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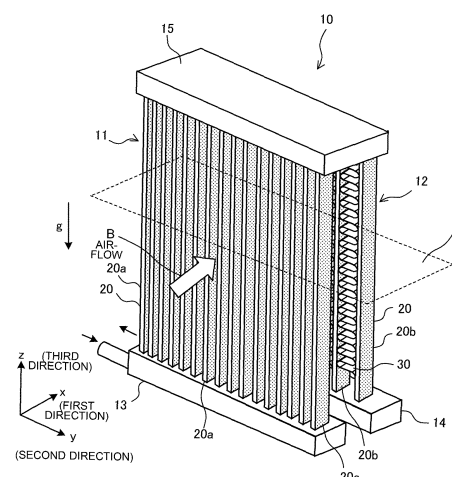
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(54) **HEAT EXCHANGER**

(57) A heat exchanger includes: a first heat exchanger configured to transfer heat between air and refrigerant; and a second heat exchanger provided in series with the first heat exchanger in a first direction that is a flow direction of the air, and configured to transfer heat between the air and the refrigerant. The first heat exchanger includes first heat transfer tubes through which the refrigerant flows, which are spaced apart from each other in a second direction intersecting the first direction, and each of which has a tube axis extending in a third direction intersecting the first and second directions. The second heat exchanger includes: second heat transfer tubes through which the refrigerant flows, which are spaced apart from each other in the second direction, and each of which has a tube axis extending in the third direction; and a corrugated fin provided between the second heat transfer tubes.

FIG. 2



Description

Technical Field

[0001] The present disclosure relates to a heat exchanger for use in a refrigeration cycle apparatus.

Background Art

[0002] As one of heat exchangers including a plurality of plate-shaped fins and elongated heat transfer tubes (hereinafter referred to as flat tubes), a vertical corrugated-fin heat exchanger is known.

[0003] In a vertical corrugated-fin heat exchanger, flat tubes are arranged in a vertical direction and joined to corrugated fins. In general, in the vertical corrugated-fin heat exchanger, in the corrugated fin, louvers (cuts) are provided in order to promote heat transfer and ensure drainage passages (for example, see Patent Literature 1).

[0004] For example, in Patent Literature 1, each of corrugated fins is a wavy plate-shaped member. The corrugated fin includes a plurality of bent portions and a plurality of fin body portions. The plate-shaped member is bent at regular intervals to have the bent portions. The fin body portions are located between adjacent ones of the bent portions. The fin body portions are provided with a plurality of louvers that are formed by cutting and raising part of the plate-shaped member.

Citation List

Patent Literature

[0005] Patent Literature 1: Japanese Unexamined Patent Application Publication No. 2019-2588

Summary of Invention

Technical Problem

[0006] As described above, an existing vertical corrugated-fin heat exchanger described in Patent Literature 1 has the following problems.

[0007] Since a large number of louvers are provided in the corrugated fin, if frost formation occurs at the corrugated fin, frost forms concentratedly at the louvers and blocks spaces between the louvers. Thus, air passages are easily blocked.

[0008] When dew condensation and frost formation occur at the corrugated fin, dew condensation water and meltwater of frost flow downward along the louvers in the vertical direction and are let out downward. However, since the fin body portions of the corrugated fin are stacked in the vertical direction such that they are spaced apart from each other, the fin body portions become obstacles in the way of the flow of water, and thus hinder the flow of the dew condensation water and meltwater of

frost, which are to be let out.

[0009] In the case where a plurality of vertical corrugated-fin heat exchangers are provided along the flow of air, frost forms concentratedly on one or ones of the heat exchangers that are located on the windward side. Consequently, air passages are blocked, and it is not possible to effectively use one or ones of the heat exchangers that are located on the leeward side.

[0010] The present disclosure is applied to solve the above problems, and relates to a heat exchanger capable of reducing occurrence of blockage of air passages that is caused by frost formation and improving a drainage performance for dew condensation water and meltwater of frost.

Solution to Problem

[0011] A heat exchanger according to an embodiment of the present disclosure includes a first heat exchanger configured to cause heat exchange to be performed between air and refrigerant, and a second heat exchanger provided in series with the first heat exchanger in a first direction that is a flow direction of the air and configured to cause heat exchange to be performed between the air and the refrigerant. The first heat exchanger includes a plurality of first heat transfer tubes through which the refrigerant flows. The plurality of first heat transfer tubes are spaced apart from each other in a second direction that intersects the first direction, and each have a tube axis extending in a third direction that intersects each of the first direction and the second direction. The second heat exchanger includes a plurality of second heat transfer tubes through which the refrigerant flows, and a corrugated fin provided between the plurality of second heat transfer tubes. The plurality of second heat transfer tubes are spaced apart from each other in the second direction, and each have a tube axis extending in the third direction.

Advantageous Effects of Invention

[0012] In a heat exchanger according to an embodiment of the present disclosure, a first heat exchanger and a second heat exchanger are provided in series with each other in a first direction. In the first heat exchanger, no fin is provided, and the ratio of the amount of latent heat exchange to the amount of total heat exchange is high. In the second heat exchanger, a corrugated fin is provided, and the ratio of the amount of sensible heat exchange to the amount of total heat exchange is high. Because of the above configuration, in the entire heat exchanger, it is possible to reduce occurrence of blockage of air passages which is caused by frost formation and to improve the performance in drainage of dew condensation water and meltwater of frost.

Brief Description of Drawings

[0013]

[Fig. 1] Fig. 1 is a circuit diagram illustrating an example of a refrigerant circuit that is formed in a refrigeration cycle apparatus 1 on which a heat exchanger 10 according to Embodiment 1 is mounted.

[Fig. 2] Fig. 2 is a perspective view illustrating an example of the configuration of the heat exchanger 10 as illustrated in Fig. 1.

[Fig. 3] Fig. 3 is a sectional view illustrating a cross-section of the heat exchanger 10 as illustrated in Fig. 2 that is taken along a cross-section A, which is an imaginary plane.

[Fig. 4] Fig. 4 is a sectional side-view illustrating an example of the configuration of a first heat exchanger 11 of the heat exchanger 10 as illustrated in Fig. 2.

[Fig. 5] Fig. 5 is a sectional side-view illustrating an example of the configuration of a second heat exchanger 12 of the heat exchanger 10 as illustrated in Fig. 2.

[Fig. 6] Fig. 6 is a partially enlarged perspective view schematically illustrating an example of the configuration of the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 1.

[Fig. 7] Fig. 7 is an explanatory view illustrating a state in which frost formation occurs at the heat exchanger 10 as illustrated in Fig. 3.

[Fig. 8] Fig. 8 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 2 that is taken along the cross-section A as illustrated in Fig. 2.

[Fig. 9] Fig. 9 is an explanatory view illustrating a state in which frost formation occurs at the heat exchanger 10 illustrated in Fig. 8.

[Fig. 10] Fig. 10 is an explanatory view illustrating a state in which frost formation has occurred in the heat exchanger 10 as illustrated in Fig. 8.

[Fig. 11] Fig. 11 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 3 that is taken along the cross-section A as illustrated in Fig. 2.

[Fig. 12] Fig. 12 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 5 that is taken along the cross-section A as illustrated in Fig. 2.

[Fig. 13] Fig. 13 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 6 that is taken along the cross-section A as illustrated in Fig. 2.

[Fig. 14] Fig. 14 is a plan view illustrating a configuration of an inter-row connection portion 15 provided in the heat exchanger 10 according to Embodiment 6.

[Fig. 15] Fig. 15 is a partial side view illustrating a configuration of the heat exchanger 10 according to Embodiment 6.

[Fig. 16] Fig. 16 is a sectional view illustrating a cross-section of the heat exchanger 10 according to a modification of Embodiment 6 that is taken along the cross-section A illustrated in Fig. 2.

[Fig. 17] Fig. 17 is an explanatory view for occurrence of dryout of the heat exchanger 10 according to Embodiment 8.

[Fig. 18] Fig. 18 is a perspective view illustrating a modification of Embodiment 8.

[Fig. 19] Fig. 19 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 9 that is taken along the cross-section A as illustrated in Fig. 2.

[Fig. 20] Fig. 20 is a plan view illustrating a configuration of the inter-row connection portion 15 of the heat exchanger 10 according to Embodiment 10.

[Fig. 21] Fig. 21 is a partial side view illustrating a configuration of the heat exchanger 10 according to Embodiment 10.

[Fig. 22] Fig. 22 is an explanatory view illustrating a configuration of a modification of heat transfer tubes 20 for use in at least one of the first heat exchanger 11 and the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 11.

[Fig. 23] Fig. 23 is an explanatory view illustrating a configuration of another modification of the heat transfer tubes 20 for use in at least one of the first heat exchanger 11 and the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 11.

[Fig. 24] Fig. 24 is an explanatory view illustrating a configuration of still another modification of the heat transfer tubes 20 for use in at least one of the first heat exchanger 11 and the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 11.

[Fig. 25] Fig. 25 is an explanatory view illustrating a configuration of a further modification of the heat transfer tubes 20 for use in at least one of the first heat exchanger 11 and the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 11.

[Fig. 26] Fig. 26 is a sectional side-view illustrating an example of the configuration of the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 12.

Description of Embodiments

[0014] Each of heat exchangers according to embodiments of the present disclosure will be described with reference to the drawings. The descriptions concerning the embodiments of the present disclosure are not limiting, and various modifications can be made without departing from the gist of the present disclosure. In addition, the present disclosure encompasses all combinations of all combinable ones of the configurations of the embodiments and modifications as described below. In each of figures to be referred to below, components that are the same as or equivalent to those in a previous figure or previous figures are denoted by the same reference signs. The same is true of the entire text of the specifi-

cation. In each of the figures, relative relationships in, for example, dimension or shape between components may differ from actual ones.

Embodiment 1

Configuration of Refrigeration Cycle Apparatus 1

[0015] Fig. 1 is a circuit diagram illustrating an example of a refrigerant circuit that is formed in a refrigeration cycle apparatus 1 in which a heat exchanger 10 according to Embodiment 1 is mounted. With reference to Fig. 1, the refrigeration cycle apparatus 1 according to Embodiment 1 will be described. As illustrated in Fig. 1, the refrigeration cycle apparatus 1 includes a compressor 2, an indoor heat exchanger 3, an indoor fan 4, an expansion device 5, the heat exchanger 10, an outdoor fan 6, and a four-way valve 7. The heat exchanger 10 may be referred to as an outdoor heat exchanger. In the case where the refrigeration cycle apparatus 1 includes an outdoor unit and an indoor unit, for example, the compressor 2, the heat exchanger 10, the expansion device 5, and the four-way valve 7 are provided in the outdoor unit, and the indoor heat exchanger 3 is provided in the indoor unit.

[0016] The compressor 2, the indoor heat exchanger 3, the expansion device 5, the heat exchanger 10, and the four-way valve 7 form a refrigerant circuit in which refrigerant can be circulated. In the refrigeration cycle apparatus 1, a refrigeration cycle in which refrigerant circulates while changing in phase in the refrigerant circuit is carried out. Hereinafter, the above components which form the refrigeration cycle apparatus 1 as illustrated in Fig. 1 will be described.

[0017] The compressor 2 has a suction port and a discharge port. The compressor 2 compresses refrigerant sucked through the suction port and discharges the compressed refrigerant through the discharge port. The compressor 2 is, for example, a rotary compressor, a scroll compressor, a screw compressor, a reciprocating compressor, or other types of compressors. The compressor 2 may be an inverter compressor. In this case, the compressor 2 may change the amount of refrigerant that is to be discharged from the compressor 2 per unit time, by arbitrarily changing, using, for example, an inverter circuit, an operation frequency of a motor that drives a compression mechanism of the compressor 2. In the case where the compressor 2 is an inverter compressor, the inverter circuit is controlled by a controller (not illustrated). The controller includes a memory and a processing circuit, such as a processor or a dedicated hardware. In the case where the processing circuit is a processor, the processor reads out and executes a program stored in the memory, thereby fulfilling an associated one of functions of the controller.

[0018] The indoor heat exchanger 3 operates as a condenser when the refrigeration cycle apparatus 1 performs a heating operation, and operates as an evaporator when the refrigeration cycle apparatus 1 performs a cooling

operation. The indoor heat exchanger 3 causes heat exchange to be performed between indoor air supplied by the indoor fan 4 and refrigerant that flows in the indoor heat exchanger 3. The indoor heat exchanger 3 is, for example, a fin-and-tube heat exchanger, a microchannel heat exchanger, a finless heat exchanger, a shell-and-tube heat exchanger, a heat pipe heat exchanger, a double-tube heat exchanger, a plate heat exchanger, or other types of heat exchangers.

[0019] The indoor fan 4 is provided for the indoor heat exchanger 3 and supplies indoor air as a heat exchange fluid to the indoor heat exchanger 3. For example, the indoor fan 4 is located to face the indoor heat exchanger 3. The rotation speed of the indoor fan 4 is controlled by a controller (not illustrated).

[0020] The expansion device 5 is a pressure reducing device that expands the refrigerant to reduce the pressure of the refrigerant. The expansion device 5 is, for example, an expansion valve. The expansion device 5 may be, for example, an electronic expansion valve that is capable of adjusting the flow rate of the refrigerant. In this case, the expansion device 5 is controlled by a controller (not illustrated). Alternatively, the expansion device 5 may be a mechanical expansion valve in which a diaphragm is employed as a pressure receiver, or may be a capillary tube, for example.

[0021] The heat exchanger 10 operates as an evaporator when the refrigeration cycle apparatus 1 performs the heating operation and operates as a condenser when the refrigeration cycle apparatus 1 performs the cooling operation. The heat exchanger 10 causes heat exchange to be performed between outdoor air supplied by the outdoor fan 6 and refrigerant that flows in the heat exchanger 10. The heat exchanger 10 includes two heat exchangers. One of these heat exchangers will be referred to as a first heat exchanger 11 (see Fig. 2) and the other will be referred to as a second heat exchanger 12 (see Fig. 2). The first heat exchanger 11 is, for example, a finless heat exchanger in which no fin is provided between adjacent ones of heat transfer tubes. The second heat exchanger 12 is a fin-and-tube type heat exchanger which includes heat transfer tubes and a fin. The heat exchanger 10 will be described in detail later. It should be noted that in the following description, all of heat exchangers in each of which no heat transfer member is not provided, for example, no fin is provided, between a plurality of heat transfer tubes are each referred to as "finless heat exchanger".

[0022] The outdoor fan 6 is provided for the heat exchanger 10 and supplies outdoor air to the heat exchanger 10. For example, the outdoor fan 6 is located to face the heat exchanger 10. The rotation speed of the outdoor fan 6 is controlled by a controller (not illustrated).

[0023] The four-way valve 7 is a flow switching device that switches a passage for refrigerant in the refrigeration cycle apparatus 1. In Fig. 1, solid lines in the four-way valve 7 indicate the state of the four-way valve 7 in the heating operation, and dashed lines in the four-way valve

7 indicate the state of the four-way valve 7 in the cooling operation and during a defrosting operation. The state of the four-way valve 7 is switched by a controller (not illustrated). When the refrigeration cycle apparatus 1 performs the heating operation, the state of the four-way valve 7 is switched to the state indicated by the solid lines in Fig. 1 such that the discharge port of the compressor 2 and the indoor heat exchanger 3 are connected to each other and the suction port of the compressor 2 and the heat exchanger 10 are connected to each other. When the refrigeration cycle apparatus 1 performs the cooling operation or the defrosting operation, the state of the four-way valve 7 is switched to the state indicated by the dashed lines in Fig. 1 such that the discharge port of the compressor 2 and the heat exchanger 10 are connected to each other and the suction port of the compressor 2 and the indoor heat exchanger 3 are connected to each other.

Configuration of Heat Exchanger 10

[0024] Next, with reference to Figs. 2 to 5, a configuration of the heat exchanger 10 will be described. Fig. 2 is a perspective view illustrating an example of the configuration of the heat exchanger 10 as illustrated in Fig. 1. Fig. 3 is a sectional view illustrating a cross-section of the heat exchanger 10 as illustrated in Fig. 2 that is taken along a cross section A which is an imaginary plane. Fig. 4 is a sectional side-view illustrating an example of the configuration of the first heat exchanger 11 of the heat exchanger 10 as illustrated in Fig. 2. Fig. 5 is a sectional side-view illustrating an example of the configuration of the second heat exchanger 12 of the heat exchanger 10 as illustrated in Fig. 2.

[0025] It should be noted that the x-direction, the y-direction, and the z-direction are defined as follows as a matter of convenience for explanation. As illustrated in Figs. 2 to 5, the heat exchanger 10 includes a plurality of heat transfer tubes 20 through which the refrigerant flows. The plurality of heat transfer tubes 20 include first heat transfer tubes 20a and second heat transfer tubes 20b. The cross-sectional shape of each of the heat transfer tubes 20, which is taken along the cross section A as illustrated in Fig. 2, is a rectangular or substantially rectangular shape that has a long side in a longitudinal direction and a short side in a width direction, as illustrated in Fig. 3. The longitudinal direction of the cross-sectional shape of the heat transfer tube 20 is the x-direction (first direction). The plurality of heat transfer tubes 20 are arranged side by side and spaced apart from each other. The direction in which the heat transfer tubes 20 are arranged is the y-direction (second direction). The width direction of the cross-sectional shape of the heat transfer tube 20 is the y-direction. The tube axial direction of the heat transfer tube 20, that is, the flow direction of the refrigerant, is the z-direction (third direction). The x-direction, the y-direction, and the z-direction intersect with each other. The flow direction of air which is a heat ex-

change fluid and supplied from the outdoor fan 6 as illustrated in Fig. 1 will be referred to as "airflow direction" as indicated by an arrow B in Fig. 2. It should be noted that the heat exchanger 10 is provided such that the x-direction coincides with the airflow direction and the z-direction coincides with the vertical direction. The vertical direction is the direction of gravity as indicated by an arrow g in Fig. 2. The cross-section A, which is an imaginary plane in Fig. 2, is an xy-plane. The x-direction, the y-direction, and the z-direction are used in common in the figures as defined above.

[0026] As illustrated in Figs. 2 and 3, the heat exchanger 10 is, for example, a heat exchanger having a two-row structure, and includes the first heat exchanger 11 and the second heat exchanger 12. In the x-direction which is the airflow direction, the first heat exchanger 11 is located on the windward side and the second heat exchanger 12 located on the leeward side. The first heat exchanger 11 is formed as a finless heat exchanger in which no fin is provided and a plurality of the first heat transfer tubes 20a are arranged independently of each other. The second heat exchanger 12 is formed as a fin-and-tube heat exchanger including a plurality of the second heat transfer tubes 20b and a corrugated fin 30 or corrugated fins 30. To be more specific, the second heat exchanger 12 is a vertical corrugated-fin heat exchanger including the second heat transfer tubes 20b and the corrugated fin 30.

[0027] Fig. 4 is a sectional side-view of the first heat exchanger 11 that is taken along a yz-plane. Fig. 4 illustrates a section as viewed in the direction indicated by an arrow B in Fig. 3. As illustrated in Figs. 2 to 4, the first heat exchanger 11 includes the plurality of first heat transfer tubes 20a. The tube axial of each of the first heat transfer tubes 20a extends in the z-direction. The first heat transfer tubes 20a are arranged side by side and spaced apart from each other in the y-direction. Since the first heat exchanger 11 is the finless heat exchanger, no heat transfer member is provided, for example, no fin is provided, between the first heat transfer tubes 20a. In a heat exchanger provided with a fin, first heat transfer tubes 20a are coupled to each other by the fin. By contrast, since the first heat exchanger 11 includes no fin, the first heat transfer tubes 20a are not coupled to each other and are thus provided independently of each other. One end (lower end) of each of the first heat transfer tubes 20a is connected to a first header portion 13. The other end (upper end) of the first heat transfer tube 20a is connected to an inter-row connection portion 15. The first header portion 13 is capable of distributing refrigerant supplied from the outside to the first heat transfer tubes 20a of the first heat exchanger 11. The refrigerants distributed by the first header portion 13 pass through the first heat transfer tubes 20a and flow into the inter-row connection portion 15. The inter-row connection portion 15 is capable of causing the refrigerants that have flowed into the inter-row connection portion 15 to join each other, and then distributing the resultant refrigerant

to the second heat transfer tubes 20b of the second heat exchanger 12. The first header portion 13 is capable of causing, in the case where refrigerants flow in the opposite directions, refrigerants that has flowed from the first heat transfer tubes 20a into the first header portion 13 to join each other, and then be let out to the outside of the first header portion 13.

[0028] Fig. 5 is a sectional side-view of the second heat exchanger 12 that is taken along a yz-plane. Fig. 5 illustrates a section as viewed in the direction indicated by the arrow B in Fig. 3. As illustrated in Figs. 2, 3, and 5, the second heat exchanger 12 includes the plurality of second heat transfer tubes 20b and the corrugated fin 30. The tube axis of each of the second heat transfer tubes 20b extends in the z-direction. The second heat transfer tubes 20b are arranged side by side and spaced apart from each other in the y-direction. The corrugated fin 30 is located between the second heat transfer tubes 20b. One end (lower end) of each of the second heat transfer tubes 20b is connected to a second header portion 14. The other end (upper end) of each of the second heat transfer tubes 20b is connected to the inter-row connection portion 15. The second header portion 14 is capable of causing refrigerants flowing thereinto from the second heat transfer tubes 20b to join each other and letting out the resultant refrigerant to the outside of the second header portion 14. The second header portion 14 is capable of distributing, when refrigerants flow in the opposite directions, the refrigerant supplied from the outside to the second heat transfer tubes 20b.

[0029] Therefore, the heat exchanger 10 has refrigerant passages in which the first header portion 13, the first heat transfer tubes 20a of the first heat exchanger 11, the inter-row connection portion 15, the second heat transfer tubes 20b of the second heat exchanger 12, and the second header portion 14 are connected in this order.

[0030] As illustrated in Fig. 5, the corrugated fin 30 of the second heat exchanger 12 is located between two adjacent second heat transfer tubes 20b. The corrugated fin 30 and the second heat transfer tubes 20b are joined to each other by brazing, for example. As illustrated in Figs. 2 to 5, the corrugated fin 30 is formed such that a wavy plate-shaped member 300 is bent at regular intervals. The plate-shaped member 300 is an elongated member having a constant length L1 (see Fig. 3) in the x-direction and a constant thickness T (see Fig. 5). As illustrated in Fig. 5, the corrugated fin 30 has fin body portions 302 (see Fig. 6) and bent portions 301 (see Fig. 6). The fin body portions 302 are spaced apart from each other in the z-direction, and each of the bent portions 301 (see Fig. 6) connects associated adjacent two of the fin body portions 302 to each other. The fin body portions 302 extend in the y-direction. The bent portions 301 extend in the z-direction. The fin body portions 302 include louvers 31. The louvers 31 are formed by cutting and raising part of the plate-shaped member 300 that forms the fin body portions 302. The louvers 31 will be described later with reference to Fig. 6.

