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(19) **United States**(12) **Patent Application Publication**
Katoh(10) **Pub. No.: US 2013/0284415 A1**(43) **Pub. Date: Oct. 31, 2013**(54) **REFRIGERANT RADIATOR**(75) Inventor: **Yoshiki Katoh**, Chita-gun (JP)(73) Assignee: **DENSO CORPORATION**, Kariya-city,
Aichi-pref. (JP)(21) Appl. No.: **13/997,895**(22) PCT Filed: **Dec. 27, 2011**(86) PCT No.: **PCT/JP2011/007297**

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(2013.01)

USPC **165/175**

(57)

ABSTRACT

A refrigerant radiator for a heat pump cycle includes a plurality of tubes. The tubes are disposed to satisfy the following relationship:

$$Re \geq A \times \theta^6 + B \times \theta^5 + C \times \theta^4 + D \times \theta^3 + E \times \theta^2 + F \times \theta + G$$

wherein θ is an inclination angle formed by a flow direction of the refrigerant flowing through the tubes and the horizontal direction; X is a dryness of the refrigerant in a predetermined position; and Re is a Reynolds number of the refrigerant in the predetermined position determined from an average flow velocity of the refrigerant flowing through the tube. The A to G are expressed by a function of θ , which suppresses the non-uniform loss in pressure of the refrigerant in the respective tubes to reduce the difference in temperature of blown air in the refrigerant radiator.

