US 20250155171A1

(19) United States
(12) Patent Application Publication
TATSUTA et al.
(10) Pub. No.: US 2025/0155171 A1
(43) Pub. Date: May 15, 2025

(54) REFRIGERANT DISTRIBUTOR, HEAT EXCHANGER, AND REFRIGERATION CYCLE DEVICE

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(21) Appl. No.: **18/721,766**

(22) PCT Filed: **Feb. 2, 2023**

(86) PCT No.: **PCT/JP2023/003414**

§ 371 (c)(1),
(2) Date: **Jun. 19, 2024**

(30) **Foreign Application Priority Data**

Feb. 9, 2022 (JP) 2022-019015

Publication Classification

(51) **Int. Cl.**
F25B 41/42 (2021.01)

(52) **U.S. Cl.**
CPC *F25B 41/42* (2021.01)

(57) **ABSTRACT**

A refrigerant distributor comprises a mixing portion having a cylindrical shape. The mixing portion has an inlet for inflow therethrough of a refrigerant at an upstream end of the mixing portion. The mixing portion has a plurality of outlets for outflow therethrough of the refrigerant at a downstream end of the mixing portion opposite to the upstream end. The mixing portion has a recess facing the inlet at the downstream end of the mixing portion. The mixing portion guides the refrigerant flowing in from the inlet to flow along the upstream end of the mixing portion and a side wall of the mixing portion to diffuse the refrigerant in a circumferential direction of the downstream end, and then sends out the refrigerant from the plurality of outlets.

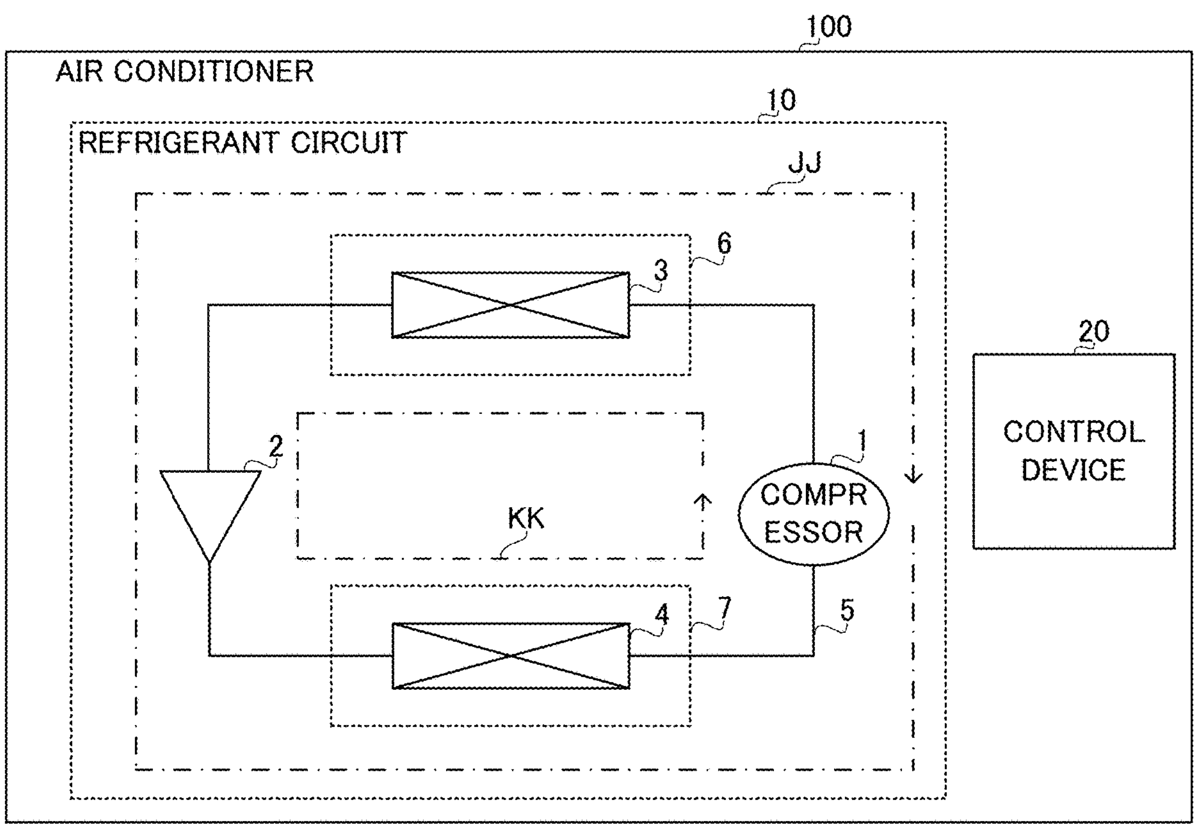


FIG.1

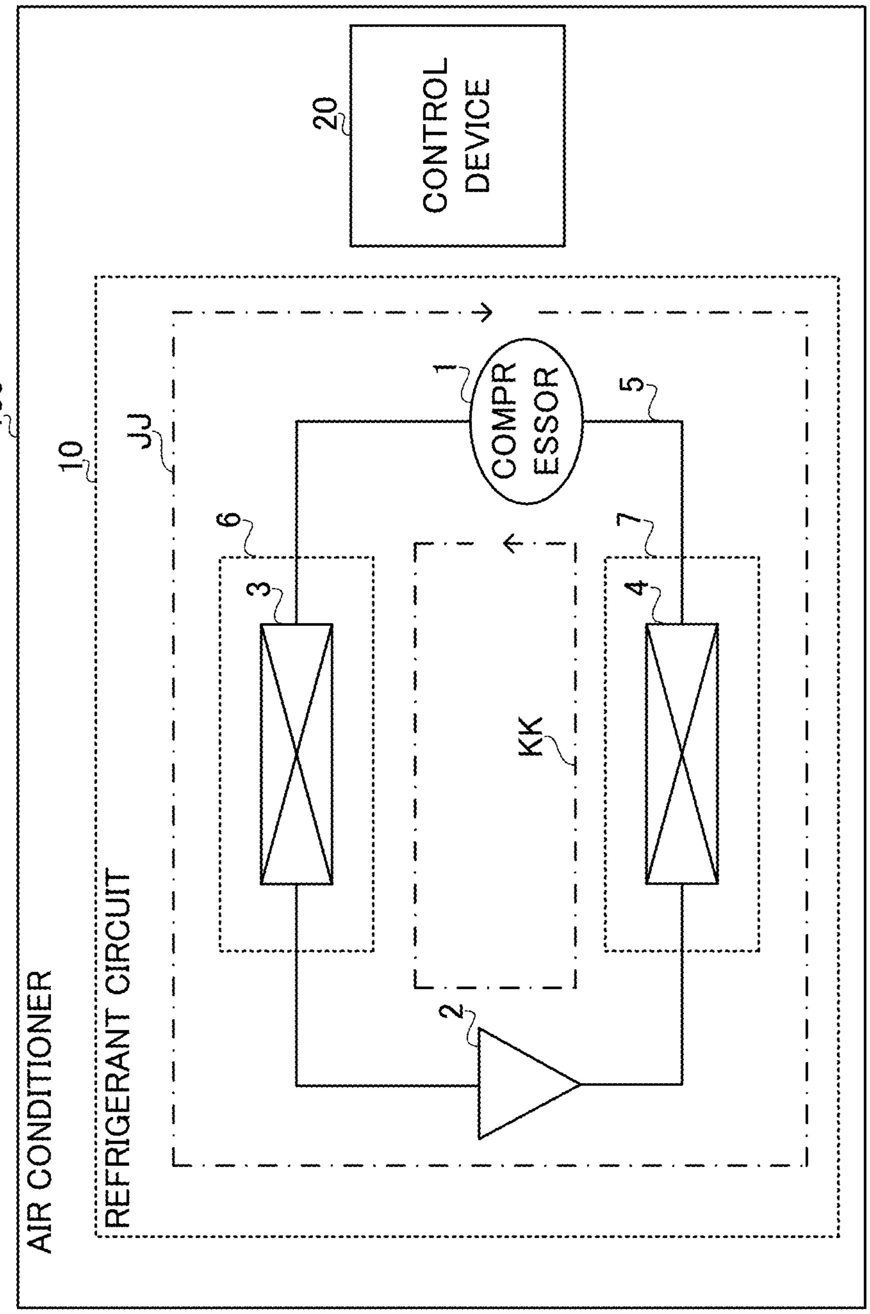

AIR CONDITIONER
100
REFRIGERANT CIRCUIT
10
JJ
KK
CONTROL DEVICE
20
COMPRESSOR
1
2
3
4
5
6
7

FIG.2

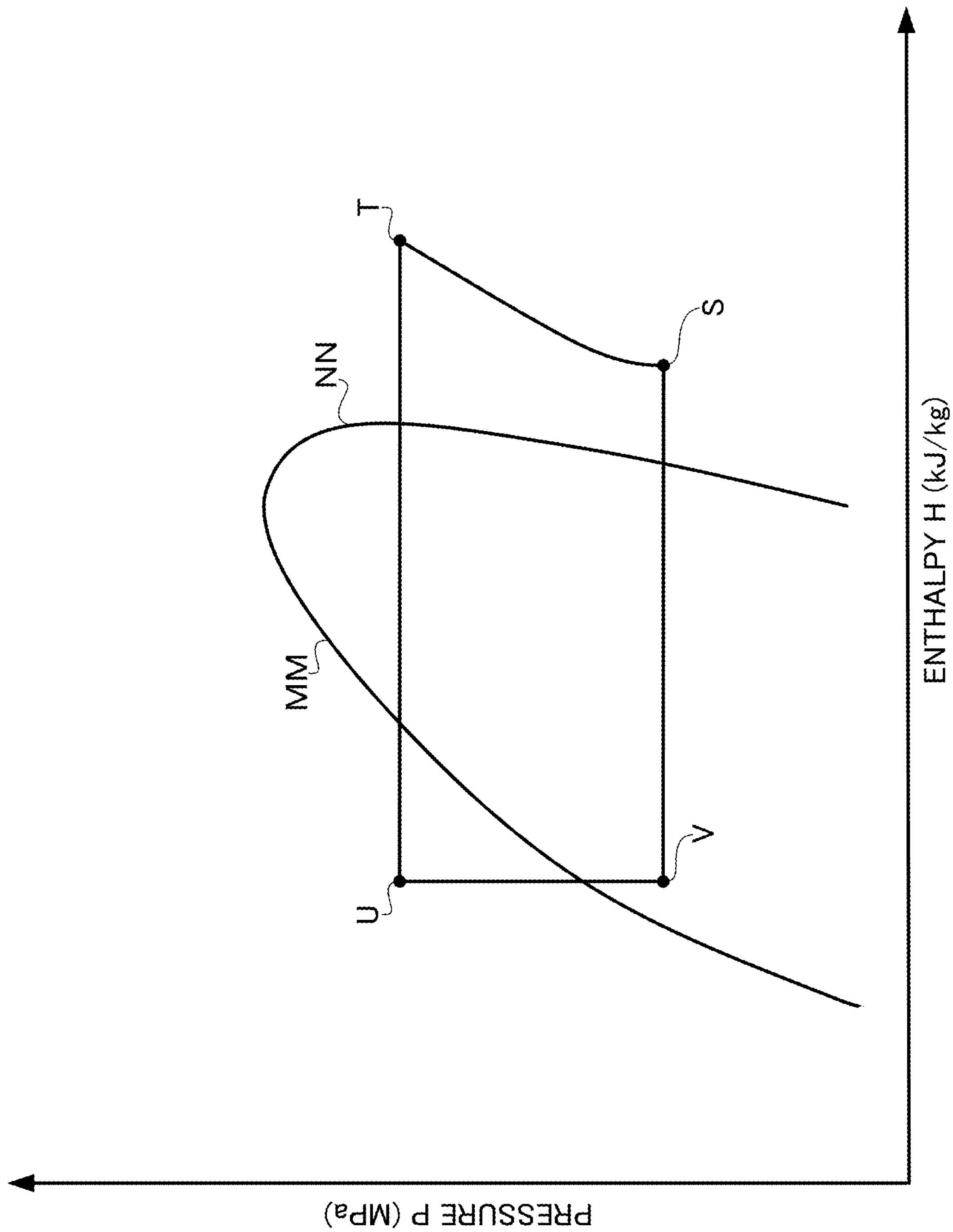


FIG.3

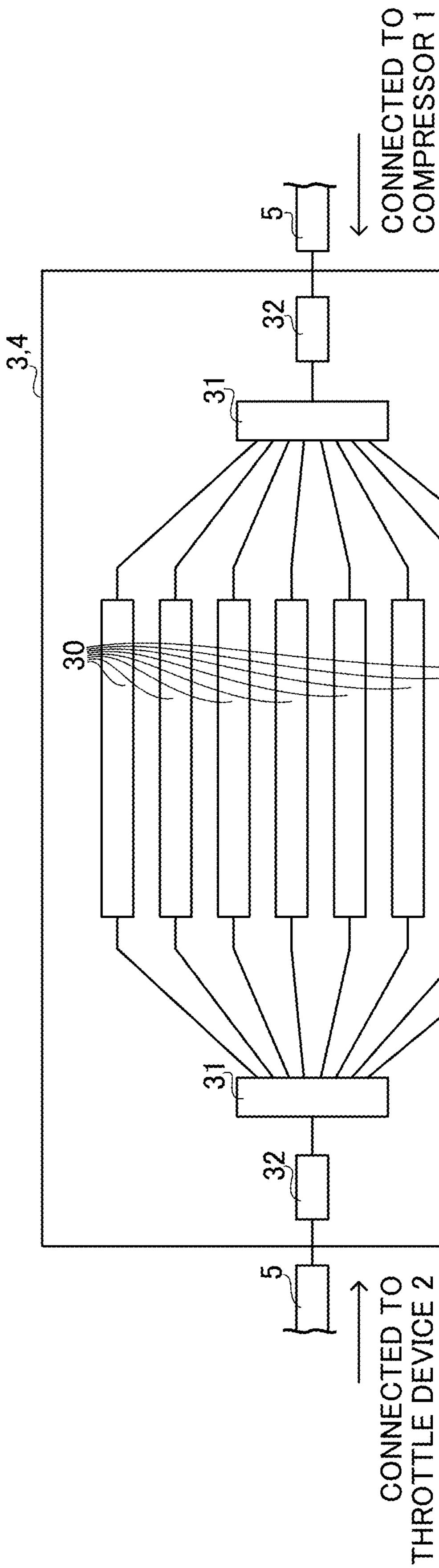

CONNECTED TO
COMPRESSOR 1
5
32
3,4
31
30
31
32
5
CONNECTED TO
THROTTLE DEVICE 2

FIG.4A

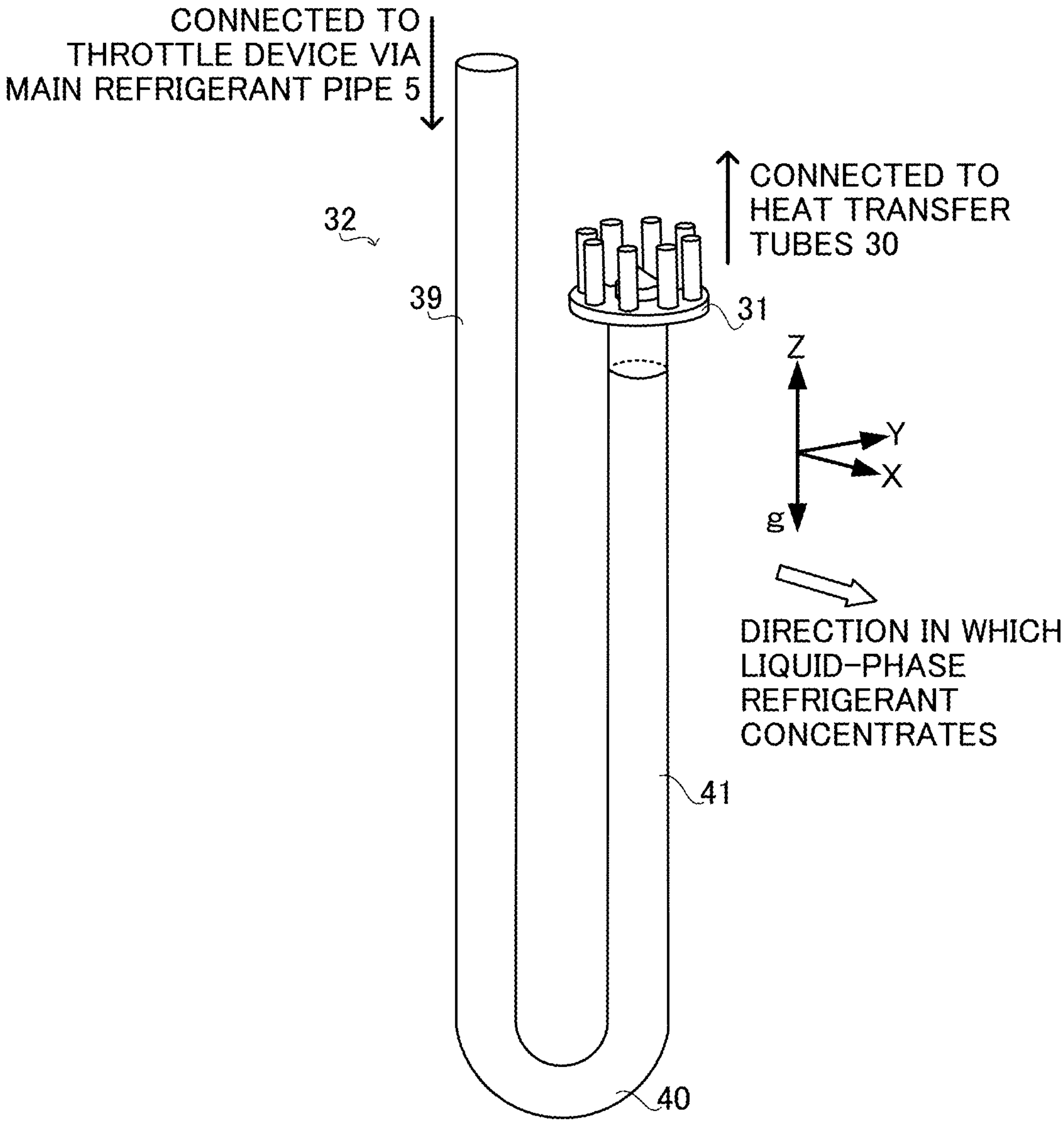

CONNECTED TO
THROTTLE DEVICE VIA
MAIN REFRIGERANT PIPE 5
CONNECTED TO
HEAT TRANSFER
TUBES 30
32
39
31
Z
Y
X
g
DIRECTION IN WHICH
LIQUID-PHASE
REFRIGERANT
CONCENTRATES
41
40