[0031] It should be noted that although Fig. 2 illustrates the heat exchanger 10 having a two-row structure, this is not limiting. The heat exchanger 10 may have at least a three-row structure. In this case, the number of heat exchangers in the structure is set depending on a heat exchange load to be applied to the heat exchanger 10. In the case where the heat exchanger 10 has a three-row structure, a finless heat exchanger, another finless heat exchanger, and a fin-and-tube heat exchanger are provided in this order from the windward side in the heat exchanger 10. In the case where the heat exchanger 10 has three or more heat exchangers in rows, one or more finless heat exchangers are provided on the windward side, and one or more fin-and-tube heat exchangers are provided on the leeward side.

[0032] As illustrated in Figs. 3 and 4, the first heat exchanger 11 includes the plurality of first heat transfer tubes 20a. As illustrated in Fig. 3, the first heat transfer tubes 20a are arranged at a predetermined first pitch PP1 in the y-direction. Thus, the first pitch PP1 may be referred to as the pitch of the first heat transfer tubes 20a. Also, as illustrated in Fig. 3, the first heat transfer tubes 20a are spaced apart from each other by a predetermined first distance PC1 in the y-direction. A space having the first distance PC1 serves as an air passage in which an outdoor air flows, the outdoor air being a heat exchange fluid which is supplied by the outdoor fan 6. The first distance PC1 is equal to the distance between any adjacent two of the first heat transfer tubes 20a. The first distance PC1 has a value that is obtained by subtracting from the first pitch PP1, the width W1 of each of the first heat transfer tubes 20a in the y-direction. Thus, the width of each of the air passages of the first heat exchanger 11 in the y-direction is referred to as the first distance PC1.

[0033] As illustrated in Figs. 3 and 5, the second heat exchanger 12 includes the plurality of second heat transfer tubes 20b. As illustrated in Fig. 3, the second heat transfer tubes 20b are arranged at a predetermined second pitch PP2 in the y-direction. Thus, the second pitch PP2 may be referred to as the pitch of the second heat transfer tubes 20b. As illustrated in Fig. 3, the second heat transfer tubes 20b are spaced apart from each other by a predetermined second distance PC2 in the y-direction. In a space having the first distance PC2 between the second heat transfer tubes 20b, the corrugated fin 30 is provided, and the space serves as an air passage in which an outdoor air flows, the outdoor air being a heat exchange fluid which is supplied from the outdoor fan 6. The second distance PC2 is equal to the distance between any adjacent two of the second heat transfer tubes 20b. The second distance PC2 has a value that is obtained by subtracting from the second pitch PP2, the width W2 of each of the second heat transfer tubes 20b in the y-direction. Thus, the width of the air passage or each of the air passages of the second heat exchanger 12 in the y-direction is referred to as the second distance PC2.

[0034] As the material of the plate-shaped member 300

forming the corrugated fin 30, for example, aluminum, an aluminum alloy, copper, or a copper alloy is used. As described above, the corrugated fin 30 includes the fin body portions 302 which extend in the y-direction. As illustrated in Fig. 5, the fin body portions 302 are arranged at a predetermined third pitch FP2 in the z-direction. Thus, the third pitch FP2 may be referred to as the pitch of the fin body portions 302. Also, as illustrated in Fig. 5, the fin body portions 302 are spaced apart from each other by a predetermined third distance FC2 in the z-direction. A space having the third distance FC2 between any adjacent two of the fin body portions 302 serves as an air passage in which the outdoor air flows, the outdoor air being a heat exchange fluid which is supplied by the outdoor fan 6. The third distance FC2 is equal to the distance between any adjacent two of the fin body portions 302. The third distance FC2 has a value that is obtained by subtracting from the third pitch FP2, the thickness T of each of the fin body portions 302 in the z-direction. Thus, air passages each having a size of $FC2 \times PC2$ are formed in the corrugated fin 30, where FC2 is the third distance in the vertical direction, and PC2 is the second distance in the width direction.

[0035] As illustrated in Fig. 3, the first heat transfer tubes 20a and the second heat transfer tubes 20b are, for example, flat tubes. Therefore, as described above, a cross-sectional shape of each of the first heat transfer tubes 20a and the second heat transfer tubes 20b, which is taken along the cross-section A as illustrated in Fig. 2, is a rectangular or substantially rectangular shape having long sides and short sides. The x-direction in which the long sides extend will also be referred to as the longitudinal direction of the cross-section of the heat transfer tube 20, and the y-direction in which the short sides extend will also be referred to as the width direction of the cross-section of the heat transfer tube 20. In order to reduce a ventilation resistance, the heat transfer tubes 20 are oriented such that the longitudinal direction of the cross-section of each of the heat transfer tubes 20 coincides with the airflow direction indicated by the arrow B. Since the heat transfer tubes 20 are formed as flat tubes, it is possible to further reduce the projected area of each of the heat transfer tubes 20 as viewed in the airflow direction, as compared with the case where circular tubes are used, and is thus possible to reduce the ventilation resistance to a low value. As illustrated in Fig. 3, each of the first heat transfer tubes 20a and the second heat transfer tubes 20b may be formed as a porous flat tube in which a plurality of refrigerant passages 21 allowing refrigerant to flow therethrough are provided. In this case, the refrigerant passages 21 are arranged side by side in the x-direction from one end portion of the cross-section of each of the heat transfer tubes 20 in the longitudinal direction toward the other end portion of the cross-section of the heat transfer tube 20 in the longitudinal direction.

[0036] As illustrated in Fig. 3, the length (the width W1 in the y-direction) of the cross-section of each of the first heat transfer tubes 20a in the width direction is smaller

than the length (the width W2 in the y-direction) of the cross-section of each of the second heat transfer tubes 20b in the width direction. As illustrated in Fig. 3, the first heat transfer tubes 20a are located at the front of the second heat transfer tubes 20b in the airflow direction indicated by the arrow B. In other words, the first heat transfer tubes 20a and the second heat transfer tubes 20b are provided in series with each other in the airflow direction indicated by the arrow B. The first heat transfer tubes 20a are located on the windward side, and the second heat transfer tubes 20b are located on the leeward side. Thus, in the x-direction, at least one of the first heat transfer tubes 20a of the first heat exchanger 11 is aligned with the second heat transfer tubes 20b of the second heat exchanger 12.

[0037] The first heat transfer tubes 20a and the second heat transfer tubes 20b are each formed of a thermally conductive metal material. As the material of each of the first heat transfer tubes 20a and the second heat transfer tubes 20b, for example, aluminum, an aluminum alloy, copper, or a copper alloy is used. Each of the first heat transfer tubes 20a and the second heat transfer tubes 20b is manufactured by extrusion in which a heated material is extruded through a hole of a die to form a cross-section shaped as illustrated in Fig. 3. It should be noted that each of the first heat transfer tubes 20a and the second heat transfer tubes 20b may be manufactured by drawing in which a material is drawn through a hole of a die to form a cross-section as illustrated in Fig. 3. A method of manufacturing each of the first heat transfer tubes 20a and the second heat transfer tubes 20b can be selected as appropriate depending on what cross-sectional shape is to be formed as the cross-sectional shape of each of the heat transfer tubes 20.

[0038] It should be noted that although it is described above that each of the first heat transfer tubes 20a and the second heat transfer tubes 20b is formed as a flat tube or a porous flat tube, it is not limiting. That is, each of the first heat transfer tubes 20a and the second heat transfer tubes 20b may be formed as, for example, a circular tube.

Operation of Heat Exchanger 10

[0039] Next, an operation of the heat exchanger 10 according to Embodiment 1 will be described. As described above, in the heat exchanger 10, the first header portion 13, the first heat transfer tubes 20a of the first heat exchanger 11, the inter-row connection portion 15, the second heat transfer tubes 20b of the second heat exchanger 12, and the second header portion 14 are connected in this order as illustrated in Fig. 2, thereby forming a refrigerant passage. When the heat exchanger 10 operates as an evaporator, the refrigerant flows therein in the above order. When the heat exchanger 10 operates as a condenser, the refrigerant flows therein in reverse order. The following description, in order to simplify the description, is made by way of example by referring to

the case where the heat exchanger 10 operates as an evaporator. However, the flow direction of the refrigerant in the case where the heat exchanger 10 operates as an evaporator or a condenser is not limited to that in the following description; that is, the refrigerant may flow in the opposite direction to the direction indicated by the description.

[0040] Meanwhile, with respect to the airflow, as illustrated in Fig. 3, air is supplied in the direction indicated by the arrow B by the outdoor fan 6. Thus, first, at the first heat exchanger 11, heat exchange is performed between air and refrigerant that flows through the first heat transfer tubes 20a. After the heat exchange, the air flows toward the second heat exchanger 12. Thus, at the second heat exchanger 12, heat exchange is performed between the air and refrigerant that flows through the second heat transfer tubes 20b.

[0041] In this case, since the first heat exchanger 11 includes no fin, the surfaces of the first heat transfer tubes 20a correspond to the outermost surface of the heat exchanger. Therefore, at the first heat exchanger 11, heat conduction through a fin does not occur, and the temperature of the outermost surface of the heat exchanger is thus close to the internal temperature of the first heat transfer tubes 20a. In general, when frost formation occurs at a heat exchanger, frost adheres mainly to a fin. However, since the first heat exchanger 11 includes no fin, frost adheres to the surfaces of the first heat transfer tubes 20a. Meltwater of the frost flows downward along the surfaces of the first heat transfer tubes 20a in the direction of gravity. Since no fin is provided at the first heat transfer tubes 20a, in drainage of meltwater of frost, the flow of water that flows downward in the direction of gravity is not hindered, and the speed of the drainage is high.

[0042] On the other hand, the second heat exchanger 12 includes the corrugated fin 30. The corrugated fin 30 serves as a main heat-exchange region. In the second heat exchanger 12, a heat exchange efficiency is higher than in the first heat exchanger 11 because of the presence of the corrugated fin 30. In addition, since the corrugated fin 30 includes the louvers 31 which has a heat-transfer enhancement effect, the heat exchange efficiency is further improved. However, when frost formation occurs at the second heat exchanger 12, frost is formed concentratedly on part of the louvers 31, although it depends on frost formation conditions. In the second heat exchanger 12, dew condensation water and meltwater of frost are drained along the louvers 31. At this time, since the fin body portions 302 of the corrugated fin 30 are arranged at a small pitch (third pitch FP2), the fin body portions 302 hinder the flow of water flowing downward in the direction of gravity, and the speed of the drainage is thus slower than in the first heat exchanger 11.

[0043] Therefore, in order to reduce occurrence of frost formation at the second heat exchanger 12, the heat exchanger 10 according to Embodiment 1 is configured such that dehumidification is performed mainly by the

first heat exchanger 11 and dry air is sent to the second heat exchanger 12. As a result, it is possible to reduce occurrence of frost formation at the second heat exchanger 12. It should be noted that if frost forms and grows, it blocks the air passages. However, it is possible to reduce occurrence of frost formation, and thus reduce occurrence of blockage of the air passages. In addition, since the first heat exchanger 11 includes no fin, even when dew condensation or frost formation occurs, in drainage of dew condensation water and meltwater of frost, the flow of water flowing downward in the vertical direction is not hindered, and the speed of the drainage is high. Therefore, the air passages between the first heat transfer tubes 20a of the first heat exchanger 11 are not blocked by dew condensation water or formed frost. This will be described in detail.

[0044] In order that advantages of the heat exchanger 10 according to Embodiment 1 be understood, an operation of the heat exchanger 10 in the case where the first heat exchanger 11 and the second heat exchanger 12 are installed independently will be described.

[0045] In the case where air of an airflow is wet air containing water vapor, sensible heat exchange (cooling) and latent heat exchange (dehumidification) at the time of cooling of the wet air, that is, at the time when a heat exchanger is used as an evaporator will be described. In the case where as a relationship between a dry-bulb temperature and a dew-point temperature, the dry-bulb temperature = the dew-point temperature is satisfied, it brings about a saturated state when a relative humidity is 100%. However, the relative humidity is generally less than 100%, that is, in many cases, the dry-bulb temperature > the dew-point temperature.

[0046] A condition under which latent heat exchange (dehumidification) occurs is that the dew-point temperature of air > the surface (water film) temperature of a heat exchanger is satisfied. When this condition is satisfied, moisture in air is condensed as water droplets, water films, or frost on a surface of the heat exchanger. Meanwhile, a condition under which sensible heat exchange (cooling) occurs is that the dry-bulb temperature of air > the surface temperature of the heat exchanger is satisfied. Therefore, when the dry-bulb temperature of air > the surface temperature of the heat exchanger > the dew-point temperature of air is satisfied, latent heat exchange does not occur and only sensible heat exchange occurs. In other words, air is cooled but is not dehumidified. In general, when a dry-bulb temperature and a dew-point temperature of wet air are equal to each other, the lower the surface temperature of the heat exchanger the same, the higher the ratio of the amount of latent heat exchange to the amount of total heat exchange.

Features of Second Heat Exchanger 12

[0047] Features of a corrugated-fin heat exchanger that corresponds to the second heat exchanger 12 will be described below. Fig. 6 is a partially enlarged per-

spective view schematically illustrating an example of the configuration of the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 1. In the corrugated fin 30, the plate-shaped member 300 may be bent to have a wavy shape such that the plate-shaped member 300 has U-shaped portions as illustrated in Fig. 6, (a). Alternatively, the plate-shaped member 300 may be bent to have a wavy shape such that the plate-shaped member 300 has angular U-shaped portions as illustrated in Fig 6, (b).

Heat Exchanging Performance and Dehumidifying Performance

[0048]

· The corrugated fin 30 is formed of the plate-shaped member 300 that is a thin plate having the thickness T. Therefore, the amount of heat that is transferred between the fin surface of the corrugated fin 30 and air is larger than the amount of heat that is transferred between the tube outer surfaces of the second heat transfer tubes 20b and air. As a result, a temperature gradient occurs due to the above difference in the amount of transferred heat, and a fin efficiency becomes smaller than 1. The fin efficiency is a ratio between the actual heat transfer amount of heat transferred from the entire surface of the corrugated fin 30 and a heat transfer amount on the assumption that all of temperatures of the fin surfaces are equal to the temperature of each of base portions 303 of the fin. Thus, the base portions 303 of the fin mean portions of the corrugated fin 30 that are joined to the second heat transfer tube 20b. Therefore, in a fin region of a portion located away from the base portion 303 of the fin, the surface temperature is raised, and the ratio of the amount of latent heat exchange to the amount of the total heat exchange is decreased and the ratio of the amount of sensible heat exchange to the total heat exchange is increased, as compared with the case where the fin efficiency is 1. As a result, the heat exchange is efficiently performed between the refrigerant and air.

· The corrugated fin 30 is provided with a plurality of the louvers 31, as illustrated in Fig. 6, (a) and Fig. 6, (b), in order to ensure a drainage passage. The louvers 31 are arranged and spaced apart from each other in the x-direction. The louvers 31 also have a heat-transfer enhancement effect. To be more specific, when an airflow collides with the louvers 31, the airflow separates from the louvers 31, thereby enhancing heat transfer on the fin side. A method of forming the louvers 31 will be described specifically. In each of the fin body portions 302 of the plate-shaped member 300, U-shaped slits are formed as illustrated in Fig. 6, (a), or two parallel slits are formed as illustrated in Fig. 6, (b). The slits extend through the fin body portions 302 having the thickness T.

Next, rectangular portions surrounded by the slits are bent such that they are raised to project from the surfaces of the fin body portions 302. Through these processes, the louvers 31 are formed. In the following description, the raised portions are referred to as projections 31a of the louvers 31. In the case where the corrugated fin 30 is provided with the louvers 31, part of air that flows along the fin body portions 302 passes through the part of the louvers 31. To be more specific, an airflow collides with the louvers 31, and separates therefrom, and part of the separated airflow flows in such a manner as to pass through a region located below the projections 31a of the louvers 31, and remaining part of the airflow flows through a region located above the projections 31a. Part of the airflow that is close to the fin body portions 302 has a higher (or lower) temperature. The greater the distance between the part of the airflow and the fin body portions 302, the lower (or higher) the temperature of the part of the airflow. As a result, a temperature boundary layer is formed in the airflow. When the airflow passes through the louvers 31, the temperature boundary layer of the airflow is reconstructed (thinned). Thus, heat transfer on the fin side is enhanced. Because of the provision of the louvers 31, the heat exchange is efficiently performed between the refrigerant and air.

Frost Formation

[0049]

· When frost forms on the second heat exchanger 12, the frost forms concentratedly on the louvers 31 of the corrugated fin 30. As a result, when the frost blocks the air passages, the air volume of air that flows in the air passages is reduced, and the load of the outdoor fan 6 is increased.

Drainage

[0050]

· Dew condensation water and meltwater of frost flow along the louvers 31 and are drained downward in the direction of gravity. It should be noted that the fin body portions 302 extend in the horizontal direction, which is perpendicular to the direction of gravity, and are arranged, as illustrated in Fig. 5, at the relatively small third pitch FP2 in the direction of gravity. Therefore, the fin body portions 302 hinder the above drainage, and as a result, the speed of the drainage is reduced, and water easily remains in the second heat exchanger 12.

Features of First Heat Exchanger 11

[0051] Features of a finless heat exchanger to which

the first heat exchanger 11 corresponds will be described below.

Heat Exchanging Performance and Dehumidifying Performance

[0052]

- Since the first heat exchanger 11 has no fin, a heat transfer area basically corresponds to only the areas of the first heat transfer tubes 20a.
- Since the tube outer surfaces of the first heat transfer tubes 20a correspond to the outermost surface of the first heat exchanger 11 and heat conduction through a fin does not occur, the outermost surface of the first heat exchanger 11 has a temperature close to the internal temperature of the first heat transfer tubes 20a. In particular, in the case where the first heat exchanger 11 is a "completely finless heat exchanger" in which the first heat transfer tubes 20a do not have projections or other similar portions,

since the temperature of each of base portions of a fin is equal to the tube surface temperature of each of the first heat transfer tubes 20a if it is assumed that the fin is present, and the fin efficiency is thus 1. In other words, in the first heat exchanger 11 having no fin, the ratio of the amount of latent heat exchange to the amount of total heat exchangers is increased, as compared with a corrugated-fin heat exchanger in which the fin efficiency is decreased by the effect of a fin. Therefore, air is efficiently dehumidified.

Drainage

[0053]

- In the first heat exchanger 11, dew condensation water and meltwater of frost flow along the first heat transfer tubes 20a and are drained downward in the direction of gravity. Since the tube axis of each of the first heat transfer tubes 20a extends in the vertical direction, dew condensation water and meltwater of frost flow downward at a high speed because of the action of the gravity. In addition, at the first heat transfer tubes 20a, since no fin is provided, the flow of water flowing downward in the direction of gravity since is not hindered. Therefore, the speed of the drainage is high, and water does not easily remain in the first heat exchanger 11.

[0054] Next, with reference to Figs. 3 and 7, an operation in the case where frost formation occurs at the heat exchanger 10 will be described. Fig. 7 is an explanatory view illustrating a state in which frost formation occurs at the heat exchanger 10 which is provided as illustrated in Fig. 3. In Fig. 7, hatched portions are regions in which frost formation occurs.

[0055] As described above, in the heat exchanger 10, first, at the first heat exchanger 11, heat exchange is performed between air and refrigerant that flows through the first heat transfer tubes 20a. Then, at the second heat exchanger 12, the air flows toward the second heat exchanger 12, and heat exchange is performed between the air and refrigerant that flows through the second heat transfer tubes 20b. At this time, in the first heat exchanger 11, the ratio of the amount of latent heat exchange to the amount of the total heat exchange is high and the air is efficiently dehumidified. Thus, air that flows toward the second heat exchanger 12 is dry air that does not contain much water vapor. Therefore, the dry-bulb temperature of air > the surface temperature of the heat exchanger > a dew-point temperature of air is satisfied, and latent heat exchange does not occur and only sensible heat exchange occurs. In other words, air is cooled but is not dehumidified. Since dehumidification is not performed at the second heat exchanger 12, moisture in air is not condensed as water droplets, water films, or frost on the heat exchanger surface of the second heat exchanger 12. Since dehumidification is performed at the first heat exchanger 11, moisture in air is condensed as water droplets, water films, or frost on the heat exchanger surface of the first heat exchanger 11. However, since drainage is efficiently performed, the air passages are not blocked by dew condensation water or formed frost. In addition, since dry air is supplied to the second heat exchanger 12 and dehumidification is not performed at the second heat exchanger 12, dew condensation or frost formation hardly occurs at the second heat exchanger 12. Therefore, in the heat exchanger 10, as illustrated in Fig. 3, generally, blockage of the air passages due to dew condensation or frost formation occurs at neither the first heat exchanger 11 nor the second heat exchanger 12.

[0056] However, frost formation may occur. Whether frost formation occurs or not depends on an environment in which the heat exchanger 10 is provided or an operation state of the heat exchanger 10. In order to simply the following description, frost formation at the first heat exchanger 11 will not be considered. As illustrated in Fig. 7, in the airflow direction, the first heat transfer tubes 20a included in the first heat exchanger 11 are located at the front of the second heat transfer tubes 20b included in the second heat exchanger 12. Therefore, air that flows toward the second heat exchanger 12 causes much frost formation on the windward side of the corrugated fin 30, on which the difference between the dew-point temperature and the surface temperature of the heat exchanger is great. In Embodiment 1, however, the frost formed on the second heat exchanger 12 is thin and does not completely block the air passages since the air dehumidified at the first heat exchanger 11 flows toward the second heat exchanger 12. Therefore, it is possible to reduce an increase in ventilation resistance and also reduce a decrease in the air volume. It is therefore possible to also reduce an increase in the load of the outdoor fan 6.

[0057] As described above, in the heat exchanger 10

according to Embodiment 1, in the first heat exchanger 11 (the finless heat exchanger on the windward side), the ratio of the amount of the latent heat exchange to the amount of the total heat exchange is high even under dew-condensation and frost-formation conditions, and dehumidification can be efficiently performed. In addition, since a large amount of air is dehumidified in the first heat exchanger 11, the amount of dew condensation or frost formation at the second heat exchanger 12 (the corrugated-fin heat exchanger on the rear row side) is relatively reduced, and it is possible to reduce occurrence of blockage of the air passages and to reduce a drainage load. Therefore, in the second heat exchanger 12, the ratio of the amount of the sensible heat exchange to the amount of the total heat exchange is high, and heat exchange is efficiently performed between refrigerant and air. In such a manner, in the heat exchanger 10 according to Embodiment 1, in the airflow direction, the first heat exchanger 11 is provided at the front row and the second heat exchanger 12 is provided at the rear row. It is therefore possible for the heat exchanger 10 as a whole to reduce occurrence of blockage of the air passages which is caused by frost formation and possible to improve the performance of draining dew condensation water and meltwater of frost. As a result, the heat exchanger 10 can cause heat exchange to be efficiently performed as a whole.

[0058] In addition, since it is possible for the heat exchanger 10 as a whole to reduce occurrence of blockage of the air passages which is caused by frost formation, although it depends on operational conditions of the refrigeration cycle apparatus 1, it is almost unnecessary to perform a defrosting operation. It is therefore possible to reduce the number of times the defrosting operation is performed, as compared with an existing heat exchanger. That is, the time period for which the heating operation is continuously performed can be increased, and discomfort caused by a drop in an indoor temperature in a defrosting operation of the related art is eliminated. It is therefore possible to improve comfort in an indoor air-conditioning space.