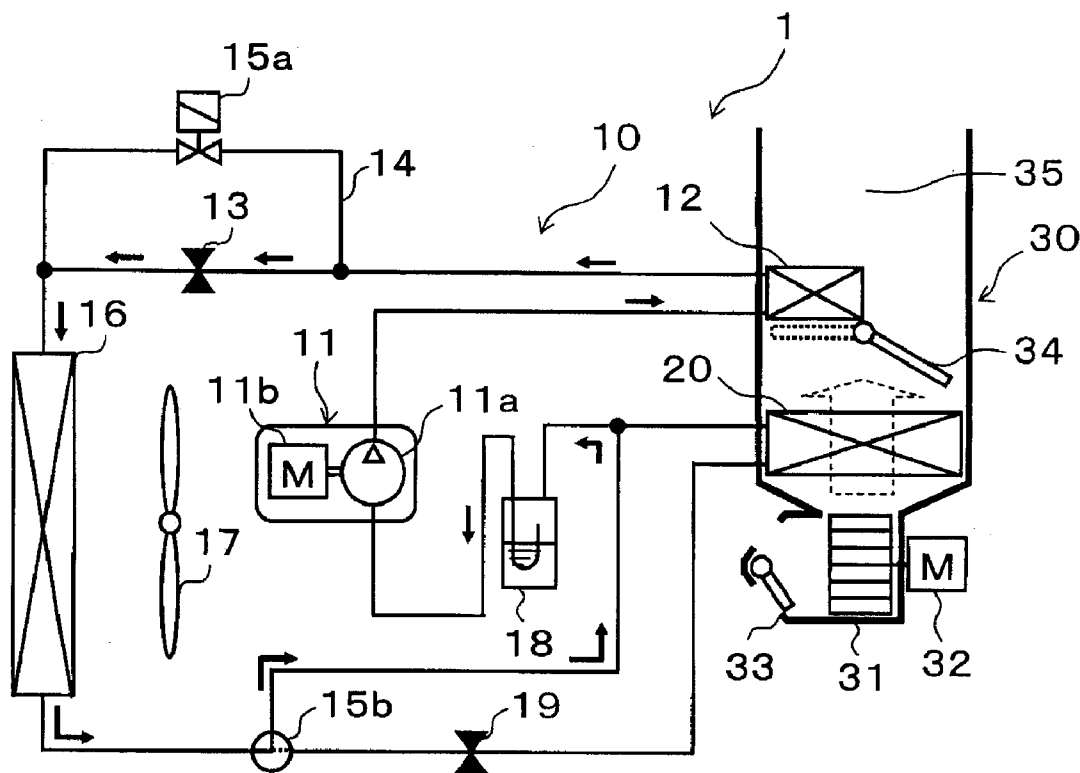
**HEATING OPERATION**

FIG. 1

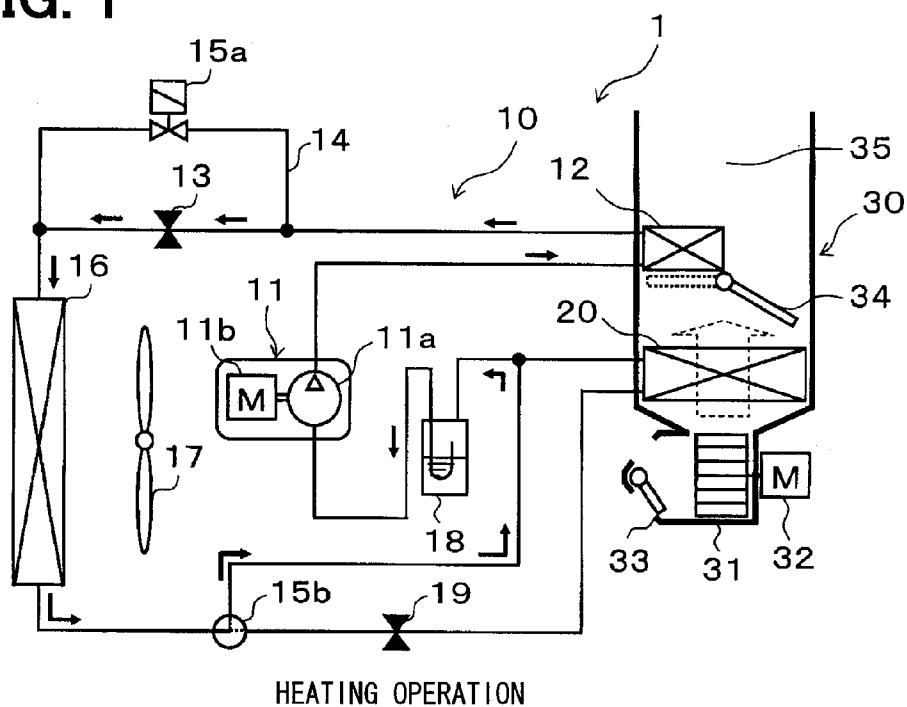


FIG. 2

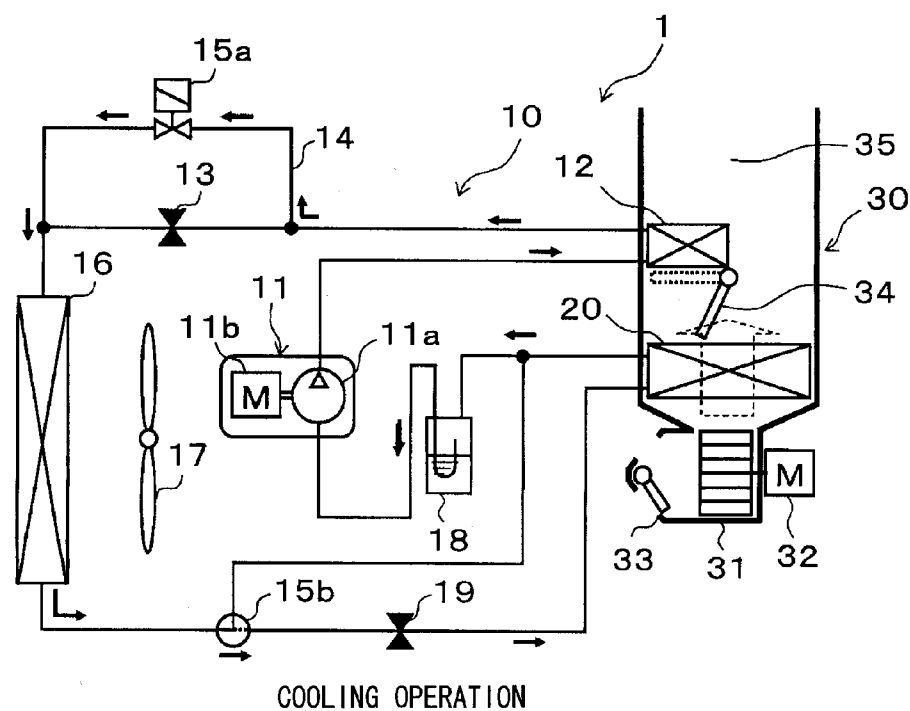


FIG. 3

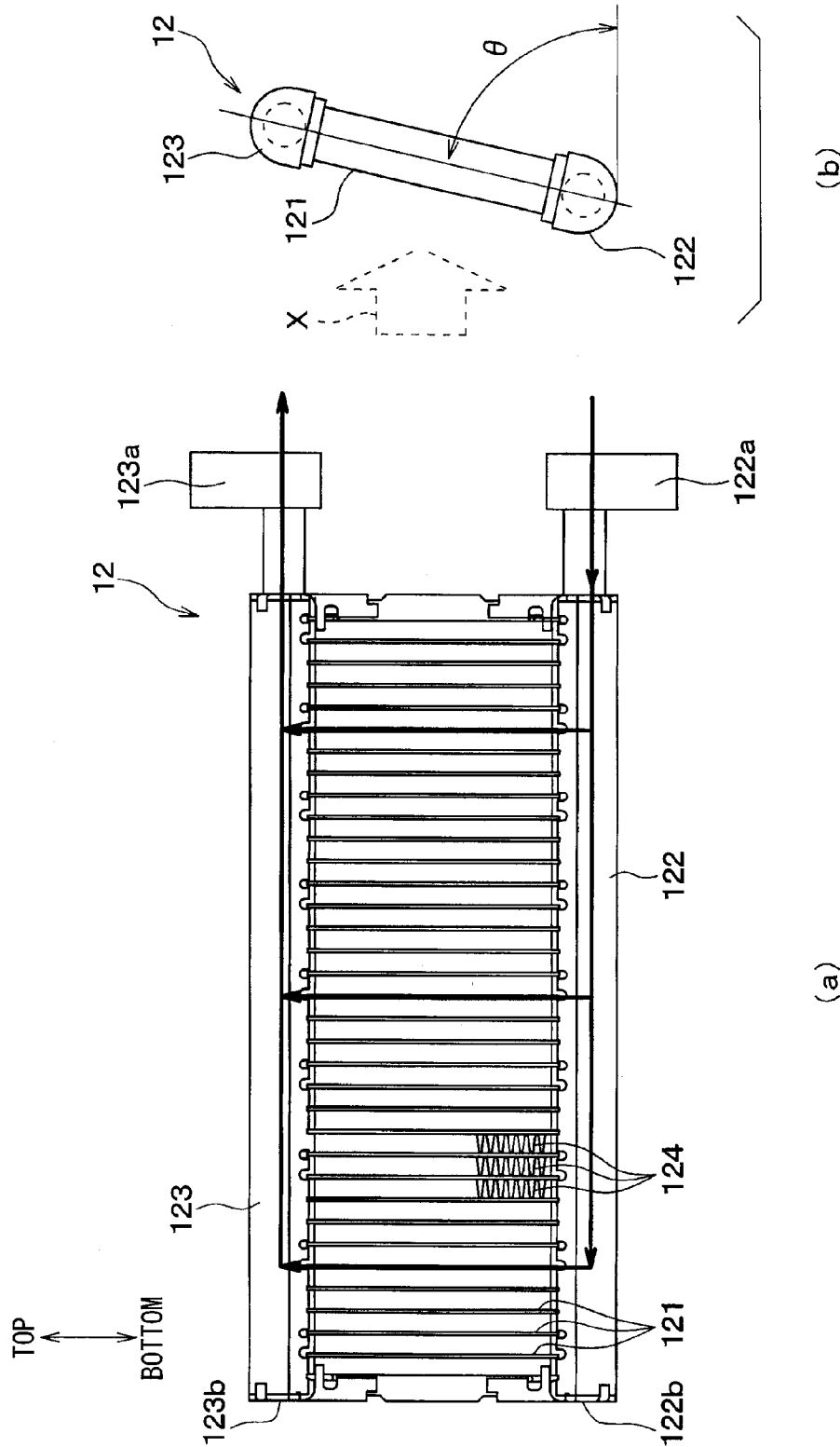


FIG. 4

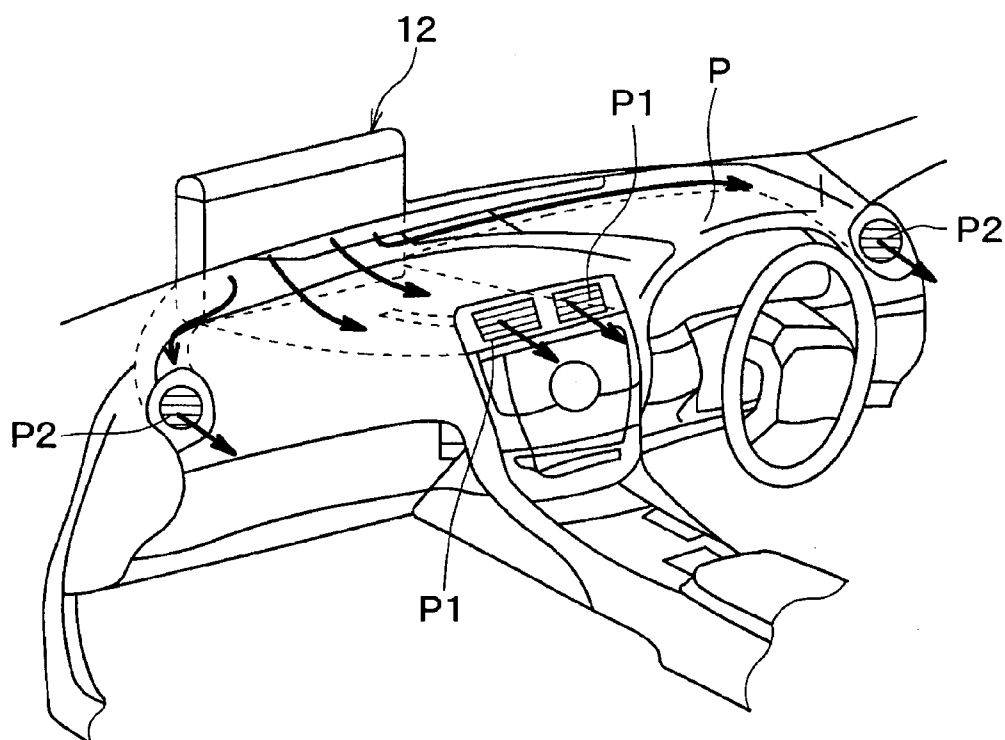


FIG. 5

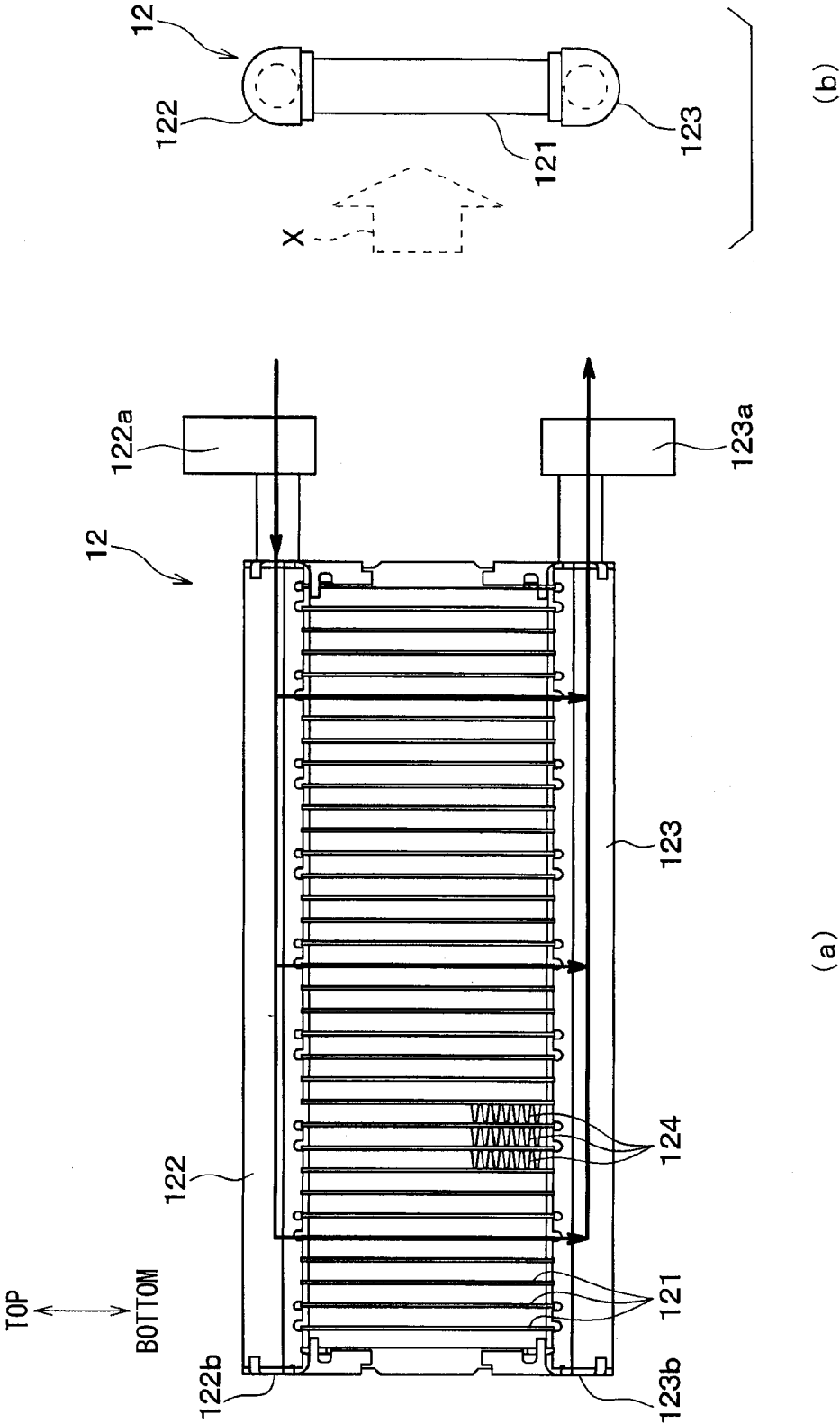


FIG. 6

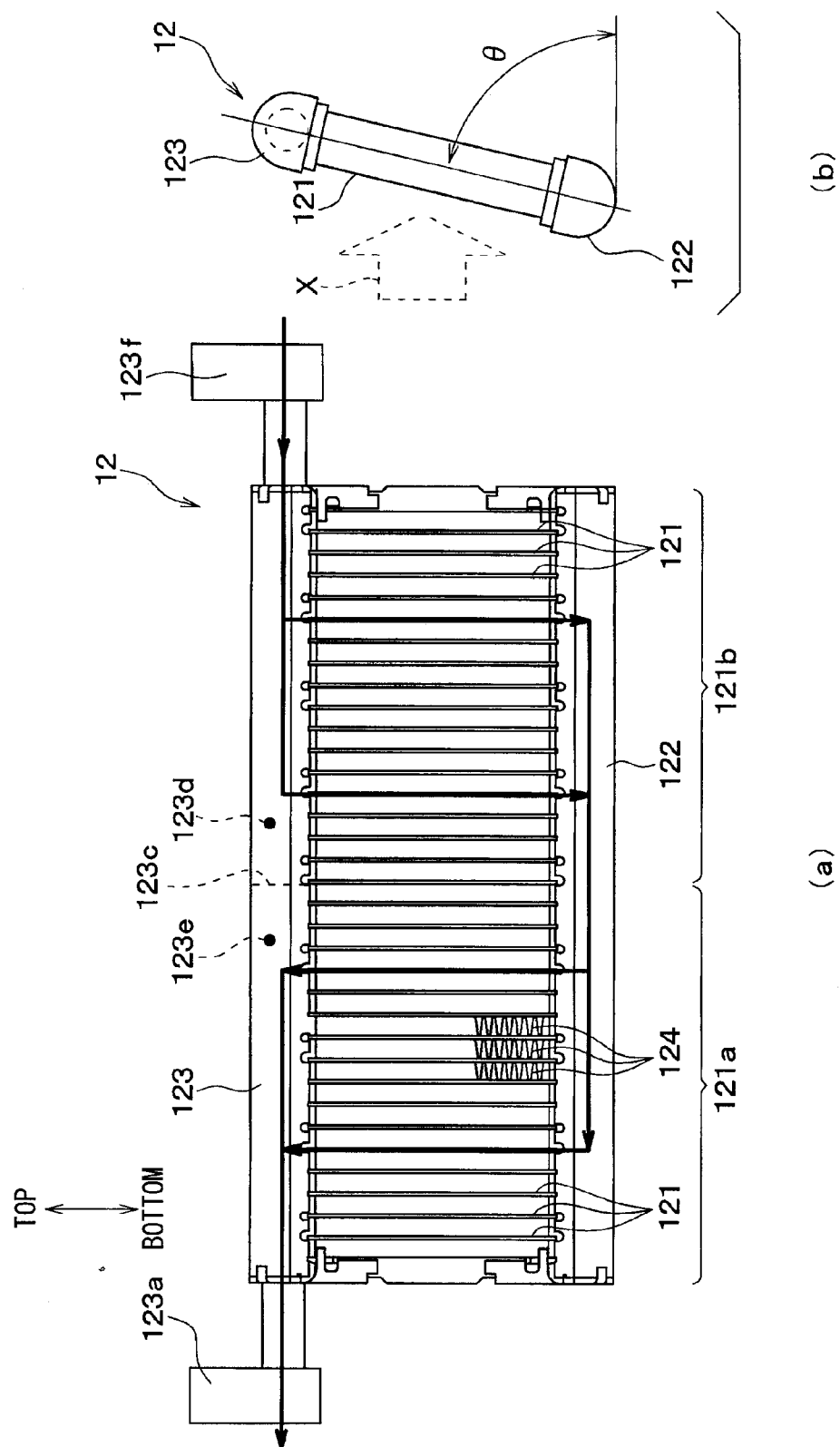


FIG. 7

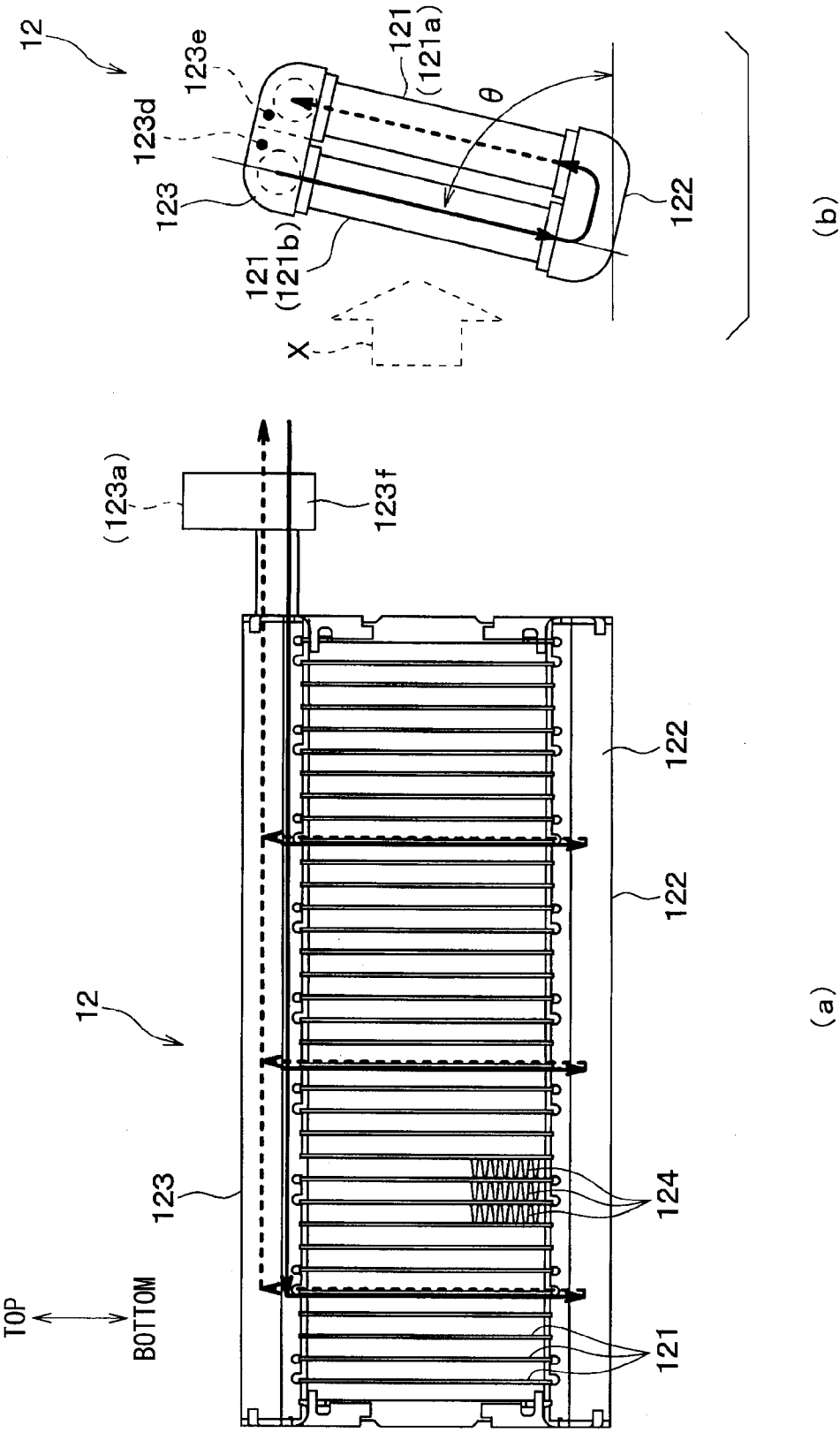


FIG. 8

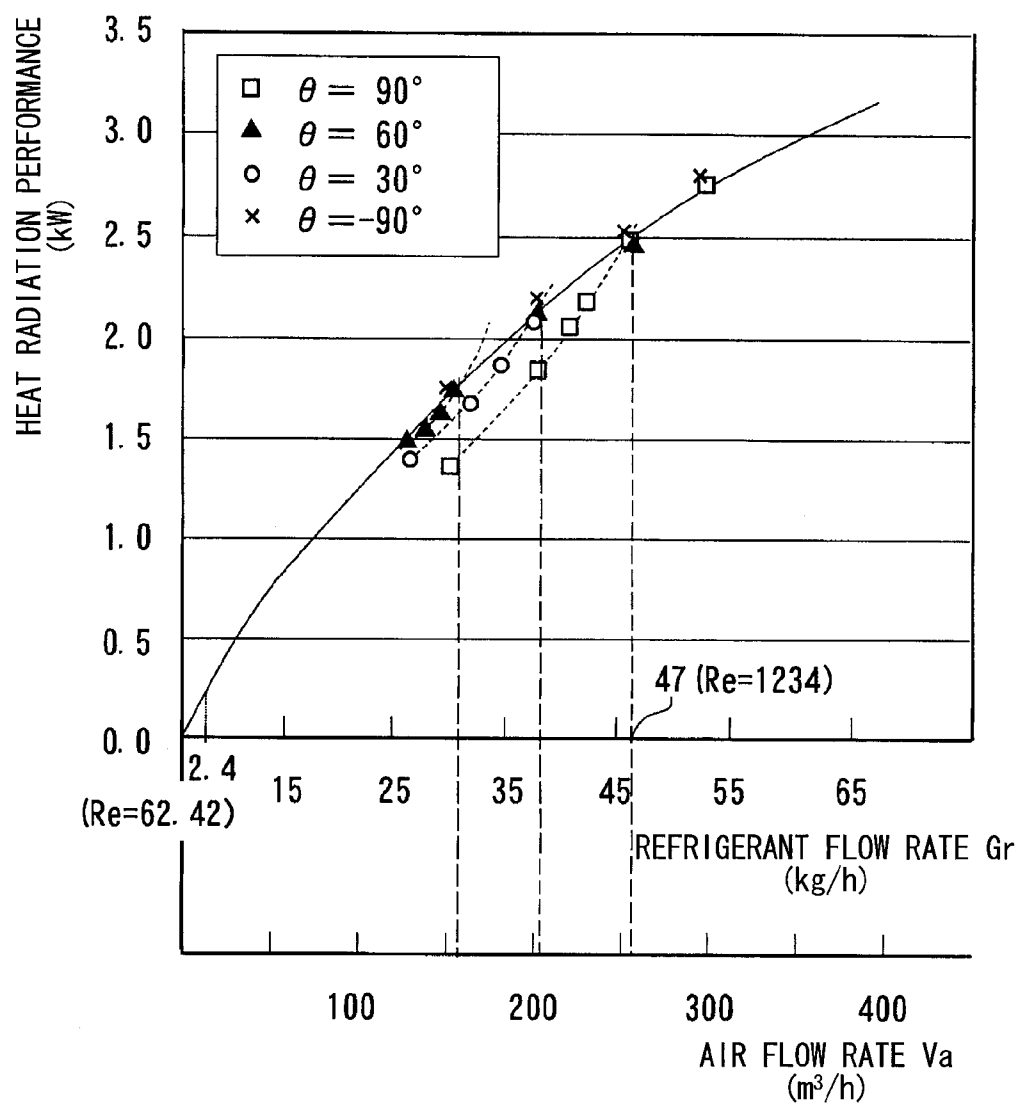
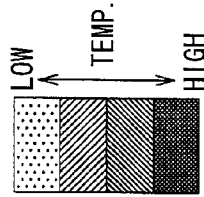
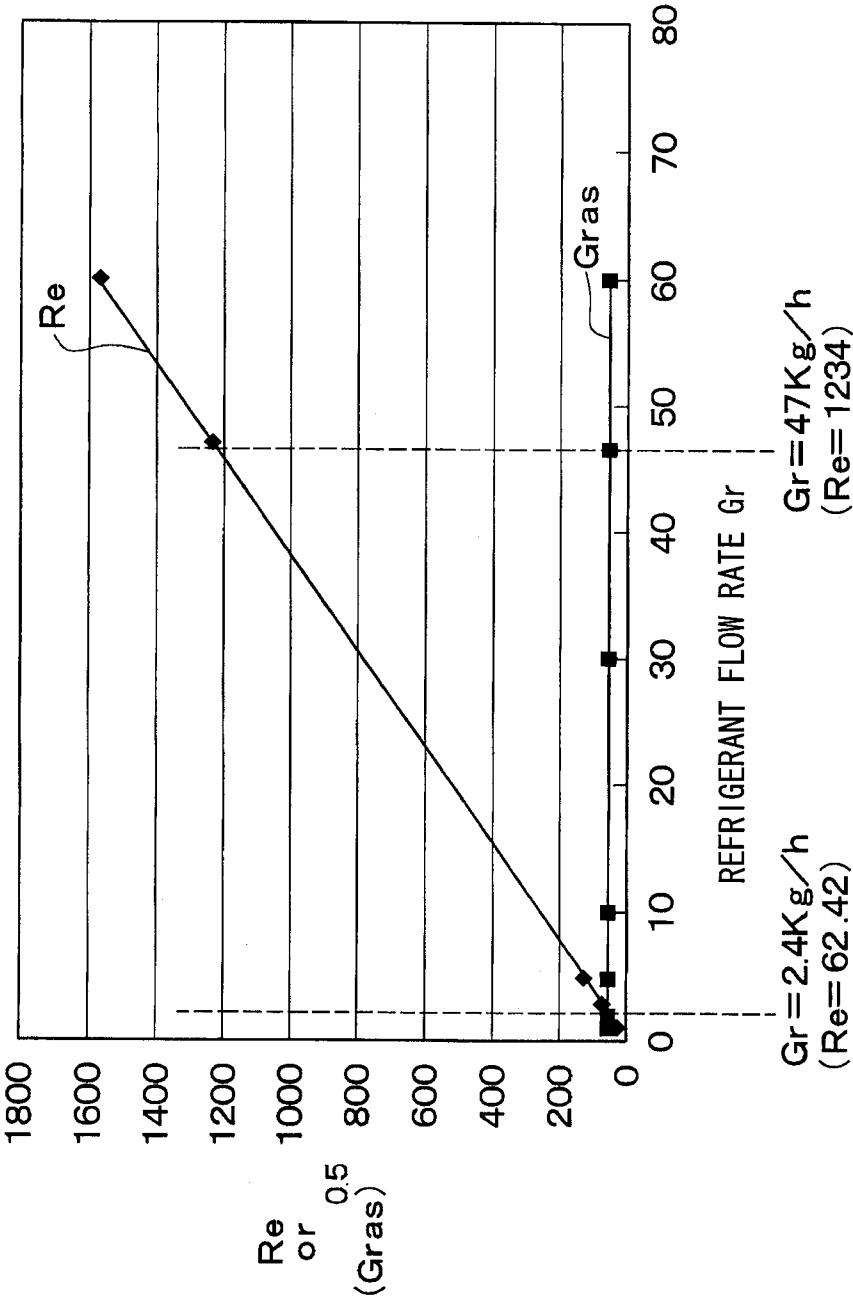


FIG. 9



	(a)	(b)	(c)	(d)																																																																
MEASUREMENT CONDITION	$V_a = 434 \text{ m}^3/\text{h}$ $Gr = 77.9 \text{ m}^3/\text{h}$ $SH = 45.8^\circ\text{C}$ $SC = 4.9^\circ\text{C}$ $T_{\text{ain}} = 10.1^\circ\text{C}$	$V_a = 386 \text{ m}^3/\text{h}$ $Gr = 56.4 \text{ m}^3/\text{h}$ $SH = 46.1^\circ\text{C}$ $SC = 6.3^\circ\text{C}$ $T_{\text{ain}} = 20.1^\circ\text{C}$	$V_a = 200 \text{ m}^3/\text{h}$ $Gr = 40.2 \text{ m}^3/\text{h}$ $SH = 44.6^\circ\text{C}$ $SC = 5.0^\circ\text{C}(4.1^\circ\text{C})$ $T_{\text{ain}} = 20.0^\circ\text{C}$	$V_a = 200 \text{ m}^3/\text{h}$ $Gr = 33.5 \text{ m}^3/\text{h}$ $SH = 43.4^\circ\text{C}$ $SC = 6.3^\circ\text{C}(5.2^\circ\text{C})$ $T_{\text{ain}} = 20.0^\circ\text{C}$																																																																
TEMP. DISTRIBUTION																																																																				
AVERAGE TEMP. OF 16 REGIONS (°C)	<table><tr><td>41.3</td><td>37.9</td><td>35.1</td><td>38.1</td></tr><tr><td>42.5</td><td>40.0</td><td>36.6</td><td>38.3</td></tr><tr><td>45.8</td><td>44.2</td><td>41.4</td><td>42.9</td></tr><tr><td>51.8</td><td>52.6</td><td>51.6</td><td>53.7</td></tr></table>	41.3	37.9	35.1	38.1	42.5	40.0	36.6	38.3	45.8	44.2	41.4	42.9	51.8	52.6	51.6	53.7	<table><tr><td>49.2</td><td>45.4</td><td>42.7</td><td>46.1</td></tr><tr><td>50.1</td><td>47.6</td><td>44.8</td><td>46.7</td></tr><tr><td>52.9</td><td>51.5</td><td>49.2</td><td>50.8</td></tr><tr><td>57.8</td><td>58.5</td><td>57.7</td><td>60.0</td></tr></table>	49.2	45.4	42.7	46.1	50.1	47.6	44.8	46.7	52.9	51.5	49.2	50.8	57.8	58.5	57.7	60.0	<table><tr><td>51.9</td><td>46.0</td><td>44.0</td><td>50.5</td></tr><tr><td>52.0</td><td>46.9</td><td>45.2</td><td>51.2</td></tr><tr><td>53.9</td><td>50.5</td><td>49.7</td><td>54.2</td></tr><tr><td>56.8</td><td>55.9</td><td>56.4</td><td>61.9</td></tr></table>	51.9	46.0	44.0	50.5	52.0	46.9	45.2	51.2	53.9	50.5	49.7	54.2	56.8	55.9	56.4	61.9	<table><tr><td>42.7</td><td>42.2</td><td>42.1</td><td>50.6</td></tr><tr><td>40.4</td><td>41.2</td><td>41.9</td><td>50.7</td></tr><tr><td>41.3</td><td>43.7</td><td>45.2</td><td>53.4</td></tr><tr><td>46.3</td><td>50.2</td><td>52.3</td><td>58.1</td></tr></table>	42.7	42.2	42.1	50.6	40.4	41.2	41.9	50.7	41.3	43.7	45.2	53.4	46.3	50.2	52.3	58.1
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46.3	50.2	52.3	58.1																																																																	
DIFFERENCE ΔT IN AVERAGE TEMP. BETWEEN LEFT AND RIGHT SIDES	2.3°C MAXIMUM TEMP. DIFFERENCE (18.6 °C)	1.9°C MAXIMUM TEMP. DIFFERENCE (17.3 °C)	0.1°C MAXIMUM TEMP. DIFFERENCE (17.9 °C)	5.8°C MAXIMUM TEMP. DIFFERENCE (16.9 °C)																																																																

FIG. 10



REFRIGERANT RADIATOR**CROSS REFERENCE TO RELATED APPLICATION**

[0001] This application is based on Japanese Patent Applications No. 2010-292599 filed on Dec. 28, 2010, and No. 2011-280337 filed on Dec. 21, 2011, the disclosures of which are incorporated herein by reference.

TECHNICAL FIELD

[0002] The present invention relates to a refrigerant radiator for radiating heat from refrigerant in a vapor-compression refrigerant cycle.

BACKGROUND ART

[0003] Conventionally, refrigerant radiators in vapor-compression refrigerant cycles have been known which radiate heat from high-temperature and high-pressure refrigerant discharged from a compressor by exchanging heat with air. For example, a refrigerant radiator disclosed in Patent Document 1 is applied to a vehicle air conditioner. The refrigerant radiator serves as a heater for heating air blowing in a vehicle compartment by exchanging heat between refrigerant discharged from the compressor and the blowing air blown into the vehicle compartment that is used as a space to be conditioned.

[0004] Specifically, the refrigerant cycle disclosed in Patent Document 1 is a so-called supercritical refrigerant cycle that uses carbon dioxide as the refrigerant and in which the pressure of refrigerant on the high-pressure side of the cycle leading from the discharge side of the compressor to the inlet side of a decompression device is equal to or higher than the critical pressure of the refrigerant. The refrigerant radiator disclosed in Patent Document 1 radiates heat from the refrigerant in a supercritical state without a phase change of the refrigerant.

[0005] The refrigerant radiator includes one heat exchange portion disposed on the windward side in the flow direction of blowing air, and the other heat exchange portion disposed on the leeward side thereof. The heat exchange portion on the leeward side allows the refrigerant discharged from the compressor to flow from one end to the other end of the radiator, and the other heat exchange portion on the windward side allows the refrigerant to flow from the leeward-side heat exchange portion to flow from the other end to the one end of the radiator. Additionally, the heat exchanging capacity of the leeward-side heat exchange portion is set lower than that of the windward-side heat exchange portion.