FIG.4B

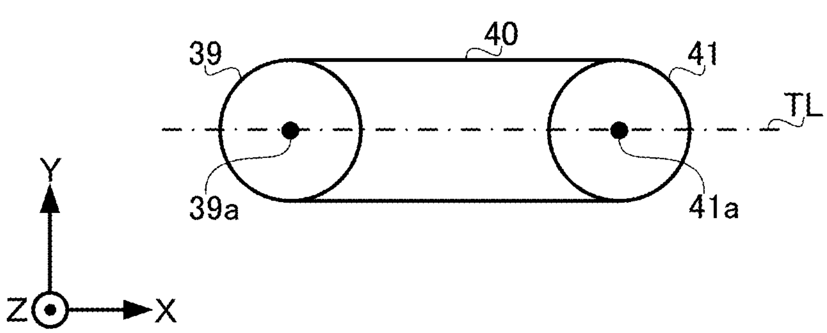
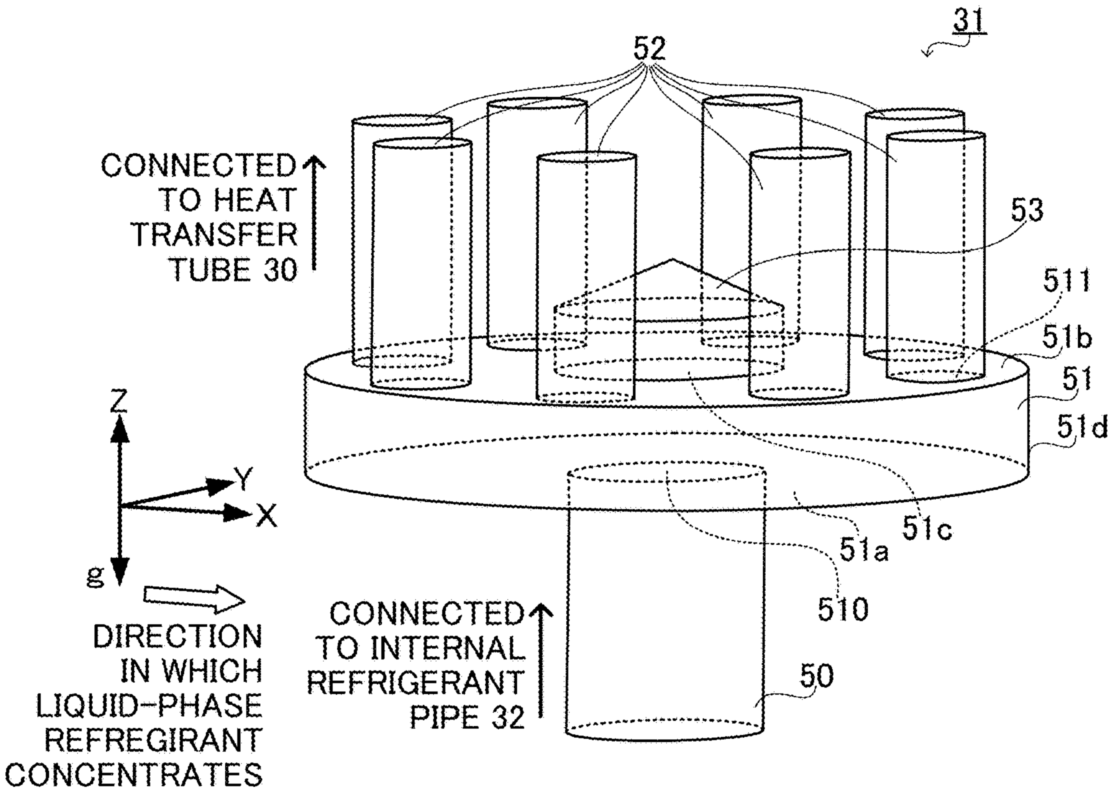
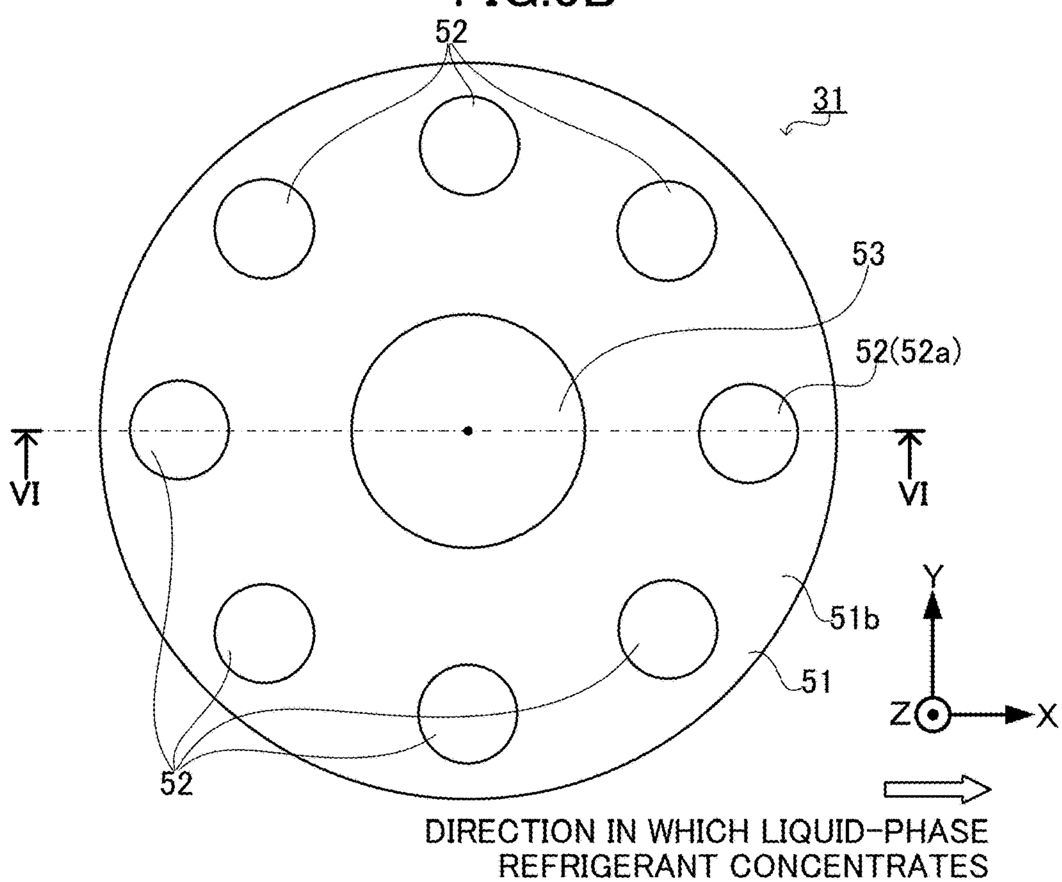

40
39
41
TL
39a
41a
Y
Z
X

FIG.5A



31
52
CONNECTED TO HEAT TRANSFER TUBE 30
53
511
51b
51
51d
Z
Y
X
g
51c
51a
510
DIRECTION IN WHICH LIQUID-PHASE REFREGIRANT CONCENTRATES
CONNECTED TO INTERNAL REFRIGERANT PIPE 32
50

FIG.5B



52
31
53
52(52a)
VI
VI
51b
51
Y
Z
X
52
DIRECTION IN WHICH LIQUID-PHASE REFRIGERANT CONCENTRATES

FIG.6

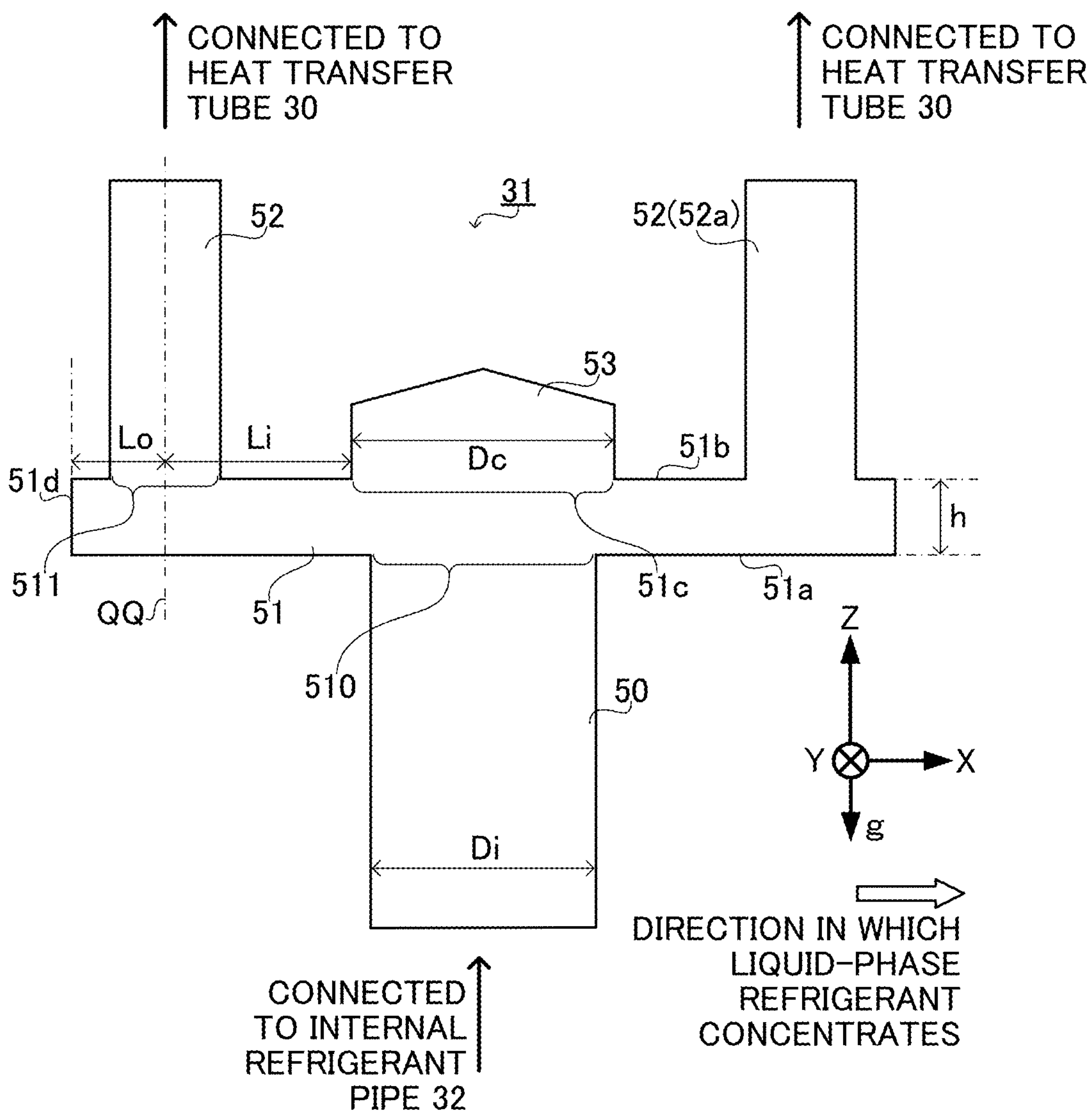

CONNECTED TO
HEAT TRANSFER
TUBE 30
CONNECTED TO
HEAT TRANSFER
TUBE 30
31
52
52(52a)
53
Lo
Li
Dc
51b
51d
h
511
51c
51a
QQ
51
Z
510
50
Y
X
g
Di
DIRECTION IN WHICH
LIQUID-PHASE
REFRIGERANT
CONCENTRATES
CONNECTED
TO INTERNAL
REFRIGERANT
PIPE 32

FIG.7

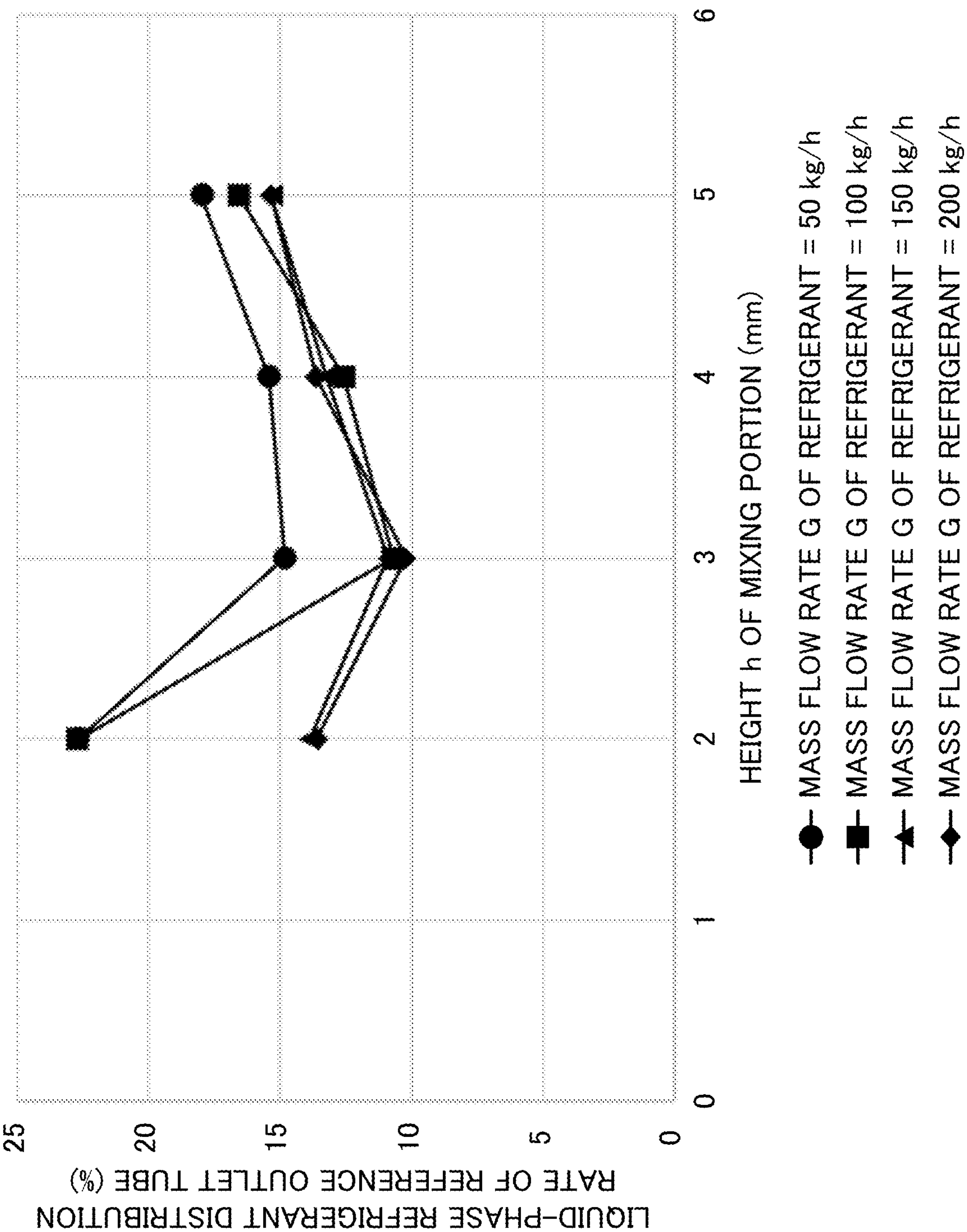

LIQUID-PHASE REFRIGERANT DISTRIBUTION RATE OF REFERENCE OUTLET TUBE (%)
0
5
10
15
20
25
0
1
2
3
4
5
6
HEIGHT h OF MIXING PORTION (mm)
MASS FLOW RATE G OF REFRIGERANT = 50 kg/h
MASS FLOW RATE G OF REFRIGERANT = 100 kg/h
MASS FLOW RATE G OF REFRIGERANT = 150 kg/h
MASS FLOW RATE G OF REFRIGERANT = 200 kg/h

FIG.8

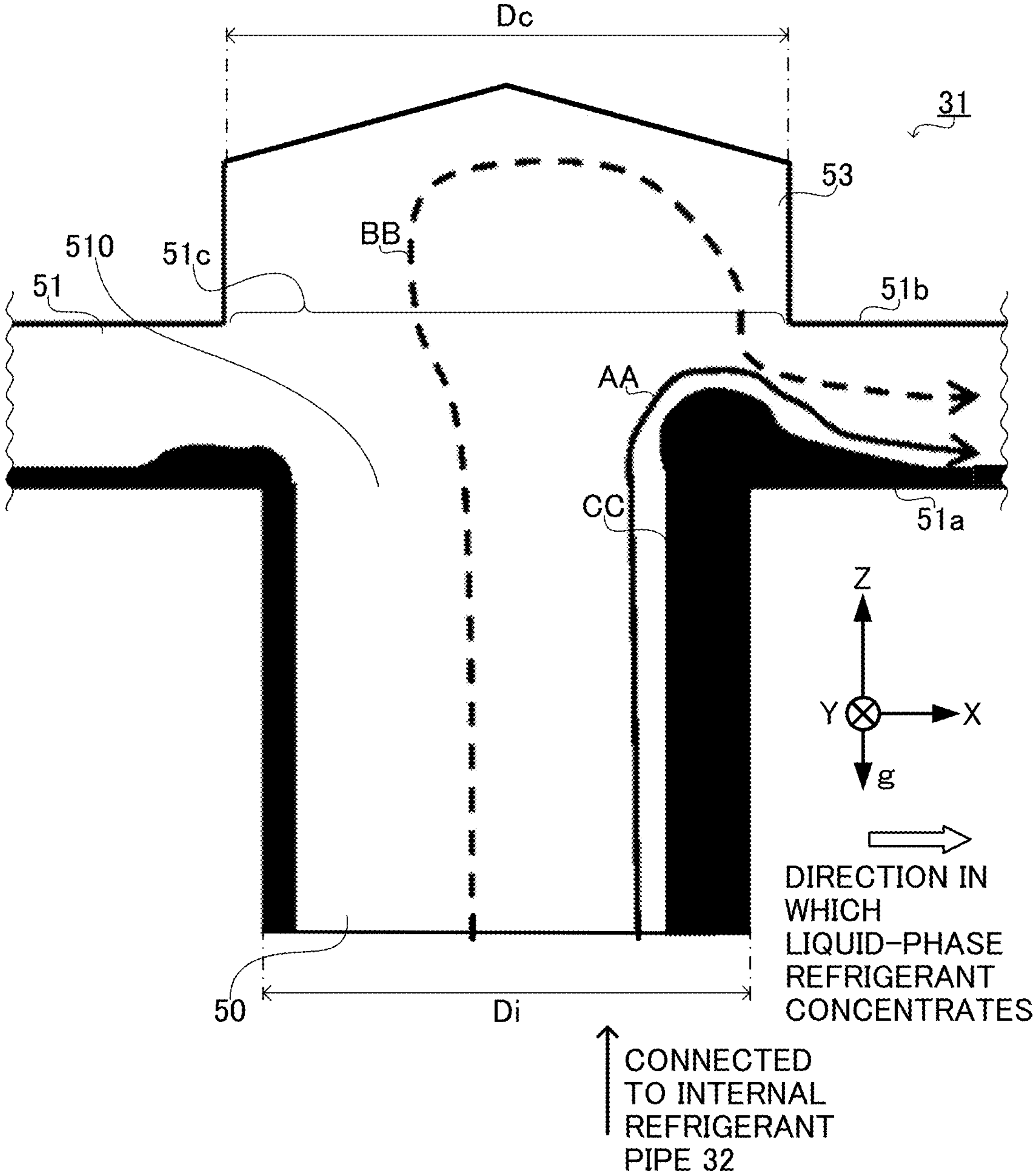

Dc
31
53
BB
510
51c
51
51b
AA
CC
51a
Z
Y
X
g
DIRECTION IN WHICH LIQUID-PHASE REFRIGERANT CONCENTRATES
50
Di
CONNECTED TO INTERNAL REFRIGERANT PIPE 32