[0059] In addition, as illustrated in Fig. 2, since the tube axis of each of the first heat transfer tubes 20a of the first heat exchanger 11 and the tube axis of each of the second heat transfer tubes 20b of the second heat exchanger 12 extend in the same direction, the headers which include the first header portion 13, the second header portion 14, and the inter-row connection portion 15, can be provided to extend in the same direction. Therefore, the inter-row connection portion 15, which connects the first heat transfer tubes 20a at the front row and the second heat transfer tubes 20b at the rear row, can be formed integrally with the first heat transfer tubes 20a and the second heat transfer tubes 20b. Incidentally, in the case where the second heat exchanger 12 is formed as, for example, a fin-and-tube heat exchanger including tubes extending in a lateral direction, the inter-row connection portion 15 cannot be formed integrally with the first heat

transfer tubes 20a or the second heat transfer tubes 20b, since the longitudinal direction of a header for the second heat exchanger 12 and the longitudinal direction of a header for the first heat exchanger 11 differ from each other.

Embodiment 2

[0060] With reference to Figs. 2 and 8 to 10, a configuration of the heat exchanger 10 according to Embodiment 2 will be described. Fig. 8 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 2 that is taken along the cross-section A as illustrated in Fig. 2. Figs. 9 and 10 are explanatory views each illustrating a state in which frost formation occurs at the heat exchanger 10 as illustrated in Fig. 8. In Figs. 9 and 10, hatched portions are portions at which frost formation occurs. Embodiment 2 will be described mainly on points on which Embodiment 2 differs from Embodiment 1. Regarding Embodiment 2, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiment 1 will be omitted.

[0061] Also in Embodiment 2, the first heat transfer tubes 20a are provided in series with the second heat transfer tubes 20b in the airflow direction, that is, in the x-direction; however, in Embodiment 2, as illustrated in Fig. 8, the first heat transfer tubes 20a are not aligned with the second heat transfer tubes 20b in the x-direction. Therefore, specifically, the corrugated fins 30 are located at the rear of the first heat transfer tubes 20a.

[0062] In such a manner, in Embodiment 2, all the first heat transfer tubes 20a are located at the front of the corrugated fins 30 of the second heat exchanger 12 in the x-direction. In Embodiment 2, as illustrated in Fig. 8, the first pitch PP1 of the first heat transfer tubes 20a is equal to the second pitch PP2 of the second heat transfer tubes 20b.

[0063] Next, a phase change during frost formation in the heat exchanger 10 according to Embodiment 2 will be described with Figs. 9 and 10.

[0064] At the first heat exchanger 11, dehumidification is performed, and moisture in air is thus condensed as water droplets, water films, or frost on surfaces of the heat exchanger. However, since drainage is efficiently performed, frost formation hardly occurs at the first heat exchanger 11. In addition, the second heat exchanger 12 is supplied with dry air, and at the second heat exchanger 12, dehumidification is not performed. Thus, frost formation thus hardly occurs at the second heat exchanger 12. Therefore, in the heat exchanger 10, as illustrated in Fig. 8, at normal times, frost formation occurs at neither the first heat exchanger 11 nor the second heat exchanger 12.

[0065] However, frost formation may occur depending on an environment in which the heat exchanger 10 is provided or an operation state of the heat exchanger 10. In order to simply the following description, frost forma-

tion at the first heat exchanger 11 will not be considered.

[0066] In the following description, as illustrated in Fig. 9, a region of the corrugated fin 30 that is aligned with the first heat transfer tube 20a provided at the front row is referred to as a first region P1, and the other regions of the corrugated fin 30 are each referred to as a second region P2.

[0067] At the first region P1, since the first heat transfer tube 20a in the first heat exchanger 11 is located at the front of the corrugated fin 30 in the airflow direction, an airflow collides with the first heat transfer tube 20a and separation of the airflow thus occurs. As a result, at the first region P1, air does not easily flow to the corrugated fin 30 at the rear row, and does not cause frost formation, or even if the air causes frost formation, the amount of formed frost is small.

[0068] By contrast, at each of the second regions P2 of the corrugated fin 30, since no first heat transfer tube 20a in the first heat exchanger 11 is located at the front of the corrugated fin 30 in the airflow direction, separation of an airflow does not occur. Therefore, at the second region P2, air flows relatively uniformly to the corrugated fin 30 at the rear row, and frost formation occurs on the windward side of the corrugated fin 30.

[0069] In Embodiment 2, in the case where air flows to the heat exchanger 10 at the same volume as in the heat exchanger 10 of Embodiment 1, the velocity of air that passes through the second region P2 is increased as compared with Embodiment 1. Therefore, the heat capacity of air locally increases, and the amounts of changes in the dry-bulb temperature and the dew-point temperature decrease. As a result, as seen from the comparison between Figs. 7 and 9, in Embodiment 2, it is possible to cause frost formation to occur over a wider area in the airflow direction, as compared with Embodiment 1 described above with reference to Fig. 7. It is therefore possible to thin frost to be formed and possible to further reduce occurrence of blockage of the air passages, as compared with Embodiment 1.

[0070] However, the amount of formed frost may increase depending on an environment in which the heat exchanger 10 is provided or an operation state of the heat exchanger 10. When the amount of frost at the second region P2 increases, as illustrated in Fig. 10, the air passages formed at the second region P2 of the corrugated fin 30 at the rear flow are blocked. In this case, the airflow at the first region P1 of the corrugated fin 30 at the rear flow is increased, and frost is accumulated also on the first region P1 of the corrugated fin 30. As described above, in Embodiment 2, at least part of the first heat transfer tubes 20a is aligned with the corrugated fin 30 in the second heat exchanger 12 in the airflow direction. Therefore, compared with Embodiment 1, it is possible to prevent frost formation from occurring at a time on the entire windward side of the corrugated fin 30 and thus to prolong the time taken for blockage of the air passages. As a result, in Embodiment 2, it is possible to further increase a proof strength against frost formation,

as compared with Embodiment 1.

Embodiment 3

[0071] With reference to Figs. 2 and 11, a configuration of the heat exchanger 10 according to Embodiment 3 will be described. Fig. 11 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 3 that is taken along the cross-section A as illustrated in Fig. 2. Embodiment 3 will be described mainly on points on which Embodiment 2 differs from Embodiment 1. Regarding Embodiment 3, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiment 1 will be omitted.

[0072] In Embodiments 1 and 2 described above, as illustrated in Figs. 3 and 8, the first pitch PP1 of the first heat transfer tubes 20a is equal to the second pitch PP2 of the second heat transfer tubes 20b. By contrast, in Embodiment 3, as illustrated in Fig. 11, the first pitch PP1 of the first heat transfer tubes 20a is smaller than the second pitch PP2 of the second heat transfer tubes 20b. That is, the first pitch PP1 of the first heat transfer tubes 20a and the second pitch PP2 of the second heat transfer tubes 20b satisfy the relationship "PP1 < PP2". In Embodiment 3, the first heat transfer tubes 20a are provided for both the windward side of the second heat transfer tubes 20b and the windward side of the corrugated fins 30. Therefore, in Embodiment 3, in the x-direction, some of the first heat transfer tubes 20a are aligned with the corrugated fins 30, and the rest of the first heat transfer tubes 20a are aligned with the second heat transfer tubes 20b.

[0073] In Embodiment 3, as illustrated in Fig. 11, the first pitch PP1 of the first heat transfer tubes 20a is smaller than that in Embodiment 1, and the number of the first heat transfer tubes 20a is thus larger than that of the first heat transfer tubes 20a in Embodiment 1. In Embodiment 3, the first heat transfer tubes 20a are provided at the front of the second heat transfer tubes 20b and the front of the corrugated fins 30. In such an arrangement, between any adjacent two of the first heat transfer tubes 20a in Embodiment 1, another first heat transfer tube 20a is further added. Therefore, it is possible that the number of the first heat transfer tubes 20a is set to be substantially twice the number of the first heat transfer tubes 20a in Embodiment 1. In this case, since an airflow flows in the direction indicated by the arrow B, and the longitudinal direction of the first heat transfer tubes 20a is the same as the airflow direction, ventilation resistance is not increased so much, though the number of the first heat transfer tubes 20a is increased. On the other hand, since the number of the first heat transfer tubes 20a is increased, the amount of heat exchange between air and refrigerant that flows through the first heat transfer tubes 20a is increased. As a result, the heat-exchanging performance of the first heat exchanger 11 is improved.

[0074] In the first heat exchanger 11, no fins are provided between the adjacent ones of the first heat transfer

tubes 20a. Therefore, the ventilation resistance to the air passages is small and the area of the heat exchanger is also small, as compared with the second heat exchanger 12. In this case, in Embodiment 3, since $PP1 < PP2$ is satisfied, it is possible to improve the heat-exchanging performance of the first heat exchanger 11 without causing the ventilation resistance to be greatly increased. As a result, it is possible to further reduce the amount of heat exchange at the second heat exchanger 12, in particular, the amount latent heat exchange at the second heat exchanger 12. In addition, in the first heat exchanger 11, since no fins are provided, the flow of water that flows downward in the direction of gravity is not hindered, as in Embodiment 1. Therefore, the speed of drainage is high, and water does not easily remain in the first heat exchanger 11. For the above reasons, in Embodiment 3, it is possible to further reduce occurrence of blockage of the air passages and reduce the drainage load, as compared with Embodiment 1.

[0075] As described above, in Embodiment 3, since $PP1 < PP2$ is satisfied, it is possible improve the heat-exchanging performance of the first heat exchanger 11 without causing the ventilation resistance to be greatly increased. As a result, it is possible to further reduce the amount of heat exchange at the second heat exchanger 12, in particular, the amount latent heat exchange at the second heat exchanger 12. Therefore, in Embodiment 3, it is possible to further reduce occurrence of blockage of the air passages and reduce the drainage load, as compared with Embodiment 1.

Embodiment 4

[0076] With reference to Figs. 2, 5, and 11, a configuration of the heat exchanger 10 according to Embodiment 4 will be described. Embodiment 4 will be described mainly on points on which Embodiment 4 differs from Embodiment 3. Regarding Embodiment 4, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiments 1 and 3 will be omitted.

[0077] In Embodiment 4, the first distance PC1 as indicated in Fig. 11 and the third distance FC2 as indicated in Fig. 5 satisfy the relationship " $PC1 > FC2$ ". As illustrated in Fig. 5, the third distance FC2 is each of the distances between the adjacent fin body portions 302 in the Z direction, that is, the dimension of each of the air passages between the adjacent fin body portions 302 in the z-direction.

[0078] The above description concerning Embodiment 3 is made with respect to the case where the first pitch PP1 of the first heat transfer tubes 20a and the second pitch PP2 of the second heat transfer tubes 20b satisfy the relationship " $PP1 < PP2$ ". In Embodiment 4, further, the first distance PC1 and the third distance FC2 satisfy the relationship " $PC1 > FC2$ ". In other words, in Embodiment 4, the first pitch PP1 is set to satisfy the relationship " $PP1 < PP2$ " and the relationship " $PC1 > FC2$ ".

[0079] In the case of setting of the first pitch PP1 of the first heat transfer tubes 20a, when the first pitch PP1 is set excessively small, the ventilation resistance to the air passages may be excessively increased. Thus, in Embodiment 4, the first pitch PP1 is set such that the first distance PC1 between the first heat transfer tubes 20a is greater than the third distance FC2, which is the dimension of each of the air passages between the adjacent fin body portions 302 in the z-direction. Therefore, it is possible to improve the heat-exchanging performance of the first heat exchanger 11 without causing the ventilation resistance to be greatly increased.

[0080] The dew-point temperature (absolute humidity) of wet air that passes through the first heat exchanger 11 is higher than that of air that passes through the second heat exchanger 12, and the fin efficiency of the first heat exchanger 11 is higher than that of the second heat exchanger 12. Thus, when the heat exchanger 10 is operated as an evaporator, frost easily forms on the first heat exchanger 11. In this case, by causing the relationship " $PC1 > FC2$ " to be satisfied, it is possible to reduce occurrence of blockage of the air passages of the first heat exchanger 11.

[0081] As described above, in Embodiment 4, by causing the relationships " $PP1 < PP2$ " and " $PC1 > FC2$ " to be satisfied, it is possible to improve the heat-exchanging performance of the first heat exchanger 11 without causing the ventilation resistance to be greatly increased. As a result, it is possible to further reduce the amount of heat exchange in the second heat exchanger 12, in particular, the amount latent heat exchange in the second heat exchanger 12. Therefore, in Embodiment 4, it is possible to further reduce occurrence of blockage of the air passages and reduce the drainage load as in Embodiment 3, as compared with Embodiment 1.

Embodiment 5

[0082] With reference to Figs. 2 and 12, a configuration of the heat exchanger 10 according to Embodiment 5 will be described. Fig. 12 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 5 that is taken along the cross-section A as illustrated in Fig. 2. Embodiment 5 will be described mainly on points on which Embodiment 5 differs from Embodiment 3. Regarding Embodiment 5, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiments 1 and 3 will be omitted.

[0083] Also, in Embodiment 5, the first heat exchanger 11 is a "finless heat exchanger" in which no fin is provided between the first heat transfer tubes 20a, as in Embodiments 1 and 3. In Embodiment 5, as illustrated in Fig. 12, the first heat transfer tubes 20a of the first heat exchanger 11 include respective first projections 51. The first projections 51 are provided on the windward side of the first heat transfer tubes 20a of the first heat exchanger in the airflow direction. The first projections 51 may be

formed integral with the first heat transfer tubes 20a or may be joined to the first heat transfer tubes 20a by brazing. Each of the first projections 51 is made of a thermally conductive metal material. As this material of the first projection 51, for example, aluminum, an aluminum alloy, copper, or a copper alloy is used. The material of the first projection 51 may be the same as or different from that of the first heat transfer tube 20a. The longitudinal direction of the first projection 51 is the same as the airflow direction, that is, the x-direction. One end of the first projection 51 is connected to a front edge portion 22a of the first heat transfer tube 20a, and the other end of the first projection 51, which is a distal end portion of the first projection 51, is a free end. That is, the first projection 51 is provided in a cantilever manner at the first heat transfer tube 20a. The first projection 51 projects from the front edge portion 22a in the opposite direction to the x-direction, that is, in a direction from the leeward side toward the windward side. The front edge portion 22a is an end portion of the first heat transfer tube 20a that is located on the windward side thereof in the x-direction.

[0084] Also in Embodiment 5, since the first heat exchanger 11 includes no fin, basically, the tube outer surfaces of the first heat transfer tubes 20a are the outermost surface of the first heat exchanger 11. Therefore, since heat conduction through a fin does not occur, the outermost surface of the first heat exchanger 11 has a temperature close to the internal temperature of the first heat transfer tubes 20a.

[0085] In Embodiment 5, the first heat transfer tubes 20a of the first heat exchanger 11 are provided with the first projections 51. Each of the first projections 51 is, for example, a plate-shaped member. A width W3 of the first projection 51, that is, the plate thickness of the first projection 51, is relatively small, and is, for example, smaller than the width W1 of each of the first heat transfer tubes 20a, as illustrated in Fig. 12. Therefore, the amount of heat transferred between air and the surface of the first projection 51 is larger than the amount of heat transferred between air and the tube outer surface of the first heat transfer tube 20a. As a result, a temperature gradient occurs due to the differences between the amounts of transferred heat between the surfaces of the first projections 51 and the tube outer surfaces of the first heat transfer tubes 20a, and the fin efficiency becomes smaller than 1. In a region of the first projection 51 that is located apart from the main body of the first heat transfer tube 20a, the surface temperature increases, and as compared with the case where the fin efficiency is 1, the ratio of the amount of the latent heat exchange to the amount of the total heat exchange decreases, and the ratio of the amount of the sensible heat exchange to the total heat exchange increases. In such a manner, since the first projections 51 are each formed of a thin plate, heat is efficiently transferred from the surface of the first projection 51, and a heat-transfer enhancement effect can be obtained.

[0086] In the above manner, in Embodiment 5, the first

projections 51 are provided at the front edge portions 22a of the first heat transfer tubes 20a of the first heat exchanger 11. Thus, it is possible to slightly reduce only the fin efficiency of the front edge portion 22a of each of the first heat transfer tubes 20a while maintaining the high fin efficiency (dehumidifying performance) of the first heat exchanger 11 as a whole. As a result, in Embodiment 5, it is possible to reduce occurrence of frost formation at the front edge portion 22a of the first heat transfer tube 20a, as compared with the other portions. It is therefore possible to reduce occurrence of blockage of the air passages which is caused by frost formation on the front edge portion 22a of the first heat transfer tube 20a.

[0087] In addition, in Embodiment 5, since the first projections 51 are provided at the first heat transfer tubes 20a of the first heat exchanger 11, it is possible to increase the heat transfer area of the first heat transfer tubes 20a, as compared with Embodiment 3. Therefore, in Embodiment 5, it is possible to further improve the heat-exchanging performance of the first heat exchanger 11, as compared with Embodiment 3. It should be noted that the first projections 51 of Embodiment 5 are also applicable to the embodiments other than Embodiment 3.

Embodiment 6

[0088] With reference to Figs. 2 and 13 to 15, a configuration of the heat exchanger 10 according to Embodiment 6 will be described. Fig. 13 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 6 that is taken along the cross-section A as illustrated in Fig. 2. Fig. 14 is a plan view illustrating a configuration of the inter-row connection portion 15 provided in the heat exchanger 10 according to Embodiment 6. Fig. 15 is a partial side view illustrating a configuration of the heat exchanger 10 according to Embodiment 6. Fig. 15 illustrates the heat exchanger 10 as viewed in the opposite direction to the y-direction in Fig. 13. Embodiment 6 will be described mainly on points on which Embodiment 6 differs from Embodiment 3. Regarding Embodiment 6, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiments 1 and 3 will be omitted.

[0089] Also, in Embodiment 6, the first heat exchanger 11 is a "finless heat exchanger" in which no fin is provided between the first heat transfer tubes 20a, as in Embodiments 1 and 3. In Embodiment 6, as illustrated in Fig. 13, the first heat transfer tubes 20a of the first heat exchanger 11 include second projections 52. Each of the second projections 52 is, for example, a plate-shaped member. A width W4 of the second projection 52, that is, the plate thickness of the second projection 52, is relatively small, and is smaller than the width W1 of each of the first heat transfer tubes 20a, as illustrated in Fig. 13, for example. Therefore, a heat-transfer enhancement effect can be achieved since heat is efficiently transferred from the surface of the second projection 52. The second

projections 52 are provided on the leeward sides of the first heat transfer tubes 20a of the first heat exchanger 11 in the airflow direction. The second projections 52 may be formed integral with the first heat transfer tubes 20a or may be joined to the first heat transfer tubes 20a by brazing. Each of the second projections 52 is made of a thermally conductive metal material. As the material of the second projection 52, for example, aluminum, an aluminum alloy, copper, or a copper alloy is used. The material of the second projection 52 may be the same as or different from that of the first heat transfer tube 20a. The second projection 52 extends in the airflow direction, that is, the x-direction. One end of the second projection 52 is connected to a rear edge portion 23a of the first heat transfer tube 20a, and the other end of the second projection 52, which is a distal end portion of the second projection 52, is a free end. That is, the second projection 52 is provided in a cantilever manner at the first heat transfer tube 20a. The second projection 52 projects from the rear edge portion 23a in the x-direction, that is, in a direction from the windward side toward the leeward side. The rear edge portion 23a is an end portion of the first heat transfer tube 20a on the leeward side thereof in the x-direction.

[0090] In Embodiment 6, since the second projections 52 are provided at the first heat transfer tubes 20a of the first heat exchanger 11, it is possible to ensure a space 15c between the first heat exchanger 11 and the second heat exchanger 12 while ensuring an independent air passage between the first heat exchanger 11 and the second heat exchanger 12.

[0091] As illustrated in Fig. 15, in the case where the inter-row connection portion 15 which connects the first heat exchanger 11 and the second heat exchanger 12 are formed integrally with the first heat exchanger 11 and the second heat exchanger 12, as indicated by dashed lines in Fig. 15, insertion holes 15a and insertion holes 15b that allow the heat transfer tubes 20 to be inserted thereinto are formed in the inter-row connection portion 15. Of the heat transfer tubes 20, the first heat transfer tubes 20a are inserted into the insertion holes 15a, and the second heat transfer tubes 20b are inserted into the insertion holes 15b. In Embodiment 6, as illustrated in a plan view of Fig. 14, since the second projections 52 are provided at the first heat transfer tubes 20a, the space 15c having a width corresponding to the length of each of the second projections 52 in the longitudinal direction can be ensured between the insertion holes 15a and the insertion holes 15b. Therefore, at the time of manufacturing the inter-row connection portion 15, the insertion holes 15a at the front row and the insertion holes 15b at the rear row need not to be provided close to each other. The inter-row connection portion 15 is molded in the above integral molding manner by press molding or other processes using a metallic mold or other tools. At this time, if the insertion holes 15a and the insertion holes 15b were close to each other, the metallic mold would be required to be formed with a high accuracy in order

to prevent the insertion holes 15a and the insertion holes 15b from being connected to each other, and it would be hard to form such a metallic mold with a high accuracy. In addition, if a thin wall between the insertion holes 15a and the insertion holes 15b were provided, the wall would be, for example, damaged at the time of removing the formed inter-row connection portion 15 from the metallic mold. In contrast, in the case where the insertion holes 15a and the insertion holes 15b are not close to each other, as in Embodiment 6, the metallic mold is not required to be formed with a high accuracy, and the metallic mold is thus easily formed. In addition, the wall between the insertion holes 15a and the insertion holes 15b has a thickness equal to the width of the space 15c and thus is not easily damaged. Therefore, work of removing the formed inter-row connection portion 15 from the metallic mold is easy. For these reasons, a work process of molding the inter-row connection portion 15 in the integral molding manner by press molding or other processes is easy, and the workability is improved.

[0092] Thus, in Embodiment 6, since the second projections are provided at the first heat transfer tubes 20a of the first heat exchanger 11, the space 15c is provided between the insertion holes 15a and the insertion holes 15b in the inter-row connection portion 15. Therefore, the work process of molding the inter-row connection portion 15 in the integral molding manner by press molding or other processes is easy, and the workability is improved.

[0093] In addition, in Embodiment 6, since the second projections are provided at the first heat transfer tubes 20a of the first heat exchanger 11, the heat transfer area of the first heat transfer tubes 20a can be increased, as compared with Embodiment 3. Therefore, in Embodiment 6, it is possible to further improve the heat-exchanging performance of the first heat exchanger 11 than in Embodiment 3. It should be noted that the second projections 52 of Embodiment 6 are also applicable to the embodiments other than Embodiment 3.

Modification of Embodiment 6

[0094] Fig. 16 is a sectional view illustrating a cross-section of the heat exchanger 10 according to a modification of Embodiment 6 that is taken along the cross-section A as illustrated in Fig. 2. As illustrated in Fig. 16, in Embodiment 6, the first projections 51 formed as described regarding Embodiment 5 may be further provided at the first heat transfer tubes 20a. That is, in the modification of Embodiment 6, each of the first heat transfer tubes 20a include the first projection 51 and the second projection 52.

