
DISTRIBUTION OF FINE PORE DIAMETERS
SAME FINE PORE DIAMETER
FIRST COLUMN
SECOND COLUMN
THIRD COLUMN

FIG. 8a

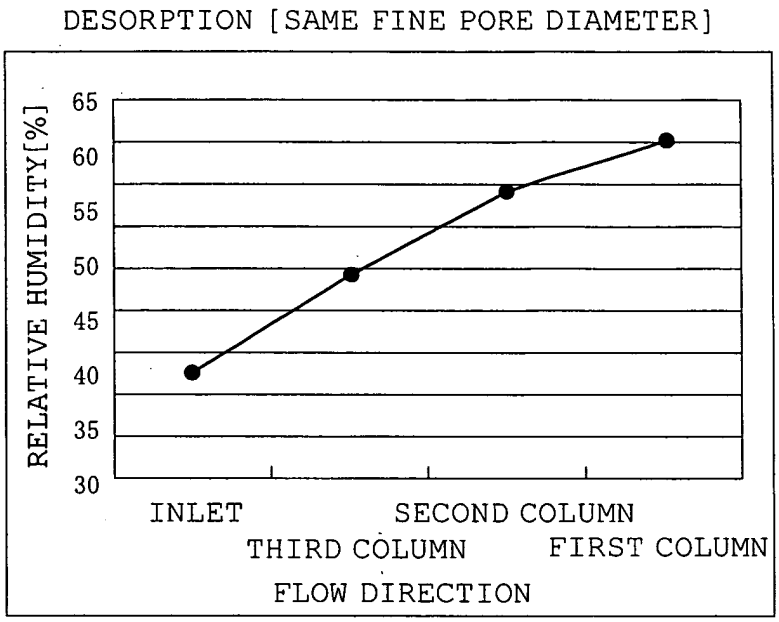

DESORPTION [SAME FINE PORE DIAMETER]
RELATIVE HUMIDITY[%]
65
60
55
50
45
40
35
30
INLET
SECOND COLUMN
THIRD COLUMN
FIRST COLUMN
FLOW DIRECTION

FIG. 8b

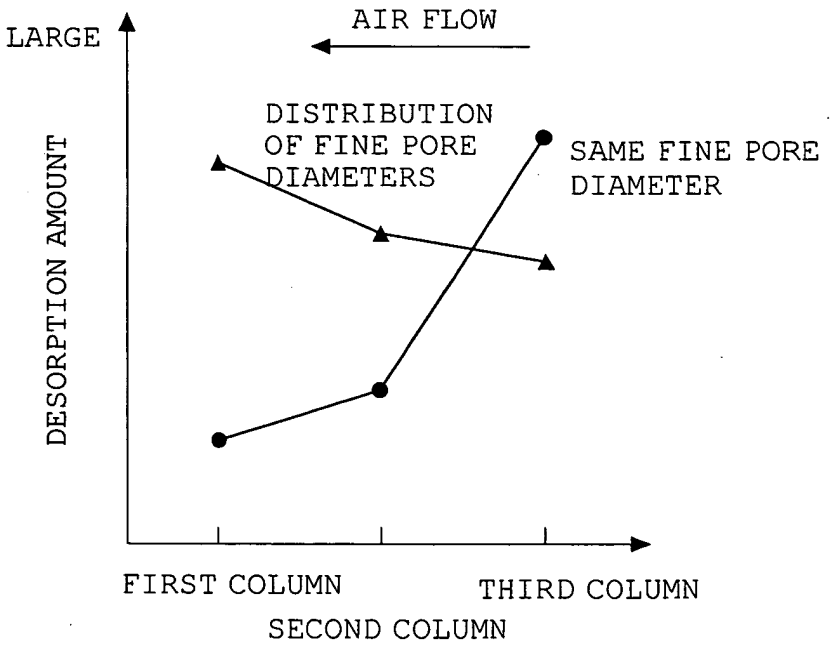

LARGE
AIR FLOW
DISTRIBUTION OF FINE PORE DIAMETERS
SAME FINE PORE DIAMETER
DESORPTION AMOUNT
FIRST COLUMN
THIRD COLUMN
SECOND COLUMN