FIG.9A

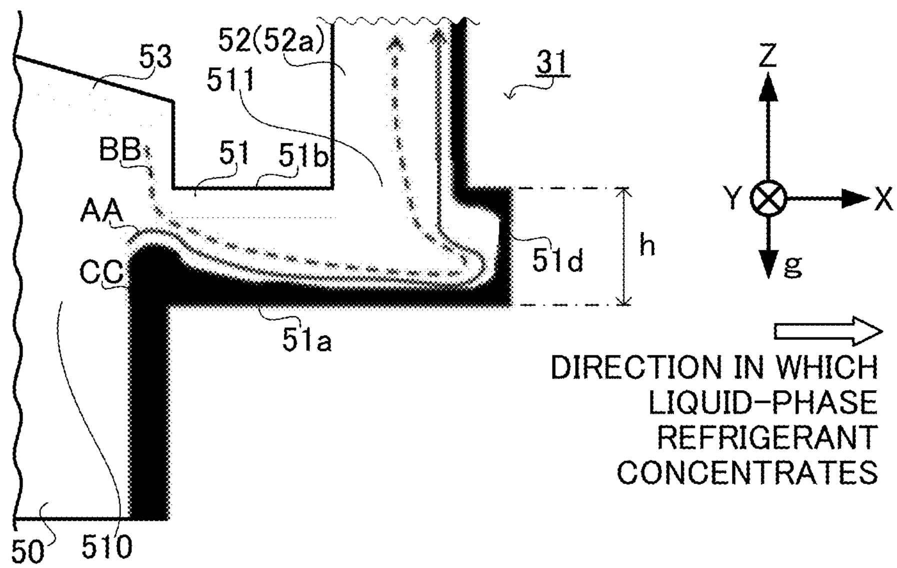

52(52a)
53
511
31
Z
BB
51
51b
AA
Y
X
h
CC
51d
g
51a
DIRECTION IN WHICH
LIQUID-PHASE
REFRIGERANT
CONCENTRATES
50
510

FIG.9B

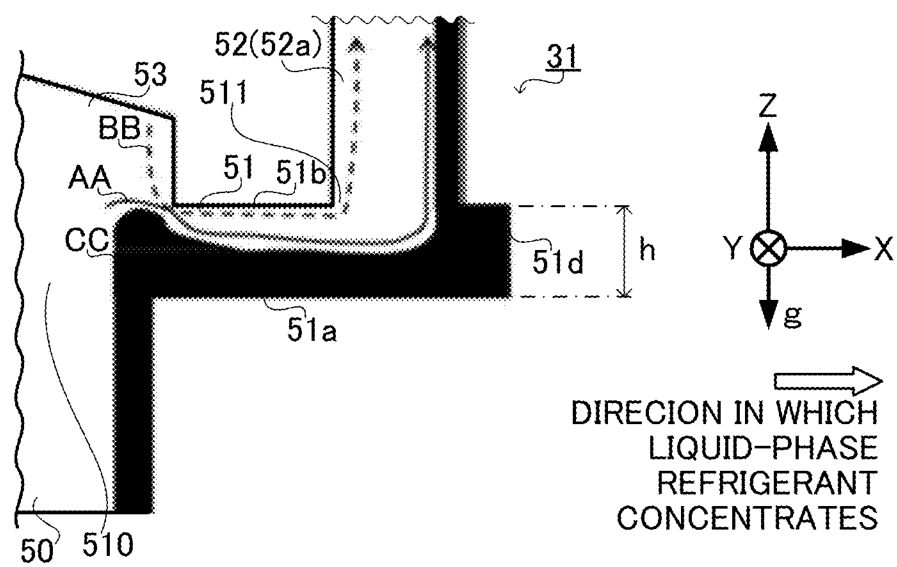

52(52a)
53
31
511
Z
BB
51
51b
AA
CC
Y
X
h
51d
g
51a
DIRECION IN WHICH
LIQUID-PHASE
REFRIGERANT
CONCENTRATES
50
510

FIG.9C

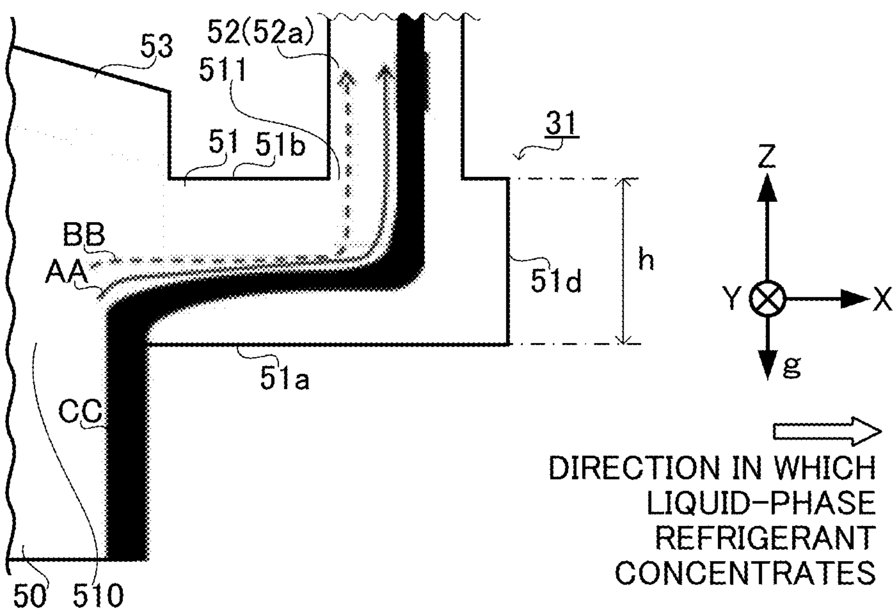

52(52a)
53
511
31
51
51b
Z
BB
AA
51d
h
Y
X
g
51a
CC
DIRECTION IN WHICH
LIQUID-PHASE
REFRIGERANT
CONCENTRATES
50
510

FIG.10

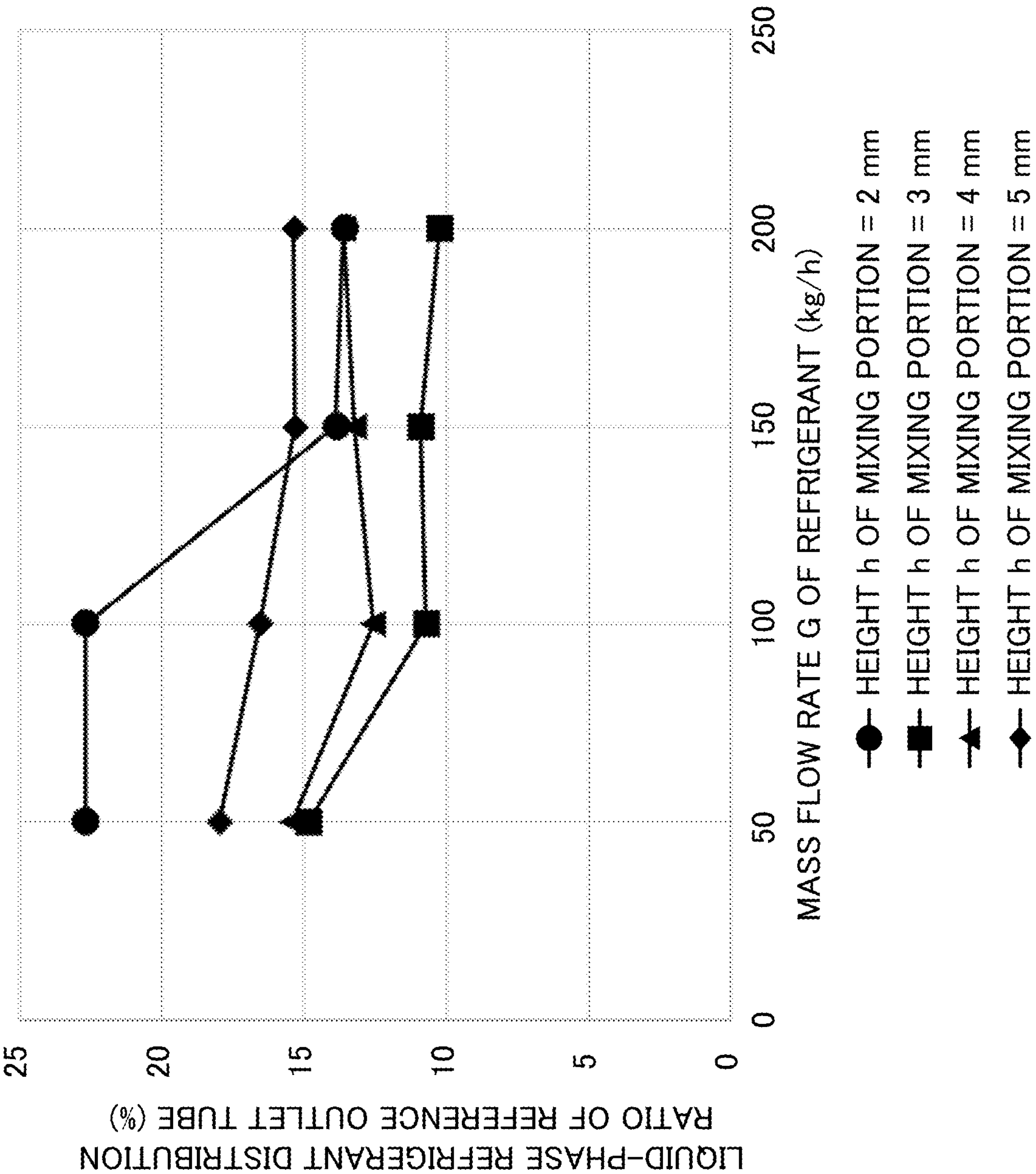

LIQUID-PHASE REFRIGERANT DISTRIBUTION RATIO OF REFERENCE OUTLET TUBE (%)
25
20
15
10
5
0
0
50
100
150
200
250
MASS FLOW RATE G OF REFRIGERANT (kg/h)
HEIGHT h OF MIXING PORTION = 2 mm
HEIGHT h OF MIXING PORTION = 3 mm
HEIGHT h OF MIXING PORTION = 4 mm
HEIGHT h OF MIXING PORTION = 5 mm

FIG.11

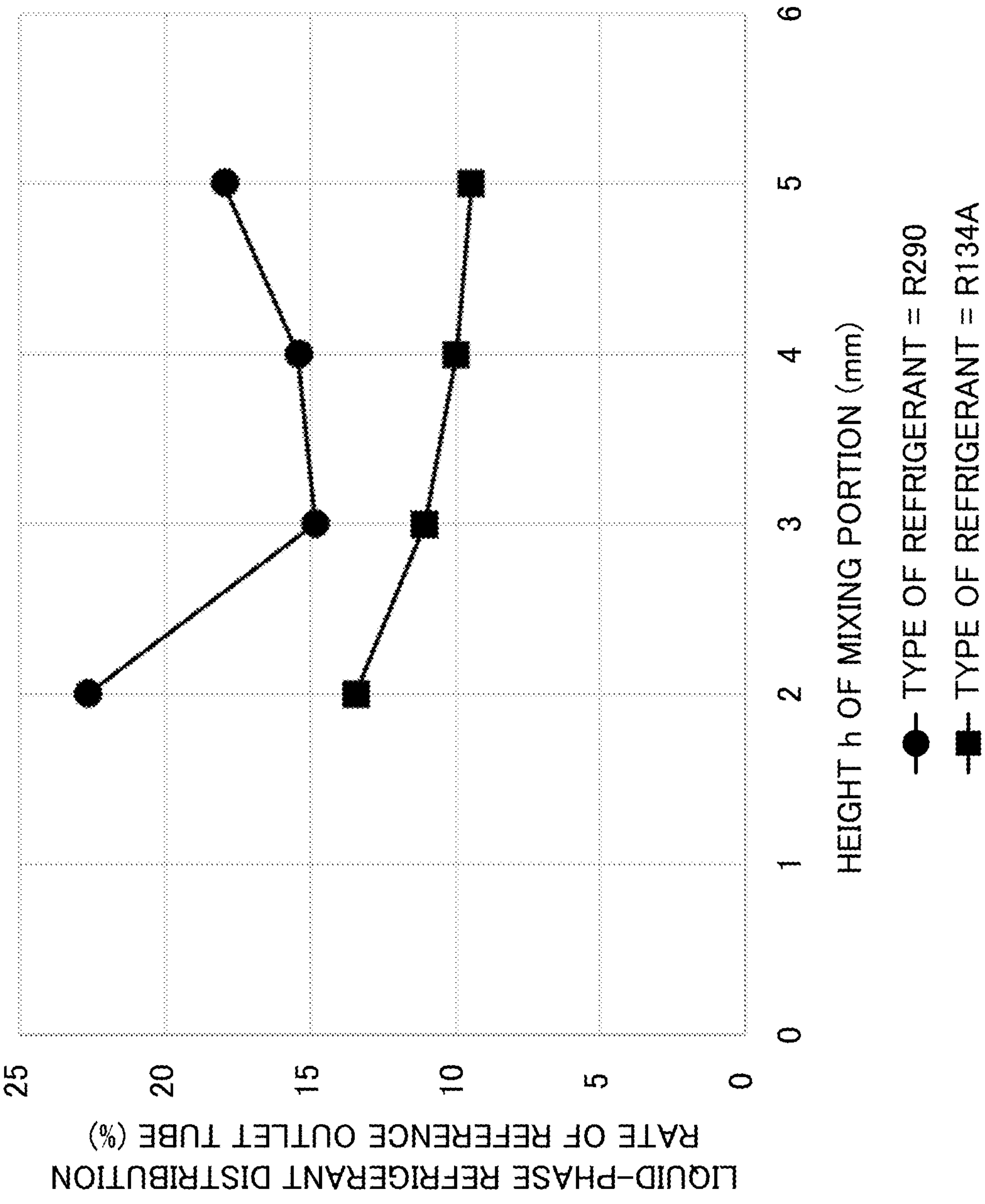

LIQUID-PHASE REFRIGERANT DISTRIBUTION RATE OF REFERENCE OUTLET TUBE (%)
25
20
15
10
5
0
0
1
2
3
4
5
6
HEIGHT h OF MIXING PORTION (mm)
TYPE OF REFRIGERANT = R290
TYPE OF REFRIGERANT = R134A

FIG.12

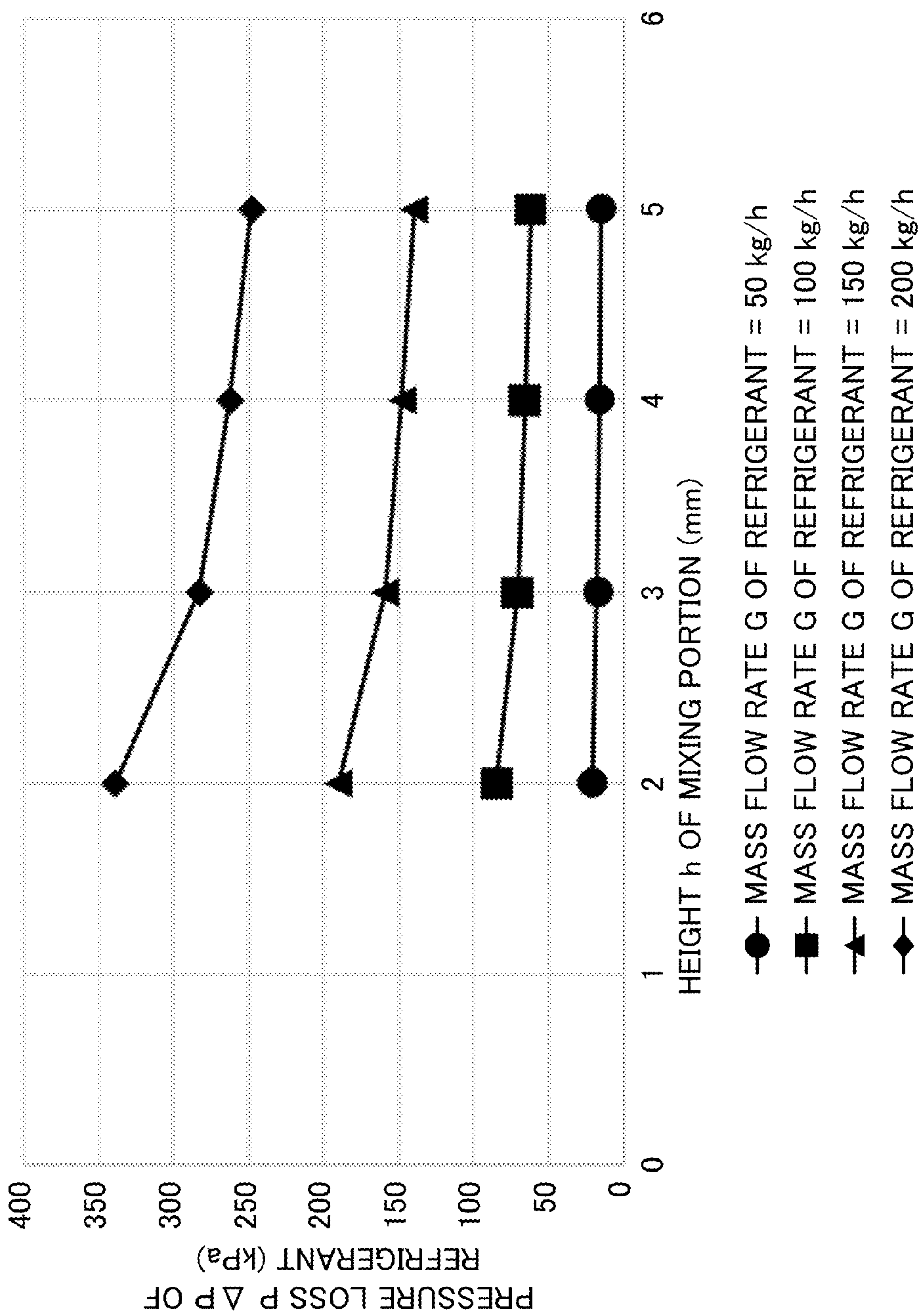

PRESSURE LOSS P ΔP OF REFRIGERANT (kPa)
400
350
300
250
200
150
100
50
0
0
1
2
3
4
5
6
HEIGHT h OF MIXING PORTION (mm)
MASS FLOW RATE G OF REFRIGERANT = 50 kg/h
MASS FLOW RATE G OF REFRIGERANT = 100 kg/h
MASS FLOW RATE G OF REFRIGERANT = 150 kg/h
MASS FLOW RATE G OF REFRIGERANT = 200 kg/h