[0095] Thus, in the modification of Embodiment 6, the first heat transfer tube 20a includes the first projection 51 described regarding Embodiment 5 and the second projection 52 described regarding Embodiment 6. Therefore, needless to say, the heat exchanger 10 of the modification of Embodiment 6 obtains the advantages of Embodiment 5 and those of Embodiment 6.

Embodiment 7

[0096] Embodiment 7 is a modification of Embodiment 3, and will thus be described with reference to Fig. 11 relating to Embodiment 3.

[0097] It should be noted that Embodiment 3 is described above with respect to the case where the first pitch PP1 of the first heat transfer tubes 20a of the first heat exchanger 11 is smaller than the second pitch PP2 of the second heat transfer tubes 20b of the second heat exchanger 12. In Embodiment 7, in the above configuration of the heat exchanger 10 of Embodiment 3, the hydraulic diameter of each of the first heat transfer tubes 20a of the first heat exchanger 11 is set smaller than that of each of the second heat transfer tubes 20b of the second heat exchanger 12. It should be noted that a hydraulic diameter R is expressed by the following formula.

Hydraulic diameter $R = 4 \times (\text{cross-sectional area of flow passage} / \text{length of wet edge})$

[0098] It should be noted that the cross-sectional area of the flow passage is a cross-sectional area of each of the first heat transfer tubes 20a and the second heat transfer tubes 20b that is taken in the radial direction; and the length of the wet edge is a perimeter of a wet portion (wet edge) of a cross-section of each of the first heat transfer tubes 20a and the second heat transfer tubes 20b, that is, the length of the wet edge is a circumferential-direction length of part of a tube inner wall of each of the first heat transfer tubes 20a and the second heat transfer tubes 20b that is in contact with the refrigerant.

[0099] The following description is made with respect to the case where the first pitch PP1 of the first heat transfer tubes 20a of the first heat exchanger 11 is smaller than the second pitch PP2 of the second heat transfer tubes 20b of the second heat exchanger 12. In this case, for example, in the case where the length of the first header portion 13 in the y-direction is equal to that of the second header portion 14 in the y-direction, the number of the first heat transfer tubes 20a of the first heat exchanger 11 is larger than that of the second heat transfer tubes 20b of the second heat exchanger 12. Therefore, in the first heat exchanger 11, it is possible to reduce the cross-sectional area of a flow passage which can maintain the pressure loss of a fluid that flows in the first heat transfer tubes 20a. Thus, for example, by reducing the hydraulic diameter of each of the first heat transfer tubes 20a, it is possible to reduce the thickness of the tube inner wall of each of the first heat transfer tubes 20a to an extent to which a proof pressure can be ensured. As a result, it is possible to mount the first heat transfer tubes 20a of the first heat exchanger 11 densely (at a small pitch) while reducing an increase in the ventilation resistance. As a result, it is possible to further improve the dehumidifying effect of the first heat exchanger 11.

Embodiment 8

[0100] With reference to Fig. 17, the heat exchanger 10 according to Embodiment 8 will be described. Fig. 17 is an explanatory view for occurrence of dryout of the heat exchanger 10 according to Embodiment 8. When the quality in the heat transfer tubes increases and causes dryout, a satisfactory heat transfer rate is not obtained. In Embodiment 8, as in Embodiment 7, the hydraulic diameter of each of the first heat transfer tubes 20a of the first heat exchanger 11 is smaller than that of each of the second heat transfer tubes 20b of the second heat exchanger 12. The following description is made by referring to by way of example the case where the heat exchanger 10 operates as an evaporator and refrigerant flows from the first heat exchanger 11 toward the second heat exchanger 12.

[0101] Fig. 17, (a), is a graph indicating timing of occurrence of dryout in a comparative example for explanation of advantages of Embodiment 8. In Fig. 17, (a), the horizontal axis indicates the quality in the heat transfer tubes, and the vertical axis indicates the in-tube heat transfer rate of the heat transfer tubes. In addition, in Fig. 17, (a), a dashed line 70 is a line for a heat exchanger in which the hydraulic diameter is small, and a dotted line 71 is a line for a heat exchanger in which the hydraulic diameter is large.

[0102] Fig. 17, (b), is a graph indicating timing of occurrence of dryout in Embodiment 8. In Fig. 17, (b), the horizontal axis indicates the quality in the heat transfer tubes, and the vertical axis indicates the in-tube heat transfer rate of the heat transfer tubes. In addition, in Fig. 17, (b), a dashed line 70 is a line for the first heat exchanger 11 in which the hydraulic diameter is small, and a dotted line 71 is a line for the second heat exchanger 12 in which the hydraulic diameter is large. In Fig. 17, (b), a thick solid line 74 is a line for the entire heat exchanger 10 including the first heat exchanger 11 and the second heat exchanger 12.

[0103] First of all, the comparative example will be described. In general, in a heat exchanger in which the hydraulic diameter is small, dryout in the heat transfer tubes occurs with a lower quality than in a heat exchanger in which the hydraulic diameter is large. When dryout occurs, the in-tube heat conduction rate in a high-quality region is reduced on and after the point of time at which the dryout occurs. This will be described specifically with respect to the comparative example indicated by Fig. 17, (a). As indicated by the dashed line 70 in Fig. 17, (a), it is assumed that dryout occurs when a quality x is x_1 in the heat exchanger in which the hydraulic diameter is small. In this case, an in-tube heat transfer rate α_i gradually decreases after the occurrence of dryout. Thus, a region in which the quality x is higher than or equal to x_1 is a dryout occurrence region 72 in the heat exchanger in which the hydraulic diameter is small. However, even if dryout occurs, it does not immediately deteriorate a heating effect. In Fig. 17, an in-tube heat transfer rate α_i

that is satisfactory in the range of a threshold value $Th\alpha$ or more is obtained, and the range is considered as a range in which the heating effect is not affected. Therefore, when the in-tube heat transfer rate αi falls below the threshold value $Th\alpha$, the heating effect greatly decreases.

[0104] In contrast, as indicated by the dotted line 71 in Fig. 17, (a), in the heat exchanger in which the hydraulic diameter is large, dryout occurs when the quality x is x_3 , which is greater than x_1 , and thereafter, the in-tube heat transfer rate rapidly decreases. Thus, a region in which the quality x is higher than or equal to x_3 is a dryout occurrence region 73 in the heat exchanger in which the hydraulic diameter is small. As described above, the dryout occurrence region can be reduced as the hydraulic diameter is increased.

[0105] As described above regarding Embodiment 7, when the hydraulic diameter of each of the heat transfer tubes of a heat exchanger is reduced, the dehumidification efficiency is increased accordingly; that is, the smaller the hydraulic diameter, the higher dehumidification efficiency. However, on the other hand, the smaller the hydraulic diameter, the larger the dryout occurrence region 72, as indicated by the dashed line 70 in Fig. 17, (a).

[0106] In such a manner, in terms of the hydraulic diameter, a trade-off relationship is satisfied between an improvement in dehumidification efficiency and reduction of the dryout occurrence region. In Embodiment 8, the heat exchanger 10 is formed to include the first heat exchanger 11 and the second heat exchanger 12 in order to improve the dehumidification efficiency and reduce the dryout occurrence region.

[0107] In Embodiment 8, the heat exchanger 10 is formed to include the first heat exchanger 11 and the second heat exchanger 12 as in Embodiments 1 to 7. Furthermore, in Embodiment 8, the hydraulic radius of each of the first heat transfer tubes 20a of the first heat exchanger 11 is set smaller than that of each of the second heat transfer tubes 20b of the second heat exchanger 12.

[0108] As indicated by the thick solid line 74 in Fig. 17, (b), when the heat exchanger 10 is operated as an evaporator, dryout occurs at the first heat transfer tubes 20a when the quality x is x_1 , and thereafter, the in-tube heat transfer rate gradually decreases. In Embodiment 8, however, since refrigerant flows from the first heat exchanger 11 toward the second heat exchanger 12, the in-tube heat transfer rate re-increases at the point of time at which the quality x is x_2 and which corresponds to a point at which the dashed line 70 and the dotted line 71 in Fig. 17, (b), intersect with each other. Therefore, the in-tube heat transfer rate αi does not fall below the threshold value $Th\alpha$. As described above, in Embodiment 8, it is possible to re-increase the in-tube heat transfer rate before the in-tube heat transfer rate greatly decreases. Then, when the quality x is x_3 , which is greater than x_2 , dryout occurs at the second heat transfer tubes 20b, and thereafter, the in-tube heat transfer rate rapidly decreases.

es. Thus, a region in which the quality x is higher than or equal to x_3 is a dryout occurrence region 75 in the heat exchanger 10.

[0109] As described above, in Embodiment 8, since the heat exchanger 10 is formed to include the first heat exchanger 11 and the second heat exchanger 12, it is possible to satisfy the two requirements that the dehumidification efficiency is improved and that the dryout occurrence region is reduced. That is, a high heat exchange efficiency is ensured by the second heat exchanger 12 while a high dehumidifying effect is maintained by the first heat exchanger 11.

[0110] Fig. 18 is a perspective view illustrating a modification of Embodiment 8. In Embodiments 1 to 7, it is described above that as illustrated in Fig. 2, the refrigerant is moved between the first heat exchanger 11 at the front row and the second heat exchanger 12 at the rear row through the inter-row connection portion 15; however, it is not limiting. For example, as illustrated in Fig. 18, an upper first header portion 13b and an upper second header portion 14b may be provided instead of the inter-row connection portion 15, and refrigerants may join each other once at the upper first header portion 13b, and be moved between the rows, and then, the refrigerant may be re-distributed at the upper second header portion 14b. Hereinafter, a modification of Embodiment 8 as illustrated in Fig. 18 will be described.

[0111] Lower ends of the first heat transfer tubes 20a of the first heat exchanger 11 are connected to a lower first header portion 13a. The lower first header portion 13a corresponds to the first header portion 13 which is provided as illustrated in Fig. 2. Upper ends of the first heat transfer tubes 20a of the first heat exchanger 11 are connected to the upper first header portion 13b. The lower first header portion 13a distributes refrigerant supplied from the outside to the first heat transfer tubes 20a of the first heat exchanger 11. The upper first header portion 13b causes refrigerants that flows from the first heat transfer tubes 20a to join each other.

[0112] A connection portion 40a of the upper first header portion 13b and a connection portion 40b of the upper second header portion 14b are connected to each other by an inter-row connection pipe 16 which is U-shaped. The refrigerant into which the refrigerants join each other to combine at the upper first header portion 13b flows into the upper second header portion 14b through the inter-row connection pipe 16.

[0113] Lower ends of the second heat transfer tubes 20b of the second heat exchanger 12 are connected to a lower second header portion 14a. The lower second header portion 14a corresponds to the second header portion 14 which is provided as illustrated in Fig. 2. Upper ends of the second heat transfer tubes 20b of the second heat exchanger 12 are connected to the upper second header portion 14b. The upper second header portion 14b distributes the refrigerant supplied from the upper first header portion 13b to the second heat transfer tubes 20b of the second heat exchanger 12. The lower second

header portion 14a causes the refrigerants that flow from the second heat transfer tubes 20b to join each other to combine into refrigerant, and then causes the refrigerant to flow out to the outside of the heat exchanger 10.

[0114] As described above, as in the modification of Embodiment 8 as illustrated in Fig. 18, refrigerants may be caused to join each other to combine into refrigerant in the upper first header portion 13b, and after being moved between the rows, the refrigerant may be re-distributed at the upper second header portion 14b. Although it is described above by way of example that the configuration of the modification as illustrated in Fig. 18 is applied to Embodiment 8, the configuration of the modification as illustrated in Fig. 18 is also applicable to any of Embodiments 1 to 7.

[0115] As described above, in Embodiment 8, the heat exchanger 10 is formed to include the first heat exchanger 11 and the second heat exchanger 12 in the case where the heat exchanger 10 is operated as an evaporator, whereby it is possible to satisfy the two requirements that the dehumidification efficiency is improved and that the dryout occurrence region is reduced. In other words, a high heat exchange efficiency can be ensured by the second heat exchanger 12 while a high dehumidifying effect can be maintained by the first heat exchanger 11.

Embodiment 9

[0116] With reference to Figs. 2 and 19, a configuration of the heat exchanger 10 according to Embodiment 9 will be described. Fig. 19 is a sectional view illustrating a cross-section of the heat exchanger 10 according to Embodiment 9 that is taken along the cross-section A as illustrated in Fig. 2. Embodiment 9 will be described mainly on points on which Embodiment 9 differs from Embodiment 2. Regarding Embodiment 9, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiments 1 and 2 will be omitted.

[0117] In Embodiment 9, as illustrated in Fig. 19, the corrugated fin 30 provided in the second heat exchanger 12 are located at only regions located leeward of front edge portions 22b of the second heat transfer tubes 20b in the airflow direction. The front edge portions 22b are end portions of the second heat transfer tubes 20b on the windward side in the x-direction. In addition, in Embodiment 9, all of the first heat transfer tubes 20a are aligned with the corrugated fins 30 in the x-direction as in Embodiment 2. Therefore, the first pitch PP1 of the first heat transfer tubes 20a is equal to the second pitch PP2 of the second heat transfer tubes 20b. The number of the first heat transfer tubes 20a in Embodiment 9 is thus smaller than that of the first heat transfer tubes 20a in Embodiment 3 which is described above with reference to Fig. 11.

[0118] In Embodiment 1, as described above, as illustrated in Fig. 3, the corrugated fin 30 is provided over the

entire length of each of the associated second heat transfer tubes 20b in the x-direction in the second heat exchanger 12. In contrast, in Embodiment 9, as illustrated in Fig. 19, the corrugated fin 30 is provided only over part of each of the associated second heat transfer tubes 20b that is located on the leeward side in the x-direction. Accordingly, the relationship " $L1 > L2$ " is satisfied, where L1 is the length from the front edge portion 22b of the second heat transfer tube 20b to a rear edge portion 23b thereof in the x-direction, and L2 is the length from a front edge portion 32 of the corrugated fin 30 to a rear edge portion 33 thereof. The rear edge portion 23b is an end portion of the second heat transfer tube 20b on the leeward side in the x-direction. The front edge portion 32 is an end portion of the corrugated fin 30 on the windward side in the x-direction, and the rear edge portion 33 is an end portion of the corrugated fin 30 on the leeward side in the x-direction.

[0119] In Embodiment 9, in the second heat exchanger 12, a region C in which the second heat transfer tube 20b is exposed is located at part of the second heat transfer tube 20b that is located on a side where the front edge portion 22b is located. In the region C, the fin efficiency of the second heat transfer tube 20b increases and the dehumidification is promoted. It is therefore possible to further reduce the dehumidification load on the corrugated fin 30, as compared with Embodiment 2. As a result, even in the case where the number of the first heat transfer tubes 20a of the first heat exchanger 11 is reduced smaller than that in Embodiment 3, it is possible to obtain the same advantages as in Embodiment 3.

[0120] It should be noted that although the above description refers to the case where Embodiment 9 is applied to the configuration in Embodiment 2, Embodiment 9 is also applicable to the other embodiments.

Embodiment 10

[0121] With reference to Figs. 20 and 21, a configuration of the heat exchanger 10 according to Embodiment 10 will be described. Fig. 20 is a plan view illustrating a configuration of the inter-row connection portion 15 of the heat exchanger 10 according to Embodiment 10. In Fig. 20, some of components are indicated by dotted lines for explanation, though they cannot be viewed. Fig. 21 is a partial side view illustrating the configuration of the heat exchanger 10 according to Embodiment 10. Fig. 21 illustrates the heat exchanger 10 as viewed in the opposite direction to the y-direction in Fig. 20. Embodiment 10 will be described mainly on points on which Embodiment 10 differs from Embodiment 1. Regarding Embodiment 10, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiment 1 will be omitted.

[0122] In Embodiment 10, as illustrated in Fig. 20, the first heat transfer tubes 20a of the first heat exchanger 11 at the front row are connected to the second heat transfer tubes 20b of the second heat exchanger 12 at

the rear row by the inter-row connection portion 15. In this case, in Embodiment 10, each of the first heat transfer tubes 20a at the front row is not aligned with the associated one of the second heat transfer tubes 20b at the rear row in the airflow direction indicated by the arrow B. It should be noted that a region which extends in the y-direction and in which the second heat transfer tube 20b is provided will be referred to as a region D. The first heat transfer tube 20a is not provided in the region D, and is provided in a region other than the region D. In an example illustrated in Fig. 20, the first heat transfer tube 20a is provided in a region adjacent to the region D. As described above, in Embodiment 10, the first heat transfer tube 20a is not aligned with the second heat transfer tube 20b in the x-direction. As is clear from Fig. 3, the corrugated fin 30 is provided in a region adjacent to the region D. Therefore, in Embodiment 10, at least part of the first heat transfer tubes 20a is aligned with the corrugated fin 30 in the x-direction.

[0123] From the comparison between Embodiment 6 as illustrated in Fig. 15 and Embodiment 10 as illustrated in Fig. 21, the width of the space 15c as illustrated in Fig. 21 appears to be smaller than the space 15c as illustrated in Fig. 14. However, actually, as illustrated in Fig. 20, the insertion hole 15a into which the first heat transfer tube 20a at the front row is inserted and the insertion hole 15b into which the second heat transfer tube 20b at the rear row is inserted are aligned with each other in neither the x-direction nor the y-direction. The insertion hole 15a and the insertion hole 15b are thus not close to each other. Therefore, in Embodiment 10, the work process of molding the inter-row connection portion 15 in the integral molding manner by press molding or other processes is easy, as in Embodiment 6, and the workability is improved.

Embodiment 11

[0124] With reference to Figs. 22 to 25, a configuration of the heat exchanger 10 according to Embodiment 11 will be described. Figs. 22 to 25 are explanatory views illustrating configurations of modifications of the heat transfer tubes 20 for use in at least one of the first heat exchanger 11 and the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 11. In the following description, the first heat transfer tubes 20a and the second heat transfer tubes 20b are collectively referred to as the heat transfer tubes 20. Embodiment 11 will be described mainly on points on which Embodiment 11 differs from Embodiments 1 to 10. Regarding Embodiment 11, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiments 1 to 10 will be omitted.

[0125] The above descriptions concerning Embodiments 1 to 10 are made by referring to by way of example the case where the heat transfer tubes 20 are each a flat tube or a porous flat tube. However, the heat transfer tube 20 is not limited to such a tube, but may be a circular

tube, an elliptical tube, or a tube formed in the shape of a square tube having a polygonal cross-section.

[0126] Furthermore, although the above descriptions concerning Embodiments 1 to 10 refer to extrusion and drawing using a die as methods of manufacturing the heat transfer tubes 20. However, the descriptions referring to such methods are not limiting, and the heat transfer tubes 20 may be each manufactured by bonding a plurality of members together.

[0127] In the modification as illustrated in Fig. 22, two irregular-shaped members 200 are bonded together to form the heat transfer tube 20. Recessed portions 200a of the irregular-shaped members 200 each have a semicircular shape. In addition, linear portions 200b are provided between adjacent ones of the recessed portions 200a. As illustrated in Fig. 22, (a), the recessed portions 200a of the two members 200 are provided to face each other, and the linear portions 200b of the two members 200 are moved in respective directions indicated by black arrows in Fig. 22, (a) and are bonded together. As a result, a heat transfer tube 20 having a plurality of the refrigerant passages 21, as illustrated in Fig. 22, (b), is formed. As material of the members 200, for example, aluminum, an aluminum alloy, copper, or a copper alloy is used.

[0128] In the modification as illustrated in Fig. 23, two irregular-shaped members 201 are bonded together to form a heat transfer tube 20. Recessed portions 201a of each of the irregular-shaped members 201 each have a polygonal shape. In addition, between adjacent ones of the recessed portions 201a, linear portions 201b are provided. In this case, as illustrated in Fig. 23, (a), the recessed portions 201a of the two irregular-shaped members 201 are located to face each other, and the linear portions 201b of the two irregular-shaped members 201 are moved in respective directions indicated by black arrows in Fig. 23, (a) and are bonded together. As a result, a heat transfer tube 20 having a plurality of refrigerant passages 21, as illustrated in Fig. 23, (b), is formed. As material of the members 201, for example, aluminum, an aluminum alloy, copper, or a copper alloy is used.

[0129] In the modification as illustrated in Fig. 24, an irregular-shaped member 200 and circular tubes 203 are bonded together to form the heat transfer tube 20. The irregular-shaped member 200 corresponds to the member 200 as illustrated in Fig. 22. In this case, as illustrated in Fig. 24, (a), the circular tubes 203 are provided at positions associated with the positions of the respective recessed portions 200a of the member 200, and the recessed portions 200a of the member 200 and the circular tubes 203 are moved in respective directions indicated by black arrows in Fig. 24, (a) and are bonded together. As a result, a heat transfer tubes 20 having a plurality of refrigerant passages 21, as illustrated in Fig. 24, (b), is formed. The refrigerant passages 21 are internal spaces of the circular tubes 203. As material of the member 200, for example, aluminum, an aluminum alloy, copper, or a copper alloy is used.

[0130] In the modification as illustrated in Fig. 25, the

heat transfer tubes 2 each include a plurality of the first heat transfer tubes 20a. The first heat transfer tubes 20a correspond to the first heat transfer tubes 20a as illustrated in Fig. 15 and described as the modification of Embodiment 6. Therefore, the first heat transfer tubes 20a each have the first projection 51 and the second projection 52. In the modification as illustrated in Fig. 25, each of the heat transfer tubes 20 includes two first heat transfer tubes 20a. The heat transfer tube 20 is formed such that the first projection 51 of one of the first heat transfer tubes 20a and the second projection 52 of the other of the first heat transfer tubes 20a are connected to each other. The first projection 51 and the second projection 52 are bonded together by, for example, brazing or other processes. Although it is described above that the heat transfer tubes 20 each include the two first heat transfer tubes 20a, the heat transfer tube 20 may include three or more first heat transfer tubes 20a. As described above, in the modification as illustrated in Fig. 25, the heat transfer tubes 20 each include a heat transfer tube group including a plurality of heat transfer tubes that are connected to each other.

[0131] As described above, in the configurations in Embodiments 1 to 10, the irregular-shaped members 200 or 201 may be bonded together as illustrated in Fig. 22 or Fig. 23 to form the heat transfer tubes 20. Alternatively, as illustrated in Fig. 24, the heat transfer tubes 20 may each include the irregular-shaped member 200 and the circular tubes 20. Furthermore, alternatively, as illustrated in Fig. 25, the heat transfer tubes 20 may include a heat transfer tube group including a plurality of heat transfer tubes 20 that are connected together.

Embodiment 12

[0132] With reference to Figs. 2 and 26, a configuration of the heat exchanger 10 according to Embodiment 12 will be described. Fig. 26 is a sectional side-view illustrating an example of the configuration of the second heat exchanger 12 of the heat exchanger 10 according to Embodiment 12. Embodiment 12 will be described mainly on points on which Embodiment 12 differs from Embodiment 1. Regarding Embodiment 12, descriptions concerning configurations, operations, and advantages that are the same as those of Embodiment 1 will be omitted.