FIG. 9a

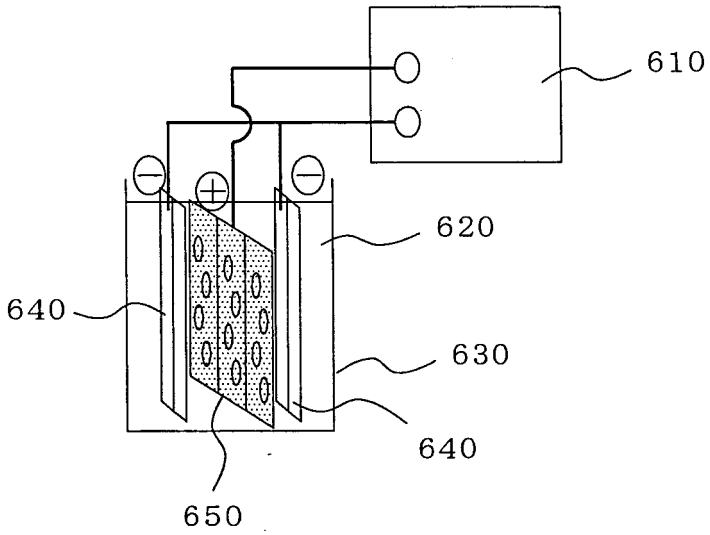


FIG. 9b

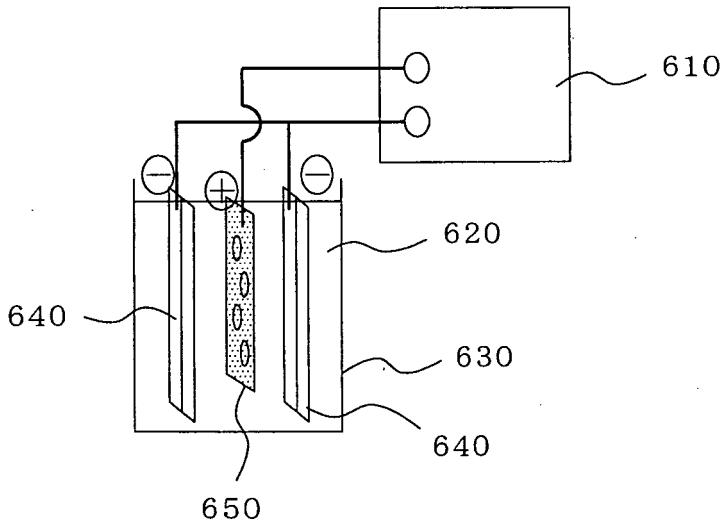


FIG. 10

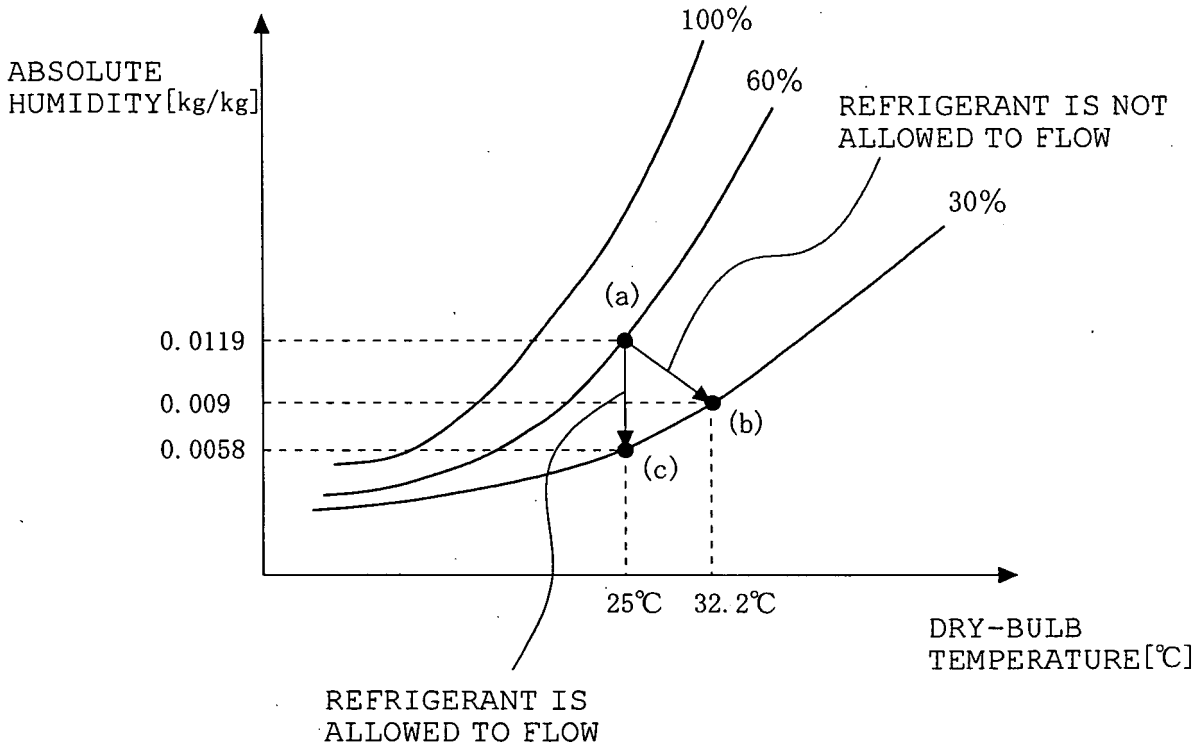

ABSOLUTE
HUMIDITY[kg/kg]
100%
60%
30%
REFRIGERANT IS NOT
ALLOWED TO FLOW
(a)
(b)
(c)
0.0119
0.009
0.0058
25℃
32.2℃
DRY-BULB
TEMPERATURE[℃]
REFRIGERANT IS
ALLOWED TO FLOW