FIG.13

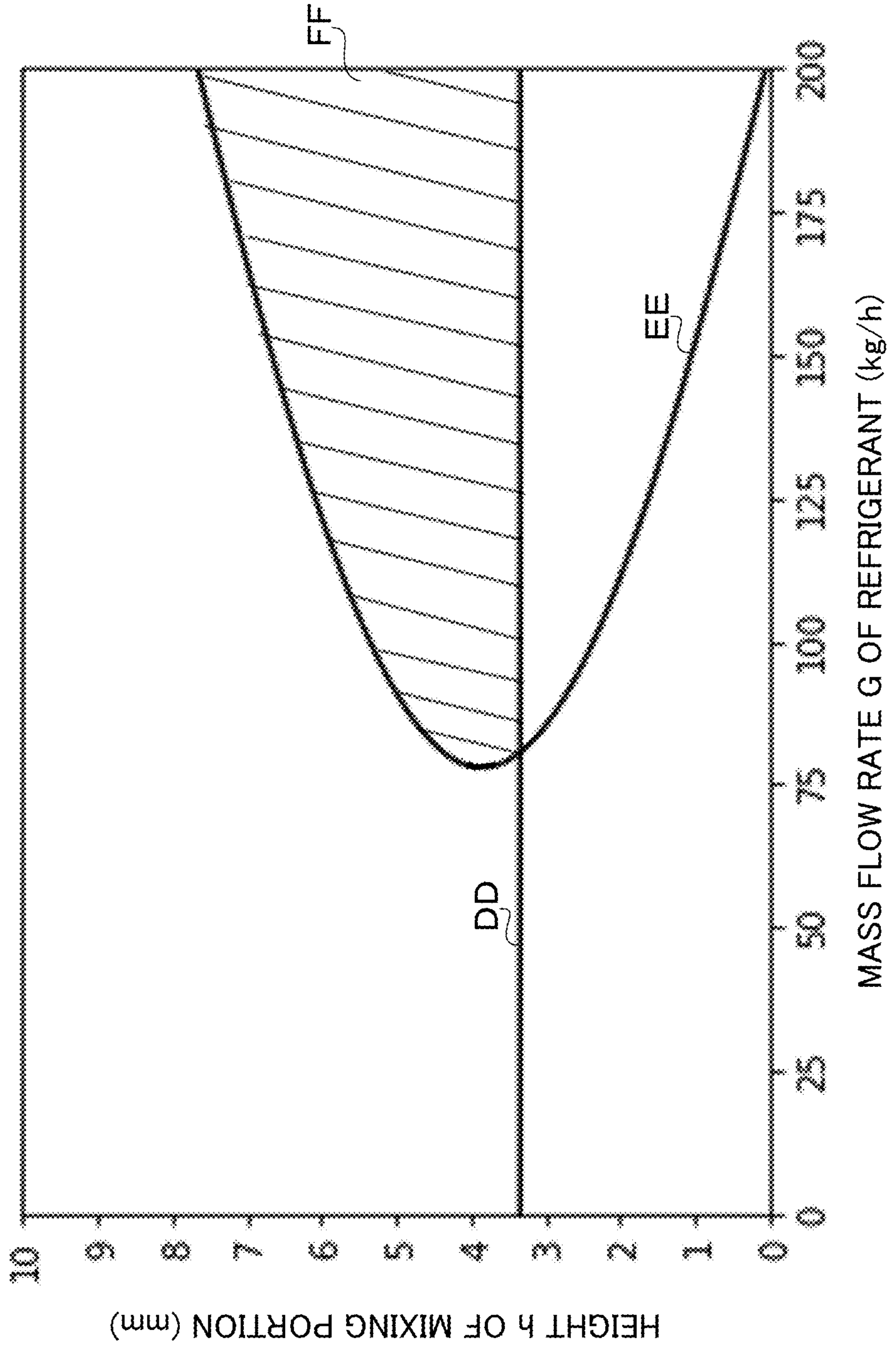

FF
EE
DD
HEIGHT h OF MIXING PORTION (mm)
MASS FLOW RATE G OF REFRIGERANT (kg/h)
0
1
2
3
4
5
6
7
8
9
10
0
25
50
75
100
125
150
175
200

FIG.14A

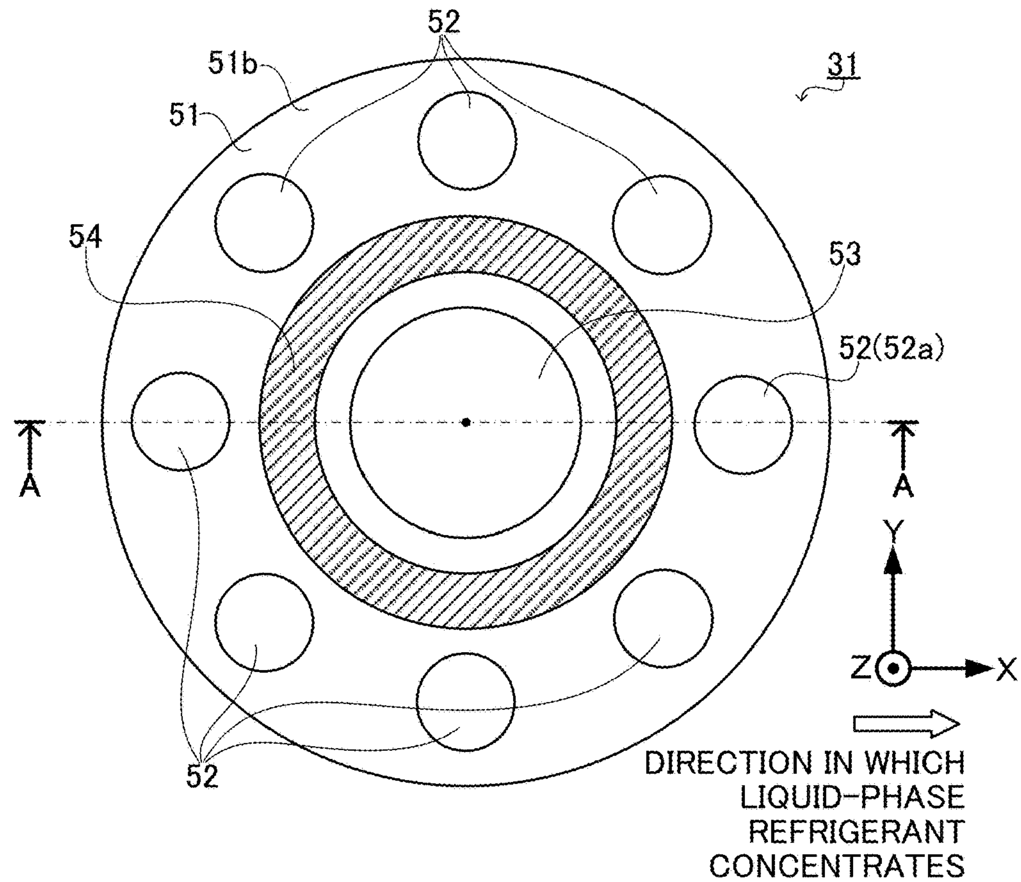


FIG.14B

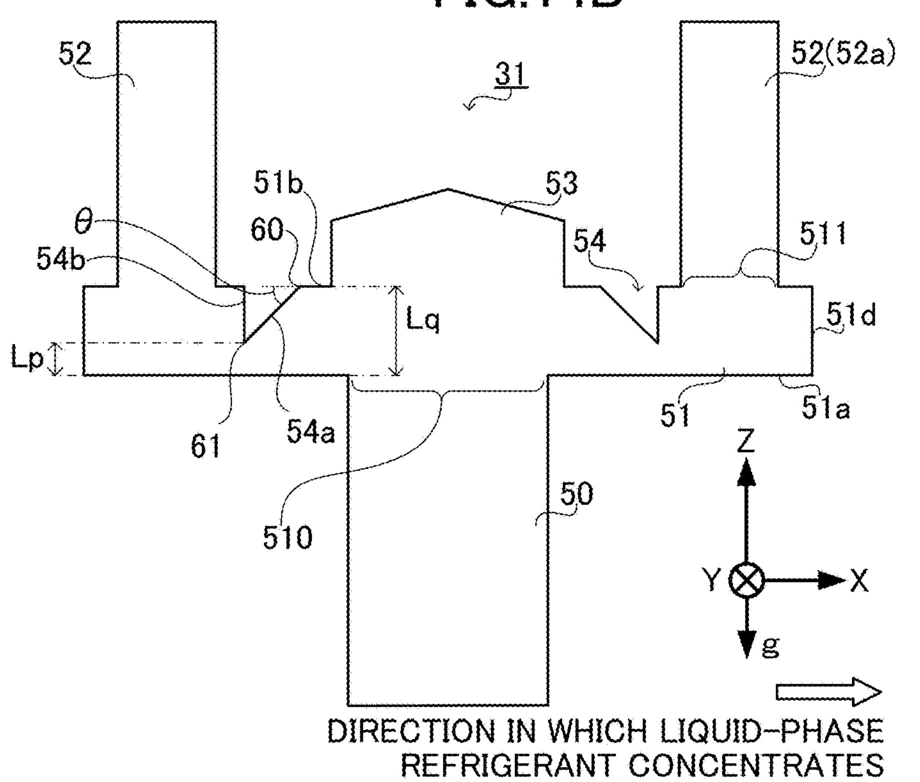


FIG.15

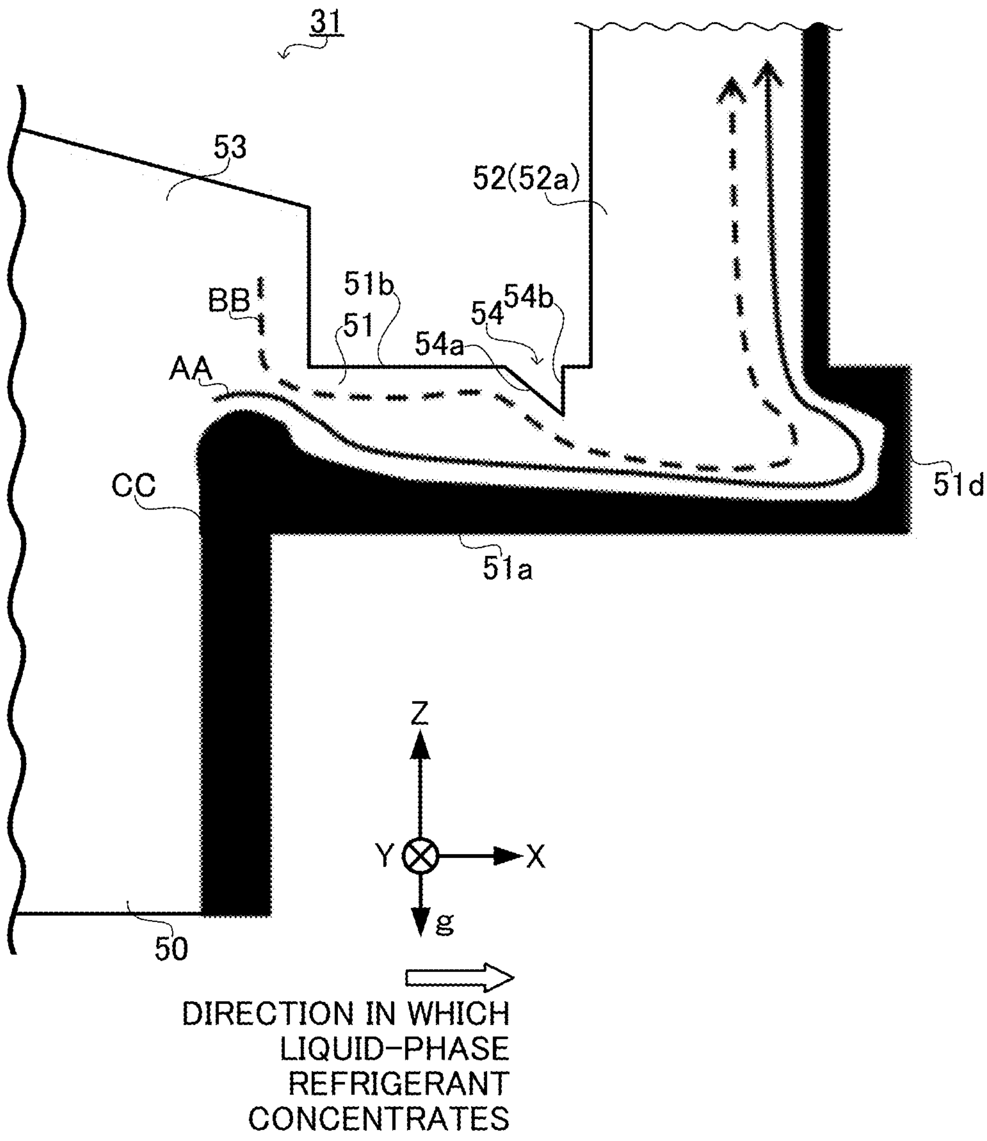
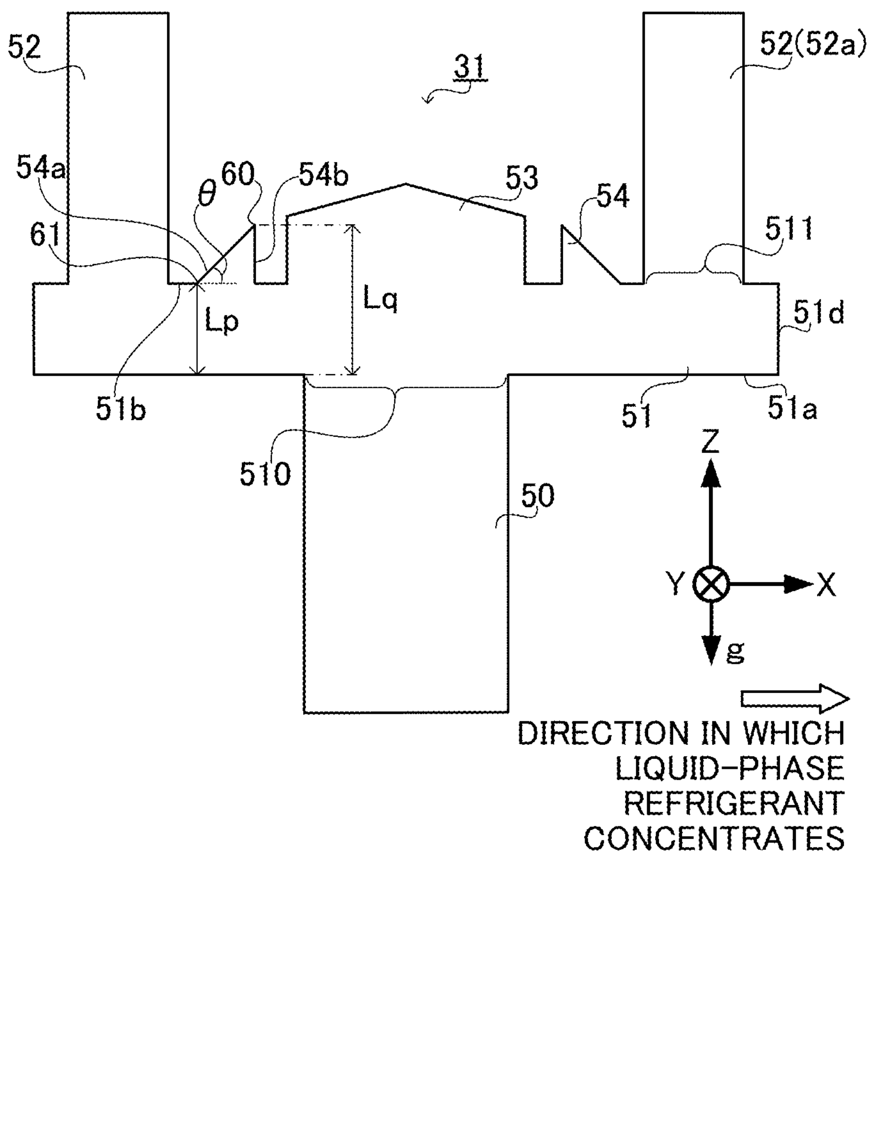

31
53
52(52a)
51b
BB
51
54
54b
54a
AA
CC
51d
51a
Z
Y
X
g
50
DIRECTION IN WHICH
LIQUID-PHASE
REFRIGERANT
CONCENTRATES

FIG.16


52
31
52(52a)
54a
θ
60
54b
53
54
61
511
Lq
Lp
51d
51b
51
51a
Z
510
50
Y
X
g
DIRECTION IN WHICH
LIQUID-PHASE
REFRIGERANT
CONCENTRATES

REFRIGERANT DISTRIBUTOR, HEAT EXCHANGER, AND REFRIGERATION CYCLE DEVICE

TECHNICAL FIELD

[0001] The present disclosure relates to a refrigerant distributor, a heat exchanger, and a refrigeration cycle device.

BACKGROUND ART

[0002] An air conditioner comprises a refrigerant circuit that circulates a refrigerant. The refrigerant circuit comprises a compressor that compresses the refrigerant, a throttle device that expands the refrigerant, an indoor heat exchanger that allows the refrigerant to exchange heat with indoor air, an outdoor heat exchanger that allows the refrigerant to exchange heat with outdoor air, and a refrigerant pipe connecting the compressor, the throttle device, the indoor heat exchanger, and the outdoor heat exchanger in a circular manner. When the air conditioner performs a cooling operation, the indoor heat exchanger functions as an evaporator that absorbs heat from outside and evaporates the refrigerant. When the air conditioner performs a heating operation, the outdoor heat exchanger functions as an evaporator.

[0003] The refrigerant circulating in the refrigerant circuit is sent out from the throttle device in a gas-liquid two-phase state and flows into the evaporator, or more specifically, the indoor heat exchanger or the outdoor heat exchanger, and exchanges heat with air while flowing through heat transfer tubes incorporated in the evaporator. When the refrigerant in the gas-liquid two-phase state exchanges heat with air, a liquid-phase refrigerant contained in the refrigerant evaporates and turns into a gas-phase refrigerant. The refrigerant thus changes from the gas-liquid two-phase state to a gas single-phase state while flowing through the heat transfer tubes. In other words, the refrigerant flows through some sections of the heat transfer tubes in the gas-liquid two-phase state and flows through the remaining sections of the heat transfer tubes in the gas single-phase state.

[0004] When a pressure loss of the refrigerant occurs inside the heat transfer tubes, the heat exchange efficiency of the evaporator decreases. The indoor heat exchanger or the outdoor heat exchanger that functions as an evaporator thus comprises multiple heat transfer tubes and a refrigerant distributor that distributes the refrigerant to the heat transfer tubes, and distributes the refrigerant to each heat transfer tube. This reduces the flow rate of the refrigerant inside each heat transfer tube, thus reducing the pressure loss of the refrigerant.

[0005] When the refrigerant pipe inside the evaporator includes a curved portion located upstream from the refrigerant distributor, a centrifugal force acts on the refrigerant in the gas-liquid two-phase state passing through the curved portion. The refrigerant thus flows into the refrigerant distributor with the liquid-phase refrigerant contained in the refrigerant concentrated. In this case, the amount of the liquid-phase refrigerant distributed to each heat transfer tube by the refrigerant distributor varies. The smaller an amount of the liquid-phase refrigerant distributed to a heat transfer tube is, the longer a section of the heat transfer tube through which the refrigerant flows in the gas single-phase state is. The heat transfer coefficient of the refrigerant in the gas single-phase state is much lower than the heat transfer coefficient of the refrigerant in the gas-liquid two-phase state. Thus, the heat exchange efficiency in the section of the heat transfer tube through which the refrigerant flows in the gas single-phase state is much lower than the heat exchange efficiency in the section through which the refrigerant flows in the gas-liquid two-phase state. Thus, the heat transfer tube with a longer section through which the refrigerant flows in the gas single-phase state has a lower heat exchange efficiency. When the heat transfer tubes have different lengths of sections through which the refrigerant flows in the gas single-phase, the heat exchange efficiency of the evaporator decreases.

[0006] In view of such issues, the refrigerant distributor described in Patent Literature 1 comprises an inlet tube into which a refrigerant in the gas-liquid two-phase state flows, a branch space in which the refrigerant in the gas-liquid two-phase state flowing from the inlet tube is mixed and split, multiple outlet tubes through which the refrigerant in the gas-liquid two-phase state split in the branch space flows out, an inflow path connecting the inlet tube to the branch space, and multiple outflow paths connecting the multiple outlet tubes to the branch space. A mixing portion with a recess at a position facing the inflow path is located in the branch space. $Ai/(\pi \times Da \times Hv) \geq 0.5$, $Av/(Apo-Api) \leq 2.0$, and $Db/Da \leq 1.0$, where Da (mm) is the inner diameter of the inflow path, Ai (mm^2) is the cross-sectional area of the inflow path, Db (mm) is the inner diameter of the mixing portion, Av (mm^2) is the area of the branch space, Hv (mm) is the height of the branch space, Apo (mm^2) is the area of a circumscribed circle of the outflow paths, and Api (mm^2) is the area of an inscribed circle of the outflow paths. With this configuration, the refrigerant distributor described in Patent Literature 1 reduces variations in the amount of the refrigerant sent out from each outlet tube to the corresponding heat transfer tube.

CITATION LIST

Patent Literature

[0007] Patent Literature 1: Unexamined Japanese Patent Application Publication No. 2014-81149

SUMMARY OF INVENTION

Technical Problem

[0008] However, the refrigerant distributor described in Patent Literature 1 may fail to reduce variations in the amount of the refrigerant distributed to each heat transfer tube depending on the flow rate, the type, and the temperature of the refrigerant flowing into the refrigerant distributor.