[0006] As viewed in the flow direction of the blowing air, the refrigerant radiator with such a structure superimposes one heat exchange portion for allowing the refrigerant with a relatively high temperature to flow therethrough, over the other heat exchange portion for allowing the refrigerant with a relatively low temperature to flow therethrough. Thus, the refrigerant radiator can suppress the drastic decrease in temperature of the refrigerant directly after flowing into the leeward-side heat exchange portion to reduce the difference in temperature of the air blown from the refrigerant radiator.

RELATED ART DOCUMENT**Patent Document****Patent Document 1**

[0007] Japanese Unexamined Patent Publication No. 2004-125346

[0008] However, when the refrigerant radiator disclosed in Patent Document 1 is applied to a so-called subcritical refrigerant cycle in which the high-pressure side refrigerant pressure is less than the critical pressure of the refrigerant, the above-mentioned effect of suppressing a non-uniform temperature distribution cannot be sufficiently exhibited. This is because the refrigerant changes from a gas phase with a degree of superheat to a gas-liquid two-phases, and further to a liquid phase with a degree of supercooling when the refrigerant radiates heat at the refrigerant radiator in the subcritical refrigerant cycle.

[0009] More specifically, in the supercritical refrigerant cycle, the refrigerant radiator radiates heat from the refrigerant kept in the supercritical state. The refrigerant flowing through the refrigerant radiator radiates heat while decreasing its temperature at a substantially certain rate. Thus, as described in Patent Document 1, the flow direction of the refrigerant in the windward-side heat exchange portion is opposed to that in the leeward-side heat exchange portion, and the refrigerant is prevented from drastically decreasing its temperature directly after flowing into the leeward-side heat exchange portion, and thereby it can reduce the difference in blowing air between the heat exchange portions of the radiator.

[0010] In contrast, in the subcritical refrigerant cycle, the refrigerant in the gas phase or liquid phase flowing through the refrigerant radiator radiates heat therefrom while decreasing its temperature (that is, decreasing both the temperature and enthalpy), whereas the refrigerant in the gas-liquid two-phase radiates heat therefrom without decreasing its temperature (that is, decreasing only the enthalpy).

[0011] As viewed in the flow direction of the blowing air, when one heat exchange portion for the refrigerant in the gas-liquid two-phase state is superimposed over the other heat exchange portion for the refrigerant in the gas-phase state or liquid-phase state in the flow direction of air, the non-uniform temperature distribution of the air flowing from the radiator cannot be sufficiently suppressed in some cases.

DISCLOSURE OF THE INVENTION

[0012] The present invention has been made in view of the above points, and it is an object of the present invention to reduce a difference in temperature of air which exchanges heat with refrigerant and is blown from a refrigerant radiator so as to cause the phase change of the refrigerant flowing therethrough to a gas-phase state, a gas-liquid two-phase state, and a liquid-phase state.

[0013] The disclosures associated with the present invention are proposed based on findings obtained by the following examination and consideration made by the inventors. The inventors of the present application have studied about a refrigerant radiator of a subcritical refrigerant cycle applied to a vehicle air conditioner, specifically, the temperature distribution of air blown from the refrigerant radiator serving as the same type of heater as that in Patent Document 1.

[0014] In this kind of vehicle air conditioner, as will be described later with reference to FIG. 4, the blowing air heated by a driver-seat side heat exchange portion of the refrigerant radiator is blown mainly toward a driver seat, while the blowing air heated by a front passenger-seat side heat exchange portion is blown mainly toward a front passenger seat. Thus, the decrease in temperature of the blowing air in the horizontal direction can reduce the difference in temperature between the air blown toward the driver seat and the air blown toward the front passenger seat.

[0015] The refrigerant flowing through the refrigerant radiator applied to the subcritical refrigerant cycle changes from the gas-phase refrigerant with a degree of superheat to the gas-liquid two-phase refrigerant, and the liquid-phase refrigerant with a degree of supercooling while flowing from the inlet side to the outlet side of the radiator, which increases the density of the refrigerant. When the flow rate of the refrigerant circulating through the cycle is constant, the mass flow rate of the refrigerant flowing through the radiator becomes constant, which decreases the velocity of the refrigerant together with the phase change.