FIG. 11

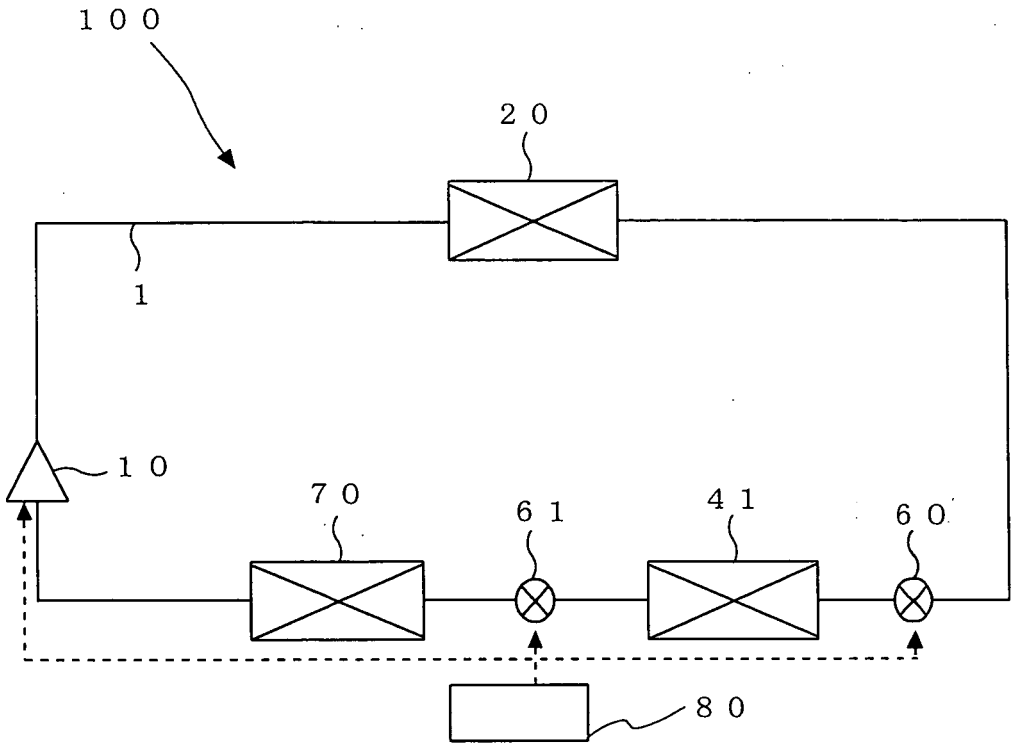

100
20
1
10
70
61
41
60
80

FIG. 12

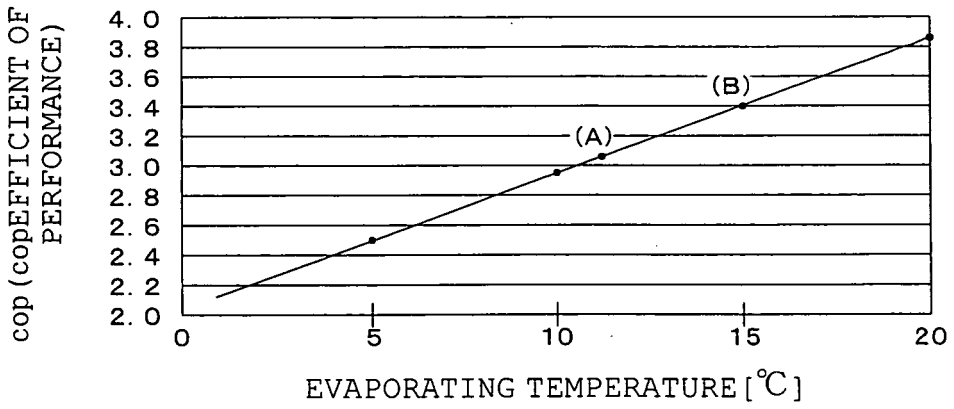

cop (copEFFICIENT OF PERFORMANCE)
4.0
3.8
3.6
3.4
3.2
3.0
2.8
2.6
2.4
2.2
2.0
(B)
(A)
0
5
10
15
20
EVAPORATING TEMPERATURE [℃]