[0009] In view of such issues, an objective of the present disclosure is to reduce variations in the amount of the refrigerant distributed to each heat transfer tube, regardless of the flow rate, the type, and the temperature of the refrigerant when the refrigerant is distributed to multiple heat transfer tubes.

Solution to Problem

[0010] To achieve the above objective, a refrigerant distributor according to the present disclosure comprises a mixing portion having a cylindrical shape. The mixing portion has an inlet for inflow therethrough of a refrigerant at a first end of the mixing portion. The mixing portion has a plurality of outlets for outflow therethrough of the refrigerant at a second end of the mixing portion opposite to the first end. The mixing portion has a recess facing the inlet at the second end of the mixing portion. The mixing portion guides the refrigerant flowing in from the inlet to flow along the first end of the mixing portion and a side wall of the mixing portion to diffuse the refrigerant in a circumferential direction of the second end, and then sends out the refrigerant from the plurality of outlets.

Advantageous Effects of Invention

[0011] The above configuration reduces variations in the amount of the refrigerant sent out from each outlet, regardless of the flow rate, the type, and the temperature of the refrigerant in the gas-liquid two-phase state flowing into the mixing portion. This reduces variations in the amount of the refrigerant flowing into any heat transfer tube connected to each outlet, regardless of the flow rate, the type, and the temperature of the refrigerant. In other words, the above configuration can reduce variations in the amount of the refrigerant distributed to each heat transfer tube, regardless of the flow rate, the type, and the temperature of the refrigerant when the refrigerant is distributed to the multiple heat transfer tubes.

BRIEF DESCRIPTION OF DRAWINGS

[0012] FIG. **1** is a diagram of a refrigerant circuit in an air conditioner according to Embodiment 1 of the present disclosure;

[0013] FIG. **2** is a Mollier diagram illustrating the state of a refrigerant circulating in the refrigerant circuit according to Embodiment 1 of the present disclosure;

[0014] FIG. **3** is a schematic diagram of an indoor heat exchanger and an outdoor heat exchanger according to Embodiment 1 of the present disclosure;

[0015] FIG. **4**A is a perspective view of a refrigerant distributor and an internal refrigerant pipe according to Embodiment 1 of the present disclosure, and FIG. **4**B is a cross-sectional view of the internal refrigerant pipe according to Embodiment 1 of the present disclosure;

[0016] FIG. **5**A is a perspective view of the refrigerant distributor according to Embodiment 1 of the present disclosure, and FIG. **5**B is a plan view of the refrigerant distributor according to Embodiment 1 of the present disclosure;

[0017] FIG. **6** is a cross-sectional view of the refrigerant distributor according to Embodiment 1 of the present disclosure taken along line VI-VI in FIG. **5**B;

[0018] FIG. **7** is a graph illustrating the relationship between the height of a mixing portion and the liquid-phase refrigerant distribution ratio of a reference outlet tube according to Embodiment 1 of the present disclosure;

[0019] FIG. **8** is a schematic diagram illustrating an example refrigerant flow inside the refrigerant distributor according to Embodiment 1 of the present disclosure;

[0020] FIG. **9**A is a schematic diagram illustrating an example refrigerant flow inside the refrigerant distributor according to Embodiment 1 of the present disclosure when the liquid-phase refrigerant distribution ratio of the reference outlet tube is suppressed, FIG. **9**B is a schematic diagram illustrating an example refrigerant flow inside the refrigerant distributor according to Embodiment 1 of the present disclosure when the mixing portion has a height too small to suppress the liquid-phase refrigerant distribution ratio of the reference outlet tube, and FIG. **9**C is a schematic diagram illustrating an example refrigerant flow inside the refrigerant distributor according to Embodiment 1 of the present disclosure when the mixing portion has a height too large to suppress the liquid-phase refrigerant distribution ratio of the reference outlet tube;

[0021] FIG. **10** is a graph illustrating the relationship between the mass flow rate of the refrigerant and the liquid-phase refrigerant distribution ratio of the reference outlet tube according to Embodiment 1 of the present disclosure;

[0022] FIG. **11** is a graph illustrating the relationship between the height of the mixing portion and the liquid-phase refrigerant distribution ratio of the reference outlet tube according to Embodiment 1 of the present disclosure;

[0023] FIG. **12** is a graph illustrating the relationship between the height of a mixing portion and a refrigerant pressure loss according to Embodiment 2 of the present disclosure;

[0024] FIG. **13** is a graph illustrating a range of the height of a mixing portion according to a modification of Embodiment 2 of the present disclosure;

[0025] FIG. **14**A is a plan view of a refrigerant distributor according to Embodiment 3 of the present disclosure, and FIG. **14**B is a cross-sectional view of the refrigerant distributor according to Embodiment 3 of the present disclosure taken along line A-A in FIG. **14**A;

[0026] FIG. **15** is a schematic diagram illustrating an example refrigerant flow inside the refrigerant distributor according to Embodiment of the present disclosure; and

[0027] FIG. **16** is a longitudinal sectional view of a refrigerant distributor according to a modification of Embodiment 3 of the present disclosure.

DESCRIPTION OF EMBODIMENTS

[0028] A refrigerant distributor, a heat exchanger, and a refrigeration cycle device according to one or more embodiments of the present disclosure are described below with reference to the drawings. In the figures, the same reference signs denote the same components.

Embodiment 1

[0029] An air conditioner **100** illustrated in FIG. **1** conditions the air inside an air-conditioning target space such as an indoor space or an interior space of an automobile. The air conditioner **100** is an example of a refrigeration cycle device. The air conditioner **100** comprises a refrigerant circuit **10** that circulates a refrigerant and a control device **20** that controls the operation of the refrigerant circuit **10**.

[0030] The refrigerant circuit **10** comprises a compressor **1** that compresses the refrigerant, a throttle device **2** that expands the refrigerant and also switches a circulation direction of the refrigerant inside the refrigerant circuit **10**, an indoor heat exchanger **3** that is incorporated in an indoor unit **6** installed inside the air-conditioning target space and causes the refrigerant to exchange heat with the air inside the air-conditioning target space, an outdoor heat exchanger **4** that is incorporated in an outdoor unit **7** installed outside the air-conditioning target space and causes the refrigerant to exchange heat with the air outside the air-conditioning target space, and a main refrigerant pipe **5** that connects the compressor **1**, the throttle device **2**, the indoor heat exchanger **3**, and the outdoor heat exchanger **4** in a circular manner, wherein the refrigerant flows through the main refrigerant pipe **5**. The indoor heat exchanger **3** and the outdoor heat exchanger **4** are examples of a heat exchanger.

[0031] The control device **20** comprises a processor that performs various processes and a memory that stores data and programs. The processor in the control device **20** executes the programs stored in the memory and thereby functions as an operation controller that controls the operation of the refrigerant circuit **10**, controlling the operations of the compressor **1**, the throttle device **2**, the indoor unit **6**, and the outdoor unit **7** by transmitting control signals.

[0032] The throttle device **2** comprises an expansion valve for expanding the refrigerant and a four-way valve for switching the circulation direction of the refrigerant. The control device **20** controls the four-way valve in the throttle device **2** to switch the circulation direction of the refrigerant, thus switching the operating state of the air conditioner **100** between a cooling operation state and a heating operation state. When the air conditioner **100** is in the cooling operation state, the four-way valve causes the refrigerant to circulate in the direction indicated by arrow JJ in FIG. **1**. This causes the indoor heat exchanger **3** to function as an evaporator that absorbs heat from outside and evaporates the refrigerant, and the outdoor heat exchanger **4** to function as a condenser that dissipates heat outside and condenses the refrigerant, thus cooling the air inside the air-conditioning target space. When the air conditioner **100** is in the heating operation state, the four-way valve causes the refrigerant to circulate in the direction indicated by arrow KK in FIG. **1**. This causes the indoor heat exchanger **3** to function as a condenser and the outdoor heat exchanger **4** to function as an evaporator, thus heating the air inside the air-conditioning target space.

[0033] FIG. **2** is a Mollier diagram illustrating the state of a refrigerant circulating in the refrigerant circuit **10**. In FIG. **2**, the horizontal axis indicates the refrigerant enthalpy, and the vertical axis indicates the refrigerant pressure. FIG. **2** illustrates a saturated liquid line MM and a saturated vapor line NN. In an area in which an enthalpy H of the refrigerant is below the saturated liquid line MM, the refrigerant is in a liquid single-phase state. In an area in which the enthalpy H of the refrigerant is above the saturated vapor line NN, the refrigerant is in a gas single-phase state, whereas in an area in which the enthalpy H of the refrigerant is above the saturated liquid line MM and below the saturated vapor line NN, the refrigerant is in a gas-liquid two-phase state.

[0034] In the cooling operation state, the refrigerant, as a low-pressure gas-phase refrigerant, is first compressed by the compressor **1** to turn into a high-pressure gas-phase refrigerant as illustrated with the path from point S to point T in FIG. **2**, and then flows into the outdoor heat exchanger **4**. The refrigerant flowing into the outdoor heat exchanger **4** exchanges heat with the air outside the air-conditioning target space to dissipate heat, is condensed to turn into a high-pressure liquid-phase refrigerant as illustrated with the path from point T to point U in FIG. **2**, and then flows into the throttle device **2**. The refrigerant is then expanded by the expansion valve in the throttle device **2** to be depressurized, turns into a low-pressure refrigerant in the gas-liquid two-phase state as illustrated with the path from point U to point V in FIG. **2**, and flows into the indoor heat exchanger **3**. The refrigerant flowing into the indoor heat exchanger **3** exchanges heat with the air inside the air-conditioning target space to absorb heat, evaporates to turn into a low-pressure gas-phase refrigerant as illustrated with the path from point V to point S in FIG. **2**, and then flows into the compressor **1**.

[0035] The indoor unit **6** comprises a housing incorporating the indoor heat exchanger **3**, a fan that feeds air to the indoor heat exchanger **3**, and a motor that drives the fan according to a control by the control device **20**. The fan is driven to rotate by the motor to cause air inside the air-conditioning target space to flow into the housing. The indoor heat exchanger **3** causes the refrigerant to exchange heat with the air flowing into the housing. The fan is driven to rotate by the motor to feed the air after heat exchange with the refrigerant into the air-conditioning target space. The air-conditioning target space is thus air-conditioned.

[0036] The outdoor unit **7** comprises a housing incorporating the outdoor heat exchanger **4**, a fan that feeds air to the outdoor heat exchanger **4**, and a motor that drives the fan according to a control by the control device **20**. The fan is driven to rotate by the motor to cause air outside the air-conditioning target space to flow into the housing. The outdoor heat exchanger **4** causes the refrigerant to exchange heat with the air flowing into the housing. The fan is driven to rotate by the motor to feed the air after heat exchange with the refrigerant out of the air-conditioning target space.

[0037] The indoor heat exchanger **3** has the same configuration as the outdoor heat exchanger **4**. As illustrated in FIG. **3**, the indoor heat exchanger **3** and the outdoor heat exchanger **4** comprise eight heat transfer tubes **30**, a pair of refrigerant distributors **31** that are connected to both ends of each heat transfer tube **30** and distribute the refrigerant to each heat transfer tube **30**, and a pair of internal refrigerant pipes **32** connecting the refrigerant distributors **31** and the main refrigerant pipe **5**. The refrigerant distributors **31** are located at the inlets of the indoor heat exchanger **3** and the outdoor heat exchanger **4**. Specifically, one of the pair of refrigerant distributors **31** is connected to the compressor **1** through the main refrigerant pipe **5** and one of the internal refrigerant pipes **32**, and the other is connected to the throttle device **2** through the main refrigerant pipe **5** and the other of the internal refrigerant pipes **32**. The refrigerant sent out from the compressor **1** or the throttle device **2** flows into a refrigerant distributor **31** through the main refrigerant pipe **5** and the internal refrigerant pipe **32**, and is distributed to each heat transfer tube **30** by the refrigerant distributor **31**. Each heat transfer tube **30** is connected to multiple radiator fins. The refrigerant distributed to each heat transfer tube **30** by the refrigerant distributor **31** exchanges heat with air through the radiator fins while flowing through each heat transfer tube **30**.

[0038] When the indoor heat exchanger **3** or the outdoor heat exchanger **4** functions as an evaporator, the refrigerant in the gas-liquid two-phase state sent out from the throttle device **2** flows into the refrigerant distributor **31** connected to the throttle device **2**, is distributed to each heat transfer tube **30** by the refrigerant distributor **31**, and exchanges heat with air while flowing through each heat transfer tube **30**. The distribution of the refrigerant to the multiple heat transfer tubes **30** reduces the flow rate of the refrigerant flowing through each heat transfer tube **30**, thus reducing the refrigerant pressure loss and improving the heat exchange efficiency of the indoor heat exchanger **3** or the outdoor heat exchanger **4** as an evaporator. When the refrigerant in the gas-liquid two-phase state exchanges heat with air, a liquid-phase refrigerant contained in the refrigerant evaporates and turns into a gas-phase refrigerant. The refrigerant thus changes from the gas-liquid two-phase state to the gas single-phase state with a much lower heat transfer coefficient while flowing through the heat transfer tubes. After changing to the gas single-phase state, the refrigerant flowing through each heat transfer tube **30** merges in the refrigerant distributor **31** connected to the compressor **1** and is sent out from the refrigerant distributor **31** to the compressor **1**.

[0039] The heat exchange efficiency of the evaporator can be improved by reducing variations in the length of the section in each heat transfer tube **30** in which the refrigerant flows in a gas single-phase state, or in other words, the section with low heat exchange efficiency. The variations in the length of the section in which the refrigerant flows in the gas single-phase state can be reduced by reducing variations in a heat load of the refrigerant in each heat transfer tube **30**. The heat load of the refrigerant in the heat transfer tube **30** is equal to the product of a mass flow rate of the refrigerant flowing through the heat transfer tube **30** and the difference in enthalpy between the refrigerant at the inlet and the refrigerant at the outlet of the heat transfer tube **30**. Variations in the heat load of the refrigerant in each heat transfer tube **30** can thus be reduced by reducing variations in the mass flow rate of the refrigerant flowing through each heat transfer tube **30**. The refrigerant distributor **31** according to the present embodiment reduces variations in the mass flow rate of the refrigerant flowing through each heat transfer tube **30**, thus reducing variations in the heat load of the refrigerant in each heat transfer tube **30**, reducing variations in the length of the section in each heat transfer tube **30** through which the refrigerant flows in the gas single-phase state, and improving the heat exchange efficiency of the evaporator.

[0040] FIG. **4**A is a perspective view of the internal refrigerant pipe **32** connected to the throttle device **2** through the main refrigerant pipe **5** and the refrigerant distributor **31** connected to the internal refrigerant pipe **32**. The internal refrigerant pipe **32** comprises a joint portion **39** that has a linear shape and is connected to the main refrigerant pipe **5**, a curved portion **40** that is U-shaped and is located downstream from the joint portion **39**, and a straight tube portion **41** that has a linear shape and is located downstream from the curved portion **40**. The straight tube portion **41** has a downstream end connected to the refrigerant distributor **31**. The refrigerant in the gas-liquid two-phase state that flows into the internal refrigerant pipe **32** from the throttle device **2** through the main refrigerant pipe **5** passes through the joint portion **39**, the curved portion **40**, and the straight tube portion **41** in this order, and then flows into the refrigerant distributor **31**. FIG. **4**B is a cross-sectional view of the internal refrigerant pipe **32** cut along a cut surface perpendicular to a direction in which the joint portion **39** and the straight tube portion **41** extend. For ease of understanding, an XYZ orthogonal coordinate system is defined as illustrated in FIGS. **4**A and **4**B. Z-axis is parallel to a gravity direction g. X-axis is perpendicular to Z-axis and parallel to straight line TL passing through an axial center **39***a* of the joint portion **39** and an axial center **41***a* of the straight tube portion **41**. Y-axis is perpendicular to X-axis and Z-axis.

[0041] When the refrigerant in the gas-liquid two-phase state flows through the internal refrigerant pipe **32**, the gas-phase refrigerant contained in the refrigerant flows through the center of the internal refrigerant pipe **32**, and the liquid-phase refrigerant contained in the refrigerant flows along the inner wall of the internal refrigerant pipe **32** in the form of a liquid film. A difference in density between the gas-phase refrigerant and the liquid-phase refrigerant causes a difference in flow velocity between the gas-phase refrigerant and the liquid-phase refrigerant. This causes shear stress to act on the liquid-phase refrigerant in the form of a liquid film at the interface between the gas-phase refrigerant and the liquid-phase refrigerant, thus causing the liquid film to be scattered as droplets. Thus, many liquid-phase refrigerants are at the center of the internal refrigerant pipe **32** as droplets together with the gas-phase refrigerant.

[0042] When the refrigerant in the gas-liquid two-phase state flows through the curved portion **40**, a centrifugal force acts on the refrigerant, and the liquid-phase refrigerant contained in the refrigerant concentrates in the direction of the centrifugal force. In other words, the liquid film of the liquid-phase refrigerant adhering to the inner wall of the internal refrigerant pipe **32** has anon-uniform thickness. This causes the refrigerant that passes through the curved portion **40** to flow into the straight tube portion **41** with the liquid-phase refrigerant contained in the refrigerant concentrated in the positive X-direction.

[0043] When the refrigerant flows through the straight tube portion **41**, a secondary flow occurs that uniformizes a thickness of the liquid film of the liquid-phase refrigerant adhering to the inner wall, thus reducing the concentration of the liquid-phase refrigerant. When the straight tube portion **41** is sufficiently long, the concentration of the liquid-phase refrigerant is eliminated while the refrigerant is flowing through the straight tube portion **41**, and the refrigerant flows into the refrigerant distributor **31** with no concentration of the liquid-phase refrigerant. In the present embodiment, however, due to structural limitations, the straight tube portion **41** is shorter than the length that can eliminate the concentration of the liquid-phase refrigerant. The refrigerant thus flows into the refrigerant distributor **31** with the liquid-phase refrigerant concentrated in the positive X-direction.

[0044] As illustrated in FIG. **5**A, the refrigerant distributor **31** comprises an inlet tube **50** connected to the internal refrigerant pipe **32**, a mixing portion **51** connected to the inlet tube **50**, multiple outlet tubes **52** connected to the mixing portion **51**, and a recess portion **53** connected to the mixing portion **51**.

[0045] An upstream end of the inlet tube **50** is connected to the straight tube portion **41** in the internal refrigerant pipe **32**, and a downstream end of the inlet tube **50** is connected to an inlet **510** formed at an upstream end **51***a* of the mixing portion **51**. In other words, the internal refrigerant pipe **32** is connected to the inlet **510** through the inlet tube **50**. The upstream end **51***a* of the mixing portion **51** is an example of a first end. An inner diameter of the inlet tube **50** is equal to an inner diameter of the inlet **510** and is equal to an inner diameter of the straight tube portion **41** in the internal refrigerant pipe **32**. The refrigerant in the gas-liquid two-phase state sent out from the throttle device **2** flows into the inlet tube **50** through the main refrigerant pipe **5** and the internal refrigerant pipe **32**. The refrigerant flowing into the inlet tube **50** from the internal refrigerant pipe **32** flows into the mixing portion **51** from the inlet tube **50** through the inlet **510**.