[0133] In Embodiment 12, as illustrated in Fig. 26, the corrugated fin 30 in the second heat exchanger 12 is provided only at part of the second heat transfer tube 20b that is adjacent to the second header portion 14.

[0134] In Embodiment 1, as illustrated in Fig. 5, in the second heat exchanger 12, the corrugated fin 30 is provided over the entire length of the second heat transfer tube 20b in the z-direction. In contrast, in Embodiment 12, as illustrated in Fig. 26, the corrugated fin 30 is provided only at part of the second heat transfer tubes 20b that is located on the lower side in the z-direction. Therefore, where L3 is the length of the second heat transfer tube 20b and L4 is the length of the corrugated fin 30 in

the z-direction, the relationship " $L3 > L4$ " is satisfied. L4 is set as appropriate to a value that falls within the range of, for example, 1/2 times the length L3 to 3/4 times the length L3. The length L3 is the length of part of the second heat transfer tube 20b in the z-direction that is located from the upper end of the second header portion 14 to the lower end of the inter-row connection portion 15.

[0135] In Embodiment 12, a region E of the second heat transfer tube 20b that is exposed is located adjacent to the inter-row connection portion 15, that is, on an upper side in the z-direction. At the region E of the second heat transfer tube 20b, the fin efficiency is increased and the dehumidification is promoted. It is therefore possible to further reduce the dehumidification load on the corrugated fin 30, as compared with Embodiment 1.

[0136] It should be noted that although the above description concerning Embodiment 12 is made with respect to the case where Embodiment 12 is applied to the configuration in Embodiment 1, Embodiment 12 is also applicable to Embodiments 2 to 11.

[0137] As described above, in Embodiment 12, the corrugated fin 30 is not provided over the entire length of the second heat transfer tube 20b, and is provided only at part of the second heat transfer tube 20b that is located on the lower side in the z-direction. Therefore, the region E of the second heat transfer tube 20b that is exposed is located on the upper side in the z-direction. At the region E of the second heat transfer tube 20b, the fin efficiency is increased and the dehumidification is promoted. It is therefore possible to reduce occurrence of blockage of the air passages that is caused by frost formation. Therefore, even when the amount of formed frost is considerably large, it is possible to prevent the air passages in the second heat exchanger 12 from being completely blocked. In addition, at the region E at which the corrugated fin 30 is not provided, an obstacle that blocks the flow of water is not present, as in the first heat exchanger 11. Thus, the speed of drainage is high, and the drainage is promoted. As described above, in Embodiment 12, because the region E at which the corrugated fin 30 is not provided is present, it is possible to further reduce occurrence of blockage of the air passages that is caused by frost formation, and possible to further improve the drainage performance, as compared with Embodiment 1.

[0138] The above descriptions concerning Embodiments 1 to 12 are made with respect to the case where the heat exchanger 10 is operated as an evaporator. On the other hand, when the heat exchanger 10 is operated as a condenser, the following advantages are obtained. As described above with reference to Fig. 1, when the refrigeration cycle apparatus 1 performs the defrosting operation, the refrigerant flows in the opposite direction to the flow direction of the refrigerant in the heating operation. At this time, the heat exchanger 10 operates as a condenser. When the heat exchanger 10 operates as a condenser, hot gas refrigerant flows from the compressor 2 into the second header portion 14 as illustrated in Fig. 2 in the opposite direction to the direction indicated

by the arrow in Fig. 2. The hot gas refrigerant distributed by the second header portion 14 and flowing in the second heat transfer tubes 20b gradually defrosts even a small amount of frost adhering to the second heat transfer tubes 20b, and meltwater of the frost flows downward along the corrugated fin 30 in the direction of gravity. Since the corrugated fin 30 is provided with the louvers 31, drainage is promoted. Thereafter, the refrigerant flows into the first heat transfer tubes 20a of the first heat exchanger 11 through the inter-row connection portion 15 as illustrated in Fig. 2. Since the first heat exchanger 11 is provided with no fin, the speed of drainage of meltwater of the frost is high. As described above, since the first heat exchanger 11 in which the speed of drainage is high is provided on the downstream side in a direction in which the hot gas refrigerant flows during defrosting, it is possible to reduce water that remains after frost at the entire heat exchanger 10 melts, and it is thus efficient. It should be noted if the second heat exchanger 12 is provided on the downstream side in the direction in which the hot gas refrigerant flows, time for draining meltwater of frost after melting of the frost ends is required. This is inefficient since time required until the defrosting operation ends is increased. Regarding the airflow, it should be noted that in the case where the heat exchanger 10 operates as a condenser, the airflow also flows as indicated by the arrow B in Fig. 2 or 18 such that the first heat exchanger 11 is located on the windward side and the second heat exchanger 12 is located on the leeward side, as in the case where the heat exchanger 10 operates as an evaporator. Thus, a counterflow in which the airflow direction and the flow direction of refrigerant are opposite to each other occurs. Therefore, the airflow heated through heat exchange at the first heat exchanger 11 flows toward the second heat exchanger 12. The hot gas refrigerant that has flowed into the second heat exchanger 12 exchanges heat with heated airflow through heat exchange at the second heat exchanger 12, and is condensed to change into liquid refrigerant or two-phase gas-liquid refrigerant. At this time, since the airflow flowing as the counterflow is heated, a temperature difference between the airflow and the refrigerant is small. Thus, the heat exchange efficiency of the condensation operation is increased, and an energy saving effect is obtained. In Embodiments 1 to 12, the heat exchange amount of the first heat exchanger 11 and the heat exchange amount of the second heat exchanger 12 are set to differ each other in accordance with the phase change of the refrigerant in a refrigerant passage extending from the second header portion 14 which serves as a refrigerant inlet to the first header portion 13 which serves as a refrigerant outlet. Therefore, the size of the first heat exchanger 11 or the second heat exchanger 12 whose heat exchange amount is smaller can be reduced depending on the heat exchange amount. In this case, it is possible to reduce the weight of the entire heat exchanger 10. As described above, in Embodiments 1 to 12, the amount of water remaining on the surface of the entire

heat exchanger 10 is reduced at the time of ending the defrosting operation, at which formed frost is completely melted. It is therefore possible to restart the heating operation in a state in which almost no water remains in the heat exchanger 10. If the heating operation is restarted in a state in which water remains in the heat exchanger 10, the remaining water is re-condensed to change into frost after the restart of the heating operation, this causes blockage of the air passages of the second heat exchanger 12, and the heat-exchanging performance of the entire heat exchanger 10 deteriorates. In Embodiments 1 to 12, since the heating operation can be restarted in a state in which no water remains in the heat exchanger 10, it is possible to prevent deterioration of the heat-exchanging performance of the entire heat exchanger 10.

[0139] The above description is made with respect to the case where the heat exchangers 10 according to each of Embodiments 1 to 12 are each used as an outdoor heat exchanger. However, needless to say, the configurations of the heat exchanger 10 according to Embodiments 1 to 12 are also applicable to the indoor heat exchanger 3 as illustrated in Fig. 1. In other words, the heat exchangers 10 according to Embodiments 1 to 12 are each also usable as the indoor heat exchanger 3 as illustrated in Fig. 1. In this case also, the same advantages can be obtained.

Reference Signs List

[0140] 1: refrigeration cycle apparatus, 2: compressor, 3: indoor heat exchanger, 4: indoor fan, 5: expansion device, 6: outdoor fan, 7: four-way valve, 10: heat exchanger, 11: first heat exchanger, 12: second heat exchanger, 13: first header portion, 13a: lower first header portion, 13b: upper first header portion, 14: second header portion, 14a: lower second header portion, 14b: upper second header portion, 15: inter-row connection portion, 15a: insertion hole, 15b: insertion hole, 15c: space, 16: inter-row connection pipe, 20: heat transfer tube, 20a: first heat transfer tube, 20b: second heat transfer tube, 21: refrigerant passage, 22a: front edge portion, 22b: front edge portion, 23a: rear edge portion, 23b: rear edge portion, 30: corrugated fin, 31: louver, 31a: projection, 32: front edge portion, 33: rear edge portion, 40a: connection portion, 40b: connection portion, 51: first projection, 52: second projection, 70: dashed line, 71: dotted line, 72: dryout occurrence region, 73: dryout occurrence region, 74: thick solid line, 75: dryout occurrence region, 200: member, 200a: recessed portion, 200b: linear portion, 201: member, 201a: recessed portion, 201b: linear portion, 203: circular tube, 300: plate-shaped member, 301: bent portion, 302: fin body portion, 303: base portion

Claims

1. A heat exchanger comprising:

- a first heat exchanger configured to cause heat exchange to be performed between air and refrigerant; and
 a second heat exchanger provided in series with the first heat exchanger in a first direction that is a flow direction of the air, and configured to cause heat exchange to be performed between the air and the refrigerant,
 wherein the first heat exchanger includes a plurality of first heat transfer tubes through which the refrigerant flows, the plurality of first heat transfer tubes being spaced apart from each other in a second direction that intersects the first direction, the plurality of first heat transfer tubes each having a tube axis extending in a third direction that intersects each of the first direction and the second direction, and wherein the second heat exchanger includes
- a plurality of second heat transfer tubes through which the refrigerant flows, the plurality of second heat transfer tubes being spaced apart from each other in the second direction, the plurality of second heat transfer tubes each having a tube axis extending in the third direction, and
 a corrugated fin provided between the plurality of second heat transfer tubes.
2. The heat exchanger of claim 1,
 wherein the first heat exchanger is provided at a front row on a windward side in the first direction, and
 wherein the second heat exchanger is located at a rear row on a leeward side in the first direction.
 3. The heat exchanger of claim 1 or 2, wherein the third direction is a vertical direction.
 4. The heat exchanger of any one of claims 1 to 3, wherein the first heat exchanger is a finless heat exchanger in which no heat transfer member is provided between the plurality of first heat transfer tubes.
 5. The heat exchanger of any one of claims 1 to 4,
 wherein the second heat exchanger is a corrugated-fin heat exchanger including the corrugated fin, and
 wherein the corrugated fin is formed of a wavy plate-shaped member that has fin body portions and a bent portion, the fin body portions being spaced apart from each other in the third direction and extending in the second direction, the bent portion connecting adjacent two of the fin body portions to each other and extending in the third direction.
 6. The heat exchanger of any one of claims 1 to 5, wherein at least part of the first heat transfer tubes is aligned with the corrugated fin of the second heat exchanger in the first direction.
 7. The heat exchanger of any one of claims 1 to 6,
 wherein the plurality of first heat transfer tubes are arranged at a first pitch PP1 in the second direction,
 wherein the plurality of second heat transfer tubes are arranged at a second pitch PP2 in the second direction, and
 wherein the first pitch PP1 is smaller than the second pitch PP2.
 8. The heat exchanger of claim 5, or claim 6 or claim 7 as dependent on claim 5,
 wherein the plurality of first heat transfer tubes are spaced apart from each other by a first distance PC1 in the second direction,
 wherein the fin body portions of the corrugated fin are spaced apart from each other by a third distance FC2 in the third direction, and
 wherein the first distance PC1 is greater than the third distance FC2.
 9. The heat exchanger of any one of claims 1 to 8, wherein each of the first heat transfer tubes of the first heat exchanger includes a first projection located at a front edge portion that is an end portion of the first heat transfer tube on a windward side, the first projection projecting from the front edge portion in the opposite direction to the first direction.
 10. The heat exchanger of any one of claims 1 to 9, wherein each of the first heat transfer tubes of the first heat exchanger includes a second projection located at a rear edge portion that is an end portion of the first heat transfer tube on a leeward side, the second projection projecting from the rear edge portion in the first direction.
 11. The heat exchanger of any one of claims 1 to 10, wherein the corrugated fin is provided only at part of each of the second heat transfer tubes that is located on a leeward side in the first direction.
 12. The heat exchanger of any one of claims 1 to 11, wherein the corrugated fin is located only at part of each of the second heat transfer tubes that is located on a lower side in the third direction.

FIG. 1

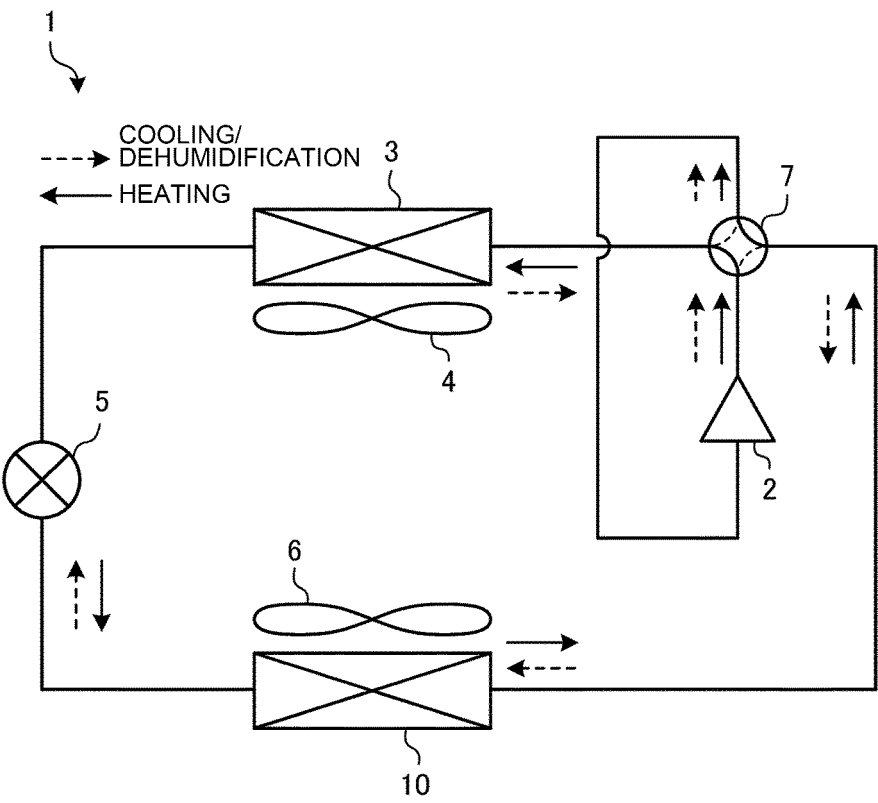


FIG. 2

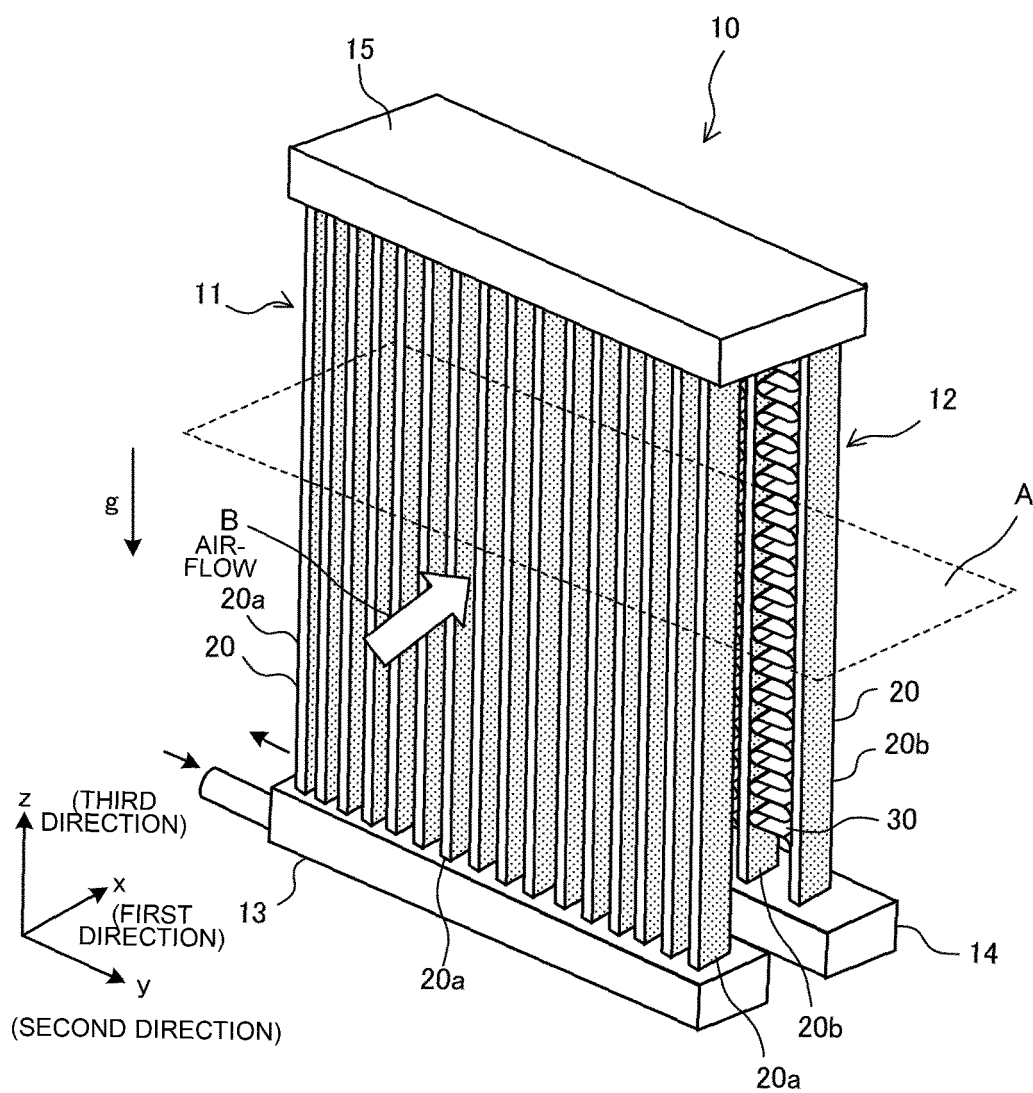


FIG. 3

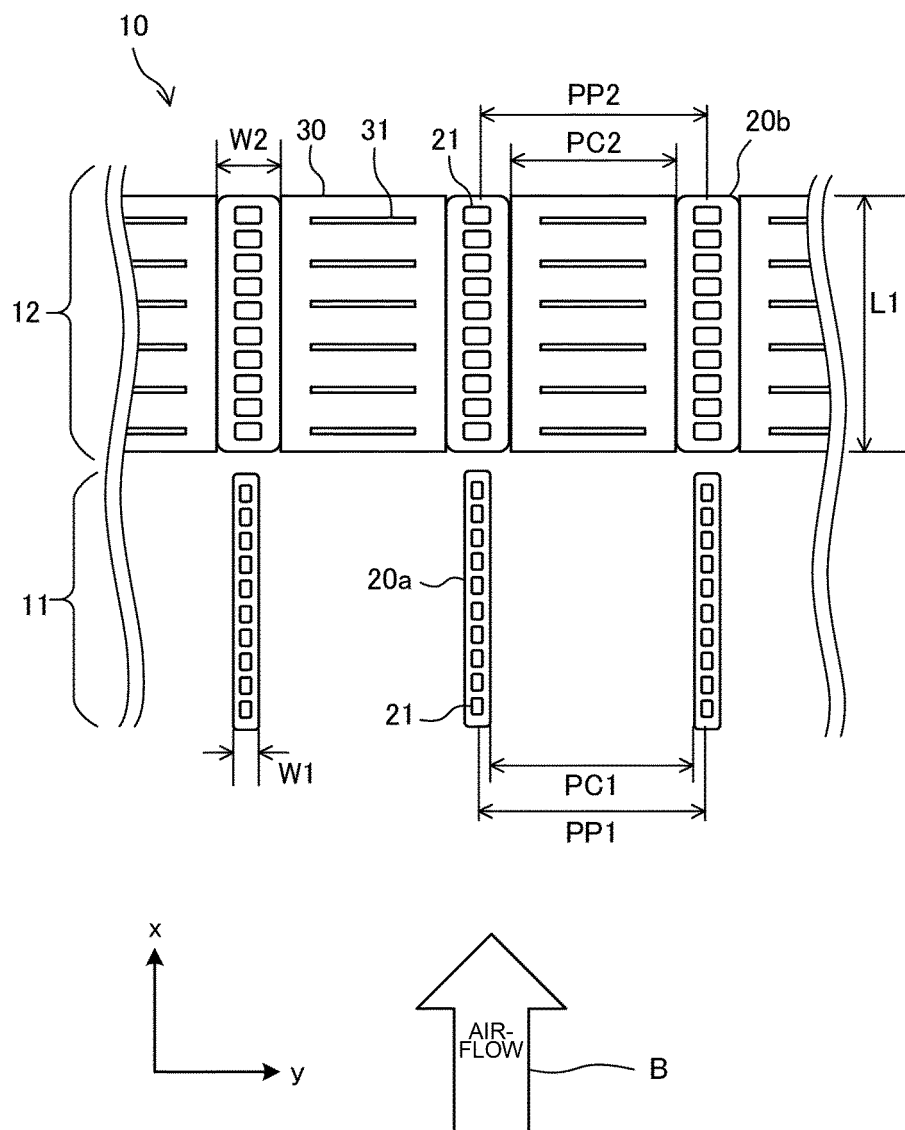


FIG. 4

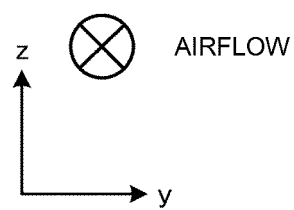
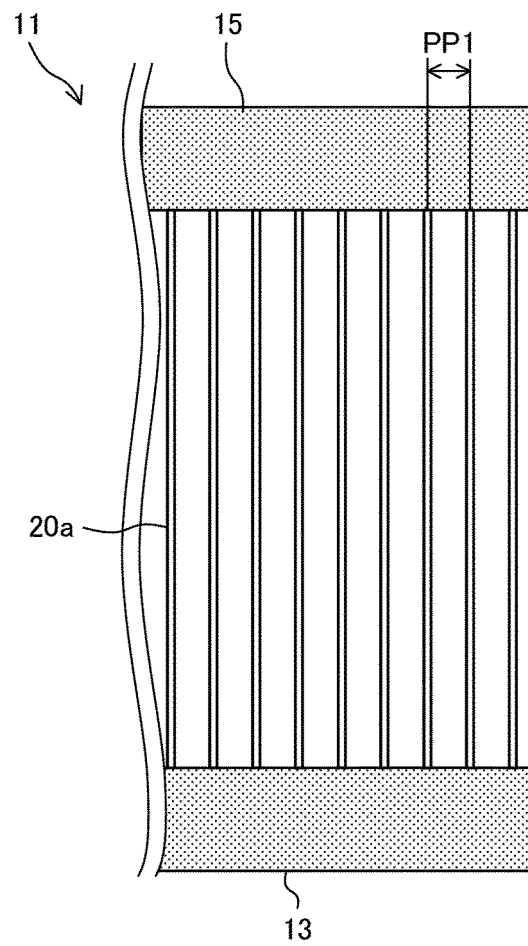