FIG. 13

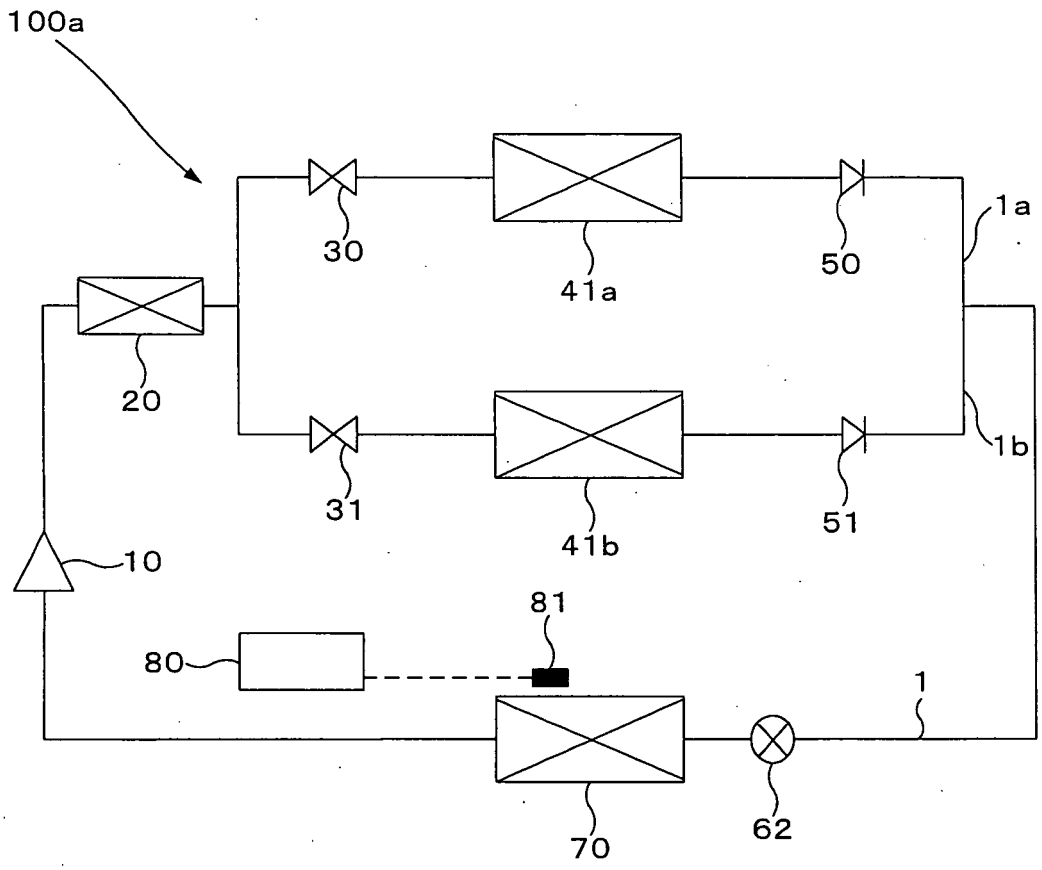


FIG. 14

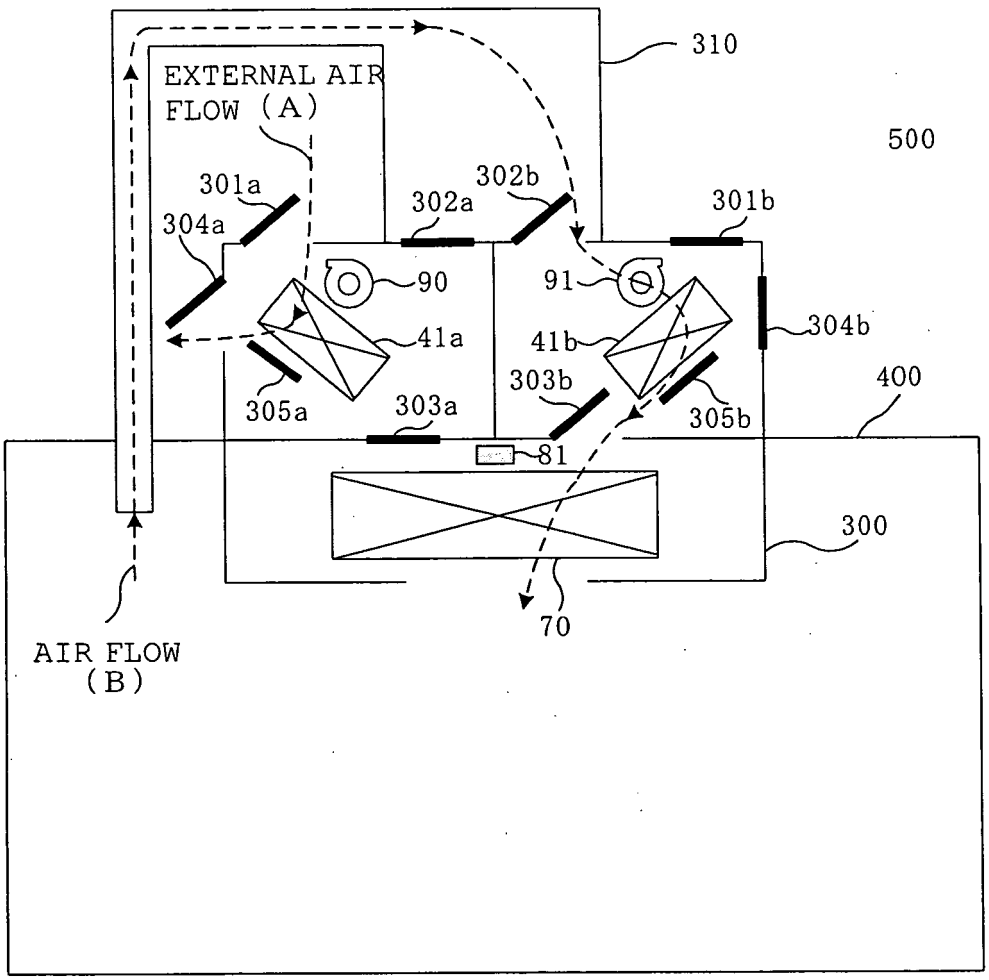
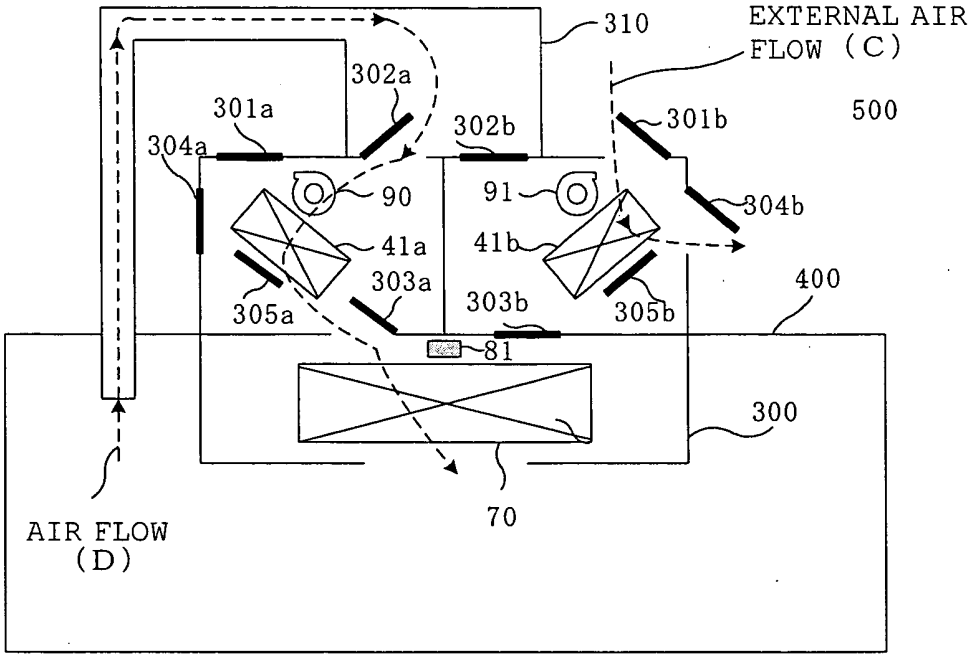

310
EXTERNAL AIR
FLOW (A)
500
301a
302b
302a
301b
304a
90
91
41a
41b
304b
305a
303a
303b
305b
400
81
300
70
AIR FLOW
(B)

FIG. 15

(a)


310
EXTERNAL AIR
FLOW (C)
500
302a
301a
302b
301b
304a
304b
90
91
41a
41b
303a
303b
305a
305b
400
81
300
70
AIR FLOW
(D)

(b)