[0046] The mixing portion **51** is hollow and has a cylindrical shape. A downstream end **51***b* of the mixing portion **51** is connected to the outlet tubes **52** and the recess portion

53. The downstream end **51***b* of the mixing portion **51** is opposite to the upstream end **51***a* and is an example of a second end. The refrigerant distributor **31** comprises eight outlet tubes **52** that are as many as the heat transfer tubes **30**, and these outlet tubes **52** are connected to the eight heat transfer tubes **30** described above. Each outlet tube **52** is connected to a different heat transfer tube **30**. Specifically, a downstream end of each outlet tube **52** is connected to the corresponding heat transfer tube **30**. An upstream end of each outlet tube **52** is end connected to one of multiple outlets **511** at the downstream end **51***b* of the mixing portion **51**. In other words, each heat transfer tube **30** is connected to the corresponding outlet **511** through the corresponding outlet tube **52**. The refrigerant flowing into the mixing portion **51** from the inlet tube **50** is sent out from the mixing portion **51** to each outlet tube **52** through the corresponding outlet **511**. In other words, the refrigerant flowing into the mixing portion **51** is distributed to each outlet tube **52**. The refrigerant flowing into each outlet tube **52** from the mixing portion **51** is sent out from the outlet tube **52** to the corresponding heat transfer tube **30** connected to the outlet tube **52**. In this manner, the refrigerant in the gas-liquid two-phase state flowing into the refrigerant distributor **31** is distributed to each heat transfer tube **30**.

[0047] An internal space of the recess portion **53** corresponds to a recess of the mixing portion **51**. The recess portion **53** communicates with the mixing portion **51** through a circular opening **51***c* formed at the downstream end **51***b* of the mixing portion **51** and facing the inlet tube **50** and the inlet **510**. In other words, the internal space of the recess portion **53**, that is the recess of the mixing portion **51**, is formed at the downstream end **51***b* of the mixing portion **51** and faces the inlet tube **50** and the inlet **510**. The recess portion **53** has a shape of a cylinder with a hollow cone connected to the cylinder, the cylinder being connected to the opening **51***c* in the mixing portion **51**. A part of the refrigerant sent out from the inlet **510** flows into the internal space of the recess portion **53**, and then flows into the mixing portion **51** from the recess portion **53**.

[0048] As described above, the refrigerant in the gas-liquid two-phase state flows into the refrigerant distributor **31** with the liquid-phase refrigerant contained in the refrigerant concentrating in the positive X-direction. If no countermeasures were taken, this would cause variations in the amount of the liquid-phase refrigerant sent out from each outlet **511** of the refrigerant distributor **31** to the corresponding outlet tube **52**. Specifically, as illustrated in FIG. 5B, the outlet tubes **52** are at positions different from each other in X-direction, and thus, if no countermeasures were taken, a larger amount of liquid-phase refrigerant would be sent out from an outlet tube **52** with a larger X-coordinate. The outlet tube **52** with the largest X-coordinate, through which the largest amount of liquid-phase refrigerant would be sent out if no countermeasures were taken, is hereafter referred to as a reference outlet tube **52***a* and distinguished from the other outlet tubes **52**. The reference outlet tube **52***a* and the other outlet tubes **52** are collectively and simply referred to as the outlet tubes **52** when there is no need to distinguish them from each other.

[0049] In the present embodiment, a height h (mm) of the mixing portion **51** satisfies Formula 1 below, where Di (mm) is the inner diameter of the inlet **510** illustrated in FIG. **6**, G (kg/h) is a mass flow rate of the refrigerant in the gas-liquid two-phase state flowing into the inlet tube **50** from the internal refrigerant pipe **32**, ρ_g (kg/m^3) is a gas-phase density of the refrigerant, and ρ_l (kg/m^3) is a liquid-phase density of the refrigerant. FIG. **6** is a cross-sectional view of the refrigerant distributor **31** taken along line VI-VI in FIG. 5B. The height h of the mixing portion **51** is a distance between the upstream end **51***a* and the downstream end **51***b* of the mixing portion **51** in the internal space of the mixing portion **51**. The gas-phase density ρ_g of the refrigerant in the gas-liquid two-phase state is the density of the gas-phase refrigerant contained in the refrigerant, and the liquid-phase density ρ_l of the refrigerant is the density of the liquid-phase refrigerant contained in the refrigerant. More specifically, in the present embodiment, the height h of the mixing portion **51** is 2.5 to 4 mm inclusive. As described later, this configuration can reduce variations in the amount of the refrigerant sent out from each outlet **511** through the corresponding outlet tube **52**, regardless of the flow rate, the type, and the temperature of the refrigerant in the gas-liquid two-phase state flowing through the inlet tube **50**.

$$\frac{\rho_l}{10\rho_g} - \sqrt{13\times 10^3\frac{\rho_l}{\rho_g}\exp\left(-18\frac{G}{D_i^2}\right) - 33.8} < h < \frac{\rho_l}{10\rho_g} + \sqrt{13\times 10^3\frac{\rho_l}{\rho_g}\exp\left(-18\frac{G}{D_i^2}\right) - 33.8} \quad (1)$$

[0050] Moreover, in the present embodiment, a diameter Dc (mm) of the recess formed in the mixing portion **51** that is the internal space of the recess portion **53** is larger than the inner diameter Di of the inlet **510**. Note that, the diameter Dc of the recess formed in the mixing portion **51** is equal to a diameter of the opening **51***c* formed in the mixing portion **51** to which the recess portion **53** is connected. As described later, this configuration can reduce variations in the amount of the refrigerant sent out from each outlet **511** through the corresponding outlet tube **52**.

[0051] Moreover, in the present embodiment, a first distance Li (mm) between the recess formed in the mixing portion **51** and an axial center QQ of each outlet **511** is larger than a second distance Lo (mm) between the axial center QQ of the outlet **511** and a side wall **51***d* of the mixing portion **51**. The axial center QQ of each outlet **511** aligns with the axial center of the corresponding outlet tube **52** connected to the outlet **511**. Note that, in the present embodiment, as illustrated in FIG. 5B, the downstream end **51***b* of the mixing portion **51** is circular, and the outlets **511** and the outlet tubes **52** are arranged apart from the recess portion **53** outwardly in the radial direction of the downstream end **51***b* of the mixing portion **51**. Specifically, the eight outlet tubes **52** and the outlets **511** to which the respective outlet tubes **52** are connected are arranged on the circumference of a single circle centered on the axial center of the recess portion **53**. Moreover, the axial center of the mixing portion **51** aligns with the axial center of the recess. Thus, the distances between the axial centers QQ of the outlets **511** and the recess are identical to one another, and the distances between the axial centers QQ of the outlets **511** and the side wall **51***d* of the mixing portion **51** are identical to one another. In other words, the first distances Li of the outlets **511** are identical to one another, and the second distances Lo of the outlets **511** are identical to one another.

[0052] This configuration facilitates processing in manufacturing the refrigerant distributor **31**, thus reducing the manufacturing cost. Specifically, for example, when the refrigerant distributor **31** is formed from a metal material such as copper or aluminum, the recess of the mixing portion **51** is formed by shaving the metal material with a drill. In this case, the shorter a distance between the outlet **511** and the recess of the mixing portion **51** is, the smaller an amount of metal material that exists between each outlet **511** and the recess of the mixing portion **51** is, the more difficult processing is, and the higher the manufacturing cost is. In the present embodiment, the first distance Li is larger than the second distance Lo, and thus the distance between each outlet **511** and the recess of the mixing portion **51** is sufficiently large, facilitating processing and suppressing the manufacturing cost. Note that, the material for the refrigerant distributor **31** is not limited to a metal material, and may be any material such as a resin. Moreover, the method for manufacturing the refrigerant distributor **31** is not limited to the method described above, and may be any method such as press forming or integral forming.

[0053] The distribution of the refrigerant performed by the refrigerant distributor **31** is described below using results of a simulation, performed using a computer, of the refrigerant flow inside the refrigerant distributor **31**. In this simulation, the refrigerant in the gas-liquid two-phase state is assumed to flow into the inlet tube **50** with the liquid-phase refrigerant contained in the refrigerant concentrating in the positive X-direction. This the simulation is performed under a condition in which the diameter Dc of the recess of the mixing portion **51** is 7 mm, the inner diameter Di of the inlet **510** is 6 mm, the first distance Li is 8.5 mm, and the second distance Lo is 2.5 mm. In the simulation, R290, i.e. propane, is used as the refrigerant unless otherwise specified. The temperature of the refrigerant in the simulation is 10° C.

[0054] FIG. **7** illustrates the relationship between the height h of the mixing portion **51** and the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* determined by simulation, when a mass flow rate G of the refrigerant flowing into the inlet tube **50** is 50 kg/h, 100 kg/h, 150 kg/h, or 200 kg/h. The liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* is a ratio of the mass flow rate of the liquid-phase refrigerant sent out from the outlet **511** connected to the reference outlet tube **52***a* through the reference outlet tube **52***a* to the sum of the mass flow rates of the liquid-phase refrigerant sent out from the respective outlets **511** through the corresponding outlet tubes **52**. When the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a*, of the eight outlet tubes **52**, through which the largest amount of liquid-phase refrigerant would be sent out if no countermeasures were taken, becomes lower, the liquid-phase refrigerant distribution ratios of the other outlet tubes **52** arranged apart from the reference outlet tubes **52***a* in the negative X-direction opposite to the direction in which the liquid-phase refrigerant concentrates become higher, and variations in the amount of the refrigerant sent out from each outlet **511** become smaller.

[0055] In FIG. **7**, polygonal lines illustrating the relationship between the height h of the mixing portion **51** and the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* are curved downward, regardless of the value of the mass flow rate G of the refrigerant. As is apparent from this fact, the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* reaches the minimum value when the height h of the mixing portion **51** is a specific value, and is greater than the minimum value when the height h is smaller than or larger than the specific value. Thus, by setting the height h of the mixing portion **51** to an appropriate value, the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* can be reduced, reducing variations in the amount of the refrigerant sent out from each outlet **511**.

[0056] FIG. **8** illustrates an example refrigerant flow inside the refrigerant distributor **31** when the refrigerant in the gas-liquid two-phase state flows into the refrigerant distributor **31** with a liquid-phase refrigerant CC contained in the refrigerant concentrating in the positive X-direction. FIG. **8** illustrates a vertical section of the refrigerant distributor **31** cut along a cut surface including the axial center of the inlet tube **50** and perpendicular to Y-direction. In FIG. **8**, arrow AA indicates a flow of the liquid-phase refrigerant CC contained in the refrigerant, and arrow BB indicates a flow of the gas-phase refrigerant contained in the refrigerant. As indicated by arrow BB, the gas-phase refrigerant flows through the center of the inlet tube **50**, flows into the recess portion **53** after being sent out from the inlet **510**, hits the inner wall of the recess portion **53**, and diffuses in the circumferential direction of the opening **51***c* in the mixing portion **51**. The gas-phase refrigerant then flows along the inner wall of the recess portion **53** and flows into the mixing portion **51**. The gas-phase refrigerant flowing into the mixing portion **51** hits, inside the mixing portion **51**, the liquid-phase refrigerant CC flowing along the inner wall of the inlet tube **50** in the form of a liquid film and flowing into the mixing portion **51** through the inlet **510**. After colliding with the liquid-phase refrigerant CC, the gas-phase refrigerant flows along the downstream end **51***b* of the mixing portion **51** toward the side wall **51***d* of the mixing portion **51**.

[0057] FIGS. **9**A to **9**C illustrate vertical sections of the refrigerant distributor **31** cut along a cut surface including the axial center of the inlet tube **50** and perpendicular to Y-direction. FIG. **9**A illustrates an example refrigerant flow inside the refrigerant distributor **31** when liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* is suppressed. In the example in FIG. **9**A, the gas-phase refrigerant hits the liquid-phase refrigerant CC inside the mixing portion **51**, thus causing the liquid-phase refrigerant CC to be pressed against the upstream end **51***a* of the mixing portion **51**. As indicated by arrow AA, the pressed liquid-phase refrigerant CC flows along the upstream end **51***a*, reaches the side wall **51***d* of the mixing portion **51**, flows along the side wall **51***d*, reaches the downstream end **51***b* of the mixing portion **51**, and is sent out from the outlet **511** to the reference outlet tube **52***a*. The liquid-phase refrigerant CC diffuses in the circumferential direction of the downstream end **51***b* of the mixing portion **51** while flowing along the upstream end **51***a* and the side wall **51***d* of the mixing portion **51**. This causes the liquid-phase refrigerant CC flowing into the mixing portion **51** while concentrating in the positive X-direction to partially move in the negative X-direction, reducing the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* and thus reducing variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511**.

[0058] FIG. **9**B illustrates an example refrigerant flow inside the refrigerant distributor **31** when the height h of the mixing portion **51** is too small to suppress the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a*. In the example in FIG. **9**B, the proportion occupied by the liquid-phase refrigerant CC in the internal space of the mixing portion **51** is higher than in the example in FIG. **9**A. Thus, in the example in FIG. **9**B, the flow rate of the gas-phase refrigerant flowing toward the side wall **51***d* of the mixing portion **51** after colliding with the liquid-phase refrigerant CC inside the mixing portion **51** is lower than in the example in FIG. **9**A, and, as indicated by arrow BB, the gas-phase refrigerant flowing toward the side wall **51***d* of the mixing portion **51** flows along the downstream end **51***b* of the mixing portion **51**, and then bends in a direction to flow into the reference outlet tube **52***a*. This causes apart of the liquid-phase refrigerant CC flowing in from the inlet **510** to be drawn by the gas-phase refrigerant bending in the direction to flow into the reference outlet tube **52***a* and flow directly into the reference outlet tube **52***a* without flowing along the upstream end **51***a* and the side wall **51***d* of the mixing portion **51**.

[0059] FIG. **9**C illustrates an example refrigerant flow inside the refrigerant distributor **31** when the height h of the mixing portion **51** is too great to suppress the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a*. In the example in FIG. **9**C, the amount of the gas-phase refrigerant flowing toward the side wall **51***d* of the mixing portion **51** without flowing into the recess portion **53** after flowing in from the inlet **510** is larger than in the example in FIG. **9**A. Thus, the force of the gas-phase refrigerant acting on the liquid-phase refrigerant CC to press the liquid-phase refrigerant CC against the upstream end **51***a* of the mixing portion **51** is lower than in the example in FIG. **9**A. Thus, as illustrated in FIG. **9**C, the liquid-phase refrigerant CC flows directly into the reference outlet tube **52***a* without flowing along the upstream end **51***a* and the side wall **51***d* of the mixing portion **51** after flowing into the mixing portion **51** from the inlet **510**.

[0060] In the examples in FIGS. **9**B and **9**C, the liquid-phase refrigerant CC flowing in from the inlet **510** flows directly into the reference outlet tube **52***a*, and thus, the liquid-phase refrigerant CC does not diffuse in the circumferential direction of the downstream end **51***b* of the mixing portion **51** before being sent out from the outlet **511** to the reference outlet tube **52***a*, the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* is not reduced, and variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511** are not suppressed. In contrast, in the example in FIG. **9**A, the liquid-phase refrigerant CC is restrained from directly flowing into the reference outlet tube **52***a*, the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* is suppressed, variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511** are suppressed.

[0061] When the height h of the mixing portion **51** is a value at which the liquid-phase refrigerant CC flows directly into the reference outlet tube **52***a*, the amount of the liquid-phase refrigerant CC sent out from each outlet **511** varies in the case where a centrifugal force acts on the refrigerant in the curved portion **40** in the internal refrigerant pipe **32**, as well as in the case where the gravity acts on the refrigerant in a direction that is not parallel to a direction of the refrigerant flow. Specifically, when the axial center of the inlet tube **50** is inclined with respect to a gravity direction g, the gravity acts on the refrigerant flowing through the inlet tube **50** in a direction that is not parallel to the direction of the refrigerant flow, and the liquid-phase refrigerant CC contained in the refrigerant concentrates. When the refrigerant flows into the mixing portion **51** from the inlet **510** with the liquid-phase refrigerant CC concentrating due to the effect of the gravity and the liquid-phase refrigerant CC flows directly into the reference outlet tube **52***a*, the amount of the liquid-phase refrigerant CC sent out from each outlet **511** varies. When the liquid-phase refrigerant CC is restrained from flowing directly into the reference outlet tube **52***a*, variations in the flow rate of the liquid-phase refrigerant CC sent out from each outlet **511** due to the inclination of the axial center of the inlet tube **50** with respect to the gravity direction g are reduced.

[0062] If the refrigerant distributor **31** were installed so that the axial center of the inlet tube **50** would be parallel to the gravity direction g when placing the refrigerant distributor **31** inside the indoor heat exchanger **3** or the outdoor heat exchanger **4**, the liquid-phase refrigerant CC would not concentrate due to the effect of the gravity described above. However, in reality, installing the refrigerant distributor **31** so that the axial center of the inlet tube **50** would be exactly parallel to the gravity direction g is difficult, and the refrigerant distributor **31** is usually installed with the axial center of the inlet tube **50** slightly inclined with respect to the gravity direction g.

[0063] Referring back to FIG. **8**, if the diameter Dc of the recess of the mixing portion **51** that is the internal space of the recess portion **53** were smaller than the inner diameter Di of the inlet **510**, an area of the gas-phase refrigerant flowing into the mixing portion **51** from the recess portion **53** contacting the liquid-phase refrigerant CC flowing into the mixing portion **51** from the inlet **510** would be small. The force with which the gas-phase refrigerant presses the liquid-phase refrigerant CC against the upstream end **51***a* of the mixing portion **51** would be thus low, and the liquid-phase refrigerant CC would be likely to flow directly into the reference outlet tube **52***a*. In the present embodiment, the diameter Dc of the recess of the mixing portion **51** is larger than the inner diameter Di of the inlet **510**. With this configuration, an area of the gas-phase refrigerant flowing into the mixing portion **51** from the recess portion **53** contacting the liquid-phase refrigerant CC flowing into the mixing portion **51** from the inlet **510** is large, and the force with which the gas-phase refrigerant presses the liquid-phase refrigerant CC against the upstream end **51***a* of the mixing portion **51** is large. This restrains the liquid-phase refrigerant CC from flowing directly into the reference outlet tube **52***a*, reducing the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* and thus reducing variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511**.

[0064] FIG. **10** illustrates the relationship between the mass flow rate G of the refrigerant in the gas-liquid two-phase state flowing into the inlet tube **50** from the internal refrigerant pipe **32** and the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* determined by simulation, with the height h of the mixing portion **51** being 2, 3, 4, or 5 mm. As illustrated in FIG. **10**, the higher the mass flow rate G of the refrigerant is, the lower the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* is, regardless of the height h of the mixing portion **51**. The higher the mass flow rate G of the refrigerant is, the larger the difference between the velocity of the gas-phase refrigerant contained in the refrigerant and the velocity of the liquid-phase refrigerant CC is, and the larger the difference between the dynamic pressure of the gas-phase refrigerant and the dynamic pressure of the liquid-phase refrigerant CC is. The larger the difference between the dynamic pressure of the gas-phase refrigerant and the dynamic pressure of the liquid-phase refrigerant CC is, the larger the force is with which the gas-phase refrigerant presses the liquid-phase refrigerant CC against the upstream end **51***a* of the mixing portion **51** when the gas-phase refrigerant flowing into the mixing portion **51** from the recess portion **53** hits the liquid-phase refrigerant CC flowing into the mixing portion **51** from the inlet **510**. The larger the force is with which the gas-phase refrigerant presses the liquid-phase refrigerant CC against the upstream end **51***a* of the mixing portion **51**, the more restrained the liquid-phase refrigerant CC is from flowing directly into the reference outlet tube **52***a*, and the lower the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* is.