[0016] In the refrigerant radiator applied to the subcritical refrigerant cycle, the rate of a heat exchange portion for exchanging heat between the blowing air and the gas-phase refrigerant with a relatively difference in temperature from the air, in the whole heat exchange portions, becomes larger in theory, so that the refrigerant radiator can radiate heat from the gas-phase refrigerant in a wide heat exchange portion.

[0017] The inventors employs as the refrigerant radiator, a full-path type tank and tube heat exchanger (multi-flow heat exchanger) with the same structure as that of the radiator to be described later in a first embodiment with reference to FIGS. 3(a) and 3(b). The refrigerant radiator is disposed such that the longitudinal direction of tubes has at least one component directed vertically. Then, the inventors have studied the temperature distribution of the blowing air in the refrigerant radiator. The term “direction having at least one component directed vertically” as used in the present application means that the tube extends in the direction perpendicular to the horizontal direction, or at an angle with respect to the horizontal direction.

[0018] The reason why the longitudinal direction of the tube has the one component directed vertically is that when the refrigerant discharged from the compressor flows into all tubes from a header tank for refrigerant distribution, the non-uniform temperature distribution of the blowing air in the vertical direction occurs, but the difference in temperature of the blowing air in the horizontal direction can be suppressed. In the refrigerant radiator, the header tank for distribution provided with a refrigerant inlet is positioned on the lower side, and a header tank for collection provided with a refrigerant outlet is positioned on the upper side.

[0019] Next, the result of consideration will be described with reference to FIGS. 8 and 9. As shown in FIG. 8, the inventors have examined a change in heat radiation performance of the refrigerant radiator obtained upon changing the flow rate Gr of the refrigerant circulating through the refrigerant cycle to which the refrigerant radiator is applied (that is, the flow rate (kg/h) of the refrigerant flowing through the refrigerant radiator).

[0020] FIG. 8 is a graph showing the change in heat radiation performance with respect to the changes in refrigerant flow rate Gr and blowing air flow rate Va (m^3/h). The scale of the refrigerant flow rate Gr and the scale of the blowing air

flow rate Va in the horizontal axis of FIG. 8 represent the respective flow rates which balance the amount of heat dissipation from the refrigerant and the amount of heat absorption of the blowing air in the refrigerant radiator. The relationship between Gr and Va can be approximated by the following formula 1.

$$Gr = -0.0002Va + 0.1997Va + 5.4994 \quad [\text{Formula 1}]$$

[0021] FIG. 8 shows the heat radiation performance at each of inclination angles θ (in units of degree) of 90° , 60° , and 30° formed by the flow direction of the refrigerant flowing through the tube and the horizontal direction as viewed in the direction perpendicular to the direction of blowing air. Also, FIG. 8 plots a heat radiation performance obtained at an inclination angle θ of -90° with the distribution header tank positioned on the upper side and the collection header tank positioned on the lower side. The inclination angle θ is defined as will be mentioned in detail in the following embodiments.

[0022] As can be seen from FIG. 8, in the refrigerant radiator used for the consideration, the heat radiation performance is greatly decreased with decreasing flow rate Gr of the refrigerant. In order to determine the reason for the decrease in heat radiation performance, as shown in FIG. 9, the inventors have examined the temperature distribution of the blowing air flowing from the radiator when changing the refrigerant flow rate Gr .

[0023] In the examination of the temperature distribution, the heat exchange portion of the refrigerant radiator is divided into 16 regions (portions). An average temperature of blowing air blown from each divided region is determined. Further, a difference in temperature between the average temperature of 8 regions of the heat exchange portion on one side (on the right side of the paper of the figure) in the horizontal direction, and the other average temperature of other 8 regions on the other side in the horizontal direction (on the left side of the paper of the figure) is regarded as a difference ΔT in average temperature between the left and right sides. The difference ΔT in average temperature between the left and right sides difference can be used as an index representing the temperature distribution of the blowing air in the horizontal direction.

[0024] In examples (a) to (d) of FIG. 9, an air flow rate Va , a refrigerant flow rate Gr , a degree of superheat SH of the refrigerant at the inlet of the radiator 12, a degree of supercooling SC of the refrigerant at the outlet of the radiator 12, and a temperature T_{air} of air flowing into the radiator 12 are simply indicated by reference characters Va , Gr , SH , SC , and T_{air} , respectively.

[0025] FIG. 9 shows that with decreasing refrigerant flow rate Gr , an area of the regions with a relatively low temperature in the heat exchange portion is increased (see the substantially center area enclosed by a broken line in the heat exchange portion shown in the examples (b) and (c) of FIG. 9). When the refrigerant flow rate Gr is further decreased, the number of regions with the relatively low temperature in the heat exchange portion is increased (see a left side part in the paper of the figure as well as the substantially center area enclosed by a broken line in the heat exchange portion shown in the example (d) of FIG. 9).

[0026] That is, as the flow rate Gr decreases, the area of the regions with the relatively low temperature in the heat exchange portion, that is, the regions that cannot sufficiently heat the blowing air in the heat exchange portion is increased, which leads to reduction in heat radiation performance as the

entire refrigerant radiator. As shown in the example (d) of FIG. 9, the formation of the regions with the relatively low temperature in the heat exchange portion increases the difference in average temperature ΔT between the left and right sides, thus degrading the uniformity of the temperature distribution of the blowing air in the horizontal direction.

[0027] The inventors have further studied and found out that the formation of the regions with the relatively low temperature in the entire heat exchange portion of the refrigerant radiator is caused by a difference in degree of condensation of the refrigerant between the respective tubes.

[0028] More specifically, the gas-phase refrigerant flows into each tube of the refrigerant radiator, so that the distribution of the refrigerant flowing into the tubes is relatively good, but it is difficult to completely uniformly distribute the refrigerant to all tubes due to influences, including a loss in pressure in the header tank, an inertial force of the refrigerant inflow, and the like. Even when the air flows relatively uniformly into all heat exchange portions of the radiator, some tubes let the less refrigerant through.

[0029] In such a specific region for heat exchange composed of the tubes for allowing the less refrigerant to pass therethrough, the refrigerant flowing through the tubes is more likely to be condensed than the refrigerant flowing through tubes forming other regions for heat exchange. At this time, when the difference in pressure of the refrigerant between the inlet and outlet of the tube is decreased with decreasing refrigerant flow rate Gr, the flow rate or velocity of the condensed refrigerant is further reduced, which makes it difficult for the condensed refrigerant to flow out of the tubes.

[0030] Thus, when the condensed refrigerant is attached to a wall of a refrigerant passage or the like to stay in the tube, an area of the refrigerant passage of each of the tubes forming the specific heat exchange portion becomes narrower than that of a refrigerant passage of tubes in other regions, which increases a loss in pressure of the refrigerant at the tubes with the refrigerant attached thereto. As a result, the tubes forming the specific heat exchange portion are less likely to allow the high-temperature refrigerant discharged from the compressor to flow thereinto, as compared to the tubes forming other regions, which might form the specific heat exchange portion with the relatively low temperature (hereinafter referred to as a "low-temperature region").

[0031] The inventors have further examined the loss in pressure of the refrigerant in the tubes forming the low-temperature region, and found out that the increase in loss of pressure of the tubes forming the low-temperature region is caused by the condensed refrigerant staying in the tubes. Then, the inventors have paid attention to the following factors, in addition to the flow velocity U of refrigerant serving as a source of energy for pushing out the refrigerant:

[0032] (1) an increase in loss of pressure due to an increase in viscosity μ caused by the condensed refrigerant; and

[0033] (2) a decrease in loss of pressure due to a decrease in density ρ caused by the condensed refrigerant.

[0034] Further, the inventors also have paid attention to the following factor in the refrigerant radiator for allowing the refrigerant to flow from the lower side to the upper side in the tube:

[0035] (3) an increase in loss of pressure calculated based on the gravity acting on the condensed refrigerant.

[0036] That is, by taking into consideration and controlling an inclination angle θ as a parameter required for calculation of the flow velocity U, viscosity μ , density ρ of the refrigerant,

and the gravity acting on the condensed refrigerant, the loss in pressure of the refrigerant at tubes 121 forming the low-temperature region can be set equal to that at tubes forming other regions to suppress the formation of the specific heat exchange portion with the relatively low temperature.

[0037] Based on the above findings, the inventors have calculated arrangement conditions of the refrigerant radiator that can suppress the formation of the specific heat exchange portion with the relatively low temperature, by simulation calculation using the inclination angle θ and a Reynolds number defined as a ratio of inertial forces to viscous forces by use of the flow velocity U, viscosity μ , and density ρ of the refrigerant, and then expressed the result by an approximation formula.

[0038] The calculation of the arrangement conditions for the refrigerant radiator employs the average flow velocity of refrigerant flowing through the tubes as the flow velocity U of the refrigerant. The refrigerant flowing into the refrigerant radiator is a gas-phase refrigerant having a 45° C. of super-cooling degree at a pressure of 2 MPa and the flow rate of air flowing into the radiator is 200 m³/h at 20° C. The viscosity μ_m of the gas-liquid two-phase fluid required for determining an increase in loss of pressure in the above formula (1) is determined using a Taylor formula expressed by the following formula (2).

$$\mu = \mu_1 \{1 + 2.5\alpha_g(\rho_l - \rho_g)/\rho_l\} \quad [\text{Formula 2}]$$

[0039] μ_m : Viscosity of gas-liquid two-phase fluid

[0040] μ_l : Viscosity of liquid-phase fluid

[0041] α_g : Void fraction

[0042] ρ_l : Density of liquid-phase fluid

[0043] ρ_g : Density of gas-phase fluid

[0044] The void fraction α_g of the gas-liquid two-phase fluid required for the Taylor formula is determined using a Levy formula (Levy's momentum minimal model) expressed by the following formula 3.

$$(1 - \alpha_g)/\alpha_g \cdot x/(1 - x) = (\rho_g/\rho_l)^{1/2} \quad [\text{Formula 3}]$$

[0045] x: Dryness

[0046] α_g : Void fraction

[0047] ρ_l : Density of liquid-phase fluid

[0048] ρ_g : Density of gas-phase fluid

[0049] As a result, at the inclination angle (θ) of $0 < \theta \leq 90^\circ$, that is, in the refrigerant radiator for allowing the refrigerant to flow through the tube from the lower side to the upper side, the arrangement conditions expressed by the following formula 4 is set, so that the loss in pressure of the refrigerant at the tubes forming the low-temperature region is equal to that at the tubes forming other regions, which can suppress the formation of the region with the relatively low temperature in the heat exchange portion.

$$Re \geq AxX^6 + BxX^5 + CxX^4 + DxX^3 + ExX^2 + FxX + G \quad [\text{Formula 4}]$$

[0050] $A = -0.0537 \times \theta^2 + 9.7222 \times \theta + 407.19$

[0051] $B = -(-0.2093 \times \theta^2 + 37.88 \times \theta + 1586.3)$

[0052] $C = -0.3348 \times \theta^2 + 60.592 \times \theta + 2538.1$

[0053] $D = -(-0.2848 \times \theta^2 + 51.53 \times \theta + 2158.2)$

[0054] $E = -0.1402 \times \theta^2 + 25.365 \times \theta + 1062.8$

[0055] $F = -(-0.0418 \times \theta^2 + 7.5557 \times \theta + 316.46)$

[0056] $G = -0.0132 \times \theta^2 + 2.3807 \times \theta + 99.73$

[0057] According to a first aspect of the present disclosure based on the above findings, a refrigerant radiator which is applied to a vapor-compression refrigerant cycle is adapted to exchange heat between a high-temperature and high-pressure refrigerant compressed by a compressor and air blown into a

space for air conditioning to thereby radiate heat from a gas-phase refrigerant with a degree of superheat to transfer the gas-phase refrigerant to a liquid-phase refrigerant with a degree of supercooling. The refrigerant radiator includes a plurality of tubes for allowing the refrigerant to flow therethrough from an upper side to a lower side; a first header coupled to ends of the tubes for distributing the refrigerant to flow into at least some of the tubes; and a second header coupled to the other ends of the tubes for collecting the refrigerants flowing from at least some of the tubes. When the phase change of the refrigerant occurs in the tube in the refrigerant radiator for allowing the refrigerant to flow through the tubes from the upper side to the lower side, the refrigerant radiator of the disclosure can suppress the non-uniform loss in pressure of the refrigerant in the tubes. Thus, the refrigerant radiator of the disclosure can prevent the formation of the specific heat exchange portion with the relatively low temperature in the entire heat exchange portion of the refrigerant radiator.

[0058] For example, according to a second aspect of the present disclosure, the refrigerant radiator used in the region satisfying the relationship of $62.42 \leq Re \leq 1234$ where the flow of refrigerant flowing through the tube becomes a forced convection is desirably a refrigerant evaporator for allowing the refrigerant to flow through the tubes from the upper side to the lower side that does not need to take into consideration an increase in loss of the pressure calculated based on the gravity acting on the condensed refrigerant.

[0059] According to a third aspect of the present disclosure, a refrigerant radiator is used for a vapor-compression refrigerant cycle, and is adapted to exchange heat between a high-temperature and high-pressure refrigerant compressed by a compressor of the vapor-compression refrigerant cycle and air blown into a space for air conditioning to thereby radiate heat from a gas-phase refrigerant having a degree of superheat so as to transfer the gas-phase refrigerant to a liquid-phase refrigerant having a degree of supercooling. The refrigerant radiator includes a plurality of tubes for allowing the refrigerant to flow therethrough, and the tube extends in a direction perpendicular to a horizontal direction, or at an angle with respect to the horizontal direction. Furthermore, the tubes are disposed so as to satisfy the following relationship:

$$Re \geq A \times X^6 + B \times X^5 + C \times X^4 + D \times X^3 + E \times X^2 + F \times X + G$$

[0060] $A = -0.0537 \times 10^{-2} + 9.7222 \times 10^{-4} + 407.19$

[0061] $B = -(-0.2093 \times 10^{-2} + 37.88 \times 10^{-4} + 1586.3)$

[0062] $C = -0.3348 \times 10^{-2} + 60.592 \times 10^{-4} + 2538.1$

[0063] $D = -(-0.2848 \times 10^{-2} + 51.53 \times 10^{-4} + 2158.2)$

[0064] $E = -0.1402 \times 10^{-2} + 25.365 \times 10^{-4} + 1062.8$

[0065] $F = -(-0.0418 \times 10^{-2} + 7.5557 \times 10^{-4} + 316.46)$

[0066] $G = -0.0132 \times 10^{-2} + 2.3807 \times 10^{-4} + 99.73$

[0067] wherein θ (°) is an inclination angle formed by a flow direction of the refrigerant flowing through the tube and the horizontal direction; X is a dryness of the refrigerant in a predetermined position where the refrigerant flowing through the tube is a gas-liquid two-phase refrigerant; and Re is a Reynolds number of the refrigerant in the predetermined position that is determined from an average flow velocity (m/S) of the refrigerant flowing through the tube, and

[0068] wherein as the flow direction of the refrigerant flowing through the tube changes from a vertically downward side to a vertically upward side, the inclination angle (θ) changes in a range of more than 0° and not more than 90° ($0 < \theta \leq 90^\circ$).