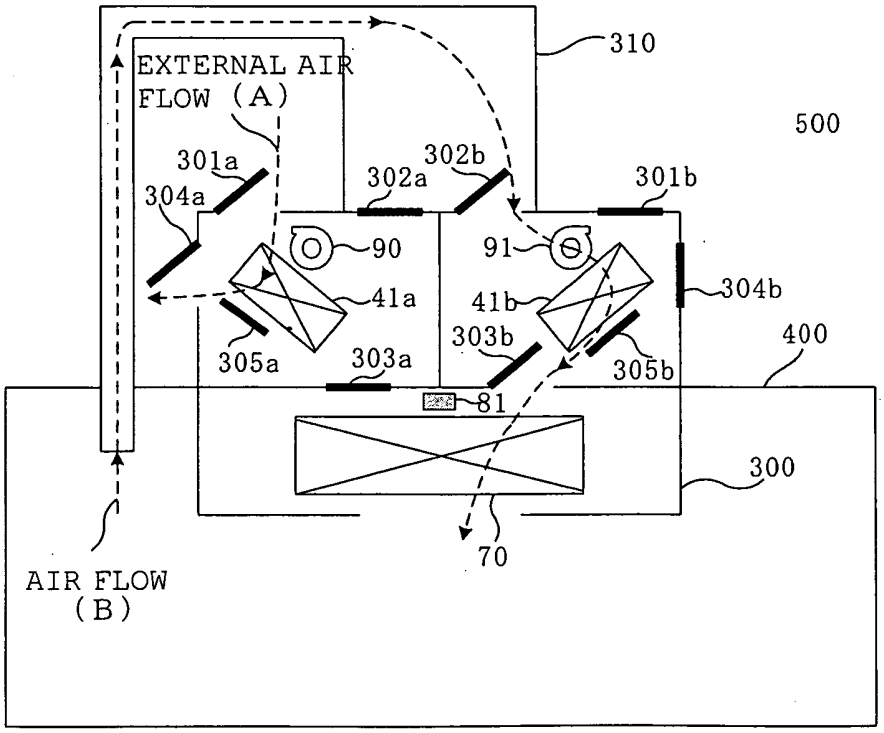

310
EXTERNAL AIR
FLOW (A)
500
301a
302b
302a
301b
304a
90
91
304b
41a
41b
303b
400
305a
303a
305b
81
300
70
AIR FLOW
(B)

FIG. 16

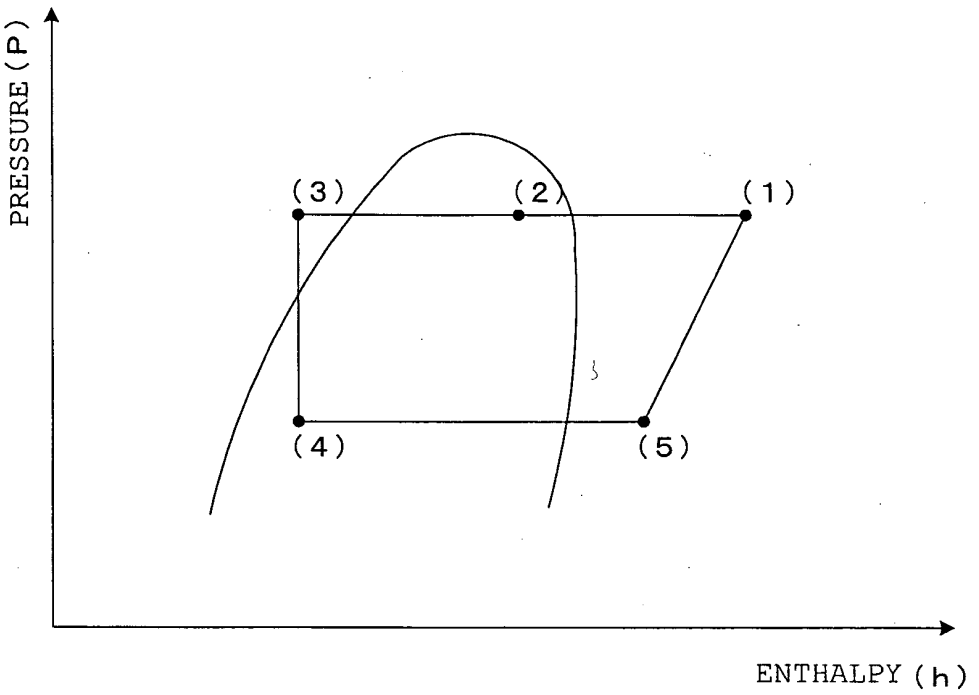

PRESSURE (P)
(3)
(2)
(1)
(4)
(5)
ENTHALPY (h)