[0065] FIG. **11** illustrates the relationship between the height h of the mixing portion **51** and the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* determined by simulation when the refrigerants used are R290 and R134A, i.e. 1, 1, 1, 2-tetrafluoroethene. In the example in FIG. **11**, a mass flow rate G of the refrigerant is 50 kg/h. If the temperatures are the same, a density ratio ρ_l/ρ_g being the ratio of the liquid-phase density ρ_l to the gas-phase density ρ_g of R134A is higher than the density ratio ρ_l/ρ_g of R290. As illustrated in FIG. **11**, when the refrigerant is R134A with a higher density ratio ρ_l/ρ_g, the height h of the mixing portion **51** at which the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* reaches a minimum value is larger than when the refrigerant is R290 with a lower density ratio ρ_l/ρ_g, and the minimum value is smaller. The larger the density ratio ρ_l/ρ_g of the refrigerant is, the difference between the velocity of the gas-phase refrigerant contained in the refrigerant and the velocity of the liquid-phase refrigerant CC is, and the larger the difference between the dynamic pressure of the gas-phase refrigerant and the dynamic pressure of the liquid-phase refrigerant CC is. The larger the difference between the dynamic pressure of the gas-phase refrigerant and the dynamic pressure of the liquid-phase refrigerant CC is, the more restrained the liquid-phase refrigerant CC is from flowing directly into the reference outlet tube **52***a*, and the lower the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* is.

[0066] The higher a mass flux, i.e. a mass flow rate per unit area, of the refrigerant sent out from the inlet tube **50** is, the lower the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* is. The mass flux of the refrigerant is proportional to the mass flow rate G of the refrigerant and inversely proportional to the square of the inner diameter Di of the inlet tube **50**.

[0067] As described above, the liquid-phase refrigerant distribution ratio of the reference outlet tube **52***a* depends on the height h of the mixing portion **51**, the mass flow rate G of the refrigerant, the density ratio ρ_l/ρ_g of the refrigerant, and the mass flux of the refrigerant. Based on the above, Formula 2 below expressing a liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a* is acquired by approximating the simulation results.

$$X = \exp\left(-10^{-3}\frac{G}{D_i^2}\right)\left\{\frac{18}{\left(\frac{\rho_l}{\rho_g}\right)}\left(h-\frac{\rho_l}{10\rho_g}\right)^2+33.8\right\} \tag{2}$$

[0068] The simulation results indicate that, regardless of the height h of the mixing portion **51**, the mass flow rate G of the refrigerant, the type of the refrigerant, and the temperature of the refrigerant, a part of the liquid-phase refrigerant CC sent out from the inlet tube **50** flows directly into the reference outlet tube **52***a* when the liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a* is greater than 0.13, and the liquid-phase refrigerant CC does not flow directly into the reference outlet tube **52***a* when the liquid-phase refrigerant distribution ratio X is less than 0.13. The range of the height h of the mixing portion **51** within which the liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a* represented by Formula 2 is less than 0.13 is expressed by Formula 1.

[0069] In the present embodiment, as described above, the height h of the mixing portion **51** satisfies Formula 1. In other words, the height h of the mixing portion **51** is a value within a range expressed by Formula 1. Thus, the liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a* expressed by Formula 2 is less than 0.13, and the liquid-phase refrigerant CC contained in the refrigerant is restrained from flowing directly into the reference outlet tube **52***a*, regardless of the mass flow rate G of the refrigerant, the type of the refrigerant, and the temperature of the refrigerant. In other words, since the height h of the mixing portion **51** satisfies Formula 1, the mixing portion **51** guides the refrigerant flowing in from the inlet **510** to flow along the upstream end **51***a* and the side wall **51***d* of the mixing portion **51** to diffuse the refrigerant in the circumferential direction of the downstream end **51***b* of the mixing portion **51**, and then sends out the refrigerant from the outlet **511**, regardless of the mass flow rate G of the refrigerant, the type of the refrigerant, and the temperature of the refrigerant. This configuration reduces variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511**, regardless of the mass flow rate G of the refrigerant, the type of the refrigerant, and the temperature of the refrigerant. Moreover, restricting liquid-phase refrigerant CC from flowing directly into the reference outlet tube **52***a* reduces variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511** due to inclination of the axial center of the inlet tube **50** with respect to the gravity direction g.

[0070] Using a computer, a simulation of the refrigerant flow inside the refrigerant distributor **31** was conducted with the height h of the mixing portion **51** satisfying Formula 1 for various values of parameters such as the number of outlet tubes **52**, the inner diameter Di of the inlet **510**, the diameter Dc of the recess of the mixing portion **51**, the first distance Li, and the second distance Lo, and the result showed that the liquid-phase refrigerant CC flowing directly into the reference outlet tube **52***a* is reduced, regardless of the values of these parameters. In other words, when the height h of the mixing portion **51** satisfies Formula 1, the liquid-phase refrigerant CC is restrained from flowing directly into the reference outlet tube **52***a*, regardless of the number of outlet tubes **52**, the inner diameter Di of the inlet **510**, the diameter Dc of the recess of the mixing portion **51**, the first distance Li, and the second distance Lo. Moreover, similar simulations were conducted with the height h of the mixing portion **51** set to various values within a range satisfying Formula 1, and the result showed that, when the height h of the mixing portion **51** is 2.5 to 4 mm inclusive, the liquid-phase refrigerant CC is significantly restrained from flowing directly into the reference outlet tube **52***a*, regardless of the values of the above parameters. In the present embodiment, as described above, the height h of the mixing portion **51** is 2.5 to 4 mm inclusive. This configuration reduces variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511**, regardless of the mass flow rate G of the refrigerant, the type of the refrigerant, the temperature of the refrigerant, the number of outlet tubes **52**, the inner diameter Di of the inlet **510**, the diameter Dc of the recess of the mixing portion **51**, the first distance Li, and the second distance Lo.

[0071] In the present embodiment, as described above, the height h of the mixing portion **51** satisfies Formula 1. The mixing portion **51** thus guides the refrigerant flowing in from the inlet **510** to flow along the upstream end **51***a* of the mixing portion **51** and the side wall **51***d* of the mixing portion **51** to diffuse the refrigerant in the circumferential direction of the downstream end **51***b*, and then sends out the refrigerant from the outlet **511**. This configuration reduces variations in the amount of the refrigerant sent out from each outlet **511**, regardless of the flow rate, the type, and the temperature of the refrigerant in the gas-liquid two-phase state flowing in from the inlet **510**. This reduces variations in the amount of the refrigerant distributed through each outlet **511** to the corresponding heat transfer tube **30** through the corresponding outlet tube **52** connected to the outlet **511**, regardless of the flow rate, the type, and the temperature of the refrigerant. In other words, this configuration can reduce variations in the amount of the refrigerant distributed to each heat transfer tube **30** when the refrigerant is distributed to the multiple heat transfer tubes **30**, regardless of the flow rate, the type, and the temperature of the refrigerant.

[0072] The refrigerant distributor **31** reduces variations in the amount of the refrigerant distributed to each heat transfer tube **30**, thus reducing variations in the heat load of the refrigerant in each heat transfer tube **30** and reducing variations in the section through which the refrigerant flows in the gas single-phase state in each heat transfer tube **30**. This improves the heat exchange efficiency of the indoor heat exchanger **3** or the outdoor heat exchanger **4** that comprises the refrigerant distributor **31** and functions as an evaporator, and improves the air conditioning efficiency of the air conditioner **100** comprising the indoor heat exchanger **3** and the outdoor heat exchanger **4**.

[0073] Moreover, in the present embodiment, the diameter Dc of the recess of the mixing portion **51** is larger than the inner diameter Di of the inlet **510**. This configuration can reduce variations in the amount of the refrigerant sent out from each outlet **511**.

[0074] Moreover, in the present embodiment, the outlets **511** are located apart from the recess of the mixing portion **51** outwardly in the radial direction of the downstream end **51***b* of the mixing portion **51**. The first distance Li that is a distance between the recess of the mixing portion **51** and the axial center QQ of each outlet **511** is larger than the second distance Lo that is a distance between the axial center QQ of each outlet **511** and the side wall **51***d* of the mixing portion **51**. This configuration facilitates processing when manufacturing the refrigerant distributor **31**, thus suppressing the manufacturing cost.

[0075] Note that, in the present embodiment, the diameter Dc of the recess of the mixing portion **51** is larger than the inner diameter Di of the inlet **510**, but this is a mere example. The diameter Dc of the recess of the mixing portion **51** may be smaller than or equal to the inner diameter Di of the inlet **510**.

[0076] Note that, in the present embodiment, the first distance Li is larger than the second distance Lo, but this is a mere example. The first distance Li may be smaller than or equal to the second distance Lo.

[0077] Note that, in the present embodiment, the height h of the mixing portion **51** is 2.5 to 4 mm inclusive, but this is a mere example. The height h of the mixing portion **51** may be any value that satisfies Formula 1.

Embodiment 2

[0078] Embodiment 2 of the present disclosure which reduces a pressure loss of the refrigerant inside the refrigerant distributor **31** is described below, focusing on the differences from Embodiment 1.

[0079] When a pressure loss of the refrigerant occurs inside the refrigerant distributor **31** included in the indoor heat exchanger **3** or the outdoor heat exchanger **4** that functions as an evaporator, the pressure of the gas-phase refrigerant flowing into the compressor **1** from the evaporator decreases. Thus, an amount of energy needed to be supplied to the gas-phase refrigerant to turn the low-pressure gas-phase refrigerant flowing into the compressor **1** into a high-pressure gas-phase refrigerant increases, and to supply this energy, a frequency of the compressor **1** needs to be increased. Increasing the frequency of the compressor **1** deteriorates the energy-saving performance of the air conditioner **100**.

[0080] The lower the gas-phase density ρ_g of the refrigerant is, the higher the flow velocity of the refrigerant in the gas-phase state is, and the larger the pressure loss of the refrigerant inside the refrigerant distributor **31** is. When the gas-phase density ρ_g of the refrigerant is lower than or equal to 20 kg/m^3, the deterioration of the energy-saving performance of the air conditioner **100** resulting from the pressure loss of the refrigerant inside the refrigerant distributor **31** is too large to be negligible. In the present embodiment, the gas-phase density ρ_g of the refrigerant is lower than or equal to 20 kg/m^3.

[0081] In the present embodiment, the height h of the mixing portion **51** satisfies Formula 1, is larger than 10/3 mm, and is smaller than or equal to 4 mm. As described later, this configuration reduces the pressure loss of the refrigerant inside the refrigerant distributor **31**, and thus improves the energy-saving performance of the air conditioner **100**.

[0082] The reduction of the pressure loss of the refrigerant inside the refrigerant distributor **31** is described below using the results of a simulation, performed using a computer, of the refrigerant flow inside the refrigerant distributor **31** when the refrigerant in the gas single-phase state flows into the inlet tube **50**. This simulation is performed under a condition in which the diameter Dc of the recess of the mixing portion **51** is 7 mm, the inner diameter Di of the inlet **510** is 6 mm, the first distance Li is 8.5 mm, and the second distance Lo is 2.5 mm. In the simulation, R290 is used as the refrigerant. The temperature of the refrigerant in the simulation is 10° C.

[0083] FIG. **12** illustrates the relationship between the height h of the mixing portion **51** and the pressure loss ΔP (kPa) of the refrigerant inside the refrigerant distributor **31** determined by simulation when the mass flow rate G of the refrigerant in the gas single-phase state flowing into the inlet tube **50** is 50, 100, 150, or 200 kg/h. The pressure loss ΔP of the refrigerant inside the refrigerant distributor **31** is the difference between a pressure of the refrigerant when the refrigerant flows into the inlet tube **50** and a pressure of the refrigerant when the refrigerant is sent out from the outlet tube **52**. As illustrated in FIG. **12**, the larger the height h of the mixing portion **51** is, the smaller the pressure loss ΔP of the refrigerant is, regardless of the mass flow rate G of the refrigerant.

[0084] The pressure loss ΔP of the refrigerant is proportional to the square of the velocity of the refrigerant. The velocity pf the refrigerant is proportional to the mass flow rate G of the refrigerant and inversely proportional to the square root of the gas-phase density ρ_g of the refrigerant. The pressure loss ΔP of the refrigerant can thus be expressed by Formula 3 below. In Formula 3, f(h) is a function using the height h of the mixing portion **51** as a variable.

$$\Delta P = f(h)\frac{G^2}{\rho_g} \tag{3}$$

[0085] Formula 4 below expressing the above function f(h) is acquired by approximating the simulation results.

$$f(h) = 5\left(1 + \frac{1}{h}\right) \tag{4}$$

[0086] As is evident from Formula 3, the smaller the value of the function f(h) is, the smaller the pressure loss ΔP of the refrigerant is. As is evident from Formula 4, the larger the height h of the mixing portion **51** is, the smaller the value of the function f(h) is. Thus, the larger the height h of the mixing portion **51** is, the smaller the pressure loss ΔP of the refrigerant is. The larger the height h of the mixing portion **51** is, the smaller a degree of bending of the refrigerant is when the refrigerant flowing in from the inlet **510** bends inside the mixing portion **51** to flow into each outlet **511**. In other words, the larger the height h of the mixing portion **51** is, the more likely it is for the refrigerant to flow inside the refrigerant distributor **31**. Thus, the larger the height h of the mixing portion **51** is, the smaller the pressure loss ΔP of the refrigerant is. The pressure loss ΔP of the refrigerant when the height h of the mixing portion **51** is infinite is hereafter referred to as a reference pressure loss. Moreover, a ratio of the pressure loss ΔP of the refrigerant to the reference pressure loss is hereafter referred to as a pressure loss ratio.

[0087] As the height h of the mixing portion **51** increases, the amount of increase in the value of the function f(h) when the height h of the mixing portion **51** increases by a unit amount gradually decreases. Thus, as the height h of the mixing portion **51** increases, the amount of decrease in the pressure loss ΔP of the refrigerant when the height h of the mixing portion **51** increases by a unit amount gradually decreases.

[0088] When the pressure loss ratio is less than or equal to 130%, the amount of decrease in the pressure loss ΔP of the refrigerant when the height h of the mixing portion **51** increases by a unit amount is extremely small. Thus, when the pressure loss ratio is less than or equal to 130%, reducing the pressure loss ΔP of the refrigerant by increasing the height h of the mixing portion **51** is extremely difficult. The pressure loss ΔP of the refrigerant thus substantially reaches the minimum value when the pressure loss ratio is less than or equal to 130%.

[0089] According to the result of the simulation, the pressure loss ratio is less than or equal to 130% when the height h of the mixing portion **51** is larger than 10/3 mm. In the present embodiment, as described above, the height h of the mixing portion **51** is larger than 10/3 mm. This configuration can suppress the pressure loss ΔP of the refrigerant to a substantially minimum value. This improves the energy-saving performance of the air conditioner **100**. Note that, when a simulation of the flow of the refrigerant inside the refrigerant distributor **31** was conducted using a computer with the height h of the mixing portion **51** being larger than 10/3 mm under variously set simulation conditions, such as the mass flow rate G of the refrigerant, the type of the refrigerant, the temperature of the refrigerant, the number of outlet tubes **52**, the inner diameter Di of the inlet tube **50**, the diameter Dc of the recess of the mixing portion **51**, the first distance Li, and the second distance Lo, the result showed that the pressure loss ratio was less than or equal to 130%, regardless of these simulation conditions. In other words, when the height h of the mixing portion **51** is larger than 10/3 mm, the pressure loss ratio is suppressed to less than or equal to 130%, regardless of the mass flow rate G of the refrigerant, the type of the refrigerant, the temperature of the refrigerant, the number of outlet tubes **52**, the inner diameter Di of the inlet tube **50**, the diameter Dc of the recess of the mixing portion **51**, the first distance Li, and the second distance Lo.

[0090] In the present embodiment, as described above, the height h of the mixing portion **51** is larger than 10/3 mm. This configuration reduces the pressure loss ΔP of the refrigerant inside the refrigerant distributor **31** and improves the energy-saving performance of the air conditioner **100**.

[0091] Note that, in the present embodiment, the height h of the mixing portion **51** is smaller than or equal to 4 mm, but this is a mere example, and the height h of the mixing portion **51** may be set to any value that satisfies Formula 1 and is larger than 10/3 mm. For example, when the diameter Dc of the recess of the mixing portion **51** is 7 mm, the inner diameter Di of the inlet **510** is 6 mm, the first distance Li is 8.5 mm, and the second distance Lo is 2.5 mm, the height h of the mixing portion **51** may be set to any value included in area FF illustrated in FIG. **13**. In the example in FIG. **13**, R290 is used as the refrigerant, and the temperature of the refrigerant is 10° C. In FIG. **13**, an area in which the height h of the mixing portion **51** is included below straight line DD is an area in which the height h of the mixing portion **51** is smaller than 10/3 mm, and an area in which the height h of the mixing portion **51** is included above straight line DD is an area in which the height h of the mixing portion **51** is larger than 10/3 mm. An area in which the mass flow rate G of the refrigerant is included below curve EE is an area in which the height h of the mixing portion **51** does not satisfy Formula 1, and an area in which the mass flow rate G of the refrigerant is included above curve EE is in an area in which the height h of the mixing portion **51** satisfies Formula 1. Area FF in which the height h of the mixing portion **51** is included above straight line DD and the mass flow rate G of the refrigerant is included above curve EE is an area in which the height h of the mixing portion **51** satisfies Formula 1 and is also larger than 10/3 mm.

Embodiment 3

[0092] Embodiment 3 of the present disclosure in which the refrigerant distributor **31** comprises a guide that guides the gas-phase refrigerant to the upstream end **51***a* of the mixing portion **51** is described below, focusing on the differences from Embodiment 1.

[0093] As illustrated in FIG. **14**A, the refrigerant distributor **31** according to the present embodiment differs from the refrigerant distributor **31** according to Embodiment 1 in that the refrigerant distributor **31** comprises a guide **54** connected to the downstream end **51***b* of the mixing portion **51**. Note that, for ease of understanding, the guide **54** is hatched in FIG. **14**A. The guide **54** has an annular shape with the axial center aligned with the axial centers of the mixing portion **51** and the recess portion **53** as viewed from the front. The guide **54** is located apart from the recess portion **53** outwardly in the radial direction of the downstream end **51***b* of the mixing portion **51**. In other words, the guide **54** is located apart from the recess of the mixing portion **51** that is the space inside the recess portion **53** outwardly in the radial direction of the downstream end **51***b* of the mixing portion **51**. Moreover, the guide **54** is located apart from each outlet **511** and each outlet tube **52** inwardly in the radial direction of the downstream end **51***b* of the mixing portion **51**.

[0094] As illustrated in FIG. **14**B, the guide **54** is provided protruding toward the upstream end **51***a* of the mixing portion **51**. FIG. **14**B is a cross-sectional view of the refrigerant distributor **31** according to the present embodiment taken along line A-A in FIG. **14**A. The guide **54** comprises a first side wall **54***a* inclined with respect to the downstream end **51***b* of the mixing portion **51** and a second side wall **54***b* perpendicular to the downstream end **51***b* of the mixing portion **51**. The first side wall **54***a* is an example of a guide side wall. The second side wall **54***b* is located more outward than the first side wall **54***a* in the radial direction of the downstream end **51***b* of the mixing portion **51**.

[0095] The first side wall **54***a* forms an angle θ of 45° or greater and less than 90° with respect to the downstream end **51***b* of the mixing portion **51**. The first side wall **54***a* has an inner end **60** and an outer end **61** located apart from the inner end **60** outwardly in the radial direction of the downstream end **51***b* of the mixing portion **51**. The inner end **60** of the first side wall **54***a* connects to the downstream end **51***b* of the mixing portion **51**. The outer end **61** of the first side wall **54***a* connects to the second side wall **54***b*. A third distance Lp that is a distance between the outer end **61** of the first side wall **54***a* and the upstream end **51***a* of the mixing portion **51** is smaller than a fourth distance Lq that is a distance between the inner end **60** of the first side wall **54***a* and the upstream end **51***a* of the mixing portion **51**. In other words, the outer end **61** of the first side wall **54***a* is located closer to the upstream end **51***a* of the mixing portion **51** than the inner end **60** of the first side wall **54***a*.