[0069] Thus, for the inclination angle (θ) of $0 < \theta \leq 90^\circ$, that is, in the refrigerant radiator for allowing the refrigerant to flow through the tubes from the lower side to the upper side, even in the phase change of the refrigerant flowing through the tubes, the non-uniform loss in pressure of the refrigerant in the respective tubes can be suppressed by taking into consideration parameters, including the flow velocity, viscosity, density of the refrigerant and the inclination angle (θ). The refrigerant radiator of the disclosure can suppress the formation of the specific heat exchange portion with the relatively low temperature in the entire heat exchange portion of the refrigerant radiator.

[0070] As a result, even if the flow rate of the refrigerant circulating through the refrigerant cycle is changed and the flow velocity of the refrigerant flowing through the tube is changed, such an arrangement can suppress the reduction in heat radiation performance of the refrigerant radiator and can also suppress the non-uniform temperature distribution in the horizontal direction of air heated and blown by the refrigerant radiator.

[0071] The term “predetermined position” as used herein can be any arbitrary position as long as the refrigerant flowing through the tube is the gas-liquid two-phase refrigerant in the position. That is, even when the dryness (X) of the refrigerant flowing through the tube is changed, the Reynolds number (Re) is calculated using the dryness (X), which can prevent the formation of the specific heat exchange portion with the relatively low temperature even in use of any arbitrary position as the predetermined position.

[0072] The phrase “tube extends in the direction having at least one component directed vertically” as used herein means not only that the entire tube extends vertically, but also that a part of the tube extends vertically.

[0073] The refrigerant radiator according to a fourth aspect of the disclosure may further include a header tank disposed on at least one end side of each of the tubes to extend in a lamination direction of the tubes, and adapted to collect or distribute the refrigerant.

[0074] In a multi-flow heat exchanger structure for collecting or distributing the refrigerants flowing through the tubes using a header tank, the flow rate of refrigerant flowing through each tube tends to be changed depending on the position of the inlet or outlet of the refrigerant provided in the header tank, and thereby it may cause the temperature distribution of the blowing air. Thus, the arrangement conditions that can reduce the difference in temperature of blowing air can be applied to such a refrigerant radiator structure, which is very effective.

[0075] According to a fifth aspect of the disclosure, the tubes include a first tube group for allowing the refrigerant to flow therethrough from a lower side to an upper side, and a second tube group for allowing the refrigerant to flow therethrough from the upper side to the lower side.

[0076] In the refrigerant radiator according to a sixth aspect of the disclosure, an internal space of the header tank is separated into a plurality of spaces, and one separated space is provided with a refrigerant inlet for allowing the gas-phase refrigerant to flow into the one separated space, and the other separated space is provided with a refrigerant outlet for allowing the liquid-phase refrigerant to flow from the other separated space.

[0077] In the refrigerant radiator according to a seventh aspect of the disclosure, the tubes may be arranged in the flow direction of the blowing air. One of the heat exchange por-

tions on the windward and leeward sides is defined as a region for allowing the refrigerant with a degree of superheat to flow therethrough (superheat-degree region), and the other on the leeward side is defined as a region for allowing the refrigerant with a degree of supercooling to flow therethrough (supercooling region). The superheat region and the supercooling region are superimposed over each other as viewed in the flow direction of the blowing air, which can reduce the difference in temperature of blowing air in the vertical direction.

[0078] In the refrigerant radiator according to an eighth aspect of the disclosure, the flow directions of the refrigerants flowing through the tubes may be the same.

[0079] In the refrigerant radiator according to a ninth aspect of the disclosure, the refrigerant cycle may be applied to a vehicle air conditioner, and the space for air conditioning may be an interior of a vehicle.

[0080] Thus, the heat radiator of the disclosure can reduce the difference in temperature in the horizontal direction of the blowing air, specifically, the difference in temperature between the air blown toward the driver seat and the air blown toward the front passenger seat.

BRIEF DESCRIPTION OF DRAWINGS

[0081] The above and other objects, structures, and advantages of the present invention will become apparent from the following detailed description of the invention, when taken in conjunction with the accompanying drawings, which respectively show:

[0082] FIG. 1 is an entire configuration diagram showing refrigerant flow paths of a heat pump cycle in a heating operation according to a first embodiment;

[0083] FIG. 2 is an entire configuration diagram showing refrigerant flow paths of the heat pump cycle in a cooling operation in the first embodiment;

[0084] FIG. 3(a) is a front view of a refrigerant radiator in the first embodiment, and FIG. 3(b) is a side view of the refrigerant radiator shown in FIG. 3(a);

[0085] FIG. 4 is a schematic diagram showing an arrangement state of the refrigerant radiator in the first embodiment;

[0086] FIG. 5(a) is a front view of a refrigerant radiator according to a second embodiment, and FIG. 5(b) is a side view of the refrigerant radiator shown in FIG. 5(a);

[0087] FIG. 6(a) is a front view of a refrigerant radiator according to a third embodiment, and FIG. 6(b) is a side view of the refrigerant radiator shown in FIG. 6(a);

[0088] FIG. 7(a) is a front view of a refrigerant radiator according to a fourth embodiment, and FIG. 7(b) is a side view of the refrigerant radiator shown in FIG. 7(a);

[0089] FIG. 8 is a graph showing a change in heat radiation performance with respect to changes in refrigerant flow rate and air flow rate of the refrigerant radiator for consideration made by the applicant(s);

[0090] FIG. 9 is a schematic diagram showing the result of experiments of temperature distribution in the refrigerant radiator for consideration made by the applicant(s); and

[0091] FIG. 10 is a graph showing a change in Reynolds number or Grashof number with respect to the refrigerant flow rate in the refrigerant radiator for consideration made by the applicant(s).

MODE FOR CARRYING OUT THE INVENTION

[0092] Some embodiments for carrying out the invention will be described below with reference to the accompanying

drawings. The same or equivalent parts of each embodiment corresponding to components described in the previous embodiment are indicated by the same reference numerals, and thus the repeated description thereof will be omitted. When only a part of the structure in each embodiment is explained, other parts of the structure can be applied to the description of the previous embodiment. The combination of the parts is not limited to those specifically described in the respective embodiments. As long as the combination does not raise any problems, the embodiments can be partly combined even though the combination is not suggested in the present specification.

First Embodiment

[0093] A first embodiment of the disclosure will be described with reference to FIGS. 1 to 4. In this embodiment, a heat pump cycle 10 (vapor-compression refrigerant cycle) with a refrigerant radiator 12 of the disclosure is applied to a vehicle air conditioner 1. FIG. 1 is an entire configuration diagram of the vehicle air conditioner 1 in this embodiment. The vehicle air conditioner 1 can be applied not only to a normal engine vehicle which obtains a driving force for traveling from an engine (internal combustion engine), but also various types of vehicle, such as a hybrid vehicle or an electric vehicle.

[0094] The heat pump cycle 10 in the vehicle air conditioner 1 serves to heat or cool the blowing air in the vehicle compartment to be blown into the vehicle interior as a space for air conditioning. That is, the heat pump cycle 10 can switch between refrigerant flow paths to thereby perform a heating operation (heater operation) and a cooling operation (cooler operation). The heating operation is performed to heat the vehicle interior by heating the blowing air in the vehicle compartment as a fluid for heat exchange. The cooling operation is performed to cool the vehicle interior by cooling the blowing air in the vehicle compartment.

[0095] In the entire configuration diagrams of FIGS. 1 and 2 showing the heat pump cycle 10, the flow of refrigerant in the heating operation and the flow of refrigerant in the cooling operation each are indicated by a solid line.

[0096] The heat pump cycle 10 of this embodiment employs HFC-based refrigerant (specifically, R134a) as the refrigerant, and forms a subcritical refrigerant cycle whose high-pressure side refrigerant pressure does not exceed the critical pressure of the refrigerant. HFO-based refrigerant (specifically, R1234yf) may be employed as long as the refrigerant forms the subcritical refrigerant cycle. Refrigerating machine oil for lubricating a compressor 11 is mixed into the refrigerant, and a part of the refrigerating machine oil circulates through the cycle together with the refrigerant.

[0097] The compressor 11 is positioned in an engine room, and is to suck, compress, and discharge the refrigerant in the heat pump cycle 10. The compressor is an electric compressor which drives a fixed displacement compressor 11a having a fixed discharge capacity by use of an electric motor 11b. Specifically, various types of compression mechanisms, such as a scroll type compression mechanism, or a vane compression mechanism, can be employed as the fixed displacement compressor 11a.

[0098] The electric motor 11b may be either an AC motor or a DC motor whose operation (number of revolutions) is controlled by a control signal output from an air conditioning controller to be described later. The control of the number of revolutions of the motor changes a refrigerant discharge

capacity of the compressor 11. Thus, in this embodiment, the electric motor 11b serves as a discharge capacity changing portion of the compressor 11.

[0099] A refrigerant discharge port of the compressor 11 is coupled to a refrigerant inlet side of a refrigerant radiator 12. The refrigerant radiator 12 is disposed in a casing 31 of an indoor air conditioning unit 30 of the vehicle air conditioner 1 to be described later. The refrigerant radiator is a heat exchanger for heating that exchanges heat between a high-temperature and high-pressure refrigerant flowing therethrough and the blowing air in the vehicle compartment having passed through a refrigerant evaporator 20 to be described later. The detailed structures of the refrigerant radiator 12 and the indoor air conditioning unit 30 will be described later.

[0100] A fixed throttle 13 for heating is coupled to a refrigerant outlet side of the refrigerant radiator 12. The fixed throttle 13 serves as a decompression portion for the heating operation that decompresses and expands the refrigerant flowing from the refrigerant radiator 12 in the heating operation. The fixed throttle 13 for heating can use an orifice, a capillary tube, and the like. The outlet side of the fixed throttle 13 for heating is coupled to the refrigerant inlet side of the outdoor heat exchanger 16.

[0101] A bypass passage 14 for the fixed throttle is coupled to the refrigerant outlet side of the refrigerant radiator 12. The bypass passage 14 causes a refrigerant flowing from the refrigerant radiator 12 to bypass the fixed throttle 13 for heating and to guide the refrigerant into the outdoor heat exchanger 16. An opening/closing valve 15a for opening and closing the bypass passage 14 for the fixed throttle is disposed in the bypass passage 14 for the fixed throttle. The opening/closing valve 15a is an electromagnetic valve whose opening and closing operations are controlled by a control voltage output from the air conditioning controller.

[0102] The loss in pressure caused when the refrigerant passes through the opening/closing valve 15a is extremely small as compared to the loss in pressure caused when the refrigerant passes through the fixed throttle 13 for heating. Thus, when the opening/closing valve 15a is opened, the refrigerant flows from the refrigerant radiator 12 into the outdoor heat exchanger 16 via the bypass passage 14 for the fixed throttle. In contrast, when the opening/closing valve 15a is closed, the refrigerant flows into the outdoor heat exchanger 16 via the fixed throttle 13 for heating.

[0103] Thus, the opening/closing valve 15a can switch between the refrigerant flow paths of the heat pump cycle 10. The opening/closing valve 15a of this embodiment serves as a refrigerant flow path switching portion. Alternatively, as such a refrigerant flow path switching portion, an electric three-way valve or the like may be provided for switching between a refrigerant circuit for coupling the outlet side of the refrigerant radiator 12 to the inlet side of the fixed throttle 13 for heating, and another refrigerant circuit for coupling the outlet side of the refrigerant radiator 12 to the inlet side of the bypass passage 14 for the fixed throttle.

[0104] The outdoor heat exchanger 16 is to exchange heat between the low-pressure refrigerant flowing therethrough and an outside air blown from a blower fan 17. Further, the outdoor heat exchanger 16 is disposed in an engine room, and is a heat exchanger which serves as an evaporator for evaporating the low-pressure refrigerant to exhibit a heat absorption effect in the heating operation, and also as a radiator for radiating heat from the high-pressure refrigerant in the cooling operation.

[0105] The blower fan 17 is an electric blower whose operating ratio, that is, whose number of revolutions (volume of blowing air) is controlled by a control voltage output from the air conditioning controller. The outdoor heat exchanger 16 has its outlet side connected to an electric three-way valve 15b. The three-way valve 15b has its operation controlled by a control voltage output from the air conditioning controller, and serves as a refrigerant flow path switch together with the above opening/closing valve 15a.

[0106] More specifically, in the heating operation, the three-way valve 15b performs switching to the refrigerant flow path for coupling the outlet side of the outdoor heat exchanger 16 to the inlet side of an accumulator 18 to be described later. In contrast, in the cooling operation, the three-way valve 15b performs switching to the refrigerant flow path for coupling the outlet side of the outdoor heat exchanger 16 to the inlet side of a fixed throttle 19 for cooling.

[0107] The fixed throttle 19 for cooling serves as a decompression portion for the cooling operation for decompressing and expanding the refrigerant flowing from the outdoor heat exchanger 16 in the cooling operation. The fixed throttle 19 has the same basic structure as that of the above fixed throttle 13 for heating. The outlet side of the fixed throttle 19 for cooling is coupled to the refrigerant inlet side of the refrigerant evaporator 20 as an indoor evaporator.

[0108] The refrigerant evaporator 20 is disposed on the upstream side of the air flow with respect to the refrigerant radiator 12 in the casing 31 of the indoor air conditioning unit 30. The refrigerant evaporator 20 is a heat exchanger for cooling that exchanges heat between the blowing air in the vehicle compartment and the refrigerant flowing therethrough to thereby cool the blowing air within the vehicle interior. A refrigerant outlet side of the refrigerant evaporator 20 is coupled to an inlet side of the accumulator 18.

[0109] The accumulator 18 is a gas-liquid separator for the low-pressure side refrigerant that separates the refrigerant flowing therein into liquid and gas phases, and which stores therein the excessive refrigerant within the cycle. A vapor-phase refrigerant outlet of the accumulator 18 is coupled to a suction side of the compressor 11. Thus, the accumulator 18 serves to suppress the suction of the liquid-phase refrigerant into the compressor 11 to thereby prevent the liquid compression of the compressor 11.