FIG. 17

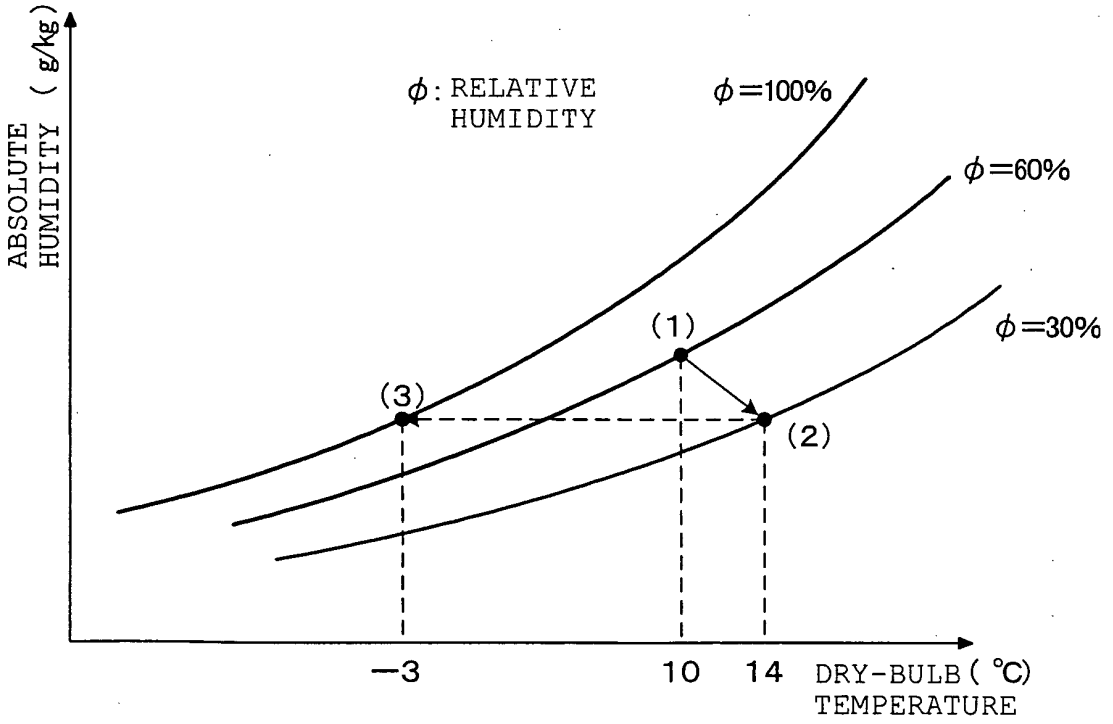

ABSOLUTE HUMIDITY (g/kg)
φ: RELATIVE HUMIDITY
φ=100%
φ=60%
φ=30%
(1)
(2)
(3)
−3
10
14
DRY-BULB (℃) TEMPERATURE

FIG. 18

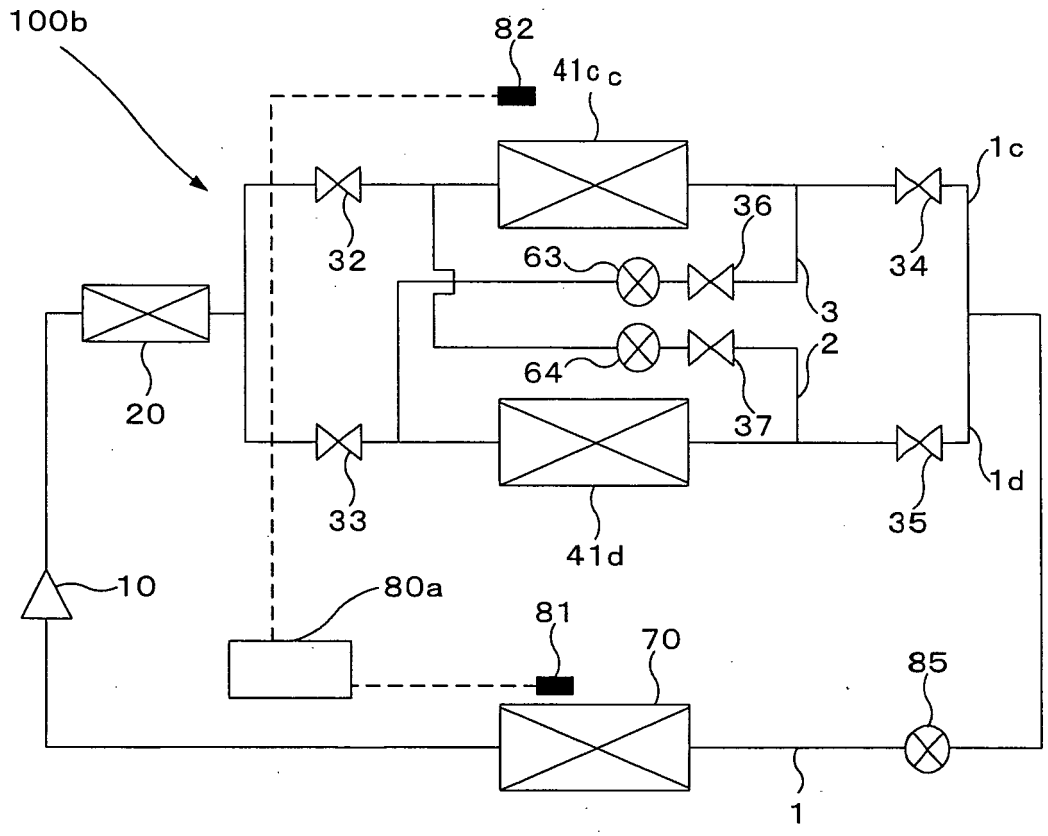


FIG. 19

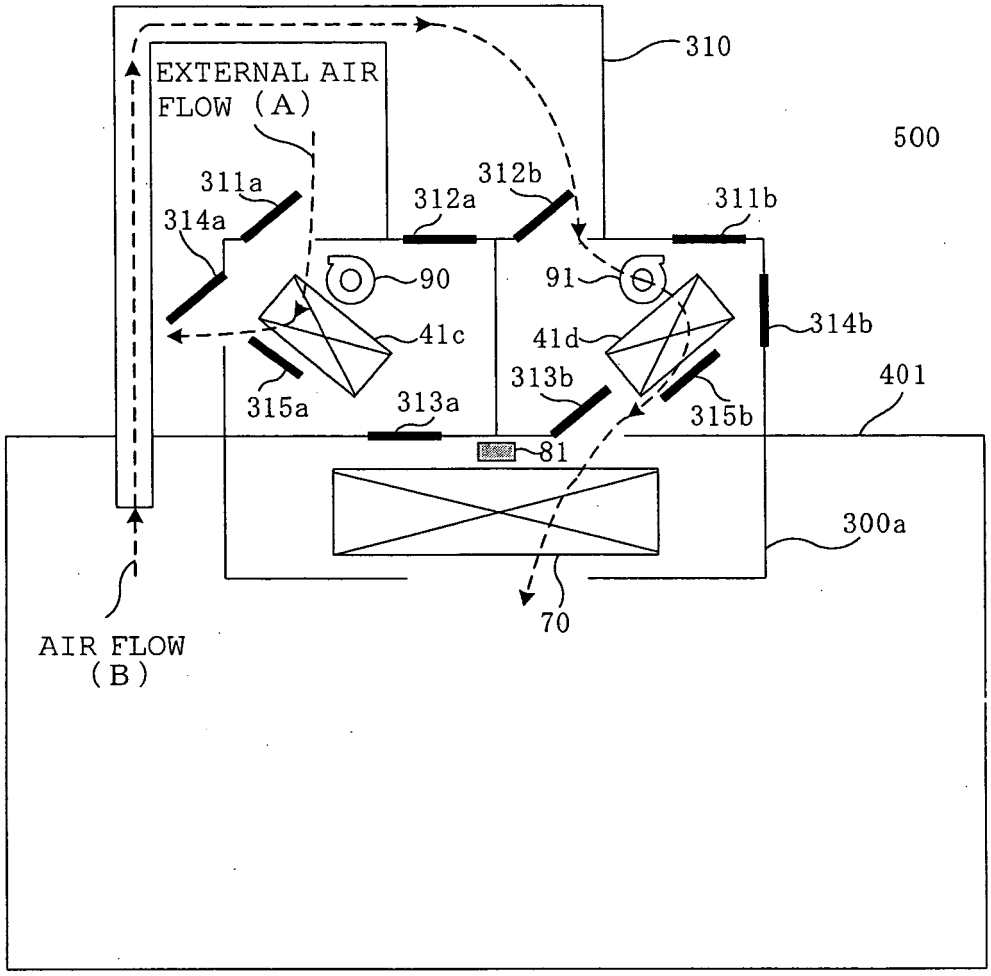
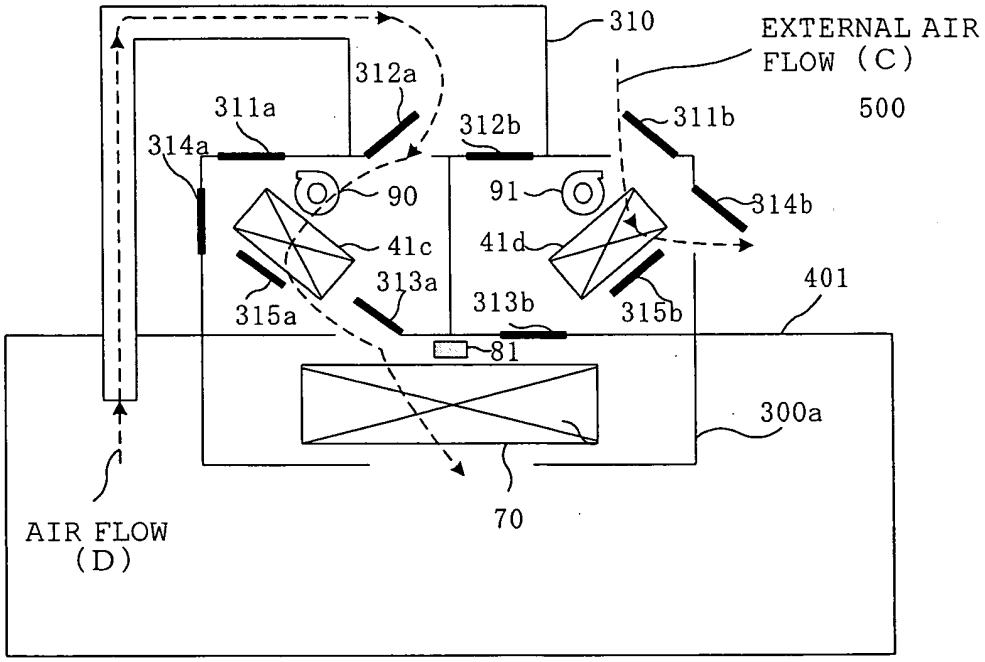