FIG. 5

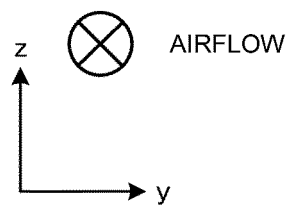
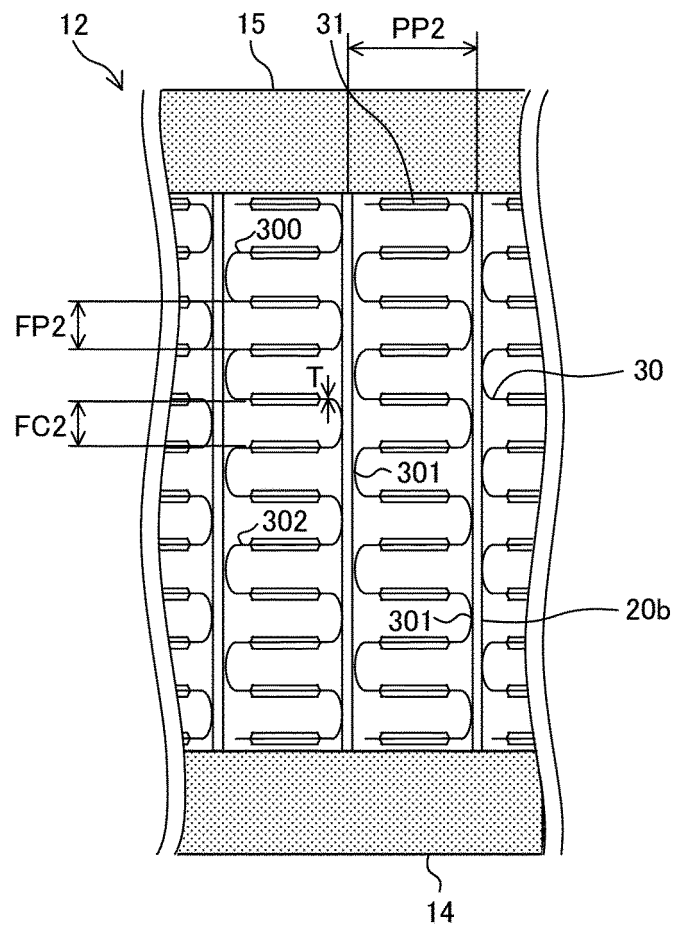
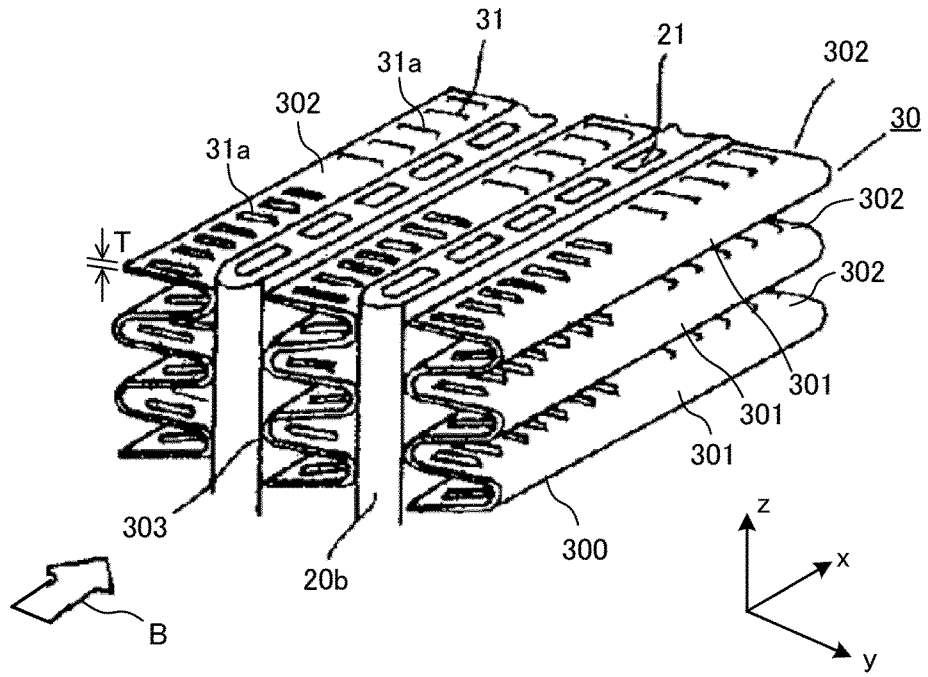


FIG. 6

(a)



(b)