[0110] Next, the detailed structure of the refrigerant radiator 12 will be described using FIGS. 3(a) and 3(b). FIG. 3(a) is a front view of the refrigerant radiator 12, and FIG. 3(b) is an exemplary side view of FIG. 3(a). For easy understanding, FIG. 3(b) omits the illustration of an inlet side connector 122a and an outlet side connector 123a to be described later.

[0111] Vertical arrows shown in FIG. 3(a) indicate the respective upward and downward directions with the refrigerant radiator 12 mounted inside the casing 31 of the indoor air conditioning unit 30. The same goes for the following drawings.

[0112] Specifically, as shown in FIGS. 3(a) and 3(b), the refrigerant radiator 12 includes a plurality of tubes 121 through which the high-temperature and high-pressure refrigerant discharged from the compressor 11 flows, and a pair of header tanks 122 and 123 disposed on both sides in the longitudinal direction of the tubes 121 for collecting or distributing the refrigerant flowing through the tubes 121. The refrigerant radiator 12 is a full-path type multi-flow heat exchanger in which the flow directions of the refrigerants flowing through the respective tubes 121 are the same.

[0113] The tube 121 is formed of metal with excellent conductivity (for example, an aluminum alloy), and is a flat tube with a flat cross section in the direction perpendicular to the flow direction of the refrigerant flowing therethrough. Further, the flat outer surface of the tube 121 (flat surface) is disposed in parallel to the flow direction X of the blowing air in the vehicle compartment. The tube 121 may be formed of a flat tube with a single or multiple holes. The tube 121 is desirably comprised of a refrigerant flow path with a diameter d ($4 \times \text{flow path sectional area} \times \text{length of wet side of flow path}$) of 0.5 to 1.5 mm in terms of diameter of the circle.

[0114] The tubes 121 are laminated in the horizontal direction such that the flat surfaces of the tubes 121 are in parallel to each other. An air passage for allowing the blowing air in the vehicle compartment to flow therethrough is formed between the adjacent tubes 121. Also, a fin 124 for promoting the heat exchange between the refrigerant and the blowing air in the vehicle compartment is disposed in between the adjacent tubes 121.

[0115] Each fin 124 is a corrugated fin formed by bending a thin plate made of the same material as the tube 121, in a wave shape. The fin 124 has its top soldered to the flat surface of the tube 121. Although FIG. 3(a) illustrates only a part of the fins 124 for easy understanding, each fin 124 is disposed over the substantially entire region between the adjacent tubes 121.

[0116] The header tanks 122 and 123 are cylindrical members extending in the lamination direction of the tubes 121 (horizontally in this embodiment). In this embodiment, with the refrigerant radiator 12 mounted inside the casing 31 of the indoor air conditioning unit 30, the lower header tank is used as the header tank 122 for distribution of the refrigerant, and the upper header tank is used as the header tank 123 for collection of the refrigerant.

[0117] Each of the header tanks 122 and 123 is composed of a separation type header tank which is formed of the same material as the tube 121. The header tank is formed in a cylindrical shape, and includes a plate soldered to the end of each tube 121 in the longitudinal direction, and a tank member combined with the plate. Alternatively, the header tanks 122 and 123 may be formed of a cylindrical member or the like.

[0118] The inlet side connector 122a is disposed on one end of the lower header tank 122 for distribution of the refrigerant. The connector 122a serves as a connecting portion to a discharge port of the compressor 11. The connector 122a is provided with a refrigerant inlet for allowing the refrigerant to flow into the header tank 122. The other end of the header tank 122 is closed by a tank cap 122b serving as a closing member.

[0119] The outlet side connector 123a is disposed on one end of the upper header tank 123 for collection of the refrigerant. The connector 123a serves as a connecting portion with an inlet side of the fixed throttle 13 for heating and an inlet side of a bypass passage 14 for the fixed throttle. The connector 123a is provided with a refrigerant outlet for allowing the refrigerant to flow from the header tank 123. The other end of the header tank 123 is closed by a tank cap 123b serving as the closing member.

[0120] Thus, in the refrigerant radiator 12, as indicated by a thick arrow of FIG. 3(a), the refrigerant discharged from the compressor 11 flows into the header tank 122 for distribution of the refrigerant via the inlet side connector 122a, and is then distributed to the tubes 121. The refrigerant entering each tube 121 exchanges heat with the blowing air in the vehicle

compartment while flowing through the tubes 121, and then flows from the tubes 121. The refrigerants flowing from the tubes 121 are collected into the header tank 123 for collection of the refrigerant to flow out via the outlet side connector 123a. That is, the refrigerant flows through the tubes 121 from the lower side to the upper side.

[0121] At this time, as mentioned above, the heat pump cycle 10 of this embodiment forms the subcritical refrigerant cycle, so that the refrigerant flowing through each tube 121 changes from a gas-phase refrigerant with a degree of superheat, to a gas-liquid two-phase refrigerant and a liquid-phase refrigerant with a degree of supercooling in that order while exchanging heat with the vehicle-interior blowing air within the tubes 121.

[0122] In the refrigerant radiator 12 of this embodiment, as shown in FIG. 3(b), the tube 121 is disposed such that the longitudinal direction of the tube is slanted with respect to the horizontal direction. That is, the longitudinal direction of the tube 121 has at least a component directed vertically (in the up-and-down direction). In short, the flow direction of the refrigerant flowing through each tube 121 is slanted or vertical with respect to the horizontal direction.

[0123] This embodiment defines an inclination angle θ ($-90^\circ \leq \theta \leq 90^\circ$) as an angle formed between a line extending from the refrigerant flow upstream side of the refrigerant radiator 12 (header tank 122 for distribution of the refrigerant in this embodiment) as a starting point toward the refrigerant flow downstream side (header tank 123 for collection of the refrigerant in this embodiment), and another line extending from the refrigerant flow upstream side of the radiator 12 as a starting point in the horizontal direction.

[0124] That is, the inclination angle θ changes from 0 to 90° as the flow direction of the refrigerant flowing through the tube 121 changes from the horizontal direction to the vertically upward direction. For example, when the flow direction of refrigerant flowing through the tube 121 is directed in the horizontal direction, the inclination angle becomes zero (inclination angle $\theta = 0^\circ$). When the flow direction of refrigerant is directed vertically upward, the inclination angle becomes 90° (inclination angle $\theta = 90^\circ$). Further, when the flow direction of refrigerant is directed vertically downward, the inclination angle becomes -90° (inclination angle $\theta = -90^\circ$).

[0125] In this embodiment, the refrigerant radiator 12 is disposed so as to satisfy the above-mentioned formula 4 when X is a dryness of the refrigerant in a predetermined position where the refrigerant flowing through the tube 121 is a gas-liquid two-phase refrigerant, and Re is a Reynolds number of the refrigerant determined from an average flow velocity (in units of m/S) of the refrigerant flowing through the tube 121.

[0126] The above predetermined position of the refrigerant radiator 12 in this embodiment can be arbitrary as long as the refrigerant flowing through the tube 121 is the gas-liquid two-phase refrigerant in the predetermined position. For example, the predetermined position can be a part on the downstream side of the refrigerant flow of the tube 121, or a part of the tube 121 closer to the header tank 123 for collection of the refrigerant than the header tank 122 for distribution of the refrigerant.

[0127] Next, the indoor air conditioning unit 30 will be described below. The indoor air conditioning unit 30 is disposed inside a gauge board (instrument panel) at the forefront of the vehicle compartment. The unit 30 accommodates in the

casing **31** forming an outer envelope, a blower **32**, the above-mentioned refrigerant radiator **12**, and the refrigerant evaporator **20**.

[0128] The casing **31** forms an air passage for the blowing air in the vehicle compartment to be blown into the vehicle interior. The casing **31** is formed of resin (for example, polypropylene) having some degree of elasticity, and excellent strength. An inside/outside air switch **33** for switching between the air (inside air) in the vehicle compartment and the outside air is disposed on the most upstream side of the vehicle-interior blowing air flow in the casing **31**.

[0129] The inside/outside air switch **33** is provided with the inside air inlet for introducing the inside air into the casing **31**, and the outside air inlet for introducing the outside air thereinto. An inside/outside air switching door is positioned inside the inside/outside air switch **33** to continuously adjust the opening areas of the inside air inlet and the outside air inlet to thereby change the ratio of volume of the inside air to the outside air.

[0130] The blower **32** for blowing the air sucked via the inside/outside air switch **33** into the vehicle interior is disposed on the downstream side of the air flow of the inside/outside air switch **33**. The blower **32** is an electric blower which includes a centrifugal multiblade fan (sirocco fan) driven by an electric motor, and whose number of revolutions (volume of blowing air) is controlled by a control voltage output from the air conditioning controller.

[0131] The refrigerant evaporator **20** and the refrigerant radiator **12** are disposed on the downstream side of the air flow of the blower **32** in that order with respect to the flow of the blowing air in the vehicle compartment. In short, the refrigerant evaporator **20** is disposed on the upstream side in the flow direction of the blowing air in the vehicle compartment with respect to the refrigerant radiator **12**.

[0132] An air mix door **34** is disposed on the downstream side of the air flow in the refrigerant evaporator **20** and on the upstream side of the air flow in the refrigerant radiator **12**. The air mix door **34** adjusts the rate of volume of the air passing through the refrigerant radiator **12** in the blowing air having passed through the refrigerant evaporator **20**. A mixing space **35** is provided on the downstream side of the air flow in the refrigerant radiator **12** so as to mix the blowing air exchanging heat with the refrigerant and heated at the refrigerant radiator **12**, and the blowing air bypassing the refrigerant radiator **12** and not heated.

[0133] Opening holes for blowing the conditioned air mixed in the mixing space **35**, into the vehicle interior as a space to be cooled are disposed on the most downstream side of the air flow in the casing **31**. Specifically, the opening holes (not shown) include a face opening hole for blowing the conditioned air toward the upper body of a passenger in the vehicle compartment, a foot opening hole for blowing the conditioned air toward the foot of the passenger, and a defroster opening hole for blowing the conditioned air toward the inner side of a front glass of the vehicle.

[0134] The air mix door **34** adjusts the rate of volume of air passing through the refrigerant radiator **12** to thereby adjust the temperature of conditioned air mixed in the mixing space **35**, thus controlling the temperature of the conditioned air blown from each opening hole. That is, the air mix door **34** serves as a temperature adjustment portion for adjusting the temperature of the conditioned air blown into the vehicle interior.

[0135] In short, the air mix door **34** serves as a heat exchanging amount adjustment portion for adjusting the amount of heat to be exchanged between the blowing air in the vehicle compartment and the refrigerant discharged from the compressor **11** in the refrigerant radiator **12**. The air mix door **34** is driven by a servo motor (not shown) whose operation is controlled based on the control signal output from the air conditioning controller.

[0136] The face opening hole, foot opening hole, and defroster opening hole have, at the respective upstream sides of the air flows thereof, a face door for adjusting an opening area of the face opening hole, a foot door for adjusting an opening area of the foot opening hole, and a defroster door for adjusting an opening area of the defroster opening hole, respectively (all doors being not shown).

[0137] The face door, foot door, and defroster door serve as an opening hole mode switching portion for switching among opening hole modes. The doors are driven by a servo motor (not shown) whose operation is controlled based on a control signal output from the air conditioning controller via a link mechanism or the like.

[0138] In contrast, the downstream sides of the air flows of the face opening hole, the foot opening hole, and the defroster opening hole are connected to a face air outlet, a foot air outlet, and a defroster air outlet provided in the vehicle compartment, respectively, via ducts forming the respective air passages. For example, as shown in FIG. 4, the face opening hole leads to a front face air outlet **P1** provided in the center of the instrument panel **P** in the horizontal direction, and side face air outlets **P2** provided on both ends of the panel **P** in the horizontal direction.

[0139] As can be seen from FIG. 4, the front face air outlets **P1** and the side face air outlets **P2** are provided in positions for a driver seat and a front passenger seat, respectively. For example, in the heating operation, the blowing air heated by the heat exchange portion on the driver seat side of the refrigerant radiator **12** is blown mainly toward the driver seat, while the blowing air heated by the heat exchanger region on the front passenger-seat side is blown mainly toward the front passenger seat.

[0140] Next, an electric controller of this embodiment will be described later. The air conditioning controller is comprised of the known microcomputer including a CPU, a ROM, and a RAM, and peripheral circuits thereof. The controller controls the operation of each of various types of air conditioning controller **11**, **15a**, **15b**, **17**, and **32** connected to its output by executing various operations and processing based on air conditioning control programs stored in the ROM.

[0141] A group of various sensors for control of air conditioning is coupled to the input side of the air conditioning controller. The sensors include an inside air sensor for detecting a temperature of the vehicle interior, an outside air sensor for detecting a temperature of the outside air, a solar radiation sensor for detecting an amount of solar radiation in the vehicle interior, and an evaporator temperature sensor for detecting a temperature of air blown from the refrigerant evaporator **20** (evaporator temperature). And, the sensors also include a discharge refrigerant temperature sensor for detecting a temperature of the refrigerant discharged from the compressor **11**, and an outlet refrigerant temperature sensor for detecting a refrigerant temperature on the outlet side of the outdoor heat exchanger **16**.

[0142] An operation panel (not shown) disposed near an instrument board at the front of the vehicle compartment is

connected to the input side of the air conditioning controller. Operation signals are input from various types of air conditioning operation switches provided on the operation panel. Various air conditioning operation switches provided on the panel include an operation switch for the air conditioner for the vehicle, a vehicle-interior temperature setting switch for setting the temperature of the vehicle interior, and a selection switch for selecting an operation mode.

[0143] The air conditioning controller includes a control portion for controlling the electric motor **11b** for the compressor **11**, the opening/closing valve **15a**, the three-way valve **15b**, and the like which are integral with each other, and is designed to control the operations of these components. In the air conditioning controller of this embodiment, the structure (hardware and software) for controlling the operation of the compressor **11** serves as a refrigerant discharge capacity control portion. The structure for controlling the operations of the respective devices **15a** and **15b** forming the refrigerant flow path switching portion serves as a refrigerant flow path control portion.

[0144] Next, the operation of the vehicle air conditioner **1** with the above arrangement in this embodiment will be described below. The vehicle air conditioner **1** of this embodiment can execute the heating operation for heating the vehicle interior, and the cooling operation for cooling the vehicle interior as mentioned above. Now, each operation will be explained in the following.