310
EXTERNAL AIR
FLOW (A)
500
311a
312b
312a
311b
314a
90
91
41c
41d
314b
315a
313a
313b
315b
401
81
300a
70
AIR FLOW
(B)

FIG. 20

(a)


310
EXTERNAL AIR
FLOW (C)
500
312a
311a
312b
311b
314a
90
91
314b
41c
41d
313a
315a
313b
315b
401
81
300a
70
AIR FLOW
(D)

(b)

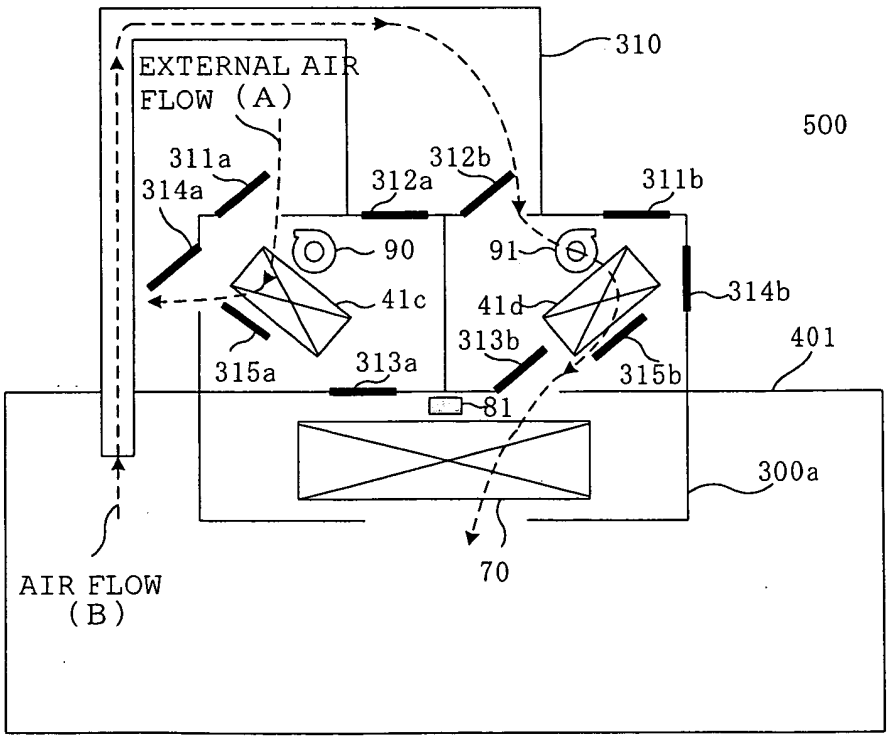

310
EXTERNAL AIR
FLOW (A)
500
311a
312b
312a
311b
314a
90
91
41c
41d
314b
313b
401
315a
313a
315b
81
300a
70
AIR FLOW
(B)

FIG. 21

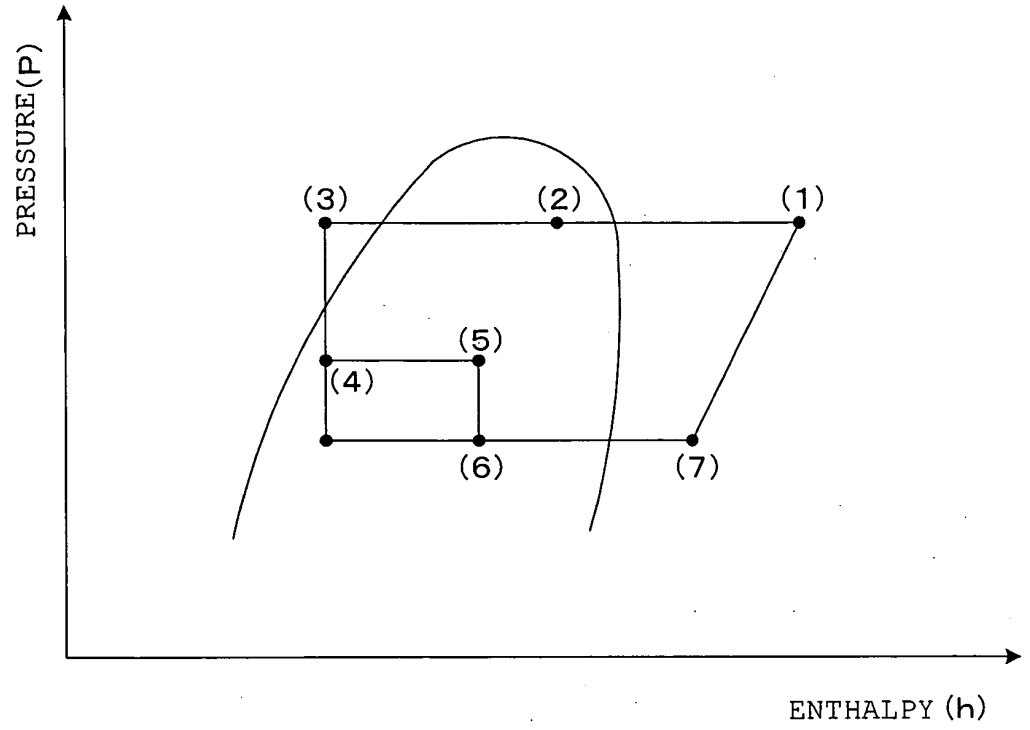

PRESSURE(P)
(3)
(2)
(1)
(5)
(4)
(6)
(7)
ENTHALPY (h)