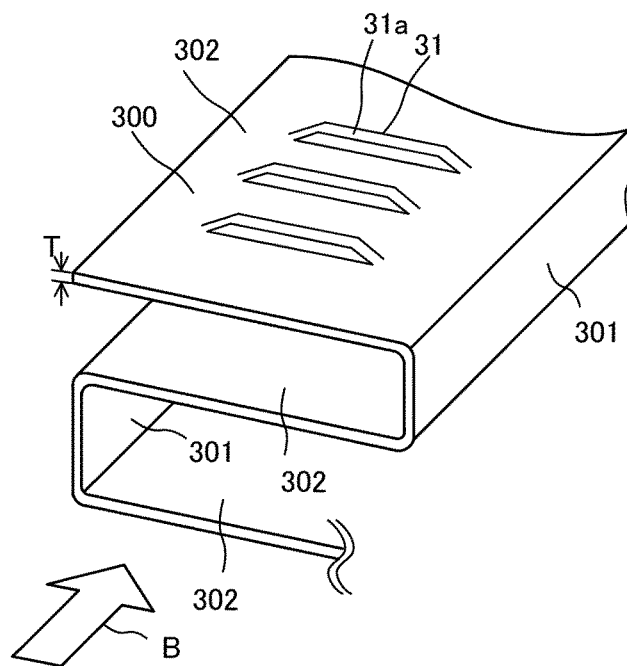


FIG. 7

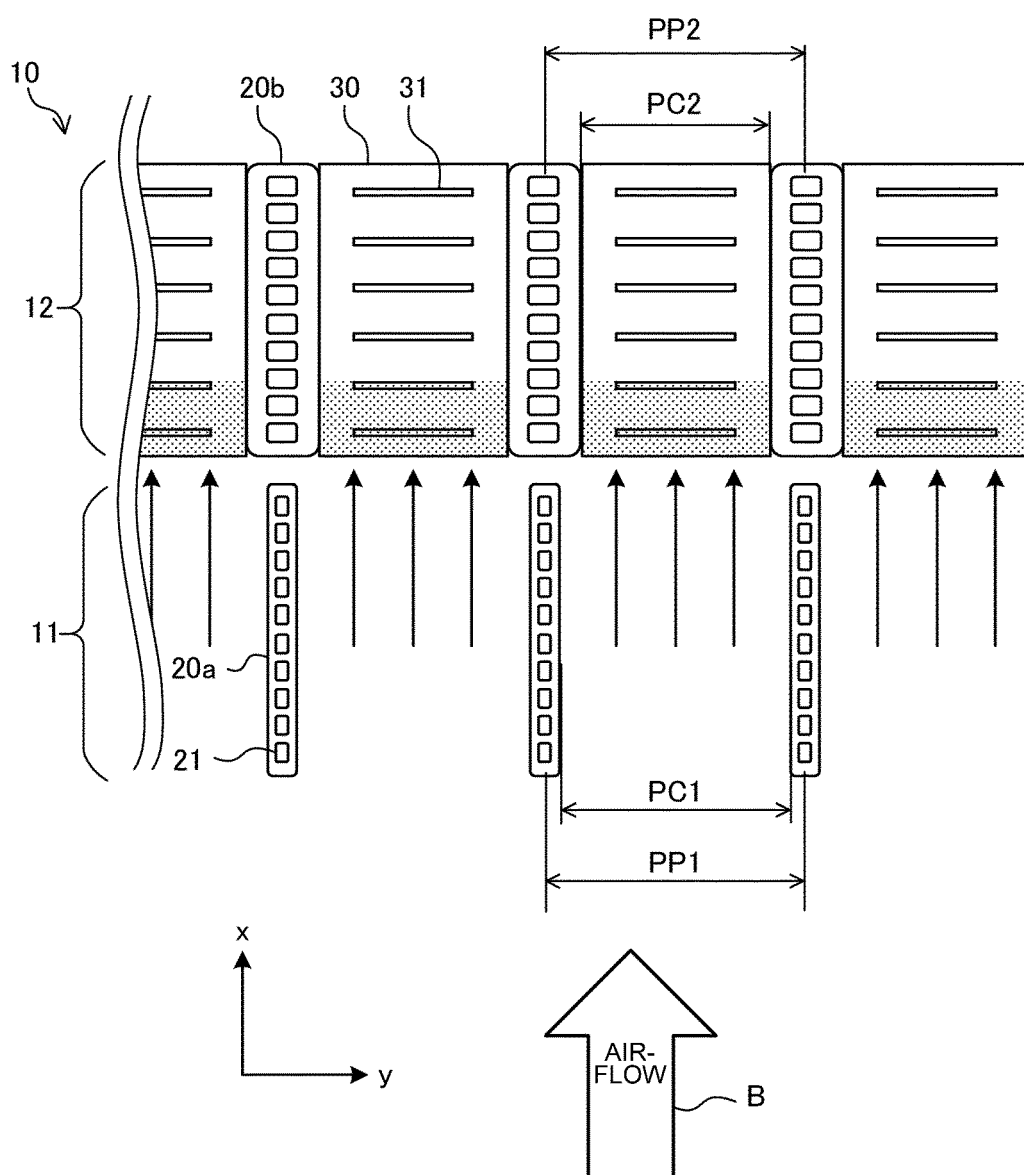


FIG. 8

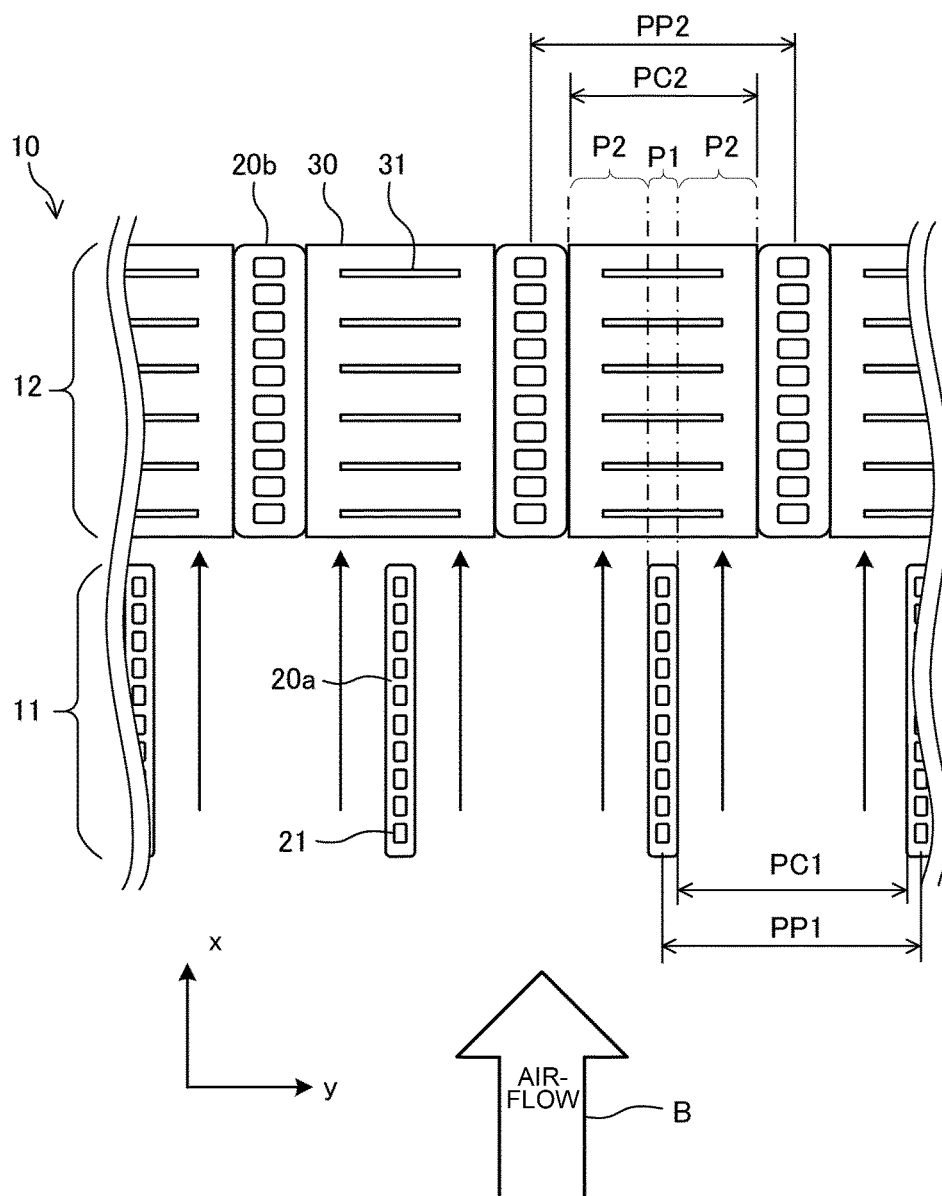


FIG. 9

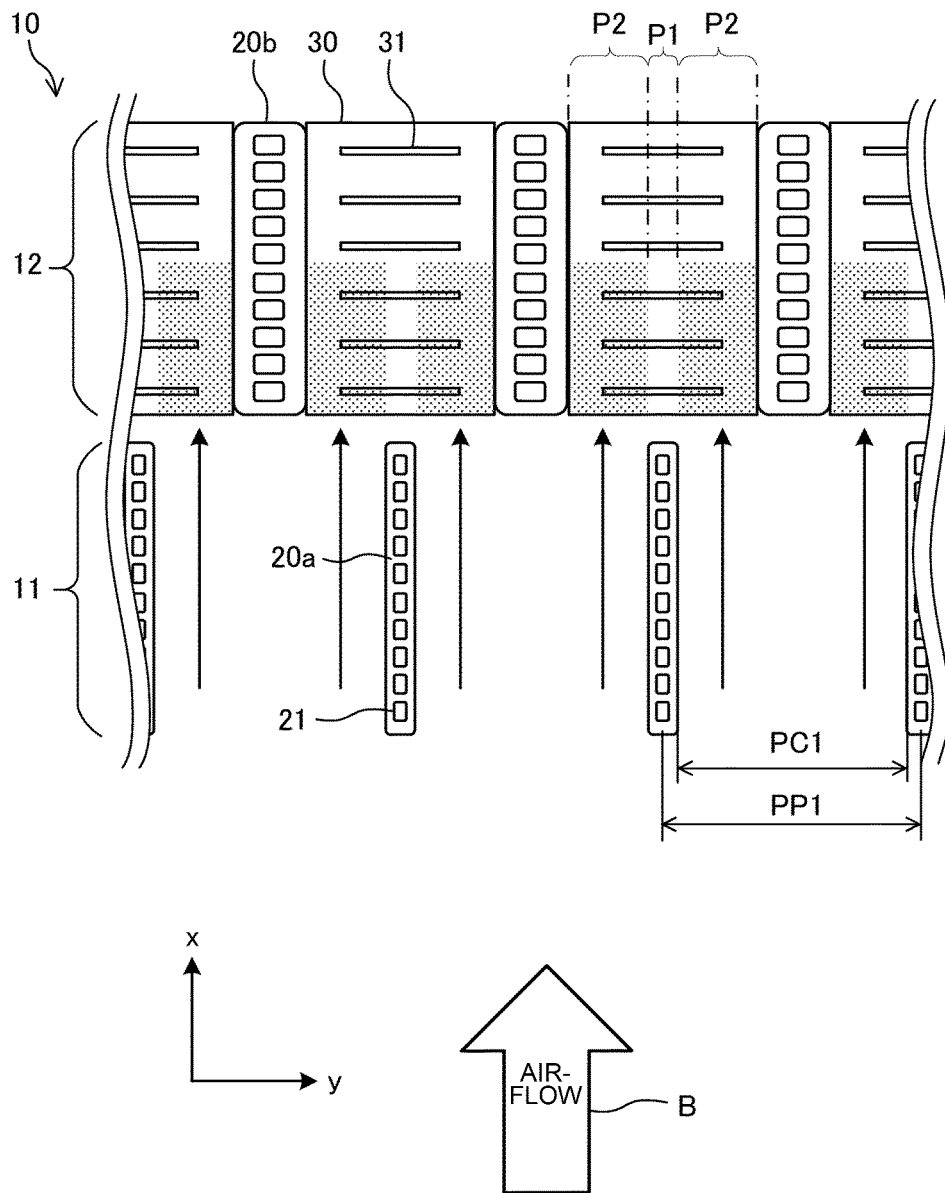


FIG. 10

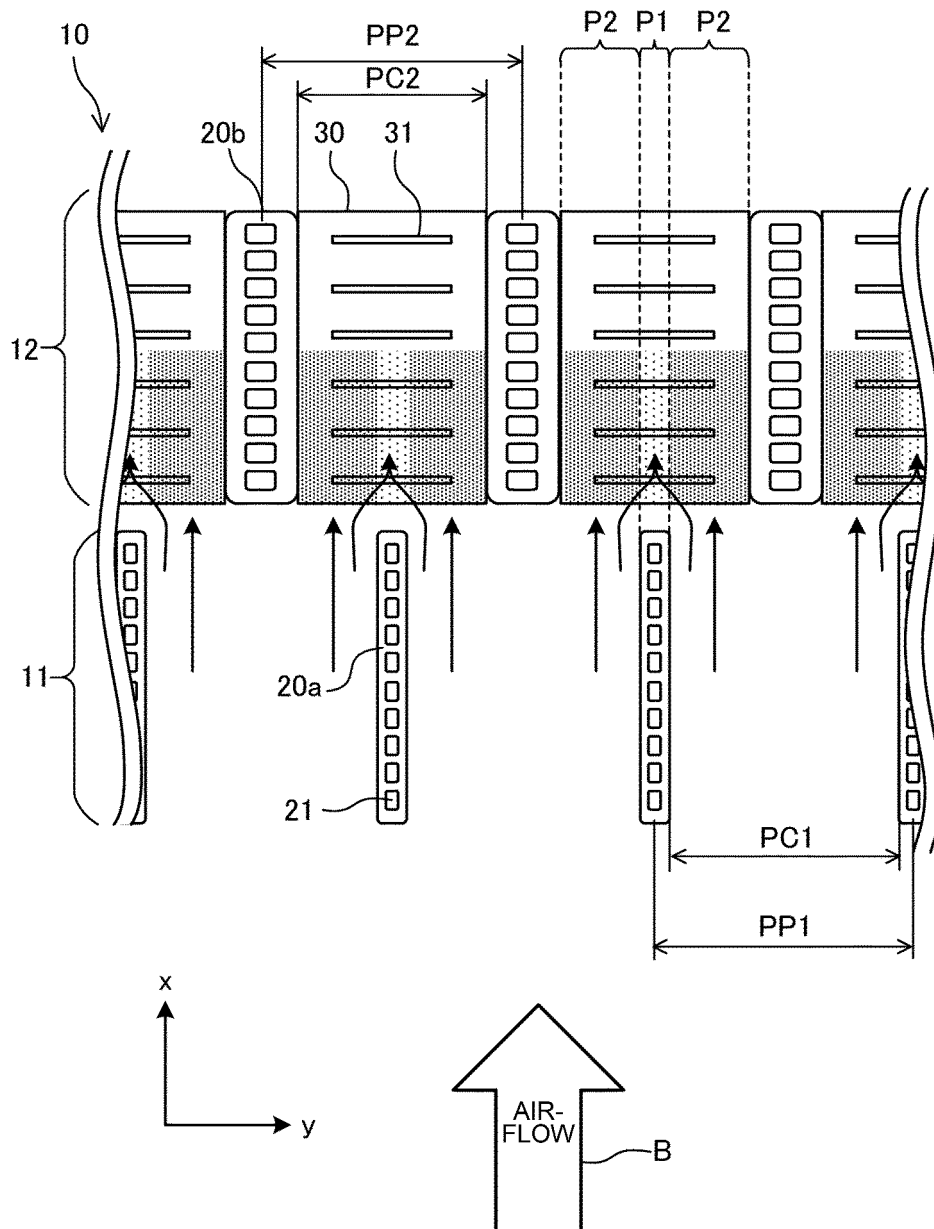


FIG. 11

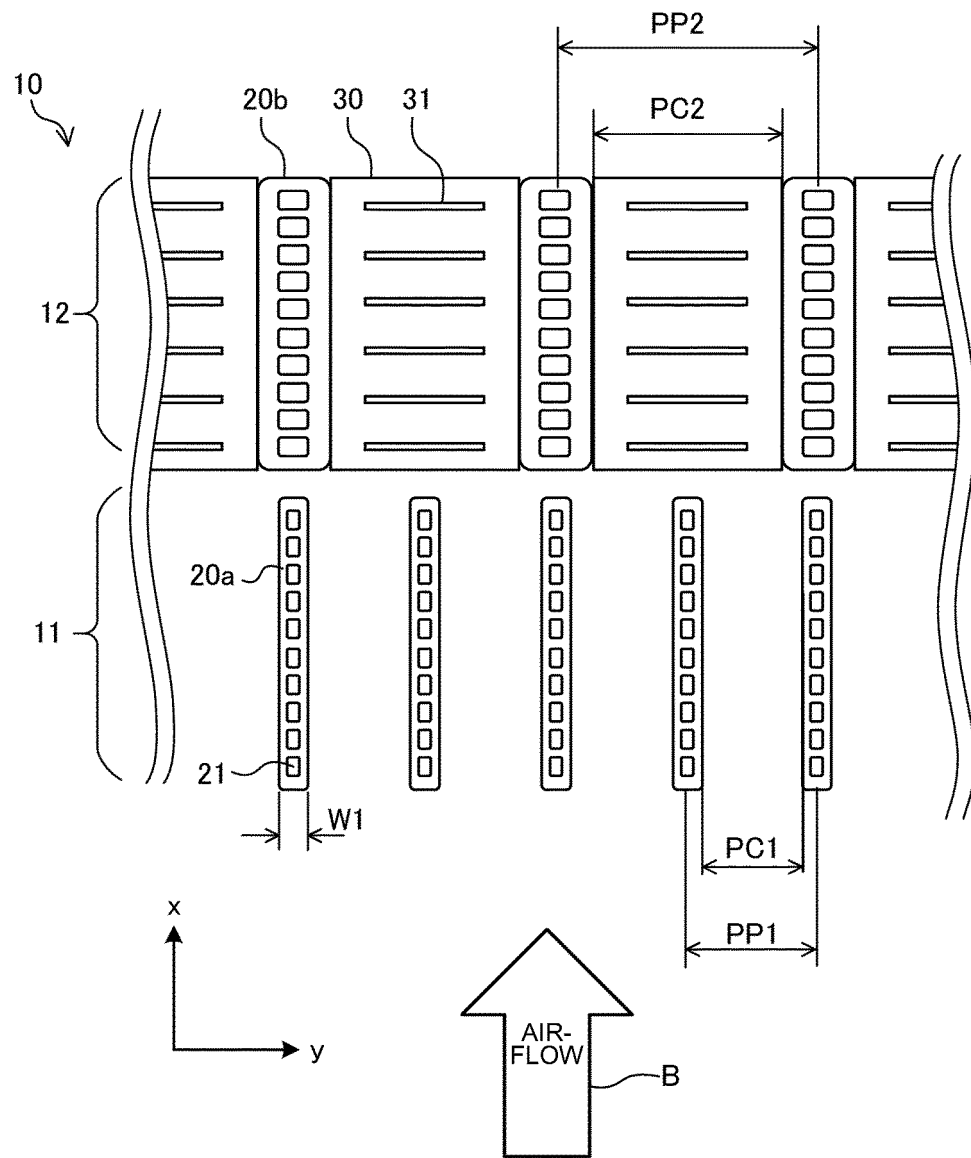


FIG. 12

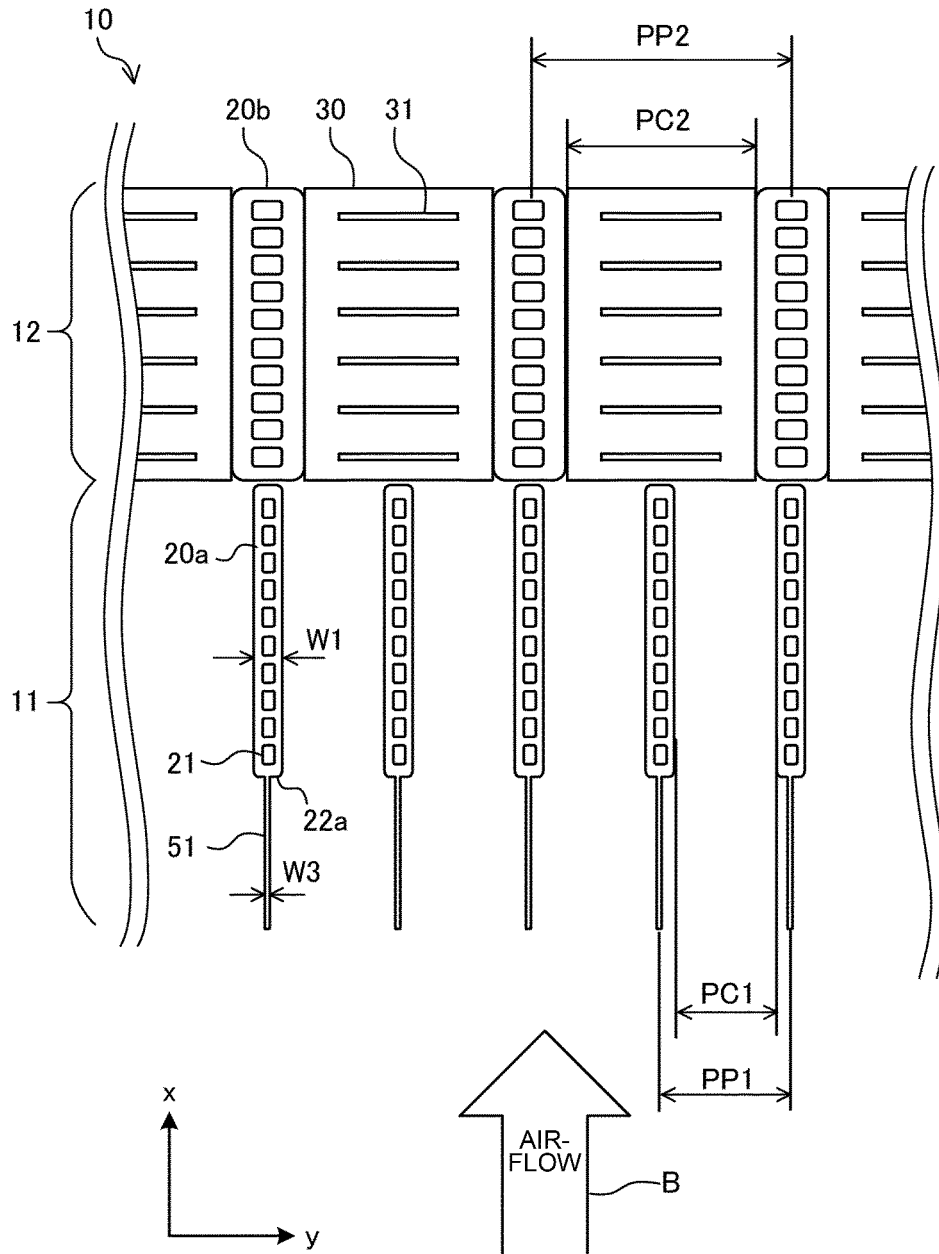


FIG. 13

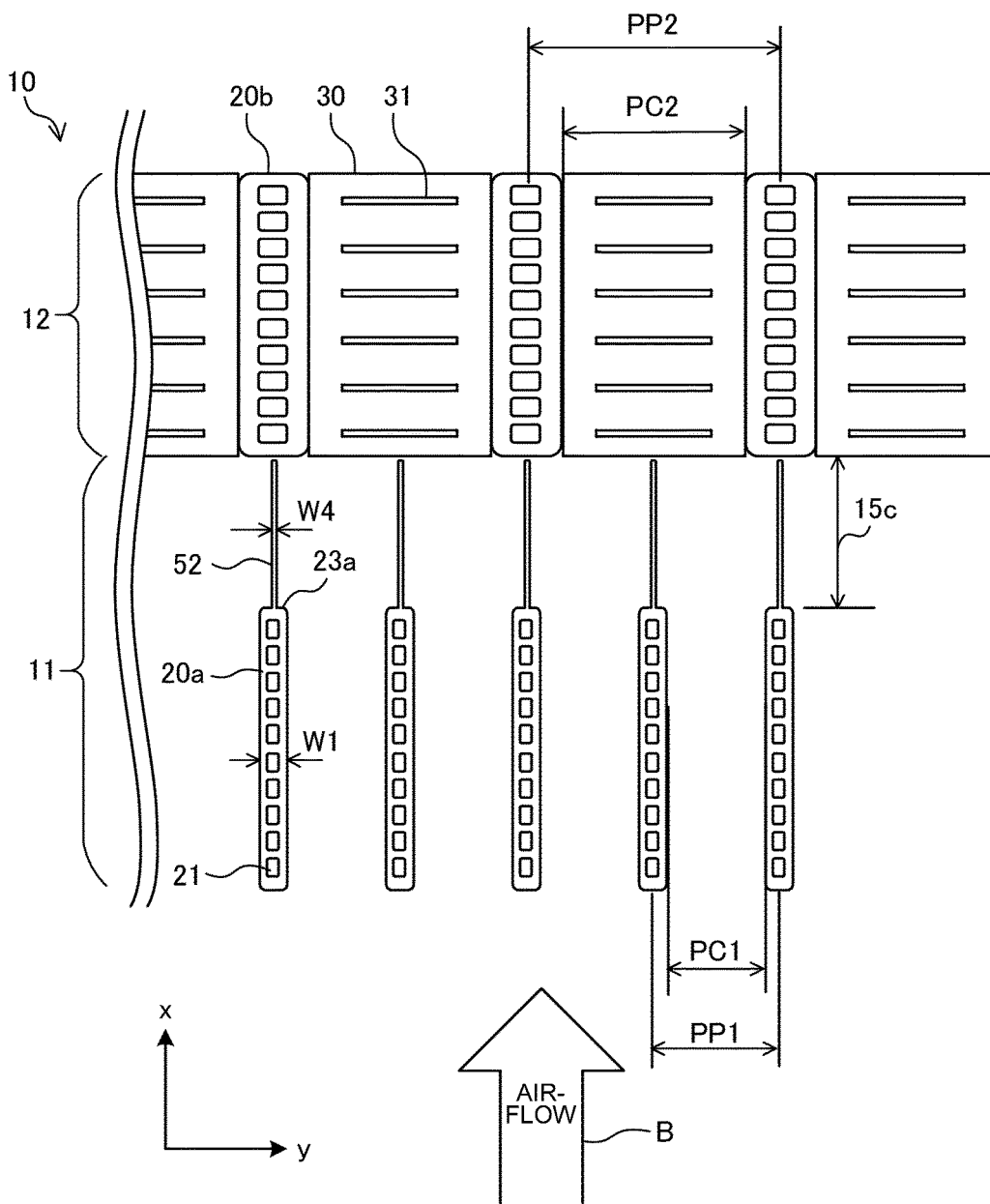


FIG. 14

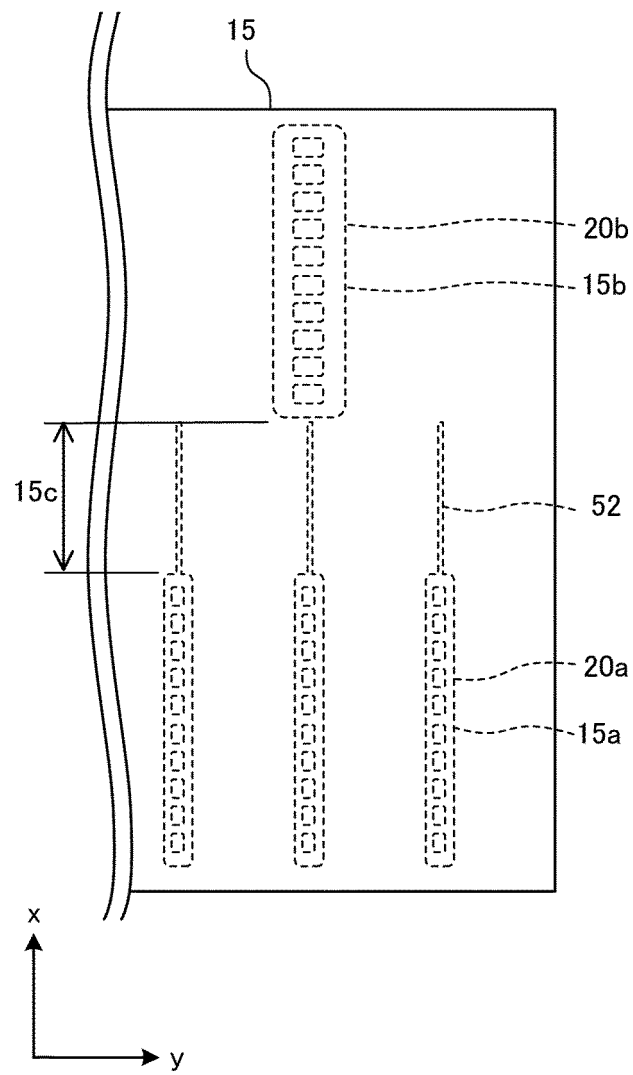


FIG. 15

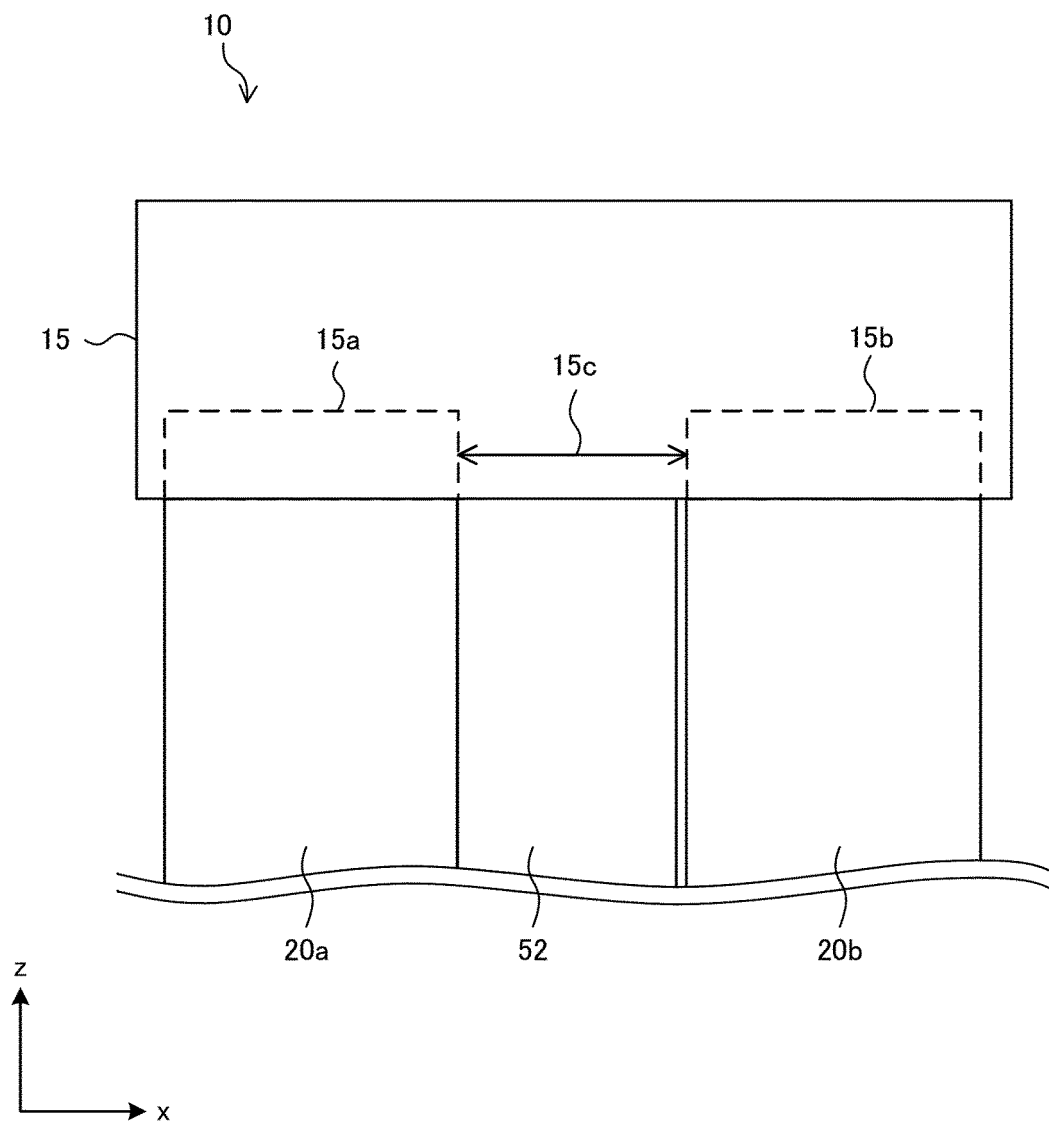


FIG. 16

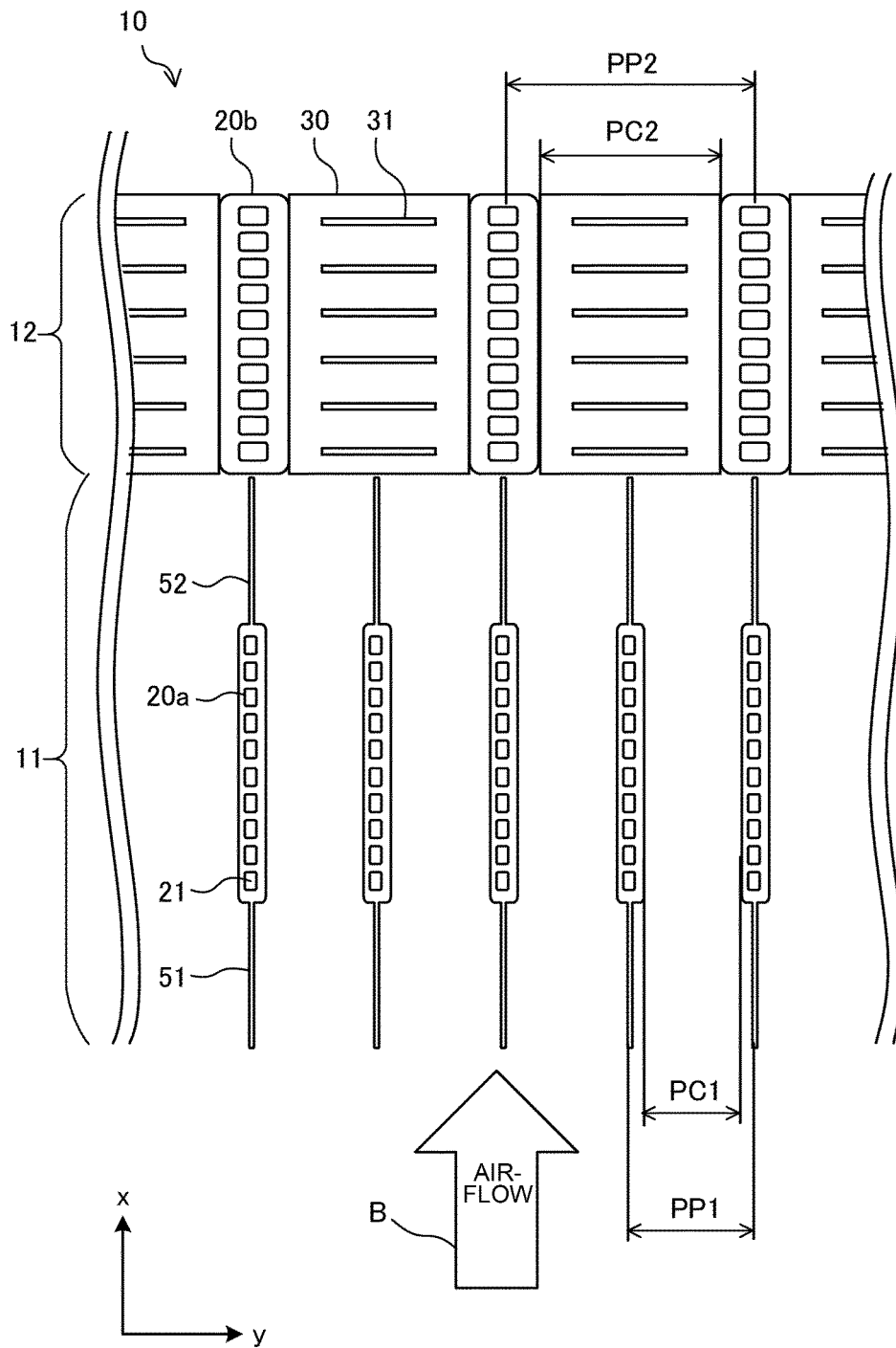


FIG. 17

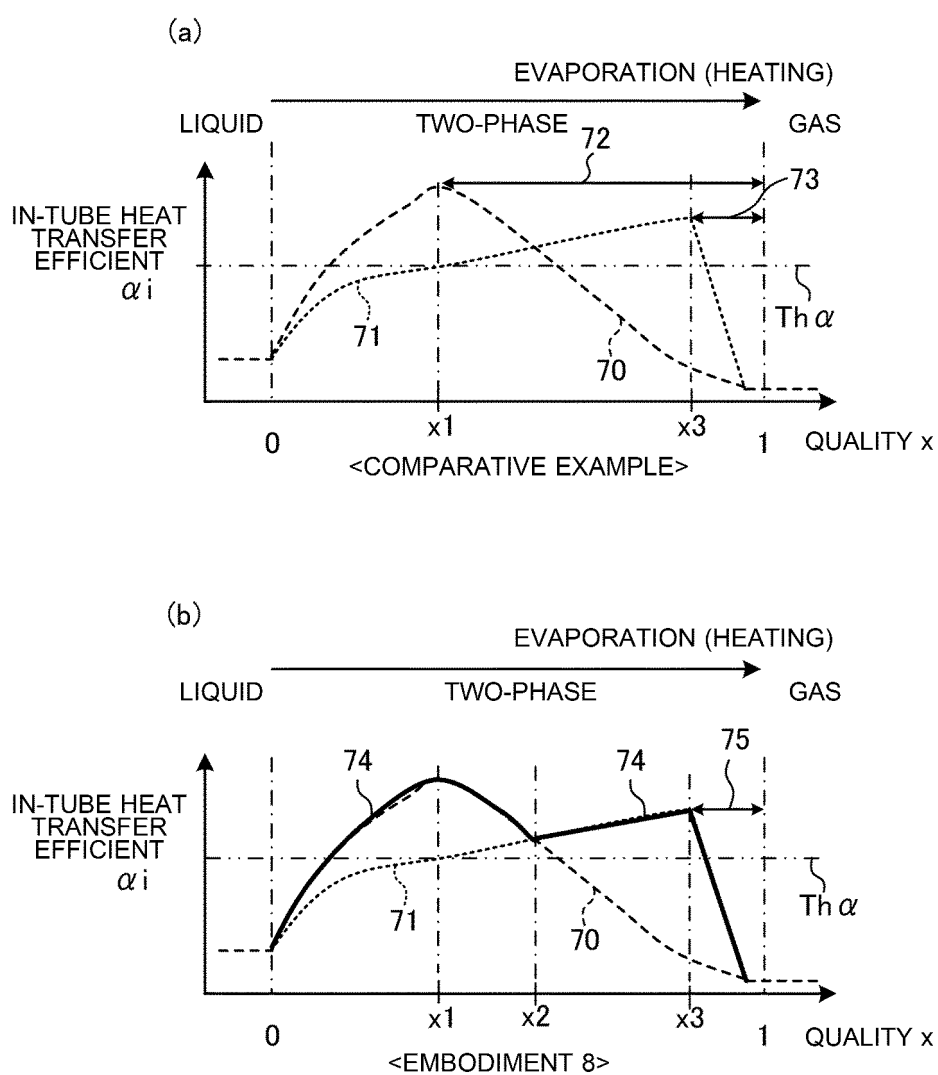


FIG. 18

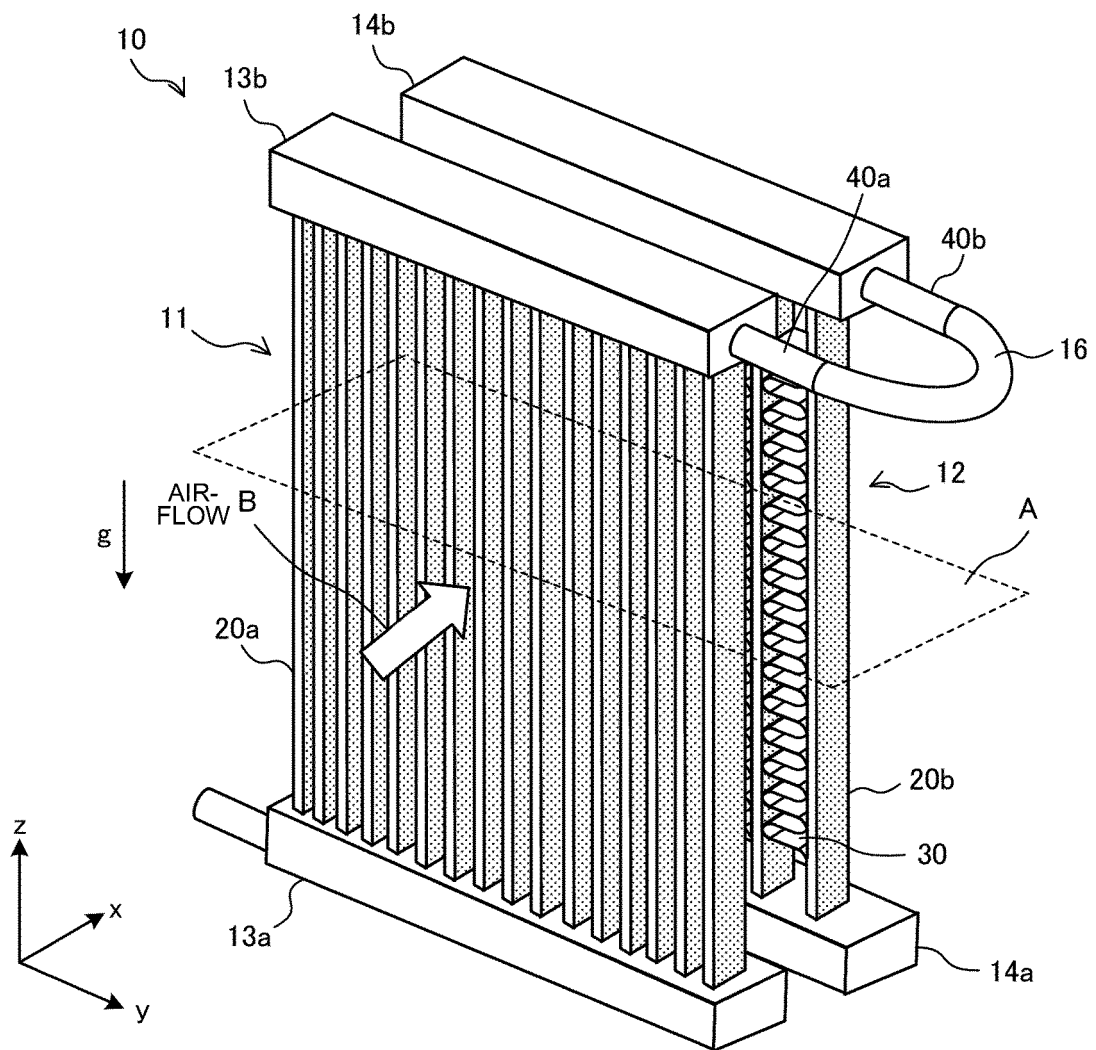


FIG. 19

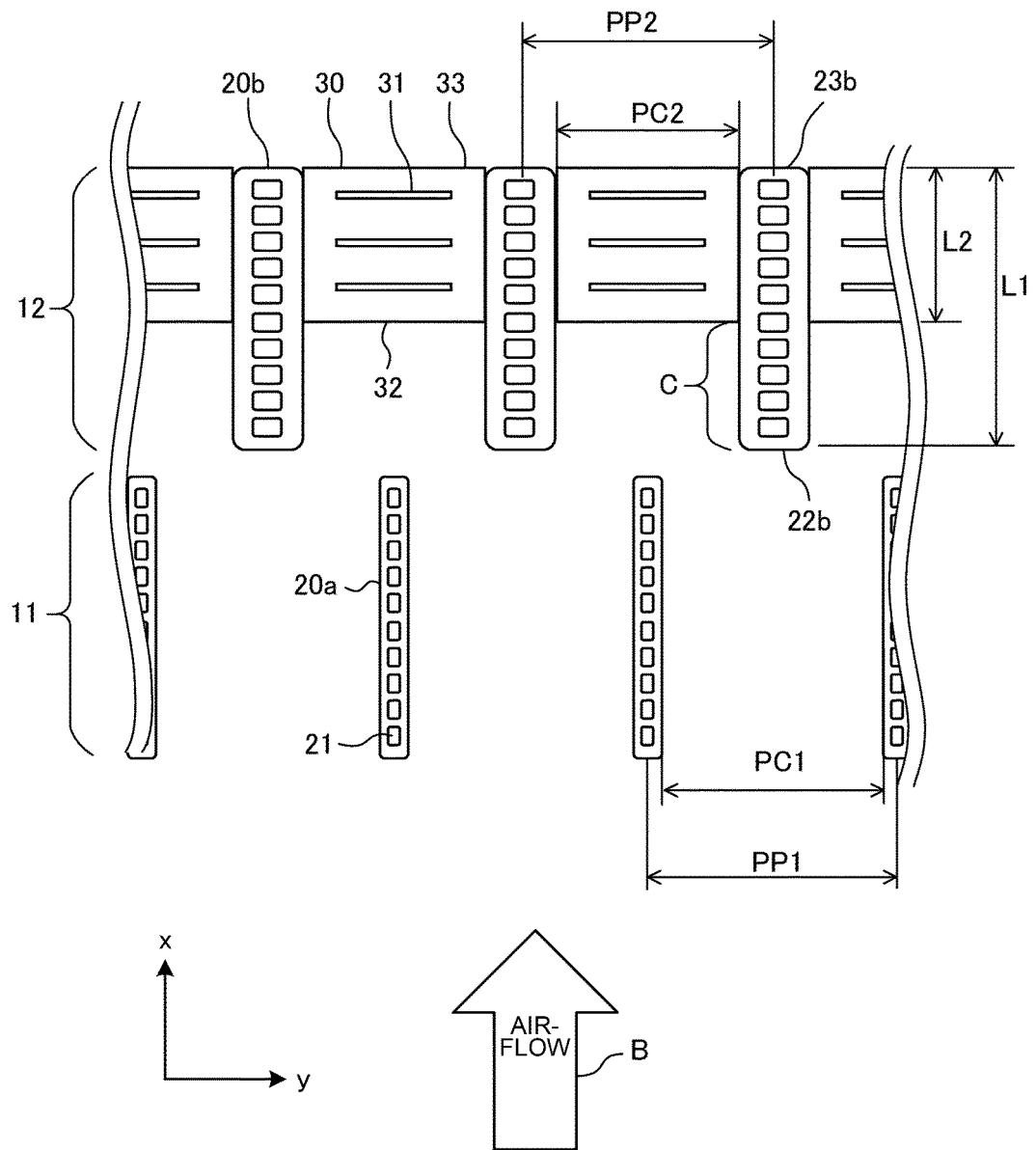


FIG. 20

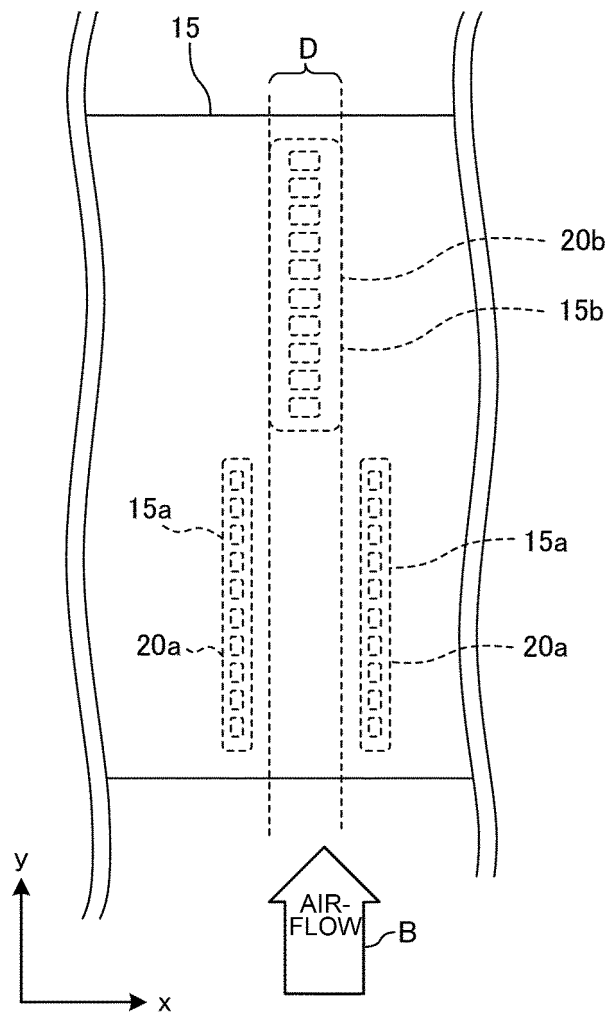


FIG. 21

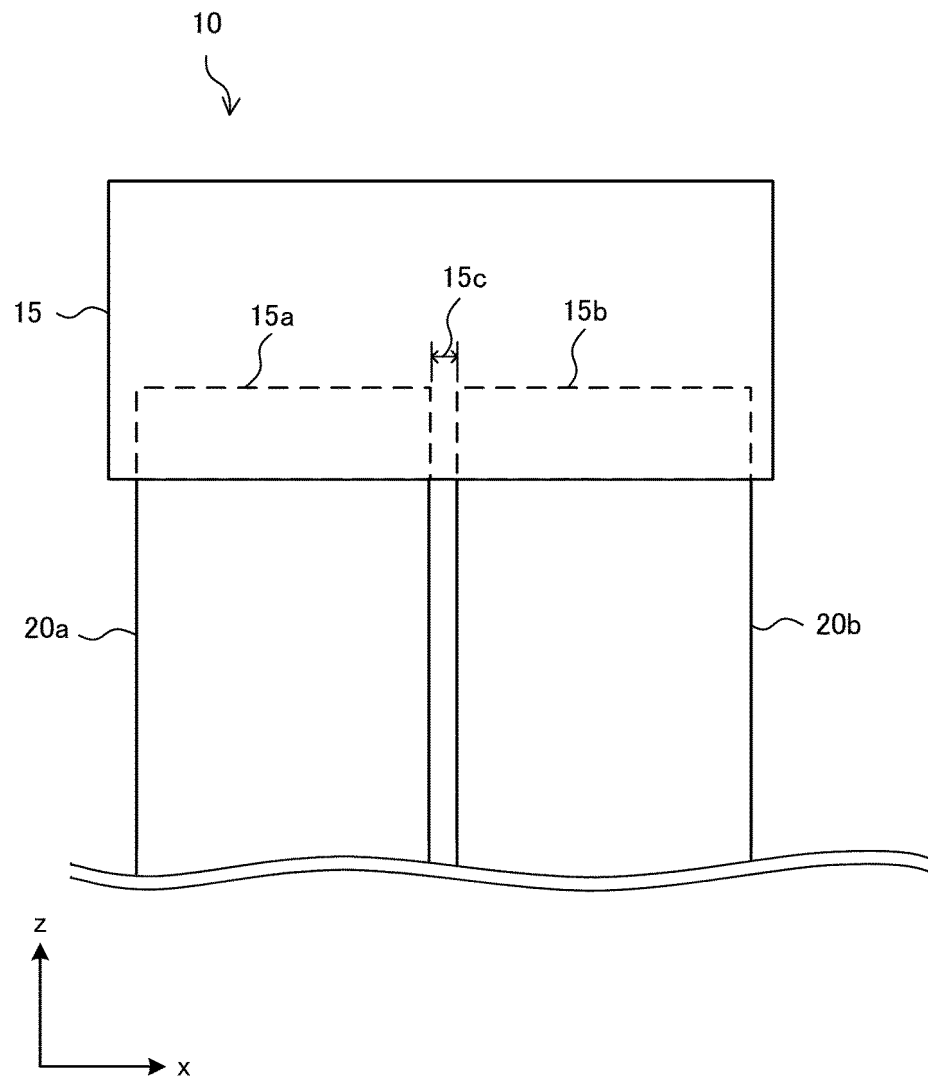


FIG. 22

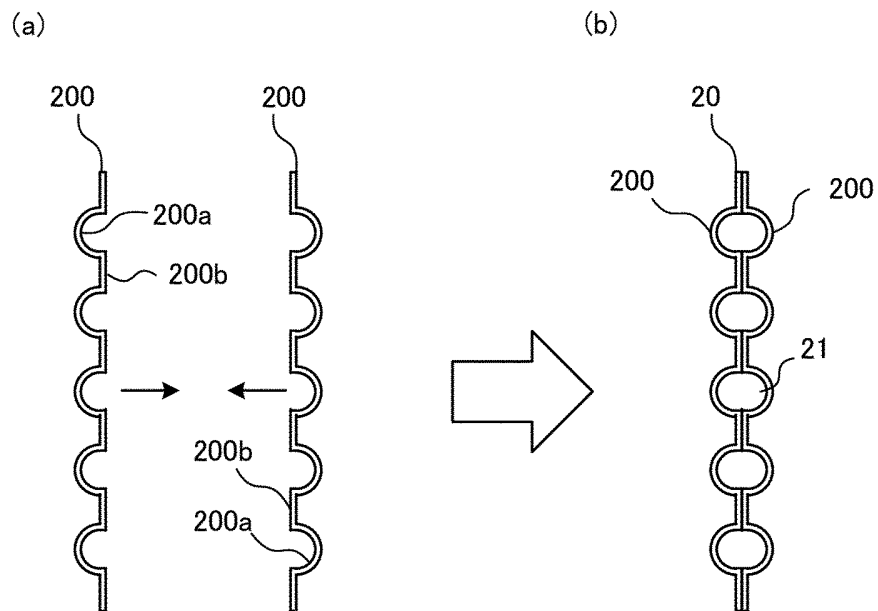


FIG. 23

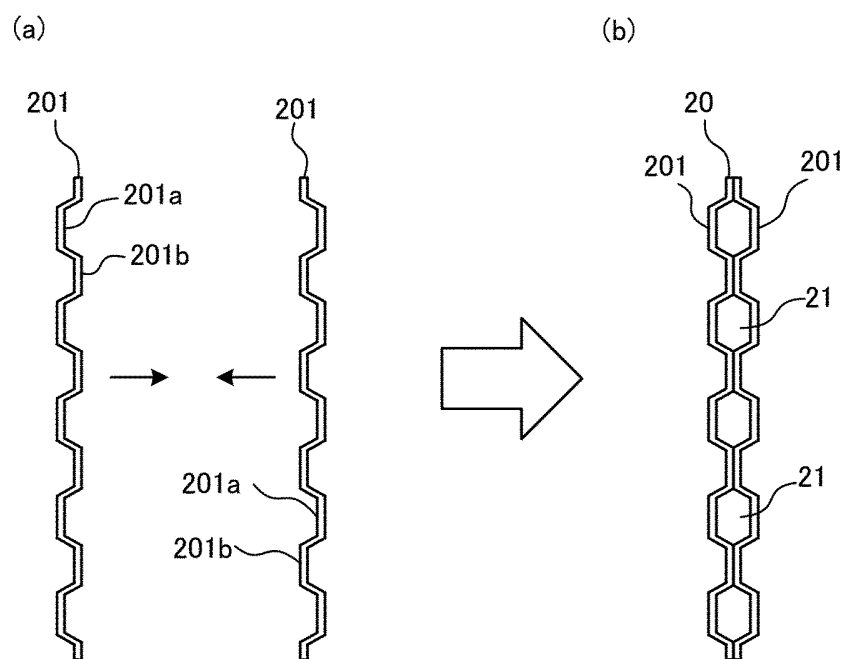


FIG. 24

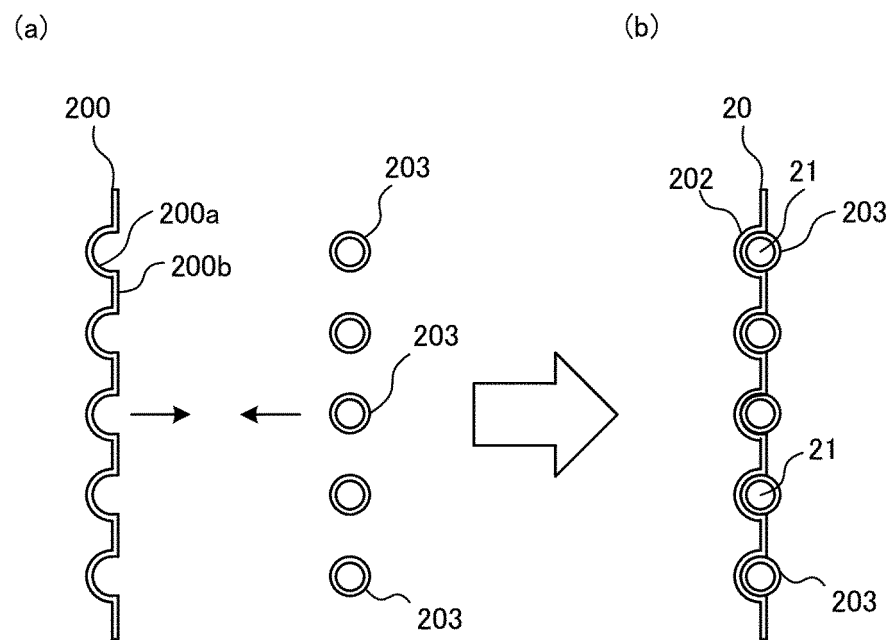


FIG. 25

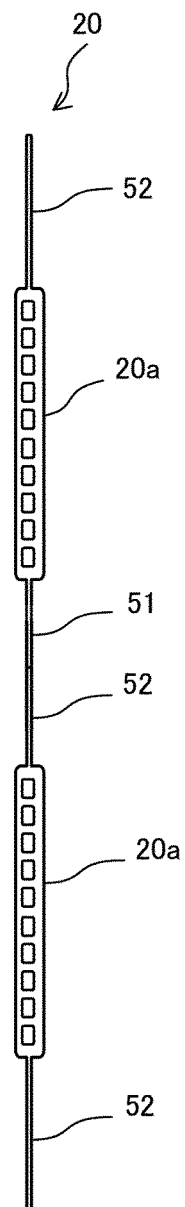
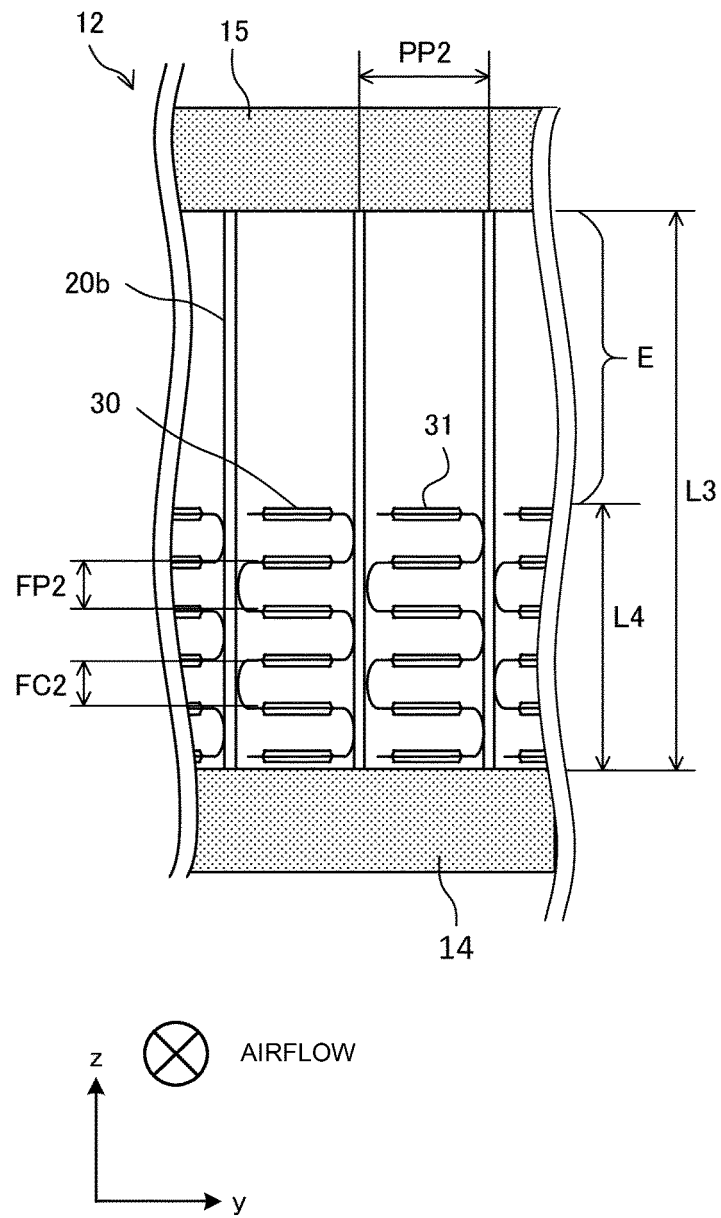


FIG. 26



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2021/016042

A. CLASSIFICATION OF SUBJECT MATTER

F28F 1/30(2006.01)i; F28D 1/053(2006.01)i; F28F 1/10(2006.01)i
FI: F28F1/30 A; F28F1/10 Z; F28D1/053 A

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F28F1/30; F28D1/053; F28F1/10

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

Published examined utility model applications of Japan 1922-1996

Published unexamined utility model applications of Japan 1971-2021

Registered utility model specifications of Japan 1996-2021

Published registered utility model applications of Japan 1994-2021

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y A	WO 2018/011873 A1 (MITSUBISHI ELECTRIC CORP) 18 January 2018 (2018-01-18) paragraphs [0014]-[0042], fig. 5-6	1-4, 6-7 5, 8-11 12
Y A	JP 2001-324290 A (DENSO CORP) 22 November 2001 (2001-11-22) paragraphs [0037]-[0041], fig. 2	5, 8 12