(a) Heating Operation

[0145] The heating operation is started when the heating operation mode is selected by the selection switch with the operation switch of the operation panel turned on (ON). In the heating operation, the air conditioning controller closes the opening/closing valve **15a**, and switches the three-way valve **15b** to the refrigerant flow path for coupling the outlet side of the outdoor heat exchanger **16** to the inlet side of the accumulator **18**. Thus, the heat pump cycle **10** is switched to the refrigerant flow path for allowing the refrigerant to flow as indicated by a solid arrow in FIG. 1.

[0146] The air conditioning controller with the above refrigerant flow paths reads a detection signal from the above sensor group for the air conditioning control and an operation signal from the operation panel. Based on the detection signal and the operation signal, a target outlet air temperature TAO is calculated as the target temperature of the air to be blown into the vehicle interior. Further, the operating states of various air conditioning control components connected to the output side of the air conditioning controller are determined based on the calculated target outlet air temperature TAO and the detection signal from the sensor group.

[0147] For example, the refrigerant discharge capacity of the compressor **11**, that is, a control signal output to the electric motor of the compressor **11** is determined as follows. First, a target evaporator outlet air temperature TEO of the refrigerant evaporator **20** is determined based on the target outlet air temperature TAO with reference to a control map previously stored in the air conditioning controller.

[0148] Based on a deviation between the target evaporator outlet air temperature TEO and the blown air temperature from the refrigerant evaporator **20** detected by the evaporator temperature sensor, the control signal to be output to the electric motor of the compressor **11** is determined such that the blown air temperature of the air blown from the refrigerant

evaporator **20** approaches the target evaporator outlet air temperature TEO by use of a feedback control method.

[0149] The control signal to be output to the servo motor of the air mix door **34** is determined based on the target outlet air temperature TAO, the blown air temperature of the refrigerant evaporator **20**, and the temperature of the refrigerant discharged from the compressor **11** which is detected by the refrigerant temperature sensor such that the temperature of air blown into the vehicle interior becomes a desired temperature set by the passenger using the vehicle-interior temperature setting switch.

[0150] As shown in FIG. 1, during the heating operation, the opening degree of the air mix door **34** may be controlled such that all the vehicle-interior blowing air blown from the blower **32** passes through the refrigerant radiator **12**.

[0151] Then, the control signals determined as described above are output to various air conditioning control components. Thereafter, until the stopping of the vehicle air conditioner is requested by the operation panel, a control routine is repeated at every predetermined control cycle. The control routine involves a series of processes: reading of the above detection signal and operation signal, calculation of the target outlet air temperature TAO, determination of the operating states of various air conditioning control components, and output of the control voltage and the control signal in that order. Such repetition of the control routine is basically performed in the cooling operation in the same way.

[0152] In the heat pump cycle **10** during the heating operation, the high-pressure refrigerant discharged from the compressor **11** flows into the refrigerant radiator **12**. The refrigerant flowing into the refrigerant radiator **12** exchanges heat with the vehicle interior blowing air blown from the blower **32** through the refrigerant evaporator **20** to radiate the heat therefrom, so that the blowing air in the vehicle compartment is heated.

[0153] The high-pressure refrigerant flowing from the refrigerant radiator **12** flows into the fixed throttle **13** for heating to be decompressed and expanded by the throttle because the opening/closing valve **15a** is closed. The low-pressure refrigerant decompressed and expanded by the fixed throttle **13** for heating flows into an outdoor heat exchanger **16**. The low-pressure refrigerant flowing into the outdoor heat exchanger **16** absorbs heat from the outside air blown by the blower fan **17** to evaporate itself.

[0154] The three-way valve **15b** is switched to a refrigerant flow path for coupling the outlet of the outdoor heat exchanger **16** to the inlet of the accumulator **18**, so that the refrigerant flowing from the outdoor heat exchanger **16** flows into the accumulator **18** to be separated into gas and liquid phases. The gas-phase refrigerant separated by the accumulator **18** is sucked into and compressed again by the compressor **11**.

[0155] As mentioned above, in the heating operation, the blowing air in the vehicle compartment is heated by the refrigerant radiator **12** with the heat contained in the refrigerant discharged from the compressor **11**, which can perform the heating operation of the vehicle interior as the space for air conditioning.

(b) Cooling Operation

[0156] The cooling operation is started when the cooling operation mode is selected by the selection switch with the operation switch of the operation panel turned on (ON). In the cooling operation, the air conditioning controller opens the

opening/closing valve **15a**, and switches the three-way valve **15b** to the refrigerant flow path for connecting the outlet side of the outdoor heat exchanger **16** to the inlet side of the fixed throttle **19** for cooling. Thus, the heat pump cycle **10** is switched to the refrigerant flow path for making the refrigerant flow as indicated by the solid arrow in FIG. 2.

[0157] In the heat pump cycle **10** during the cooling operation, the high-pressure refrigerant discharged from the compressor **11** flows into the refrigerant radiator **12**, and exchanges heat with the blowing air in the vehicle compartment blown from the blower **32** and having passed through the refrigerant evaporator **20** to radiate heat therefrom. The high-pressure refrigerant flowing from the refrigerant radiator **12** flows into the outdoor heat exchanger **16** via the bypass passage **14** for the fixed throttle because the opening/closing valve **15a** is opened.

[0158] The low-pressure refrigerant flowing into the outdoor heat exchanger **16** further radiates heat toward the outside air blown by the blower fan **17**. The three-way valve **15b** is switched to the refrigerant flow path for coupling the outlet side of the outdoor heat exchanger **16** to the inlet side of the fixed throttle **19** for cooling, so that the refrigerant flowing from the outdoor heat exchanger **16** is decompressed and expanded by the fixed throttle **19** for cooling.

[0159] The refrigerant flowing from the fixed throttle **19** for cooling flows into the refrigerant evaporator **20**, and absorbs heat from the blowing air in the vehicle compartment blown by the blower **32** to evaporate itself. In this way, the blowing air in the vehicle compartment can be cooled. The refrigerant flowing from the refrigerant evaporator **20** flows into the accumulator **18**, and is then separated into liquid and gas phases by the accumulator **18**.

[0160] The gas-phase refrigerant separated by the accumulator **18** is sucked into and compressed by the compressor **11** again. As mentioned above, during the cooling operation, the low-pressure refrigerant absorbs heat from the blowing air in the vehicle compartment and evaporates itself at the refrigerant evaporator **20** to thereby cool the blowing air in the vehicle compartment, which can perform cooling of the vehicle interior.

[0161] During the cooling operation, when the passenger sets the temperature higher than the vehicle interior temperature by the vehicle-interior temperature setting switch, the opening degree of the air mix door **34** is adjusted such that the temperature of the blowing air in the vehicle compartment is higher than the vehicle interior temperature. In such a case, the refrigerant evaporator **20** can cool the blowing air in the vehicle compartment and reduce the absolute humidity to achieve the dehumidification and heating of the vehicle interior.

[0162] As mentioned above, the vehicle air conditioner **1** of this embodiment can perform switching between the refrigerant flow paths of the heat pump cycle **10** to execute the heating operation, the cooling operation, and the dehumidification and heating operation.

[0163] In this embodiment, the refrigerant radiator **12** is arranged in the indoor air conditioning unit **30** so as to satisfy the relationship expressed by the above formula 4. Such arrangement taking into consideration the parameters, including the flow velocity U , viscosity μ , and density ρ of the refrigerant, and the inclination angle θ , can prevent the condensed refrigerant from remaining in the specific tube **121** even in the refrigerant radiator **12** that causes the phase change of the refrigerant flowing through the tubes **121**.

[0164] Thus, this embodiment can suppress the non-uniform loss in pressure caused in the refrigerant flowing through the tubes **121** to thereby prevent the formation of the specific heat exchange portion with the relatively low temperature in the heat exchange portions of the refrigerant radiator. As a result, even when fluctuations in load on air conditioning of the heat pump cycle **10** or the like change the flow velocity of the refrigerant flowing through the tubes **121**, the reduction in heat radiation performance of the refrigerant radiator **12** can be suppressed to reduce the non-uniform temperature distribution in the horizontal direction of air heated and blown by the refrigerant radiator **12**.

[0165] As mentioned above, in the vehicle air conditioner **1** of this embodiment, the air heated by the driver-seat side heat exchange portion of the refrigerant radiator **12** is blown mainly toward the driver seat, whereas the air heated by the front passenger-seat side heat exchange portion is blown mainly toward the front passenger seat. Thus, this embodiment can provide the refrigerant radiator **12** for decreasing the difference in temperature in the horizontal direction of blowing air flowing from the refrigerant radiator **12**, which is very effective in reduction of the difference in temperature between the blowing air blown toward the driver seat, and the blowing air blown toward the front passenger seat.

[0166] In the multi-flow heat exchanger, like the refrigerant radiator **12** of this embodiment, the flow rate of refrigerant flowing through each tube **121** is more likely to be changed depending on the positions of the refrigerant inlet of the inlet side connector **122a** and the refrigerant outlet of the outlet side connector **123a** provided in the header tanks **122** and **123**, respectively, which leads to the non-uniform temperature distribution of the blowing air. Thus, the application of the arrangement conditions that can reduce the difference in temperature of the blowing air in the refrigerant radiator **12** is very effective.

[0167] In the case of the refrigerant radiator that allows the refrigerant to flow through the tube **121** from the lower side to the upper side, the refrigerant radiator **121** is desirably disposed so as to satisfy the above formula 4.

[0168] Thus, as indicated by the above formula 4, the parameters A to G are represented by a function of an inclination angle θ , which can suppress the non-uniform temperature distribution in the horizontal direction of the air blown from the refrigerant radiator at any inclination angle θ when the flow direction of refrigerant flowing through the tube **121** is directed upward.

Second Embodiment

[0169] In this embodiment, as shown in FIGS. **5(a)** and **5(b)**, some changes are made to the positions of the inlet side connector **122a** and the outlet side connector **123a** of the first embodiment, by way of example. FIGS. **5(a)** and **5(b)** correspond to FIGS. **3(a)** and **3(b)**, respectively, and the same or equivalent parts as those in FIGS. **3(a)** and **3(b)** are represented by the same reference characters. The same goes for the following drawings.

[0170] In the refrigerant radiator **12** of this embodiment, the inlet side connector **122a** is positioned on the upper side, and the outlet side connector **123a** is positioned on the lower side, so that the refrigerant flows through the tubes **121** from the upper side to the lower side.

[0171] The refrigerant radiator for allowing the refrigerant to flow through the tubes **121** from the upper side to the lower side does not need to take into consideration the loss in

pressure due to the gravity acting on the condensed refrigerant, and can suppress the non-uniform loss in pressure caused by the refrigerant flowing through the tubes **121** without considering the inclination angle θ to thereby prevent the formation of the specific heat exchanger range with the relatively low temperature in the heat exchange portions of the refrigerant radiator.

[0172] Thus, it is not necessary to place the tubes **121** such that the longitudinal direction of the tube **121** is slanted with respect to the horizontal direction. In short, the tube **121** is disposed to have its longitudinal direction directed substantially vertically, causing the refrigerant to flow through the tubes **121** substantially vertically.

[0173] As shown in FIG. 10, a parameter similarly representing the state of flow (influence of gravity) with respect to the refrigerant flow includes a Grashof number Gr calculated by a gravitational acceleration g , a cubical expansion force β , a viscosity μ , a density ρ , and the like. In a region satisfying the following relationship between the Reynolds number Re and the Grashof number Gr of the refrigerant flowing through the tubes **121**: $Re^2 > Gr$, that is, in a region satisfying the formula of $62.42 \leq Re$, the flow of refrigerant flowing through the tubes **121** becomes a forced convection, which results in a large flow velocity of the refrigerant passing through the tubes **121**, thus making the flow velocities of the refrigerant in the tubes **121** non-uniform to cause the non-uniform temperature distribution of the respective heat exchange portions.

[0174] As shown in FIG. 8, in a region with a refrigerant flow rate Gr of 47 kg/h or less, in other words, in a region of $Re \leq 1234$, the refrigerant evaporator for allowing the refrigerant to flow through the tube **121** from the lower side to the upper side decreased its heat radiation performance, regardless of the inclination angle θ of the tube **121**, as compared to the case where the refrigerant flows through the tube **121** from the upper side to the lower side.

[0175] Thus, in the region that satisfies the formula of $62.42 \leq Re \leq 1234$, the flow of refrigerant flowing through the tubes **121** is desirably directed from the upper side to the lower side with less influence of the gravity on the condensed refrigerant.

Third Embodiment

[0176] In this embodiment, as shown in FIGS. 6(a) and 6(b), some changes are made to the structure of the refrigerant radiator **12** of the first embodiment. In the refrigerant radiator **12** of this embodiment, a separator **123c** is disposed in the upper header tank **123** to separate the internal space of the header tank **123** into two sections in the longitudinal direction, namely, a distribution space **123d** and a collection space **123e**. FIGS. 6(a) and 6(b) correspond to FIGS. 3(a) and 3(b), and 5(a) and 5(b) of the first embodiment, respectively.

[0177] The tubes **121** of this embodiment are classified into a first tube group **121a** coupled to the collection space **123e** of the upper header tank **123**, and a second tube group **121b** coupled to the distribution space **123d** thereof. The upper header tank **123** is further coupled to an inlet side connector **123f** for allowing the refrigerant discharged from the compressor **11** to flow into the distribution space **123d**, and to an outlet side connector **123a** for allowing the refrigerant to flow from the inside of the collection space **123e**.

[0178] Thus, in the refrigerant radiator **12** of this embodiment, as indicated by a thick arrow of FIG. 6, the refrigerant discharged from the compressor **11** flows into the distribution

space **123d** of the upper header tank **123** via the inlet side connector **123f** to be distributed to the tubes **121** forming the second tube group **121b**.

[0179] Then, the refrigerant flowing into the tubes forming the second tube group **121b** exchanges heat with the blowing air in the vehicle compartment in passing through the tubes **121**, and flow out of the tubes **121**. The refrigerants flowing from the tubes **121** forming the second tube group **121b** are collected into the lower header tank **122**, and then are distributed to the tubes **121** forming the first tube group **121a**.

[0180] Further, the refrigerant flowing into the tubes **121** forming the first tube group **121a** exchanges heat with the blowing air in the vehicle compartment in passing through the tubes **121**, and flows out of the tubes **121**. The refrigerant flowing from the tubes **121** forming the first tube group **121a** are collected in the collection space **123e** of the upper header tank **122**, and then flow from the space via the outlet side connector **123a**.