FIG. 22

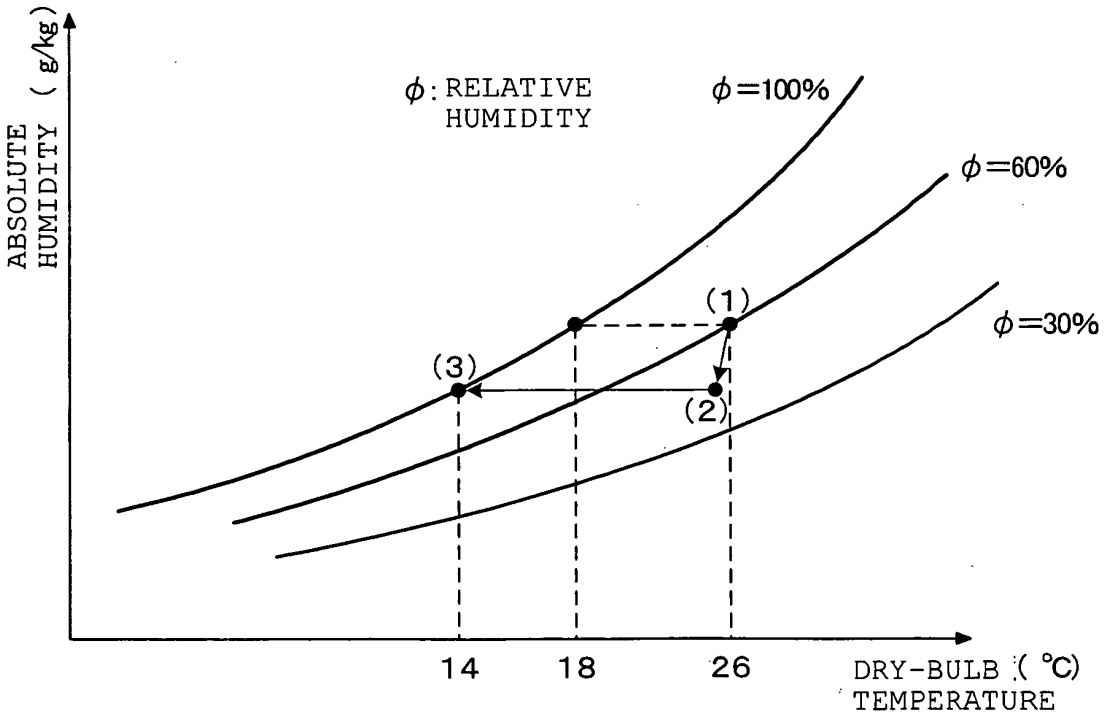
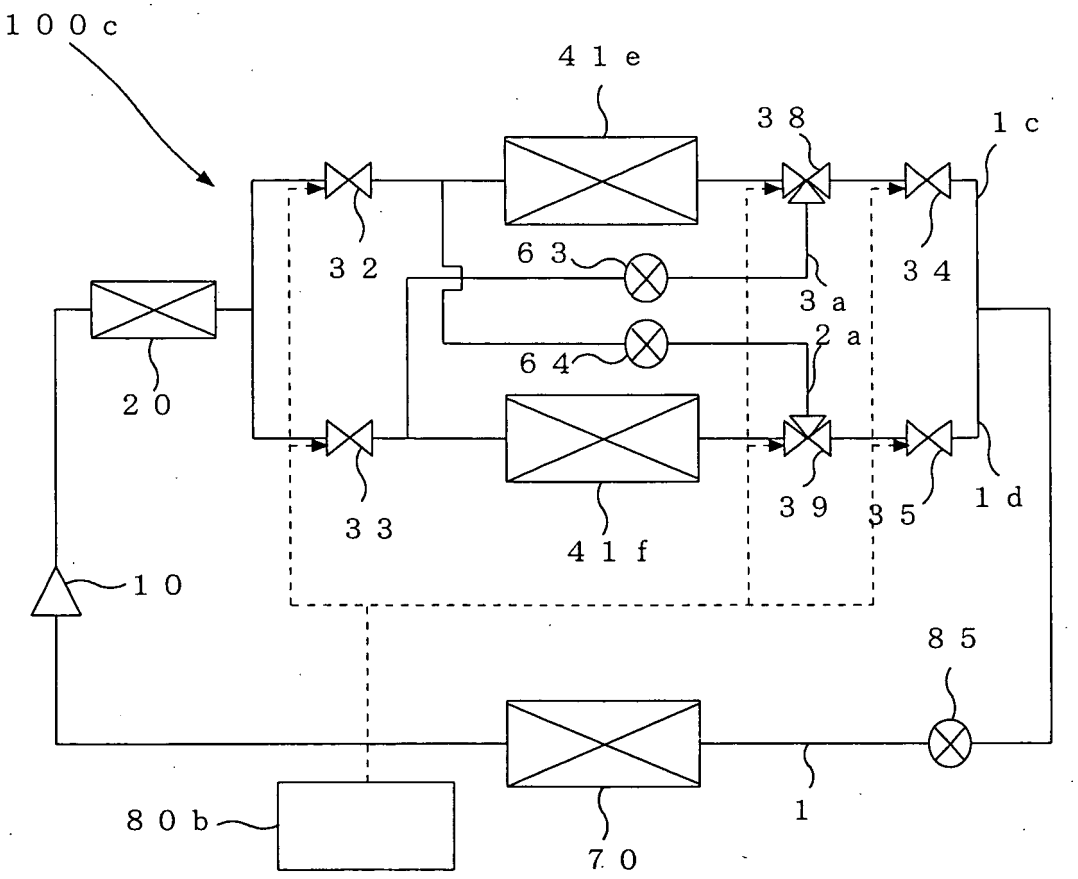

ABSOLUTE HUMIDITY (g/kg)
φ: RELATIVE HUMIDITY
φ=100%
φ=60%
φ=30%
(1)
(3)
(2)
14
18
26
DRY-BULB TEMPERATURE (℃)

FIG. 23



REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2000140625 A **[0002]**
- DE 102005007516 A1 **[0003]**
- JP 2001149735 A **[0004]**
- JP 2006349342 A **[0005]**
- JP 2004353887 A **[0011]**

Non-patent literature cited in the description

- Latest humidity control technology. *lecture meeting of Japan Society of Refrigerating and Air conditioning Engineers,* 25 May 2005, 5-6 **[0032]**