Y A	WO 2020/012549 A1 (MITSUBISHI ELECTRIC CORP) 16 January 2020 (2020-01-16) paragraphs [0036]-[0046], [0063], fig. 6	9-10 12
Y A	WO 2019/026243 A1 (MITSUBISHI ELECTRIC CORP) 07 February 2019 (2019-02-07) paragraphs [0068]- [0073], fig. 23	11 12
A	JP 2007-93023 A (SHOWA DENKO KK) 12 April 2007 (2007-04-12) paragraphs [0025]-[0031], fig. 2-3	5, 8, 12

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

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"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search
28 June 2021 (28.06.2021)Date of mailing of the international search report
13 July 2021 (13.07.2021)Name and mailing address of the ISA/
Japan Patent Office
3-4-3, Kasumigaseki, Chiyoda-ku,
Tokyo 100-8915, Japan

Authorized officer

Telephone No.

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2021/016042

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT		
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 9-159301 A (DENSO CORP) 20 June 1997 (1997-06-20) paragraphs [0038]-[0041], fig. 8	12

Form PCT/ISA/210 (continuation of second sheet) (January 2015)

INTERNATIONAL SEARCH REPORT
 Information on patent family members

International application no.

PCT/JP2021/016042

Patent Documents referred in the Report	Publication Date	Patent Family	Publication Date
WO 2018/011873 A1 JP 2001-324290 A	18 Jan. 2018 22 Nov. 2001	(Family: none) US 6339937 B1 column 6, line 65 to column 7, line 42, fig. 2 EP 1058070 A2 CN 1276507 A CN 1525120 A KR 10-2001-0007153 A	
WO 2020/012549 A1 WO 2019/026243 A1	16 Jan. 2020 07 Feb. 2019	(Family: none) US 2020/0217588 A1 paragraphs [0092]- [0097], fig. 23 EP 3663677 A1 CN 110998201 A	
JP 2007-93023 A	12 Apr. 2007	US 2007/0068661 A1 paragraphs [0036]- [0042], fig. 2-3 CN 1940450 A	
JP 9-159301 A	20 Jun. 1997	(Family: none)	

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2019002588 A [0005]