[0181] That is, in the refrigerant radiator **12** of this embodiment, the refrigerant flowing through the second tube group **121b** flows from the upper side to the lower side, whereas the refrigerant flowing through the first tube group **121a** flows from the lower side to the upper side.

[0182] In the refrigerant radiator **12** of this embodiment, the refrigerant in the gas-phase state exchanges heat during flowing through the second tube group **121b**, and becomes a gas-liquid two-phase refrigerant during flowing from an intermediate part to a downstream part (which is enclosed by a circle indicated by a broken line of FIG. 6) of the first tube group **121a** in the refrigerant flow direction. Then, the refrigerant becomes a liquid-phase refrigerant on the downstream side of the first tube group.

[0183] Thus, in this embodiment, the predetermined position can be any arbitrary position from the intermediate part to the downstream part of the first tube group **121a** in the refrigerant flow direction. In the above position, the refrigerant flows from the lower side to the upper side, and thus the inclination angle θ of the refrigerant radiator **12** of this embodiment is set to the same as that in the first embodiment. The structure and operation of other components of the vehicle air conditioner **1** are the same as those in the first embodiment.

[0184] In the refrigerant radiator **12** of this embodiment, in the heat exchange portion formed by the second tube group **121b**, the refrigerant radiates heat therefrom in the gas-phase state as it is, which hardly causes the degradation in heat radiation performance due to the difference in condensation of the refrigerant between the tubes **121**. As a result, the temperature of blowing air in the vehicle compartment blown from the heat exchange portion formed by the second tube group **121b** is less likely to vary between the tubes.

[0185] In contrast, in the heat exchange portion formed by the first tube group **121a**, the arrangement conditions represented by the above formula 4 are satisfied, which can provide the same effects as those of the first embodiment.

[0186] As a result, even when the flow velocity of refrigerant flowing through the tubes **121** varies depending on fluctuations in load on air conditioning of the heat pump cycle **10**, the entire refrigerant radiator **12** can suppress the reduction in heat radiation performance and also can decrease the difference in temperature in the horizontal direction of air heated and blown by the refrigerant radiator **12**.

[0187] Also, in the refrigerant radiator **12** of this embodiment, even when the flow direction of refrigerant is directed

from the upper side to the lower side in the predetermined position where the refrigerant flowing through the tubes **121** becomes the gas-liquid two-phase refrigerant, the refrigerant radiator **12** is arranged to satisfy the relationship represented by the above formula 4, which can provide the same effects.

Fourth Embodiment

[0188] In this embodiment, as shown in FIGS. 7(a) and 7(b), some changes are made to the structure of the refrigerant radiator **12** of the first embodiment by way of example. In the refrigerant radiator **12** of this embodiment, the internal space of the upper header tank **123** is separated in the flow direction of the blowing air into a distribution space **123d** and a collection space **123e**. FIGS. 7(a) and 7(b) correspond to FIGS. 3(a) and 3(b) and FIGS. 5(a) and 5(b) of the first embodiment, respectively.

[0189] Like the second embodiment, the tubes **121** of this embodiment are also classified into the first tube group **121a** coupled to the collection space **123e**, and the second tube group **121b** coupled to the distribution space **123d**. Further, the first tube group **121a** is positioned on the downstream side of the second tube group **121b** in the flow direction X of the blowing air in the vehicle compartment. In short, the tubes **121** are arranged in a plurality of lines (in this embodiment, in two lines) in the flow direction X of the blowing air in the vehicle compartment.

[0190] The header tank **123** is provided with an inlet side connector **123f** for allowing the refrigerant discharged from the compressor **11** to flow into the distribution space **123d**, and an outlet side connector **123a** for allowing the refrigerant to flow from the inside of the collection space **123e**.

[0191] Thus, also in the refrigerant radiator **12** of this embodiment, as indicated by a thick arrow of FIG. 7(a), the refrigerant discharged from the compressor **11** flows from the distribution space **123d** of the upper header tank **123** to the tubes **121** on the upstream side of air flow included in the second tube group **121b**, the lower header tank **122**, the tubes **121** on the downstream side of air flow included in the first tube group **121a**, and the collection space **123e** of the upper header tank **122** in that order, and then flows from the radiator via the outlet side connector **123a**.

[0192] In the refrigerant radiator **12** of this embodiment, the refrigerant exchanges heat in the gas-phase state during flowing through the second tube group **121b** on the upstream side of the air flow, and becomes a gas-liquid two-phase refrigerant during flowing from an intermediate part to a downstream part in the refrigerant flow direction of the first tube group **121a** located on the downstream side of the air flow. Then, the refrigerant becomes a liquid-phase refrigerant on the downstream side of the first tube group.

[0193] Thus, in this embodiment, the predetermined position can be any arbitrary position from the intermediate part to the downstream part of the first tube group **121a** in the refrigerant flow direction. The inclination angle θ of the refrigerant radiator **12** of this embodiment is set to the same as that in the first embodiment. The structure and operation of other components of the vehicle air conditioner **1** are the same as those in the first embodiment.

[0194] The refrigerant radiator **12** of this embodiment is structured as mentioned above. In the heat exchange portion on the air flow upstream side formed by the second tube group **121b**, the refrigerant radiates heat therefrom in the gas-phase state as it is, which hardly causes the degradation in heat radiation performance due to the difference in condensation

of the refrigerant, resulting in less non-uniform temperature distribution of the vehicle-interior blowing air blown from the heat exchange portion.

[0195] In contrast, in the heat exchange portion on the air flow downstream side of the first tube group **121a**, the arrangement conditions represented by the above formula 4 are satisfied, which can provide the same effects as those of the first embodiment. Thus, the entire refrigerant radiator **121** can suppress the reduction in heat radiation performance of the radiator and also can suppress the formation of the non-uniform temperature distribution in the horizontal direction of air heated and blown by the refrigerant radiator **12**.

[0196] In the refrigerant radiator **12** of this embodiment, as shown in FIG. 7(b), the refrigerant distributed by the distribution space **123d** of the upper header tank **123** turns around via the lower header tank **122** to return to the collection space **123e** of the upper header tank **123**.

[0197] Thus, a region (superheat region) through which the gas-phase refrigerant with a relatively high degree of superheat flows can be formed on the upper side in the windward-side heat exchange portion, and a region (supercooling region) through which the liquid-phase refrigerant with a relatively low degree of supercooling can be formed on the upper side in the leeward-side heat exchange portion. Thus, the superheat region and the supercooling region can be superimposed over each other as viewed in the flow direction X of the blowing air, which can also suppress the non-uniform temperature distribution of the blowing air in the vertical direction.

[0198] The refrigerant radiator **12** of this embodiment allows the refrigerant to flow from the tubes **121** on the upstream side of the blowing air flow, and to turn around into the downstream side tube **121**, by way of example. Alternatively, the refrigerant flowing from the downstream side tube **121** may be turned around into the upstream side tube **121**.

[0199] Also, even when the flow direction of refrigerant in the predetermined position, which flows through the tube **121** in the gas-liquid two-phase state, is directed from the upper side to the lower side, the refrigerant radiator **12** of this embodiment can also arrange the refrigerant radiator **12** so as to satisfy the above relationship represented by the formula 4 to provide the same effects.

Other Embodiments

[0200] The present invention is not limited to the above embodiments, and various modifications and changes can be made to the disclosed embodiments without departing from the scope of the invention.

[0201] (1) Although in the above embodiments, the tube **121** extending in one direction is used as the tube **121** of the refrigerant radiator **12** by way of example, the tube **121** applicable to the refrigerant radiator **12** of the disclosure is not limited thereto. That is, any other tube with at least one component extending vertically may be used even though the tube is formed in a meandering shape or the like.

[0202] For example, the tube **121** may be curved in a U-like shape, and the inlet and outlet of the tube **121** may be positioned on the same side of the tube **121** in the longitudinal direction. Such a tube can be used to achieve the substantially same refrigerant radiator **12** as that of the third embodiment, which can remove the header tank **122** disposed on the lower side.

[0203] (2) The refrigerant radiator **12** in the above embodiments is adapted to exchange heat between the refrigerant and

the blowing air in the vehicle compartment by way of example, but the heat radiator 12 of the disclosure is not limited thereto. For example, the heat radiator may be adapted to be capable of exchanging heat among a plurality of kinds of fluids, including refrigerant, blowing air in the vehicle compartment, another heat medium, and the like.

[0204] A heat exchanger that can exchange heat among the plurality of fluids includes refrigerant tubes for allowing the refrigerant to flow therethrough, and heat medium tubes for allowing the heat medium to flow therethrough, both the refrigerant tubes and the heat medium tubes being alternately laminated over each other. An air passage through which the blowing air flows is formed between the adjacent refrigerant tube and heat medium tube. And fins are in the air passages to be coupled to both the refrigerant tubes and the heat medium tubes to promote the heat exchange between the refrigerant and the blowing air, and between the heat medium and the blowing air, while enabling the heat transfer between the refrigerant and the heat medium.

[0205] (3) In the above embodiments, the refrigerant radiator 12 is applied to the vehicle air conditioner by way of example. Some devices mounted on the vehicle change the position thereof with respect to the horizontal direction when the entire vehicle inclines upon acceleration or deceleration, turning left or right, or stopping or parking on an upslope. Thus, each of the above embodiments desirably satisfies the relationship represented by the above formula 4 with respect to the inclination angle θ in the entire range of $\theta \pm \Delta\theta$, taking into consideration a change $\Delta\theta$ due to the inclination of the entire vehicle mentioned above.

[0206] (4) Although the above embodiments have described the application of the heat pump cycle 10 with the refrigerant radiator 12 of the disclosure to the vehicle air conditioner by way of example, the application of the heat pump cycle 10 with the refrigerant radiator 12 of the disclosure is not limited thereto. For example, the heat pump cycle 10 with the refrigerant radiator 12 may be applied to a stationary air conditioner, a low temperature storage, a cooling and heating device for a vending machine, and the like.

[0207] The present invention has been disclosed with reference to the preferred embodiments. However, it is to be understood that the present invention is not limited to the preferred embodiments and the structures disclosed above. The invention is intended to cover various modifications and equivalent arrangements. In addition, other preferred embodiments which include one additional element or which lose one element with respect to the disclosed embodiments, or various other combinations of the embodiments also fall within the scope and spirit of the present invention.

What is claimed is:

1. A refrigerant radiator, which is used for a vapor-compression refrigerant cycle, the refrigerant radiator being adapted to exchange heat between a high-temperature and high-pressure refrigerant compressed by a compressor of the vapor-compression refrigerant cycle and air blown into a space for air conditioning to thereby radiate heat from a gas-phase refrigerant having a degree of superheat so as to transfer the gas-phase refrigerant to a liquid-phase refrigerant having a degree of supercooling, the refrigerant radiator comprising:

- a plurality of tubes for allowing the refrigerant to flow therethrough from an upper side to a lower side;
- a first header coupled to ends of the tubes to distribute the refrigerant to flow into at least a part of the tubes; and

a second header coupled to the other ends of the tubes to collect the refrigerant flowing from at least a part of the tubes.

2. The refrigerant radiator according to claim 1, wherein the tubes are disposed to satisfy the following relationship:

$$62.42 \leq Re \leq 1234$$

wherein Re is a Reynolds number of the refrigerant in a predetermined position that is determined from an average flow velocity of the refrigerant flowing through the tube.

3. A refrigerant radiator, which is used for a vapor-compression refrigerant cycle, the refrigerant radiator being adapted to exchange heat between a high-temperature and high-pressure refrigerant compressed by a compressor of the vapor-compression refrigerant cycle and air blown into a space for air conditioning to thereby radiate heat from a gas-phase refrigerant having a degree of superheat so as to transfer the gas-phase refrigerant to a liquid-phase refrigerant having a degree of supercooling, the refrigerant radiator comprising:

a plurality of tubes for allowing the refrigerant to flow therethrough,

wherein the tube extends in a direction perpendicular to a horizontal direction, or at an angle with respect to the horizontal direction,

wherein the tubes are disposed so as to satisfy the following relationship:

$$Re \geq AxX^6 + BxX^5 + CxX^4 + DxX^3 + ExX^2 + FxX + G$$

$$A = -0.0537 \times \theta^2 + 9.7222 \times \theta + 407.19$$

$$B = -(-0.2093 \times \theta^2 + 37.88 \times \theta + 1586.3)$$

$$C = -0.3348 \times \theta^2 + 60.592 \times \theta + 2538.1$$

$$D = -(-0.2848 \times \theta^2 + 51.53 \times \theta + 2158.2)$$

$$E = -0.1402 \times \theta^2 + 25.365 \times \theta + 1062.8$$

$$F = -(-0.0418 \times \theta^2 + 7.5557 \times \theta + 316.46)$$

$$G = -0.0132 \times \theta^2 + 2.3807 \times \theta + 99.73$$

wherein $\theta(^{\circ})$ is an inclination angle formed by a flow direction of the refrigerant flowing through the tube and the horizontal direction; X is a dryness of the refrigerant in a predetermined position where the refrigerant flowing through the tube is a gas-liquid two-phase refrigerant; and Re is a Reynolds number of the refrigerant in the predetermined position that is determined from an average flow velocity of the refrigerant flowing through the tube, and

wherein as the flow direction of the refrigerant flowing through the tube changes from a vertically downward side to a vertically upward side, the inclination angle changes in a range of more than 0° and not more than 90° ($0 < \theta \leq 90^{\circ}$).

4. The refrigerant radiator according to claim 3, further comprising a header tank disposed at least at one side end of each of the tubes to extend in a lamination direction of the tubes, to collect or distribute the refrigerant.

5. The refrigerant radiator according to claim 3, wherein the tubes include a first tube group for allowing the refrigerant to flow therethrough from a lower side to an upper side, and a second tube group for allowing the refrigerant to flow therethrough from the upper side to the lower side.

6. The refrigerant radiator according to claim 1, wherein an internal space of the header tank is separated into a plurality of spaces,

one separated space is provided with a refrigerant inlet for allowing the gas-phase refrigerant to flow into the one separated space, and

the other separated space is provided with a refrigerant outlet for allowing the liquid-phase refrigerant to flow therefrom.

7. The refrigerant radiator according to claim 1, wherein the tubes are arranged in a flow direction of the air.

8. The refrigerant radiator according to claim 1, wherein the flow direction of the refrigerant flowing through the tubes is on the same direction.

9. The refrigerant radiator according to claim 1, wherein the refrigerant cycle is used for a vehicle air conditioner, and the space for air conditioning is an interior of a vehicle.

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