[0096] FIG. **15** illustrates an example refrigerant flow inside the refrigerant distributor **31** when the refrigerant in the gas-liquid two-phase state flows into the refrigerant distributor **31** with the liquid-phase refrigerant CC contained in the refrigerant concentrating in the positive X-direction. FIG. **15** illustrates a vertical section of the refrigerant distributor **31** cut along a cut surface including the axial center of the inlet tube **50** and perpendicular to Y-direction. In FIG. **15**, arrow AA indicates the flow of the liquid-phase refrigerant CC and arrow BB indicates the flow of the gas-phase refrigerant. The gas-phase refrigerant flowing into the recess portion **53** and then flowing into the mixing portion **51** from the recess portion **53** hits, inside the mixing portion **51**, the liquid-phase refrigerant CC flowing into the mixing portion **51** from the inlet **510**. After colliding with the gas-phase refrigerant, the liquid-phase refrigerant CC flows along the upstream end **51***a* of the mixing portion **51** toward the side wall **51***d* of the mixing portion **51** as indicated by arrow AA. On the other hand, after colliding with the liquid-phase refrigerant CC, the gas-phase refrigerant flows along the downstream end **51***b* of the mixing portion **51** toward the side wall **51***d* of the mixing portion **51** and reaches the guide **54** as indicated by arrow BB.

[0097] The gas-phase refrigerant that has reached the guide **54** flows from the downstream end **51***b* of the mixing portion **51** toward the upstream end **51***a* along the first side wall **54***a* of the guide **54**. In other words, the gas-phase refrigerant is guided from the downstream end **51***b* of the mixing portion **51** to the upstream end **51***a* by the first side wall **54***a* of the guide **54**. Thus, the gas-phase refrigerant is restrained from flowing into the reference outlet tube **52***a*, and the liquid-phase refrigerant CC is restrained from being drawn by the gas-phase refrigerant flowing into the outlet **511** to flow directly into the reference outlet tube **52***a*. Restraining the liquid-phase refrigerant CC from flowing directly into the reference outlet tube **52***a* suppresses the liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a*, thus reducing variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511**.

[0098] With this configuration, the gas-phase refrigerant is restrained from flowing into the reference outlet tube **52***a* by the guide **54** even when the height h of the mixing portion **51** is smaller than a value necessary to suppress the liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a* as in the example in FIG. **9**B described above. This restrains the liquid-phase refrigerant CC from flowing directly into the reference outlet tube **52***a*, reducing the liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a*.

[0099] As described above, the first side wall **54***a* of the guide **54** has an angle θ of 45° or greater and less than 90° with respect to the downstream end **51***b* of the mixing portion **51**. With this configuration, the gas-phase refrigerant is more likely to be guided to the upstream end **51***a* of the mixing portion **51** by the first side wall **54***a*, and thus the gas-phase refrigerant is more effectively restrained from flowing into the reference outlet tube **52***a*. This restrains the liquid-phase refrigerant CC from flowing directly into the reference outlet tube **52***a* more effectively, suppresses the liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a* more effectively, and suppresses variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511** more effectively.

[0100] In the present embodiment, as described above, the refrigerant distributor **31** guides, with the guide **54** connected to the downstream end **51***b* of the mixing portion **51**, the gas-phase refrigerant flowing into the mixing portion **51** from the downstream end **51***b* of the mixing portion **51** to the upstream end **51***a*. With this configuration, the liquid-phase refrigerant CC can be restrained from flowing directly into the reference outlet tube **52***a*, the liquid-phase refrigerant distribution ratio X of the reference outlet tube **52***a* can be suppressed, and variations in the amount of the liquid-phase refrigerant CC sent out from each outlet **511** can be reduced.

[0101] Note that, in the present embodiment, the guide **54** is provided protruding toward the upstream end **51***a* of the mixing portion **51**, but this is a mere example. The guide **54** may be provided protruding away from the upstream end **51***a* of the mixing portion **51** as illustrated in FIG. **16**. FIG. **16** is a vertical section of the refrigerant distributor **31** according to a modification cut along a cut surface including the axial center of the inlet tube **50** and perpendicular to Y-direction. In the modification illustrated in FIG. **16**, the second side wall **54***b* of the guide **54** is located more inward than the first side wall **54***a* in the radial direction of the downstream end **51***b* of the mixing portion **51**. In the present modification, the first side wall **54***a* of the guide **54** forms an angle θ of 45° or greater and less than 90° with respect to the downstream end **51***b* of the mixing portion **51** as in Embodiment 3 described above. The inner end **60** of the first side wall **54***a* connects to the second side wall **54***b*. The outer end **61** of the first side wall **54***a* connects to the downstream end **51***b* of the mixing portion **51**. In the present modification, as in Embodiment 3 described above, the third distance Lp that is a distance between the outer end **61** of the first side wall **54***a* and the upstream end **51***a* of the mixing portion **51** is smaller than the fourth distance Lq that is a distance between the inner end **60** of the first side wall **54***a* and the upstream end **51***a* of the mixing portion **51**. In other words, the outer end **61** of the first side wall **54***a* is located closer to the upstream end **51***a* of the mixing portion **51** than the inner end **60** of the first side wall **54***a*.

[0102] Note that, in the present embodiment, the second side wall **54***b* of the guide **54** is perpendicular to the downstream end **51***b* of the mixing portion **51**, but this is a mere example. The second side wall **54***b* of the guide **54** may have an angle less than 90° with respect to the downstream end **51***b* of the mixing portion **51**.

Modifications

[0103] While one or more embodiments of the present disclosure have been described above, the present disclosure is not limited to the above embodiments, and may be modified in various manners without departing from the spirit and scope of the present disclosure.

[0104] For example, in Embodiments 1 to 3 described above, the air conditioner **100** is used as a specific example of the refrigeration cycle device, but this is a mere example. The refrigeration cycle device according to one or more embodiments of the present disclosure may be a refrigeration cycle device other than an air conditioner, such as a heat pump water heater, a refrigerator, or a freezer.

[0105] In Embodiments 1 to 3 described above, the indoor heat exchanger **3** and the outdoor heat exchanger **4** that are examples of a heat exchanger cause the refrigerant to exchange heat with air, but this is a mere example. The heat exchanger in one or more embodiments of the present disclosure may cause the refrigerant to exchange heat with any substance. For example, when the heat exchanger in one or more embodiments of the present disclosure is included in a heat pump water heater, the heat exchanger causes the refrigerant to exchange heat with water.

[0106] In Embodiments 1 to 3 described above, the refrigerant distributors **31** are located at the inlets of the indoor heat exchanger **3** and the outdoor heat exchanger **4**, but this is a mere example. The refrigerant distributor according to one or more embodiments of the present disclosure may be located in the middle of a heat exchanger. Specifically, a reheat dehumidification air conditioner is known in which a throttle device is located in the middle of refrigerant paths of an indoor heat exchanger, and during the cooling operation, multiple refrigerant paths located upstream from the throttle device function as a condenser, and multiple refrigerant paths located downstream from the throttle device function as an evaporator. In such a reheat dehumidification air conditioner, the refrigerant distributor according to one or more embodiments of the present disclosure may be located in the middle of the refrigerant paths of the indoor heat exchanger and distribute the refrigerant to the multiple refrigerant paths downstream from the throttle device.

[0107] In Embodiments 1 to 3 described above, the number of heat transfer tubes **30** and outlet tubes **52** are set to be eight, but this is a mere example. The number of heat transfer tubes **30** and outlet tubes **52** may be any number greater than two.

[0108] In Embodiments 1 to 3 described above, the recess portion **53** has a shape of a cylinder with a cone connected to the cylinder, but this is a mere example. The recess portion **53** may have any shape. For example, the recess portion **53** may have a hemispherical shape.

[0109] In Embodiments 1 to 3 described above, the inner diameter of the inlet tube **50** is equal to the inner diameter of the straight tube portion **41** in the internal refrigerant pipe **32**, but this is a mere example. The inner diameter of the inlet tube **50** may be different from the inner diameter of the straight tube portion **41**. Note that, in this case, the inlet tube **50** and the straight tube portion **41** may be connected in any manner. For example, the inlet tube **50** and the straight tube portion **41** may be connected with a pipe having a tapered shape in which the inner diameter gradually decreases. Alternatively, the inlet tube **50** and the straight tube portion **41** may be connected with a stepped rod-like pipe.

[0110] In Embodiments 1 to 3 described above, the internal refrigerant pipe **32** is connected to the inlet **510** through the inlet tube **50**, but this is a mere example. The internal refrigerant pipe **32** may be directly connected to the inlet **510**. In this case, an end of the internal refrigerant pipe **32** functions as the inlet tube **50**.

[0111] In Embodiments 1 to 3 described above, each heat transfer tube **30** is connected to the corresponding outlet **511** through the corresponding outlet tube **52**, but this is a mere example. Each heat transfer tube **30** may be directly connected to the corresponding outlet **511**. In this case, an end of each heat transfer tube **30** functions as the outlet tube **52**.

[0112] Embodiments 1 to 3 described above may be combined with one another. For example, the refrigerant distributor **31** according to Embodiment 2 may be provided with the guide **54** according to Embodiment 3. This configuration reduces variations in the amount of the refrigerant sent out from each outlet tube **52** and also reduces the pressure loss of the refrigerant inside the refrigerant distributor **31**, thus improving the energy-saving performance of the air conditioner **100**.

[0113] The foregoing describes some example embodiments for explanatory purposes. Although the foregoing discussion has presented specific embodiments, persons skilled in the art will recognize that changes may be made in form and detail without departing from the broader spirit and scope of the invention. Accordingly, the specification and drawings are to be regarded in an illustrative rather than a restrictive sense. This detailed description, therefore, is not to be taken in a limiting sense, and the scope of the invention is defined only by the included claims, along with the full range of equivalents to which such claims are entitled.

[0114] This application claims the benefit of Japanese Patent Application No. 2022-019015, filed on Feb. 9, 2022, the entire disclosure of which is incorporated by reference herein.

REFERENCE SIGNS LIST

[0115] **1** Compressor
[0116] **2** Throttle device
[0117] **3** Indoor heat exchanger
[0118] **4** Outdoor heat exchanger
[0119] **5** Main refrigerant pipe
[0120] **6** Indoor unit
[0121] **7** Outdoor unit
[0122] **10** Refrigerant circuit
[0123] **20** Control device
[0124] **30** Heat transfer tube
[0125] **31** Refrigerant distributor
[0126] **32** Internal refrigerant pipe
[0127] **39** Joint portion
[0128] **39***a* Axial center of joint portion
[0129] **40** Curved portion
[0130] **41** Straight tube portion
[0131] **41***a* Axial center of straight tube
[0132] **50** Inlet tube
[0133] **51** Mixing portion
[0134] **51***a* Upstream end
[0135] **51***b* Downstream end
[0136] **51***c* Opening
[0137] **51***d* Side wall
[0138] **52** Outlet tube
[0139] **52***a* Reference outlet tube
[0140] **53** Recess portion
[0141] **54** Guide
[0142] **54***a* First side wall
[0143] **54***b* Second side wall
[0144] **60** Inner end
[0145] **61** Outer end
[0146] **100** Air conditioner
[0147] **510** Inlet
[0148] **511** Outlet
[0149] AA Flow of liquid-phase refrigerant
[0150] BB Flow of gas-phase refrigerant
[0151] CC Liquid-phase refrigerant
[0152] DD Straight line
[0153] EE Curve
[0154] FF Area
[0155] Dc Diameter of recess of mixing portion
[0156] Di Inner diameter of inlet
[0157] h Height of mixing portion
[0158] Li First distance
[0159] Lo Second distance
[0160] Lp Third distance
[0161] Lq Fourth distance
[0162] MM Saturated liquid line
[0163] NN Saturated vapor line
[0164] QQ Axial center of outlet
[0165] TL Straight line passing through axial center of joint and axial center of straight tube

1. A refrigerant distributor comprising a mixing portion having a cylindrical shape, wherein

the mixing portion has an inlet for inflow therethrough of a refrigerant at a first end of the mixing portion,

the mixing portion has a plurality of outlets for outflow therethrough of the refrigerant at a second end of the mixing portion opposite to the first end,

the mixing portion has a recess facing the inlet at the second end of the mixing portion,

the mixing portion guides a gas-phase refrigerant contained in the refrigerant flowing into the recess from the inlet to hit a liquid-phase refrigerant contained in the refrigerant flowing in from the inlet, thereby pressing the liquid-phase refrigerant against the first end of the mixing portion to guide the liquid-phase refrigerant to flow along the first end of the mixing portion and a side wall of the mixing portion to diffuse the liquid-phase refrigerant in a circumferential direction of the second end, and then sends out the liquid-phase refrigerant from the plurality of outlets,

the refrigerant distributor further comprises a guide located at the second end of the mixing portion, and

the guide guides, from the second end of the mixing portion toward the first end of the mixing portion, the gas-phase refrigerant guided by the mixing portion to hit the liquid-phase refrigerant.

2. (canceled)

3. The refrigerant distributor according to claim **1**, wherein

the refrigerant is in a gas-liquid two-phase state, and

the mixing portion has a height h (mm) satisfying

$$\frac{\rho_l}{10\rho_g} - \sqrt{13\times10^3\frac{\rho_l}{\rho_g}\exp\left(-18\frac{G}{D_i^2}\right) - 33.8} < h < \frac{\rho_l}{10\rho_g} + \sqrt{13\times10^3\frac{\rho_l}{\rho_g}\exp\left(-18\frac{G}{D_i^2}\right) - 33.8} \quad 1)$$

where h (mm) is a distance between the first end and the second end in an internal space of the mixing portion, Di (mm) is an inner diameter of the inlet, G (kg/h) is a mass flow rate of the refrigerant, ρ_g (kg/m^3) is a gas-phase density of the refrigerant, and ρ_l (kg/m^3) is a liquid-phase density of the refrigerant.

4. (canceled)

5. The refrigerant distributor according to claim **1**, wherein

the plurality of outlets are located apart from the recess outwardly in a radial direction of the second end of the mixing portion, and

a first distance being a distance between the recess and an axial center of each of the plurality of outlets is larger than a second distance being a distance between the axial center of each of the plurality of outlets and the side wall of the mixing portion.

6. The refrigerant distributor according to claim **1**, wherein a height h of the mixing portion is 2.5 to 4 mm inclusive.

7. The refrigerant distributor according to claim **1**, further comprising:

an inlet tube connected to the inlet; and

outlet tubes connected to the plurality of outlets.

8. The refrigerant distributor according to claim **1**, wherein a height h of the mixing portion is larger than 10/3 mm.

9. The refrigerant distributor according to claim **1**, wherein the guide is located apart from the recess outwardly in a radial direction of the second end of the mixing portion and apart from the plurality of outlets inwardly in the radial direction of the second end of the mixing portion,

the guide includes a guide side wall inclined with respect to the second end of the mixing portion,

the guide side wall has an inner end and an outer end located apart from the inner end outwardly in the radial direction of the second end of the mixing portion, and

the outer end of the guide side wall is located closer to the first end of the mixing portion than the inner end of the guide side wall.

10. A heat exchanger comprising:

the refrigerant distributor according to claim **1**; and

a plurality of heat transfer tubes connected to the plurality of outlets.

11. A refrigeration cycle device comprising a refrigerant circuit to circulate the refrigerant, the refrigerant circuit comprising:

the heat exchanger according to claim **10**,

a compressor to compress the refrigerant, and

a throttle device to expand the refrigerant.

12. A heat exchanger comprising:

the refrigerant distributor according to claim **1**; and

an internal refrigerant pipe connected to the inlet, wherein the internal refrigerant pipe comprises

a straight tube portion that has a linear shape and is connected to the inlet, and

a curved portion that is located upstream from the straight tube portion, is U-shaped, and is connected to the straight tube portion.

13. A refrigeration cycle device comprising a refrigerant circuit to circulate the refrigerant, the refrigerant circuit comprising

the heat exchanger according to claim **12**,

a compressor to compress the refrigerant, and

a throttle device to expand the refrigerant.

14. A refrigerant distributor comprising a mixing portion having a cylindrical shape, wherein

the mixing portion has an inlet for inflow therethrough of a refrigerant at a first end of the mixing portion,

the mixing portion has a plurality of outlets for outflow therethrough of the refrigerant at a second end of the mixing portion opposite to the first end,

the mixing portion has a recess facing the inlet at the second end of the mixing portion,

the mixing portion guides the refrigerant flowing in from the inlet to flow along the first end of the mixing portion and a side wall of the mixing portion to diffuse the refrigerant in a circumferential direction of the second end, and then sends out the refrigerant from the plurality of outlets, and

a diameter of the recess is larger than an inner diameter of the inlet.

15. The refrigerant distributor according to claim **14**, wherein

the refrigerant is in a gas-liquid two-phase state, and

the mixing portion has a height h (mm) satisfying

$$\frac{\rho_l}{10\rho_g} - \sqrt{13\times10^3\frac{\rho_l}{\rho_g}\exp\left(-18\frac{G}{D_i^2}\right) - 33.8} < h < \frac{\rho_l}{10\rho_g} + \qquad 2)$$

-continued

$$\sqrt{13\times10^3\frac{\rho_l}{\rho_g}\exp\left(-18\frac{G}{D_i^2}\right) - 33.8}$$

where h (mm) is a distance between the first end and the second end in an internal space of the mixing portion, Di (mm) is an inner diameter of the inlet, G (kg/h) is a mass flow rate of the refrigerant, ρ_g (kg/m^3) is a gas-phase density of the refrigerant, and ρ_l (kg/m^3) is a liquid-phase density of the refrigerant.

16. The refrigerant distributor according to claim **14**, wherein

the plurality of outlets are located apart from the recess outwardly in a radial direction of the second end of the mixing portion, and

a first distance being a distance between the recess and an axial center of each of the plurality of outlets is larger than a second distance being a distance between the axial center of each of the plurality of outlets and the side wall of the mixing portion.

17. The refrigerant distributor according to claim **14**, wherein a height h of the mixing portion is larger than 10/3 mm.

18. The refrigerant distributor according to claim **14**, further comprising a guide located at the second end of the mixing portion, wherein

the guide is located apart from the recess outwardly in a radial direction of the second end of the mixing portion and apart from the plurality of outlets inwardly in the radial direction of the second end of the mixing portion,

the guide includes a guide side wall inclined with respect to a main surface of the second end of the mixing portion,

the guide side wall has an inner end and an outer end located apart from the inner end outwardly in the radial direction of the second end of the mixing portion, and

the outer end of the guide side wall is located closer to the first end of the mixing portion than the inner end of the guide side wall.

19. A heat exchanger comprising:

the refrigerant distributor according to claim **14**; and

a plurality of heat transfer tubes connected to the plurality of outlets.

20. A refrigeration cycle device comprising a refrigerant circuit to circulate the refrigerant, the refrigerant circuit comprising

the heat exchanger according to claim **19**,

a compressor to compress the refrigerant, and

a throttle device to expand the refrigerant.

21. A heat exchanger comprising:

the refrigerant distributor according to claim **14**; and

an internal refrigerant pipe connected to the inlet, wherein the internal refrigerant pipe comprises

a straight tube portion that has a linear shape and is connected to the inlet, and

a curved portion that is located upstream from the straight tube portion, is U-shaped, and is connected to the straight tube portion.

22. A refrigeration cycle device comprising a refrigerant circuit to circulate the refrigerant, the refrigerant circuit comprising

the heat exchanger according to claim **21**,
a compressor to compress the refrigerant, and
a throttle device to expand the refrigerant.

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